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Research on Disturbance Suppression Based Motion Axis Control of Machine Tools

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Abstract

For the high-precision tracking performance of machine tool motion axis control, this paper establishes a system mathematical model considering the elastic deformation of ball screws, and then designs a sliding mode controller to suppress the influence of uncertainty on control performance. Next, an extended state observer was designed to observe the system state and disturbances, and feed them back to the sliding mode controller for position control. Finally, the correctness of the designed sliding mode control and extended state observer was demonstrated through MATLAB simulation analysis.

Keywords

Disturbance Suppression, Tracking Control, Sliding Mode Control, Extended State Observer.

1. Introduction

PMSM (Permanent magnet synchronous motors) have the advantages of small size, reliable structure, high efficiency, and simple control, and are widely used in fields such as CNC (computer numerical control) machine tools, robots, aerospace, etc. The motion axis servo system drives the worktable to move according to a given input through a permanent magnet synchronous motor, which is commonly used in the position control system of CNC machine tools. This requires the control system to have good dynamic performance and steady-state accuracy. However, the machine tool servo system includes motor science, mechanical dynamics, control science, etc. It is a complex control system, and there are many factors that affect its performance. Among the many influencing factors, interference and uncertainty make it difficult to establish accurate mathematical models. Classify the disturbances and uncertainties experienced by the system into unmodeled dynamics, parameter uncertainties, and external disturbances (Yan, Yang, Sun et al., 2018). The unmodeled dynamics are mainly determined by their complex internal structure; Parameter uncertainty is due to the fact that some mechanical and electrical parameters of the system are often time-varying; External interference, as the most serious factor affecting the system, mainly includes load force and friction force. In response to these disturbances and uncertainties, traditional PID (proportion–integral–derivative) control has been widely used in the field of industrial control due to its simple control structure, ease of implementation, independence from mathematical models of objects, and certain robustness. But when the internal parameters and external disturbances of the system vary greatly, PID control cannot meet the performance requirements of the system well. To improve the robustness of the system and achieve satisfactory control performance, it is necessary to design an advanced control method to suppress the disturbances and uncertainties experienced by the system.

In recent decades, some advanced control strategies based on PID feedback control have been proposed and applied in various control fields. For permanent magnet synchronous motor servo systems, there are mainly self disturbance rejection control (Hebertt, Jesus, Carlos et al., 2014), adaptive control (Hu, Qiu, Lu, 2016), robust control (Kim, Lee, J., & Lee, K, 2018), sliding mode control (Zhang, Sun, Zhao et al., 2013), etc. And with the deepening of research, some advanced control strategies are combined with each other to form new control strategies with complementary advantages, such as adaptive sliding mode control (Zhang & Li, 2017), neural network sliding mode control (Wang & Tsai, 2018), etc. Among them, sliding mode control is widely used in high-performance control of servo motors due to its advantages such as insensitivity to disturbances and uncertainties, good robustness, and ease of implementation (Zhang, Zhang, He et al., 2016).

In this paper, a sliding mode control method is designed to track the position of the motion axis of machine tool under the condition of disturbance and uncertainty. Firstly, the system plant is modeled, and then the sliding mode controller is designed for the plant. To solve the problem of unknown disturbances and further improve the performance of sliding mode control, the extended state observer is designed. Finally, the simulation analysis of the above design verifies the correctness of the design.

2. Modeling of Motion Axis Servo System

In the positioning control of CNC machine tools, the modeling of the system has the most important impact on control performance. The motion axis servo system mainly consists of a motor drive unit, a mechanical transmission unit, and a signal detection unit. The motor drive unit drives the servo motor to rotate, and the mechanical transmission unit converts the rotational motion of the motor into linear motion of the worktable. The signal detection unit detects the working status of the worktable or motor during operation and feeds it back to the controller. The system structure is shown in the Fig.1, where v is the speed of the worktable, f is the friction force and Fd is the external disturbance of the worktable.

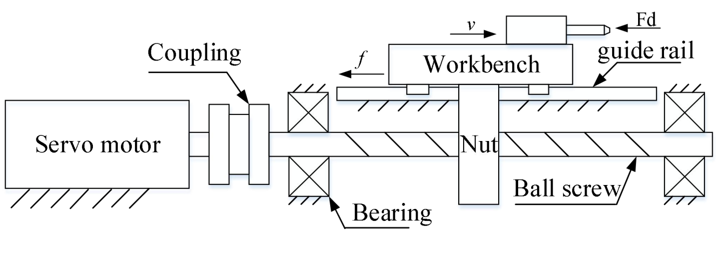

Figure 1. Schematic Diagram of System Structure

For the surface-mount PMSM, in the d-q rotating coordinate system, when $i_d = 0$, the mechanical equation of PMSM is

$$\begin{cases} T_e - T_L = J_m\ddot{\theta}_m + B_m\dot{\theta}_m \\ T_e = k_i i_q \end{cases} \tag{1}$$

Where T_e is output electromagnetic torque of the motor, T_L is load torque of motor, J_m is motor shaft moment of inertia, B_m is motor damping coefficient, θ_m is output angle of motor shaft, i_q is current of motor q axis, and k_i is the constant of motor torque.

For the machine tool motion axis system structure diagram in the Fig.1, there are mainly two degrees of freedom: the rotation of the motor shaft and the movement of the worktable. The following Lagrange equation can be established by dynamic analysis of the system.

$$\frac{d}{dt}\left(\frac{\partial L}{\partial \dot{q}_j}\right) - \frac{\partial L}{\partial q_j} + \frac{\partial D}{\partial \dot{q}_j} = Q_j \qquad j = 1, 2 \tag{2}$$

Where q is generalized coordinate of the system, that is, the two degrees of freedom of the system

$$q = \left[\theta_m \;\; \mathrm{x}_l\right]^T \tag{3}$$

Where x_l is the linear displacement of the worktable.

L is Lagrange function, which is the difference between kinetic energy T and potential energy V of the system. The kinetic energy mainly includes kinetic energy of motor and kinetic energy of worktable. The potential energy is mainly the elastic potential energy produced by the torsion deformation of the ball screw.

$$L = T - V = \frac{1}{2} J_m\dot{\theta}_m^2 + \frac{1}{2} M\dot{x}_l^2 - \frac{1}{2} K_T\left(\frac{x_l}{i} - \theta_m\right)^2 \tag{4}$$

Where M is the total mass converted to the workbench, KT is torsional stiffness of the ball screw, and i is screw transmission ratio.

D is dissipation function of the system, mainly including the dissipation energy produced by viscous damping in the system.

$$D = \frac{1}{2} B_1\dot{\theta}_m^2 + \frac{1}{2} B_2\dot{x}_l^2 \tag{5}$$

Where B1 is the viscous damping coefficient of the rotating motion part (i.e. the rotational motion of the motor shaft and screw), and B2 is the viscous damping coefficient of the linear moving part.

Q is generalized force of the system, mainly the electromagnetic torque Te of the motor and the external disturbance Fd of the workbench

$$Q = \begin{bmatrix} T_e & F_d \end{bmatrix}^T \tag{6}$$

Substituting equation (3) ~ (6) into equation (2) gives

$$\begin{cases} J_m \ddot{\theta}_m + K_T\left(\theta_m - \dfrac{x_l}{i}\right) + B_1\dot{\theta}_m = T_e \\ M\ddot{x}_l - \dfrac{1}{i}K_T\left(\theta_m - \dfrac{x_L}{i}\right) + B_2\dot{x}_l = F_d \end{cases} \tag{7}$$

Then the torque balance equation of the servo motor is

$$T_e = k_i i_q = J_m\dot{\theta}_m + B_m\dot{\theta}_m + i\left(M\ddot{x}_l + B\dot{x}_l - F_d\right) \tag{8}$$

To analyze the mechanism of the system more clearly, based on the Lagrange equation and the torque balance equation of the motor established above, the structural diagram of the motion axis servo system model can be obtained as shown in the Figure 2, The first half of Figure 2 is the mathematical model structure diagram of the rotating part. The combined force of electromagnetic torque Te and elastic torque Tds of the ball screw is used to drive the rotation of the servo motor rotating part, output angular displacement θm, and overcome its frictional torque. The latter half of Figure 2 is the mathematical model structure diagram of the linear moving part in the mechanical transmission mechanism. Due to the flexibility of the ball screw, there is a difference in displacement between the input end (i.e., the servo motor connection end) and the output end (i.e., the screw nut end) of the ball screw, resulting in elastic deformation. The elastic torque generated when the screw undergoes elastic deformation is the load torque for the rotating part of the motor, while it is the active torque for the ball screw nut pair. The screw nut overcomes the friction force of the worktable and the cutting load force of the tool holder according to the elastic driving force it receives, causing the worktable to move and produce displacement xl.

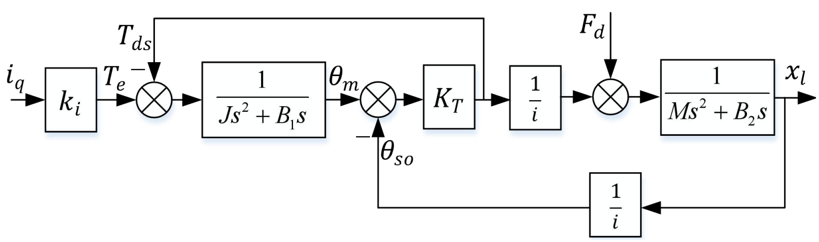

Figure 2. Structural Diagram of Servo Feed System Model for Machine Tool Motion Axis

Let the input u=iq. The state vector x=[x1 x2 x3 x4]T =[x_l θ_m $\dot{x}_l$ $\dot{\theta}_m$] are the output displacement of the workbench, the output angle of the motor shaft, the linear speed of the workbench and the angular velocity of the motor shaft respectively. Then the equation of state space is written as follows

$$\begin{bmatrix} \dot{x}_l \\ \dot{\theta}_m \\ \ddot{x}_l \\ \ddot{\theta}_m \end{bmatrix} = \begin{bmatrix} 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \\ \frac{-K_T}{i^2 M} & \frac{K_T}{iM} & \frac{-B_2}{M} & 0 \\ \frac{K_T}{iJ} & \frac{-K_T}{J} & 0 & \frac{B_1}{J} \end{bmatrix} \begin{bmatrix} x_l \\ \theta_m \\ \dot{x}_l \\ \dot{\theta}_m \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ 0 \\ \frac{k_i}{J} \end{bmatrix} u + \begin{bmatrix} 0 \\ 0 \\ \frac{1}{M} \\ 0 \end{bmatrix} F_d \tag{9}$$

Based on the parameters of the ball screw worktable and servo motor, the system object parameter table shown in Table.1 can be obtained.

Table 1. Object Parameter Table of Servo Feed System

Parameter	symbol	Value	unit
Electromagnetic torque constant	ki	0.85	N·m·A-1
Moment of inertia of motor shaft	Jm	0.00084	kg·m2
Inertia of screw rotation	Js	0.00074	kg·m2
Total inertia of rotating parts	J	0.00158	kg·m2
Screw torsional stiffness	KT	5430	N·m·rad-1
Screw transmission ratio	i	0.00127	m·rad-1
Total weight of workbench	M	25	kg
Equivalent damping of moving parts	B2	3.0	N·m-1·s

In servo motor control, it is often used as a three loop structure, that are current loop, speed loop and position loop in turn from inside to outside. The output of the position and speed controller can only output current iq by the current loop, and then produce electromagnetic torque. Because the electromagnetic time constant is far less than the mechanical time constant, the response speed of the current loop is much faster than that of the velocity loop and the position loop. Therefore, it is assumed that the current loop is a proportional link. The output of the outer loop controller is proportional to the electromagnetic torque (Chang, Chen, Ting et al., 2010).

3. System Controller Design

3.1 Design of Sliding Mode Controller

Sliding mode control is not sensitive to system disturbance and parameter perturbation, and easy to implement, thus the sliding mode controller is selected to control the system. Sliding mode control generally consists of the following three steps.

Determine the switching function s(x), that is, determine the sliding mode switching surface.

Determine corresponding control function u so that the sliding mode of the system exists, and reach switching surface within a limited time.

Determine the system is stable

According to the state space equation of the system mentioned above, the sliding surface in sliding mode control can be designed as follows, where C = [c1 c2 c3 1].

$$s(\boldsymbol{x})=\boldsymbol{C}^T\boldsymbol{x}=\sum_{i=1}^{4}c_i x_i=\sum_{i=1}^{3}c_i x_i+x_4 \tag{10}$$

The problem of position tracking of servo system is essentially to control the error signal. The error signal and the sliding mode function is designed as follows

$$\begin{cases} e_1=x_1^*-x_1=x_l^*-x_l \\ e_2=x_2^*-x_2=\theta_m^*-\theta_m \\ e_3=x_3^*-x_3=\dot{x}_l^*-\dot{x}_l \\ e_4=x_4^*-x_4=\dot{\theta}_m^*-\dot{\theta}_m \end{cases} \tag{11}$$

$$s=c_1e_1+c_2e_2+c_3e_3+e_4 \tag{12}$$

Where C1, C2 and C3 must satisfy Hurwitz condition, so that, let C1 = λ3, C2 = 3λ2, C3 = 3λ, and λ > 0.

The Lyapunov function is defined as follow

$$V_1=\frac{1}{2}s^2 \tag{13}$$

Then

$$\begin{aligned} \dot{s}=c_1\dot{e}_1+c_2\dot{e}_2+c_3\dot{e}_3+\dot{e}_4=c_1(\dot{x}_l^*-\dot{x}_l)+c_2(\dot{\theta}_m^*-\dot{\theta}_m)+ \\ c_3[\ddot{x}_l^*-(\frac{K_T}{iM}\theta_m-\frac{K_T}{i^2M}x_l-\frac{B_2}{M}\dot{x}_l+\frac{1}{M}F_d)]+[\ddot{\theta}_m^*-(-\frac{K_T}{J}\theta_m-\frac{B_1}{J}\dot{\theta}_m+\frac{K_T}{iJ}x_l+\frac{k_i}{J}u)] \end{aligned} \tag{14}$$

In order to reduce the chattering effect of sliding mode control, the exponential reaching law is adopted

$$\dot{s}=-\varepsilon\,\mathrm{sgn}s-ks \qquad s>0,k>0 \tag{15}$$

By substituting the above formula into equation (14), the sliding mode control rate is obtained as follows

$$\begin{aligned} u=\frac{J}{k_i}\left\{\begin{matrix}\varepsilon\,\mathrm{sgn}\,s+ks+\ddot{\theta}_m^*+c_1(\dot{x}_l^*-\dot{x}_l)+c_2(\dot{\theta}_m^*-\dot{\theta}_m)+ \\ c_3\left[\ddot{x}_l^*-\left(\frac{K_T}{iM}\theta_m-\frac{K_T}{i^2M}x_l-\frac{B_2}{M}\dot{x}_l+\frac{1}{M}F_d\right)\right]\end{matrix}\right\} \\ +\frac{K_T}{k_i}\theta_m+\frac{B_1}{k_i}\dot{\theta}_m-\frac{K_T}{ik_i}x_l \end{aligned} \tag{16}$$

In the above control rate, the control variable u cannot be realized because the disturbance Fd is unknown. When the upper and lower bounds of Fd are known, the control rate can be designed by the bounds of Fd

$$u=\frac{J}{k_i}\left\{\begin{array}{l}\varepsilon\,\mathrm{sgn}\,s+ks+\ddot{\theta}_m^*+c_1(\dot{x}_l^*-\dot{x}_l)+c_2(\dot{\theta}_m^*-\dot{\theta}_m)\\+c_3\left[\ddot{x}_l^*-\left(\frac{K_T}{iM}\theta_m-\frac{K_T}{i^2M}x_l-\frac{B_2}{M}\dot{x}_l\right)\right]\end{array}\right\}+\frac{K_T}{k_i}\theta_m+\frac{B_1}{k_i}\dot{\theta}_m-\frac{K_T}{ik_i}x_l-\frac{c_3J}{k_iM}d_c \tag{17}$$

Where dc is a quantity related to the boundary of disturbance Fd, Substituting equation (17) into equation (15) yields:

$$\dot{s}=-\varepsilon\,\mathrm{sgn}\,s-ks-\frac{c_3}{M}F_d+\frac{c_3}{M}d_c=-\varepsilon\,\mathrm{sgn}\,s-ks+\frac{c_3}{M}\left(d_c-F_d\right) \tag{18}$$

In order to satisfy the stability condition, let

$$d_c=\frac{d_U+d_L}{2}-\frac{d_U-d_L}{2}\mathrm{sgn}s \tag{19}$$

$$d_L\le F_d\le d_U \tag{20}$$

When s (t)>0, $\dot{s}=-\varepsilon\,\mathrm{sgn}\,s-ks+\frac{c_3}{M}(d_L-F_d)<0$, then $\dot{s}(t)s(t)<0$

When s (t)<0, $\dot{s}=-\varepsilon\,\mathrm{sgn}\,s-ks+\frac{c_3}{M}(d_U-F_d)>0$, then $\dot{s}(t)s(t)<0$

Therefore, the system is stable at this time, and can be achieved through the boundary of interference while satisfying the sliding mode arrival condition.

3.2 Design of Extended State Observer

The state observer can observe the internal state of the plant according to its input and output, this makes it easy to get some state variables that are difficult to measure in the system, such as velocity and acceleration, for the sliding mode control rate of the system, the speed of the workbench and the angular speed of the motor can be obtained by the observer. However, the control rate u also contains the unknown disturbance information of the system, so the extended state observer can be designed to observe the system state and unknown disturbance. The extended state observer takes the disturbance as the expanded state, and feeds back the observed value to the controller for control compensation.

The extended state observer is designed as follows

$$
\begin{cases}
e = \hat{x}_l - x_l \\
\dot{\hat{x}}_l = \hat{\dot{x}}_l + L_1 e \\
\dot{\hat{\theta}}_m = \hat{\dot{\theta}}_m + L_2 e \\
\dot{\hat{\dot{x}}}_l = -\dfrac{K_T}{i^2 M}\hat{x}_l + \dfrac{K_T}{iM}\hat{\theta}_m - \dfrac{B_2}{M}\hat{\dot{x}}_l + \dfrac{1}{M}\hat{F}_d + L_3 e \\
\dot{\hat{\dot{\theta}}}_m = \dfrac{K_T}{iJ}\hat{x}_l - \dfrac{K_T}{J}\hat{\theta}_m - \dfrac{B_M}{J}\hat{\dot{\theta}}_m + \dfrac{k_i}{J}u + L_4 e \\
\dot{\hat{F}}_d = L_5 e
\end{cases}
\tag{21}
$$

Where $\hat{x}_l, \hat{\theta}_m, \hat{F}_d$ are the observed values of x_l, θ_m, F_d, L1，L2，L3，L4，L5 are observer gains. By designing appropriate observer coefficients, when t →∞, there are:

$$
\hat{x}_l \to x_l \;,\; \hat{\theta}_m \to \theta_m \;,\; \hat{\dot{x}}_l \to \dot{x}_l \;,\; \hat{\dot{\theta}}_m \to \dot{\theta}_m \;,\; \hat{F}_d \to F_d
\tag{22}
$$

Then

$$
u = \frac{J}{k_i}\left\{
\begin{array}{l}
\varepsilon \operatorname{sgn} s + ks + \ddot{\theta}_m^* + c_1(\dot{x}_l^* - \hat{\dot{x}}_l) + c_2(\dot{\theta}_m^* - \hat{\dot{\theta}}_m) + \\
c_3\left[\ddot{x}_l^* - \left(\dfrac{K_T}{iM}\hat{\theta}_m - \dfrac{K_T}{i^2 M}\hat{x}_l - \dfrac{B_2}{M}\hat{\dot{x}}_l + \dfrac{1}{M}\hat{F}_d\right)\right]
\end{array}
\right\}
+ \frac{K_T}{k_i}\hat{\theta}_m + \frac{B_1}{k_i}\hat{\dot{\theta}}_m - \frac{K_T}{ik_i}\hat{x}_l
\tag{23}
$$

The control block diagram of sliding mode control based on extended state observer is shown in Fig.3.

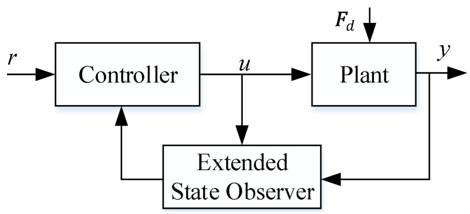

Figure 3. Block Diagram of Sliding Mode Control Based on Extended State Observer

4. Simulation And Analysis

Conduct Simulink simulation analysis to verify the correctness of the design. Design the S-function of the object model based on the mathematical model of the controlled object, and then design the S-function of the controller according to the design idea of sliding mode control function. Connect the forward channel and feedback channel in sequence according to the signal transmission relationship as shown in Fig.4. The system conducts simulation experiments on the given position input as a step signal and a sine signal, where the step signal input is r (t)=1, t ≥ 0; the sine signal input is r (t)=sin (t), t ≥ 0. Assuming that the system is subjected to a disturbance force of 1000N at t=10s. In the sliding mode control rate, λ=5 is taken. For the parameters ε and k of the exponential approaching law, in order to ensure that the sliding mode approaching law has a fast approaching speed and minimize chattering, ε should be taken as a small value and k as a large value. In this paper, ε=0.1 and k=20 are taken.

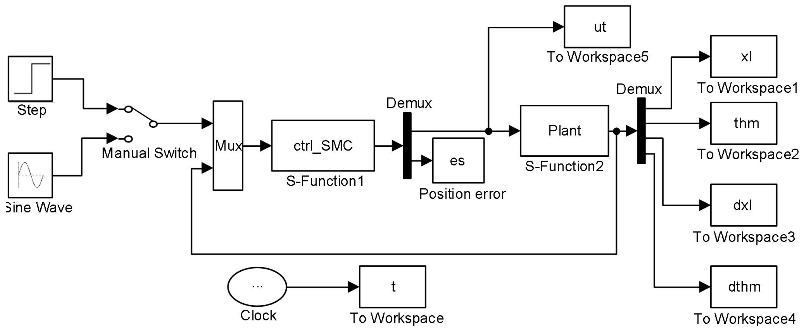

Figure 4. Simulation Diagram of Sliding Mode Control

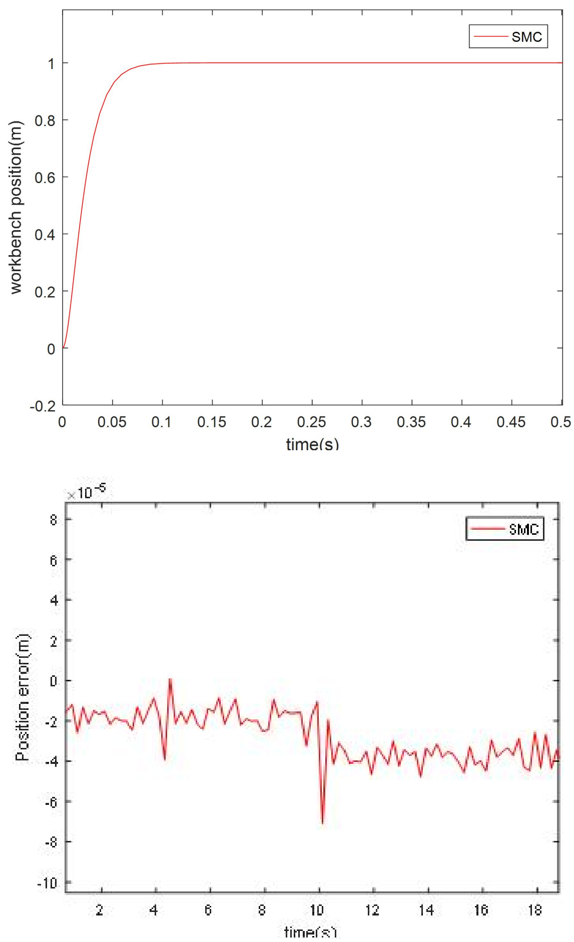

Figure 5. Simulation Diagram of Sliding Mode Control under Step Input

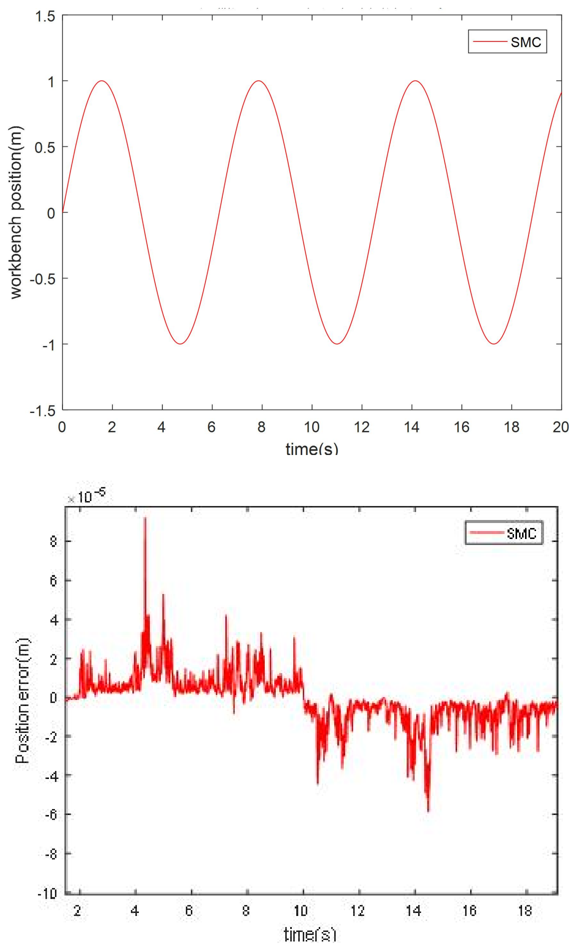

Figure 6. Simulation Diagram of Sliding Mode Control under Sinusoidal Input

From the simulation curve, it can be seen that position sliding mode control (SMC) has high control accuracy for step response and sine response, and the control error is within the range of 10-5m. In the case of sudden disturbance (1000N), the position mutation caused by disturbance is small, the adjustment is rapid, and it has good disturbance suppression ability.

In simulation, a certain deterministic force is applied to the actual controlled object to verify the disturbance estimation capability of the extended state observer. In terms of the specific magnitude of the interference force, the first step is to apply a constant force of 1000N at t=10s to verify its ability to observe and estimate forces of constant magnitude. The specific simulation results are shown in Figure 7.

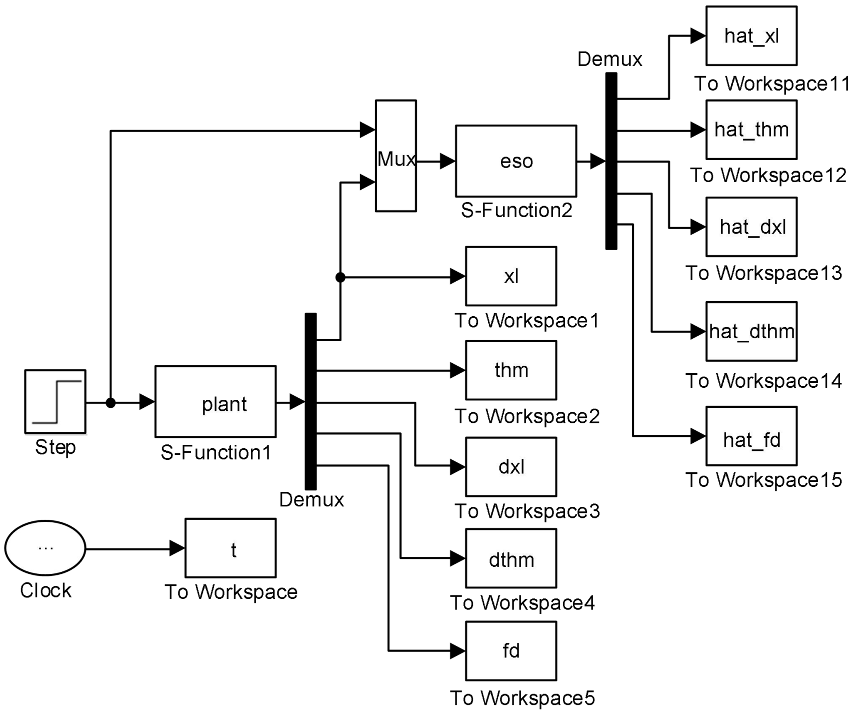

Figure 7. Simulation Structure of Extended State Observer

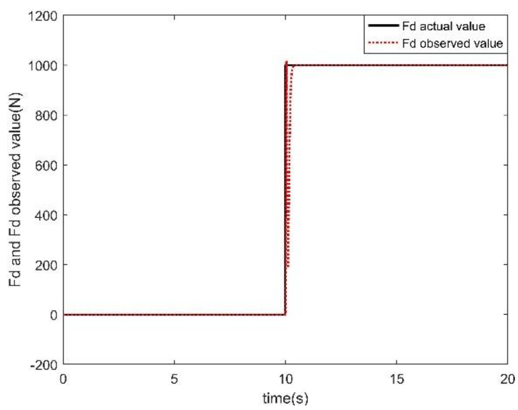

Figure 8. Extended State Observer Disturbance Observation Diagram

From the simulation results, it can be seen that the designed extended state observer not only has fast convergence for the observation and estimation of system object disturbances, but also has high observation and estimation accuracy. Therefore, an extended state observer can be used to observe and estimate system disturbances.

Then, assuming that the value of disturbance is known, the Fd is 1000N, the sliding mode control based on the boundary of disturbance and sliding mode control based on extended state observer are respectively adopted for simulation analysis. For the given position of 0.02t, the results are shown in the fig.9. It can be seen from the figure that the sliding mode control based on the boundary of disturbance can't estimate the real-time situation of the disturbance, so it can only consider that the disturbance force of 1000N acts on the plant in the whole process, which results in the greater tracking error of the first half part and the phenomenon of jitter. The sliding mode control based on the extended state observer can observe the disturbance in real time and carry out compensation control, the control effect is better and the jitter is not obvious.

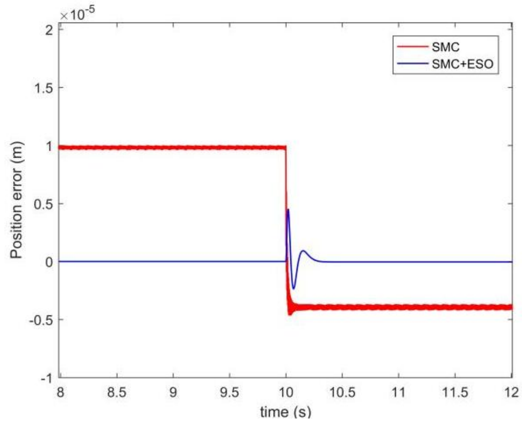

Figure 9. Position Errors of Sliding Mode Control Based on the Boundary of Disturbance (SMC) and Extended State Observer (SMC+ESO)

5. Conclusion

A system model considering the elastic deformation of ball screw is established for the position tracking control of machine tool motion axis servo system, and a fourth-order mathematical model of the system is obtained. Then, a sliding mode controller is designed to overcome the problem of unknown system disturbances, and an extended state observer is designed to observe the internal state and disturbances of the system. Finally, simulations are conducted on two sliding mode control methods, and the comparison show that sliding mode control based on extended state observer can effectively observe and suppress disturbances.

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