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Disturbance Observer- Based Sliding Mode Control for the Precise Mechanical System with the Bristle Friction Model

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ABSTRACT

Tracking control schemes on the precise mechanical system in presence of nonlinear dynamic friction is proposed. A nonlinear dynamic friction is regarded as the bristle friction model to compensate for effects of friction. The conventional SMC method often shows poor tracking performance in high-precision position tracking application since it cannot completely compensate for the friction effect below a certain precision level. Thus to improve the precise position tracking performance, we propose the SMC method combined with the disturbance observer having tunable transient performance. Then this control scheme has the high precise tracking performance as well as a good transient response when it is compared with the conventional SMC method and the similar types of observers. The experiment on the XY ball-screw drive system with the nonlinear dynamic friction confirms the feasibility of the proposed control scheme.

Key Words : Bristle friction model, Sliding mode control, Model-based disturbance observer, XY ball-screw drive table

1. Introduction

Friction is a phenomenon that appears in almost every mechanical system contacted each other. It affects the tracking performance of the servo system such as machine tools and robots, etc. Specially, the tracking performance can be worse in the range of the low velocity when the precise motion is carried out. Thus, many researches have been executed to understand the mechanism of friction and to compensate the effects of the friction problem. The friction effects at moderate velocity are somewhat predictable. In this velocity range, friction model called as a classical friction model is built by the combination of the Coulomb and viscous friction that considers the friction model as a static relation of friction force and velocity. A classical friction model, however, fails to capture the low velocity effects such as the Stribeck effect, stick-slip motion, pre-sliding motion, break-away force, etc, which play a significant role in high precision position tracking applications. Besides these friction effects, through many researches, it has been revealed that friction is also influenced by the factors such as interior temperature of friction surface, contacting time, magnitude of load, operating distance, etc. The dynamic friction depending on the internal dynamics of friction mechanism, which is very difficult to model, causes these low velocity friction effects.

From the researches on the more complete friction model, Canudas de Wit [1] *et al.* present a new dynamic friction model which captures dynamic friction effects in low velocity as well as steady-state friction effects. They propose a state variable bristle model, called the bristle model, to describe the friction between two contacting surfaces. Now, this bristle model is often adopted as the friction model by many researchers for the friction compensation. The bristle model explains that in low velocity, the bristles between two contacting surfaces maintain the elastic property within tiny relative displacement and when they deflect beyond the elastic

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range, the sliding motion appears. The bristle deflections, however, cannot be measured directly and so must be estimated by the friction observer. In this paper, the dynamic friction of the ball-screw drive servo system is modeled as the bristle model.

In general, the control methods for a compensation of nonlinear friction divide two sides of approaches: friction model-based method and non-friction model-based method. The latter approach is mainly used when the exact friction model cannot be constructed and no need precise tracking performance. In low precision level, this approach can be applied conveniently since the procedure of the friction parameter identification is not needed and the structure of controller is also simple. The neural network control[2] and sliding mode control [3] (SMC) correspond to non-model based method. While the model-based method can be applied to the case that the identification for friction model is possible in a certain range of precision level, a more precise tracking performance can be obtained but cannot avoid a little complexity of control system and difficulty of the exact identification of friction parameters. PID/friction observer-based [1,4,5] control method, Lyapunov-based method[6] and adaptive control method [7,8] are included as this friction model-based method.

In this paper, we propose the friction observer-based SMC method to improve the position tracking performance of ball-screw drive table that is frequently used in machine tool in the presence of dynamic nonlinear friction. The proposed SMC method is constituted by combining the reducing chattering structure [9] SMC of integral type [10,11] with friction model-based disturbance observer. Then, it will be shown that the proposed method has a more precise position tracking performance compared with non-model based SMC method and similar friction observer-based control methods via real time experiments.

2. Design of SMC and Disturbance Observer

2.1 System dynamics

The dynamic model for the mechanical system in the presence of friction

$$J\ddot{q} + C_t\dot{q} + T_f = u \tag{1}$$

where J is a moment of inertia, C_t is viscous friction coefficient, $u(t)$ is the control input and $T_f(t)$ is the nonlinear dynamic friction. In bristle friction model, the interface between two surfaces is modeled by contact between sets of elastic bristles. When a tangential torque is applied, the elastic bristle will deflect like spring that gives rises to the friction torque. If the torque is increased beyond a certain magnitude, some of the elastic bristles deflect so much and they will slip. The average deflection of the elastic bristle is defined by $z(t)$ and it dynamic is represented as follows [1]:

$$\dot{z}(t) = \dot{q}(t) - f(\dot{q})z(t) \tag{2}$$

where

$$f(\dot{q}) = \frac{|\dot{q}|}{g(\dot{q})}. \tag{3}$$

In Eq. (3), $\dot{q}$ is the relative velocity between two contact surfaces. The function $g(\dot{q})$ is positive and depends on many factors such as the material properties, lubrication, and temperature. The dynamic friction term excluding viscous friction torque is described by

$$T_f(t) = \sigma_0\, z(t) + \sigma_1\, \dot{z}(t) \tag{4}$$

where σ_0 is the stiffness of the elastic bristle, σ_1 is a damping coefficient in elastic range. In Eq. (2) and Eq. (4), $T_f(t)$ can be represented by

$$T_f(t) = \chi(\dot{q})z + \sigma_1\dot{q} \tag{5}$$

where the auxiliary function $\chi(\dot{q})$ is defined as follows:

$$\chi(\dot{q}) = \sigma_0 - \sigma_1 f(\dot{q}) \tag{6}$$

We substitute Eq. (5) into Eq. (1)

$$J\ddot{q} + C_{eq}\,\dot{q} + T_z = u \tag{7}$$

where $C_{eq} = C_t + \sigma_1$ and $T_z = \chi(\dot{q})z$.

In Eq. (3), the function $g(\dot{q})$ is parameterized as follows:

$$g(\dot{q}) = \frac{1}{\sigma_0}[T_c + (T_s - T_c)e^{-(\dot{q}/\dot{q}_s)^2}] \tag{8}$$

For steady-state motion, the static friction torque containing viscous friction torque is given by

$$T_{fss}(\dot{q}) = [T_c + (T_s - T_c)e^{-(\dot{q}/\dot{q}_s)^2}]\mathrm{sgn}(\dot{q}) + C_t\dot{q} \tag{9}$$

where T_c is Coulomb friction level, T_s is stiction level, $\dot{q}_s$ is Stribeck velocity.

2.2 Design of sliding mode controller

We define the sliding surface of the SMC of the integral type and the position tracking error $e(t)$ as follows [10,11]:

$$s = \lambda_1 e + \dot{e} + \lambda_2 \int e dt \tag{10}$$

$$e = q_d - q \tag{11}$$

where λ_1 and λ_2 is the positive constant and q_d represents the desired position trajectory. Our objective is to choose the control force $u(t)$ such that the system state is driven to the sliding surface $s = 0$ regardless of friction. Choosing the Lyapunov function $V = \frac{1}{2}s^2$, the time-derivative of V is obtained by

$$\begin{aligned} \dot{V} &= s\dot{s} = s(\lambda_1 \dot{e} + \ddot{e} + \lambda_2 e) \\ &= s\left[\lambda_1 \dot{e} + \ddot{q}_d + \lambda_2 e + \frac{C_{eq}\dot{q} + T_z - u}{J}\right]. \end{aligned} \tag{12}$$

Let us choose the control force $u(t)$ of the form with unified smooth control law[9] instead of the traditional variable control law with switching or boundary layer term as follows:

$$\begin{aligned} u &= J(\lambda_1 \dot{e} + \ddot{q}_d + \lambda_2 e) + C_{eq}\dot{q} + \frac{\beta s}{\phi} \\ &= u_{eq} + u_f \end{aligned} \tag{13}$$

where $u_{eq} = J(\lambda_1 \dot{e} + \ddot{q}_d + \lambda_2 e) + C_{eq}\dot{q}$, $u_f = \frac{\beta s}{\phi}$, β is a positive constant whose magnitude depends on the system uncertainties and disturbances and ϕ is a positive constant relating to the thickness of the boundary layer. It was reported that the smooth control law has the advantages such as chattering free behavior, better robustness and ease of adaptability than the traditional variable control law with switching or boundary layer term. Let us substitute Eq.(13) into Eq.(12)

$$\dot{V} = -\frac{s}{J}\left(\frac{\beta s}{\phi} - T_z\right). \tag{14}$$

Now, β and ϕ must be chosen such that $|T_z| < \frac{\beta}{\phi}|s|$ in order to become $\dot{V} < 0$. Thus, the position tracking errors asymptotically go to zero even in the presence of friction. But in case that precision tracking performance is required, the condition of $|T_z| < \frac{\beta}{\phi}|s|$ leads to excessive level in the control input magnitude even though the smooth control law is introduced. In addition, a tracking performance can be worse for a certain precise tracking level if not compensating the friction effects.

Thus, in order to compensate the friction torque, let us add the estimated friction term $\hat{T}_z$ in the control input Then the control input can be written by

$$\begin{aligned} u &= J(\lambda_1 \dot{e} + \ddot{q}_d + \lambda_2 e) + C_{eq}\dot{q} + \frac{\beta s}{\phi} + \hat{T}_z \\ &= u_{eq} + u_f + \hat{T}_z \end{aligned} \tag{15}$$

and Eq. (14) is rewritten as

$$\dot{V} = -\frac{s}{J}\left(\frac{\beta s}{\phi} + \hat{T}_z - T_z\right). \tag{16}$$

From Eq. (16), if the friction torques are estimated such that $\hat{T}_z \to T_z$, then Eq. (16) is expressed as

$$\dot{V} = -\frac{\beta}{J\phi}s^2 \tag{17}$$

and then the tracking error $e(t)$ will go to zero asymptotically fast.

2.3 Design of disturbance observer

Since the state $z(t)$ cannot be measured directly, the observer which dynamically estimate unmeasurable state $z(t)$ for use with SMC system. Now, we suggest the exponentially stable observer with tunable transient performance in order to estimate the friction. Let us substitute Eq. (15) into Eq. (7) and rearrange it in terms of tracking error as follows:

$$J[(\ddot{q}_d - \ddot{q}) + \lambda_1 \dot{e} + \lambda_2 e] = T_z - \hat{T}_z - \frac{\beta s}{\phi} \quad (18)$$

Let us define the observation error for unmeasurable friction state as follows:

$$\tilde{z} = z - \hat{z} \quad (19)$$

Then

$$T_z - \hat{T}_z = \chi(\dot{q})\tilde{z} \quad (20)$$

so the following closed-loop error system is obtained

$$\dot{s} = \frac{1}{J}(\chi(\dot{q})\tilde{z} - \frac{\beta s}{\phi}) \quad (21)$$

Define the Lyapunov function as the following nonnegative function:

$$V_s = \frac{1}{2} J s^2 \quad (22)$$

Taking the time-derivative of Eq. (22) and substituting it into Eq. (21), the following expression can be obtained:

$$\dot{V}_s = s[\chi(\dot{q})\tilde{z} - \frac{\beta s}{\phi}]. \quad (23)$$

An observer, which exponentially estimates the state $z(t)$, is given by.

$$\hat{z} = w + \frac{J}{\sigma_1} s + (k_1 e + k_2 \dot{e}) \quad (24)$$

where w is an auxiliary variable and satisfy the following equation.

$$\begin{aligned} \dot{w} = \frac{1}{\sigma_1}[&-\sigma_0 w + (-C_{eq} + \sigma_1)\dot{q} - J\frac{\sigma_0}{\sigma_1} s + u \\ &+ \chi(\dot{q})s - \sigma_0(k_1 e + k_2 \dot{e}) \\ &- J(\ddot{q}_d + \lambda_1 e + \lambda_2 \dot{e})] - (k_1 \dot{e} + k_2 \ddot{e}) \end{aligned} \quad (25)$$

In order to prove exponential stability, let us redefine the following nonnegative Lyapunov function

$$V_{s2} = V_s + \frac{1}{2}\sigma_1 \tilde{z}^2 \quad (26)$$

In order to formulate the error dynamics, we first take the time-derivative of Eq. (26) and substitute Eq. (23) into it, the resulting expression is obtained as

$$\dot{V}_{s2} = s\left[\chi(\dot{q})\tilde{z} - \frac{\beta s}{\phi}\right] - \sigma_1 \tilde{z}\dot{\tilde{z}}. \quad (27)$$

Next, substituting Eq. (25) into the result of taking the time-derivative of Eq. (24), then

$$\begin{aligned} \dot{V}_{s2} &= s\left[\chi(\dot{q})\tilde{z} - \frac{\beta s}{\phi}\right] - \sigma_1 \tilde{z}\left[\frac{\chi(\dot{q})s}{\sigma_1} + \frac{\sigma_0}{\sigma_1}\tilde{z}\right] \\ &= -\frac{\beta s^2}{\phi} - \sigma_0 \tilde{z}^2 \end{aligned} \quad (28)$$

In Eq. (26), V_{s2} is positive-definite and in Eq. (28), since β, ϕ and σ_0 are positive, $\dot{V}_{s2}$ is negative-definite. Then, $\tilde{z}(t)$ go to zero exponentially fast and in Eq. (16), T_z approaches fast to $\hat{T}_z$. Therefore the effect of friction torque is almost disappeared and the observer-based SMC system maintains a good tracking performance under small scale of control input compared with non-observer based SMC system. Fig. 1 indicates the block diagram of the proposed controller.

Remark 1: The observer in Eq. (24) and (25) is similar to the exponentially stable observer suggested by Vedagarbha, *et al.* [5], where they proposed the following exponentially stable observer:

$$\hat{z} = p - \frac{J}{\sigma_1}\dot{q} \quad (29)$$

where p is an auxiliary variable and satisfy the following equation.

$$\dot{p} = \frac{1}{\sigma_1}\left[-\sigma_0 p + (-C_{eq} + \sigma_1 + J\frac{\sigma_0}{\sigma_1})\dot{q} + u + \chi(\dot{q})r\right] \quad (30)$$

But since the transient response of the observer suggested by Vedagarbha, *et al.* is limited by the value of the dynamic friction parameter (i.e., σ_0, σ_1), the transient performance cannot be adjusted independently. Thus they presented an observer that can improve the transient performance[5]. An observer that estimates the state $z(t)$ with tunable transient performance is given by

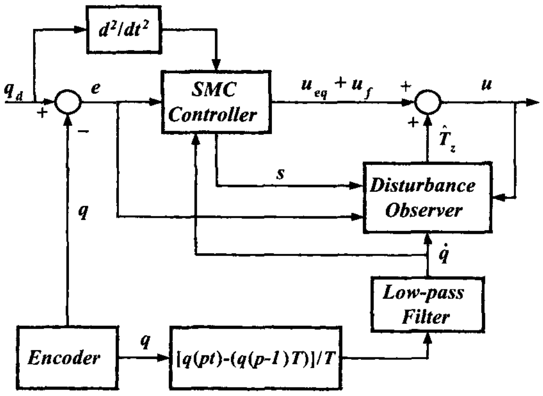

Fig. 1 Block diagram of the proposed controller

$$\hat{z} = p - \kappa_0 J \int_0^{\dot{q}} C(\dot{q}) d\dot{q} \quad (31)$$

where $p(t)$ is the auxiliary variable and is updated according to

$$\begin{aligned}\dot{p} = &-\kappa_0 C(\dot{q})[C_{eq}\dot{q} + \chi(\dot{q})(p - \kappa_0 J \textstyle\int_0^{\dot{q}} C(\dot{q})d\dot{q}) - u] \\ &- f(\dot{q})(p - \kappa_0 J \textstyle\int_0^{\dot{q}} C(\dot{q})d\dot{q}) + \chi(\dot{q})r + \dot{q}\end{aligned} \quad .(32)$$

In Eq. (30) and (32), the filtered tracking error $r(t)$ is defined as follows [5]:

$$r = \dot{e} + L^{-1}\left\{\frac{1}{s}K_F(s)e(s)\right\} \quad (33)$$

where L^{-1} denotes the inverse Laplace transform operation, s denotes the Laplace transform variable, and $K_F(s)$ denotes a linear filter which is selected to ensure that the transfer function given by

$$\frac{e(s)}{r(s)} = \frac{s}{s^2 + K_F(s)} \quad (34)$$

is strictly proper and exponentially stable. For example, a proportional integral derivative (PID) feedback law can be designed by defining $K_F(s)$ can be as follows:

$$K_F(s) = \alpha s + \beta \quad (35)$$

where α and β are a positive, scalar, constant control gains.

Remark 2: The observer of Eq. (31) and (32) suggested by Vedagarbha, *et al.*, has the limitation that $C(\dot{q})$ must be constructed to ensure the condition that $C(\dot{q})\chi(\dot{q}) \geq 0$ for guaranteeing exponential convergence of $\tilde{z}(t)$. Also the form of $C(\dot{q})$ is not unique since many different functions can be constructed to satisfy the requisite conditions and it is unknown how a particular choice of $C(\dot{q})$ from many different functions to satisfy the requisite conditions effects on the observer performance although Vedagarbha, *et al.* suggest that $C(\dot{q})$ can be constructed graphically only.

Remark 3: Our proposed observer has the tunable structure compared with the observer of Eq. (29) and Eq. (30), where the transient performance of the observer is fixed by the value of system parameter. Also no limitation exists such as $C(\dot{q})\chi(\dot{q}) \geq 0$ for guaranteeing exponential convergence of $\tilde{z}(t)$ in the observer of Eq. (31) and Eq. (32). Simply tuning the gains k_1 and k_2 such as PD controller, the transient performance can be handled easily. As a result, our observer has the structure to combine two observer suggested by Vedagarbha, *et al.* without any limitation for convergence of the estimation error $\tilde{z}(t)$.

3. Results of Experiment for XY Ball-screw Drive System

3.1 System description

The bristle friction model is characterized by six parameters, $\sigma_0\ (kg_f/cm)$, $\sigma_1\ (kg_f\sec/rad)$, $T_c\ (kg_f cm)$, $C_t\ (kg_f\sec/rad)$, $T_s\ (kg_f cm)$), $\dot{q}_s\ (rad/\sec)$ and these parameters must be identified via experimental process. In this paper, for the ball-screw drive table supported by LM guide, the experimental identification is executed to estimate the parameters of the bristle friction model. Sweeping slowly input voltage on the DC servo motor, the input current and velocity from the signal of encoder attached at each motor axis are measured. The measured input currents are converted to the friction torque using torque coefficients of DC servo motor. Through twenty-five experiments, the relation curve between the friction torque and velocity is obtained by the least square estimate. From these relation curves, the average value of four friction parameters T_c, T_s, C_t, and $\dot{q}_s$ is estimated.

However, steady-state friction data yields no information about σ_0, which represents bristle stiffness. Then, in pre-sliding range, the linear relation between the input torque and corresponding small pre-displacement until the sliding motion starts can be approximately searched. From the slope of this linear relation, the value of σ_0 can be obtained. σ_1, which is related to damping during friction transients, is computed by pre-sliding motion equation with assuming a damping factor 0.5^6. The identified values of the bristle friction model are given in Table 1.

Fig. 2 shows the photograph of the ball-screw drive table system. The LM guide supports the linear motion. In one side of axis, the DC servo motor is attached with the reduction gear. The rotary incremental type encoder reads the angular position with the resolution of *1000* pulse/rev.

In the computer, the control system is implemented by Simulink of Mathwork company and the DSP 1102 system of dSPACE company. The control signal is to send to the motor amplifier through the interface system to drive the servo motor. Since the angular velocity cannot be measured directly, we obtain the angular velocity $\dot{q}$ applying the backward difference algorithm to the position signal from encoder as follows with resulting signal being filtered by the low pass filter:

Table 1 The values of estimated friction parameters

	T_c	T_s	C_t	$\dot{q}_s$	σ_0	σ_1
X-axis	*0.9*	*1.13*	*1.1*	*0.056*	*86.4*	*4.7*
Y-axis	*0.93*	*1.1*	*0.006*	*3.32*	*11.2*	*0.8*

Table 2 The parameters of XY ball-screw drive system

	J ($kg_f \cdot cm \cdot \sec^2 / rad$)	pitch (mm)	gear ratio	amp. gain	torque constant
X-axis	*0.0251*	*0.5*	*20:1*	*2.73*	*2.27*
Y-axis	*0.0137*	*0.8*	*1:1*	*3.26*	*2.27*

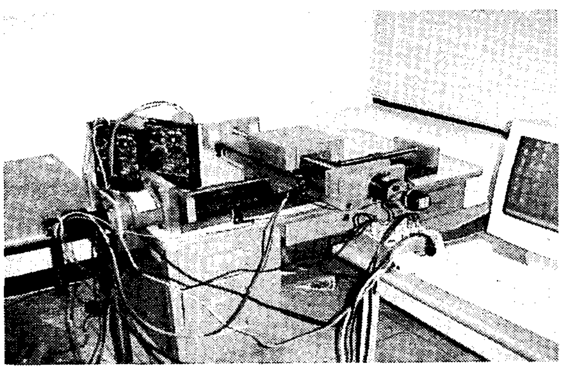

Fig. 2 Photograph of the ball-screw system

$$\dot{q}_i = \frac{q_i(pT) - q_i[(p-1)T]}{T}, \quad p = 1, \cdots, n \qquad (36)$$

where p, T and nT denote the sampling instance, sampling interval, and total experimental time, respectively.

3.2 Conventional SMC system without observer

Firstly, the position tracking experiment is executed for the circle command input (radius of $50\,\mu m$) with the design parameters shown in Table 3.

Table 3 The design parameters of the conventional SMC system

	λ_{1i}	λ_{2i}	β_i / ϕ_i
X-axis	*30*	*60*	*10/2*
Y-axis	*15*	*60*	*1/10*

In this case, the results of precise circle tracking for the conventional SMC system show that only the SMC

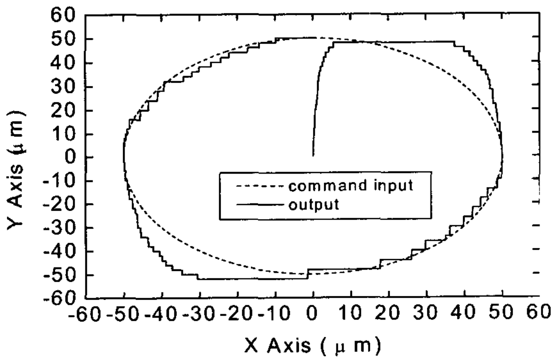

Fig. 3 Circle command input and tracking output of the SMC system without the observer

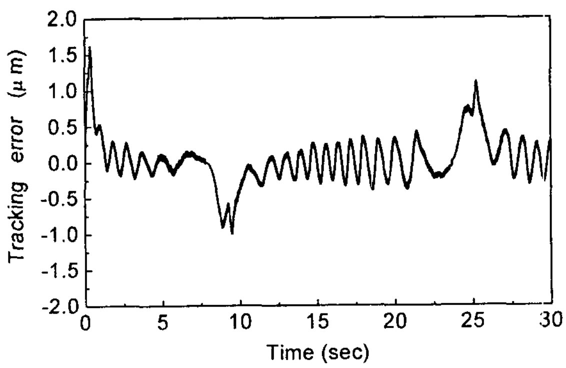

Fig. 4 Tracking error of X-axis of the SMC system without the observer for circle command input

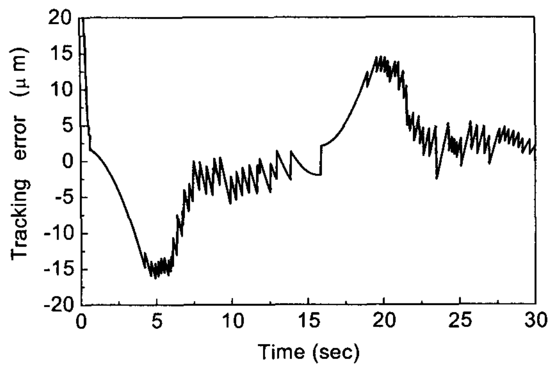

Fig. 5 Tracking error of Y-axis of the SMC system without the observer for circle command input

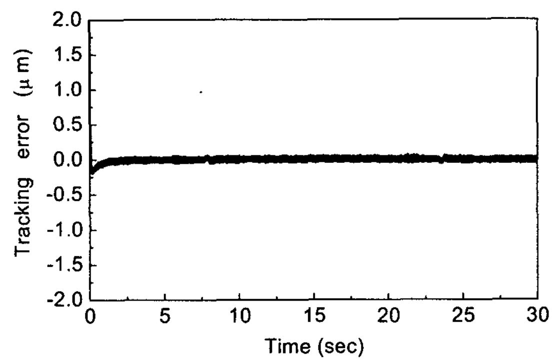

Fig. 7 Tracking error of X-axis of the SMC system with the proposed observer for circle command input

controller does not compensate the frictions completely as shown in Fig. 3, Fig. 4 and Fig. 5. In Fig. 4 and Fig. 5, the tracking error size of Y-axis is much larger than that of X-axis since the position resolution of Y-axis ball-screw is lower than that of X-axis.

3.3 SMC system with the proposed observer

Next, using the SMC system with the proposed tunable observer, we execute the position tracking experiment for the circle command input that is same as the previous case. The design parameters are selected as shown in Table 4.

Table 4 The design parameters of the SMC with the proposed observer

	λ_{1i}	λ_{2i}	β_i / ϕ_i	k_{1i}	k_{2i}
X-axis	*30*	*60*	*10/2*	*0.3*	10^{-5}
Y-axis	*15*	*60*	*1/10*	*2.5*	10^{-5}

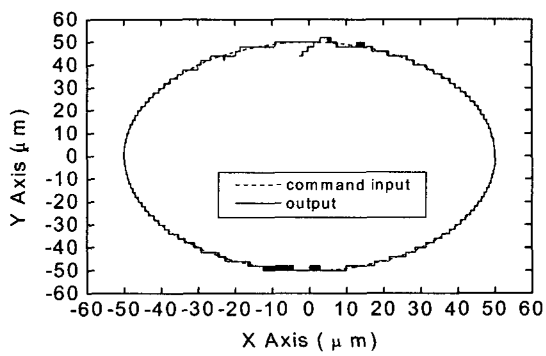

Fig. 6 Circle command input and tracking output of the SMC system with the proposed observer

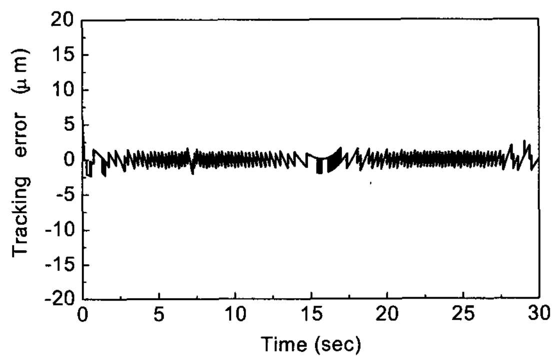

Fig. 8 Tracking error of Y-axis of the SMC system with the proposed observer for circle command input

In Fig. 5, Fig. 6 and Fig. 7, it is shown that the tracking performance is significantly improved when compared with the case of the SMC system without the observer. While the tracking errors of the SMC system without observer are stable within about $\pm 1.1\,\mu m$ (X-axis) and $\pm 17\,\mu m$ (Y-axis), the tracking errors of the SMC system with the proposed observer are stable within about $\pm 0.05\,\mu m$ (X-axis) and $\pm 2.5\,\mu m$ (Y-axis).

3.4 Transient performance of SMC system with the proposed observer

In this section, we examine the transient performance of the SMC system with the proposed observer for the X-axis ball-screw system. For two observers proposed by Vedagarbha, *et al.*, the command input $q_d = 0.0062832\ \cos(1.25t)$ (rad) (maximum amplitude of $5\,\mu m$) is selected to examine transient response when $\alpha = 30$ and $\beta = 60$. The transient outputs of the observer system of fixed parameter in Eq. (29) are shown in Fig. (9). But overshoot is very high and this overshoot cannot be adjusted by the observer itself without changing control parameter α and β. Then for the observer of Eq. (31), which can adjust the transient response, the same experiment is executed with respect to variations of tuning parameter κ_0. In Fig. (10), the transient response can be adjusted according to the value of κ_0, but overshoot is high as before.

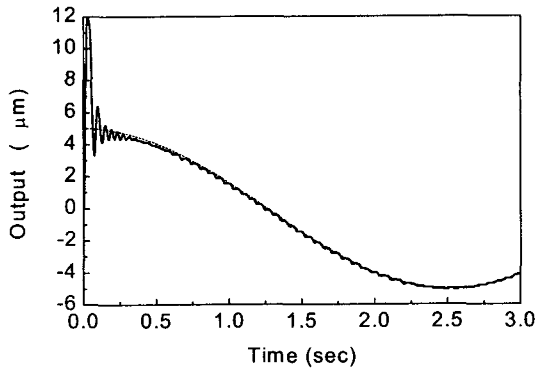

Fig. 9 Transient outputs of fixed parameter observer

For our proposed observer system, the same input command as the previous case is selected to examine transient response for fixed value of $\lambda_1 = 30$ and $\lambda_2 = 60$. In Fig. 11, it is shown that transient response can be adjusted by tuning proportional gain k_1, while k_2 is fixed. However, the rising time is a little longer than two observers proposed by Vedagarbha, *et al.*, but this time can be shortened by increasing value of λ_1 as shown in Fig. 12 without increasing control input excessively higher. This fact can be shown in Fig. 13 and Fig. 14. In Fig. 13

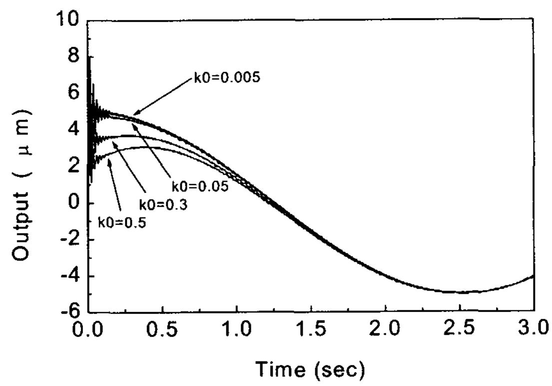

Fig. 10 Transient outputs of tuning parameter observer as variations of κ_0

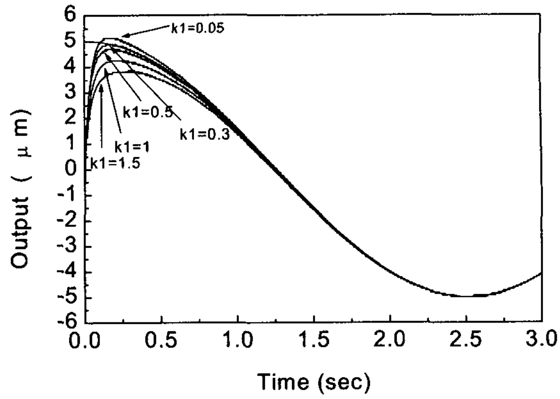

Fig. 11 Transient outputs of our proposed observer as variations of k_1

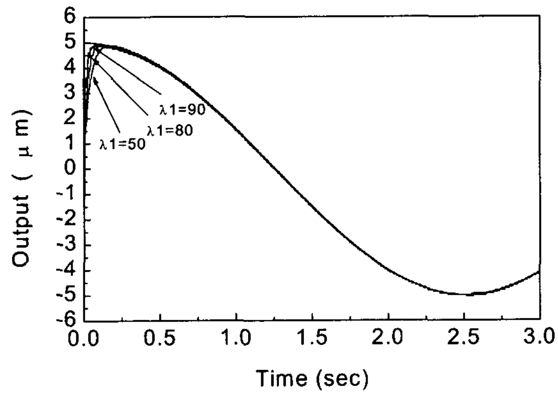

Fig. 12 Transient outputs of our proposed observer as variations of λ_1

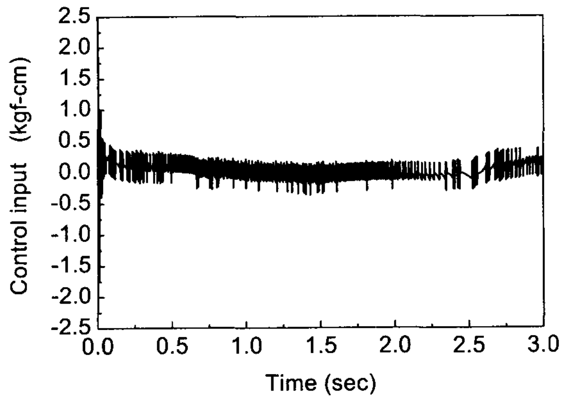

Fig. 13 Control input of the our proposed observer system for a cosine input command with $k_1 = 0.3$, $\lambda_1 = 90$

and Fig. 14, the magnitude of the transient control input of our proposed observer system with $k_1 = 0.3$, $\lambda_1 = 99$

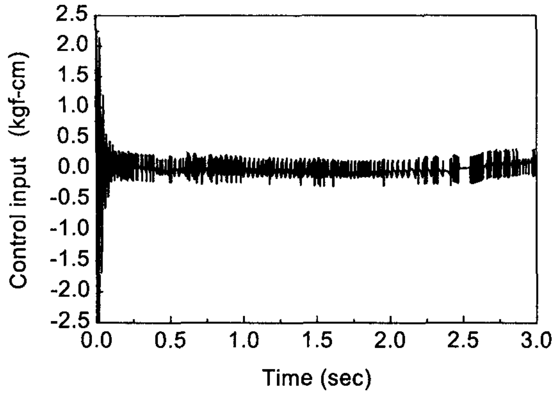

Fig. 14 Control input of the observer system proposed Vedagarbha, *et al.* for a cosine input command with $\kappa_0 = 0.05$, $\alpha = 30$

is smaller than the tunable observer proposed by Vedagarbha, *et al.* with $\kappa_0 = 0.05$, $\alpha = 30$. Thus, the adjusting of transient response can be more flexibly done in our proposed method under mild control input.

4. Conclusions

For a ball-screw drive system in the presence of nonlinear dynamic friction, a tunable observer-based sliding mode control scheme is proposed. Taking nonlinear dynamic friction as a bristle friction model, the parameters of bristle friction model are estimated by experiment identifications. When a precise level of position tracking is required, a conventional SMC system without friction observer cannot guarantee a good position tracking performance. Thus, we propose the SMC system combined with friction model-based observer with simple tunable structure such as PD control to improve the transient response as well as position tracking performance under a mild control input.

To show the effectiveness of our proposed control system, the experiment on the ball-screw drive system is executed via DSP system. The experiments show that the precise position tracking performance is accomplished and the transient state of output can be improved by tuning parameters of observer comparing with a similar observer proposed Vedagarbha, *et al.*.

Therefore, our proposed control scheme can be applied to control the dynamic friction satisfying precise tracking position performance as well as the adjusting transient response. As a future research, a design of observer that can estimate the friction state and velocity simultaneously will be needed since the velocity information obtained by the backward difference algorithm is weakly exact than directly obtained velocity information.

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