Research Journal of Applied Sciences, Engineering and Technology 4(21): 4237-4244, 2012
ISSN: 2040-7467

Submitted: December 20, 2011 Accepted: April 23, 2012 Published: November 01, 2012

Improvement of Hydraulic Edge Position Control System by Proportion Sliding Mode of Self-tuning Switching Gain

Jin Baoquan and Chen Dongbing
Key Lab of Ministry of Education for Advanced ,Transducers and Intelligent Control System,Taiyuan University of Technology, China

Abstract: Aiming at the problem of tracking performance degradation of hydraulic EPC system caused by time-varying inertia parameters, nonlinear and external disturbances, the proportion sliding mode control of fuzzy self-tuning gain was proposed. The EPC system state space model on deviation parameters was established and the main feedback sliding mode switching algorithm was designed. The fuzzy method was used to dynamically adjust the proportion sliding mode switching gain by product of the switching function and its derivative state and to adaptive compensate for the uncertainty of the system. At the same time to ensure the effectiveness of the design strategy, the controller model and physical model worked together to simulate the actual conditions. The fixed switching gain switch was, respectively greater and smaller and compared with the fuzzy self-tuning gain, in which the latter achieves a fast and coordinated control of chattering. The results show that after comprehensive consideration all interference the system is stable, fast response, high accuracy and to solve chattering problem caused by the traditional large switching gain of proportion sliding mode.

Keywords: Co-simulation, fuzzy, hydraulic EPC, proportion sliding mode, switching gain

INTRODUCTION

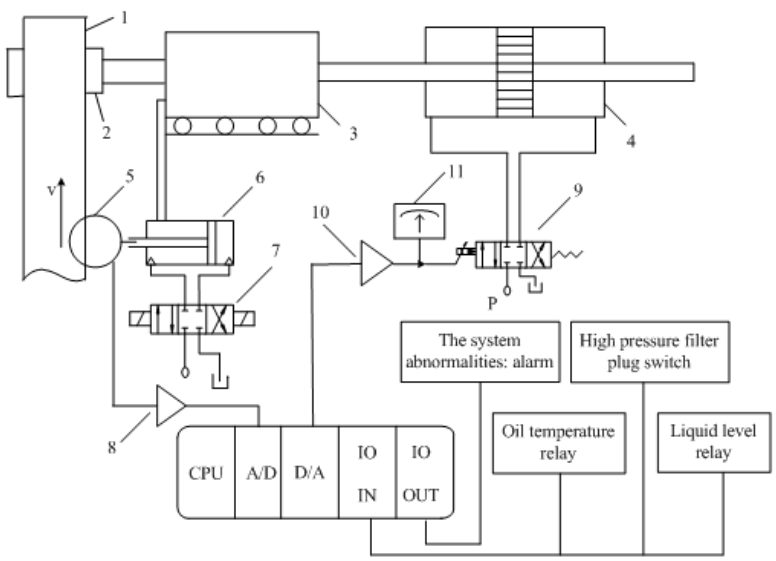
Electro-hydraulic servo control system has some significant advantages of compact structure, small size, light weight, good acceleration, the large volume elastic modulus, low-speed stability, etc. Therefore it has occupied an important position in the automation field and has been largely used in the defense industry and civilian industry. However, there are many unfavorable factors for system performance in electro-hydraulic servo system, such as the nature nonlinear of hydraulic servo systems, parameter uncertainty, mechanical resonance, un-modeled dynamics, sensor response characteristics (Zhong and He, 2008; Su *et al*., 2010). The 'robustness' of variable structure control systems is generally stronger than the conventional continuous control system. But for a real sliding mode system, the 'chattering' is caused by limited control, the system inertia, switching time and space lags, measurement error and the quasi-sliding mode' formed with discrete systems, etc. At home and abroad many scholars have carried out research around the elimination of chattering. The system parameter perturbation and external disturbances invariance controlled with sliding mode variable structure are achieved by high-frequency chattering. Due to the high-frequency chattering in theory is infinitely fast, but in practice real executive mechanism can not be achieved. Therefore, the chattering can be weakened and not be completely eliminated. If the chattering is removed, the perturbation and anti-disturbance ability of sliding mode system are eliminated. At the same time, its performance of high-frequency input is easy to stimulate the mechanical system resonance or un-modeled characteristics, thus affects the actual control performance of the system. In practice, if the design of sliding mode controller can weaken chattering, it will be able to greatly expand its application areas and obtain the properties to meet the technical requirements (Luo *et al*., 2009; Jin *et al*., 2007; Wang *et al*., 2008). Servo systems for improved sliding mode control research has yielded some results, such as the research results of adaptive variable structure are: multi-model switching adaptive sliding mode, adaptive bipolar sigmoid function, Backstepping reverse recursive method, etc Fang *et al*. (2010), Zhang and Wang (2009) and Guan and Zhu (2005). The method of a combination with intelligent control is adopted, such as the research combining with fuzzy theory: variable universe adaptive fuzzy sliding mode with dead-time compensation, the boundary layer thickness dynamically adjusted by the fuzzy system, adaptive fuzzy sliding compensated with friction torque etc. Bai *et al*. (2009), Gao *et al*. (2007) and Liu *et al*. (2010). These studies have greatly expanded the application scope of sliding mode control and obtained better control effect. If in practice the design of the sliding mode algorithms is complex, it is only simulated by the numerical phantom and then is difficult to realize in engineering. However fuzzy control is easy to implement with computer software and do not rely on accurate

Corresponding Author: Jin Baoquan, Key Lab of Ministry of Education for Advanced ,Transducers and Intelligent Control System, Taiyuan University of Technology, China

mathematical model and it will further enhance the control effect if it combines with the sliding mode. If the design of fuzzy sliding mode control can meet the real-time while retaining their robustness, the entire sliding mode quality is assured and improved and the chattering is effectively inhibited. Than that has the important engineering value. As proportion sliding mode retains some useful properties of linear control, it has high precision characteristics. The sliding mode control system is constituted by the nonlinear and uncertainty electro hydraulic system used with all phase variables and then that has many shortcomings of higher-order derivative and switch quantity, so it could easily lead to some problems of poor noise suppression and complex structure. Proportion switch sliding with the main feedback has the advantages of simple structure and easy project implementation, but also it more suitable for constituting a closed-loop control of electro hydraulic servo system. In this study the fuzzy sliding mode controller is design by using the main feedback proportion switch sliding and then satisfactory control performance is obtained by the complementary of the control characteristics. According to system requirements of real-time and robustness, fuzzy sliding mode control is designed. The input signals designed with fuzzy sliding mode control are always two dimensional, so the complex structure of fuzzy control can be simplified. In electro hydraulic control system the impact factors are complex, so an accurate model can be used to describe its all features and then the simplified models are often adopted in the control system analysis and design. But in the design ignored factors may cause the quality deterioration of the control system, even lead to instability. In light of this, the state space model of the system based on deviation variable is used, controller parameters are initially identified by numerical simulation, the AMEsim physical models and numerical controller model of the actual hydraulic mechanical system are used, the co-simulation of the electro-hydraulic position system of hydraulic EPC valve controlled cylinder is implemented, finally the control effects of the realistic system are more achieved.

THE EPC ELECTROHYDRAULIC SERVO SYSTEM AND THE STATE SPACE MODEL

The working mechanism of coiling EPC electro hydraulic servo system: The tension coiler is generally in the pickling, annealing, cutting, coating and other finishing machines group of cold rolling line. In the rolling process the forward and backward tension can be established by relying on such coiling equipment, which can reduce the rolling pressure and improve the rolling stability of roller. The strip deviation is caused by tension, deviation and vibration of roller set, roller system error, the strip geometry and other factors. The deviation not only affects the quality of products, but will also cause equipment damage or belt production. Figure 1 shows the principle of coiler electro hydraulic servo system. In the practical production due to the impact of the



1: Strip
2: Drum
3: Coiler
4: Floating cylinder
5: Photoelectric detector
6: Detector positioning cylinders
7: Solenoid directional control valve
8: Photoelectric signal amplifier
9: Electro hydraulic servo valve
10: Servo amplifier
11: Servo current instructions

Fig. 1: Schematic diagram of electro-hydraulic servo take-up EPC

above-mentioned interference factors, the accuracy deviations can not meet the requirements of the process quality and in severe equipments shut down.

The state-space model of coiling EPC electro hydraulic servo system: The simplified model of the coiling EPC electro hydraulic servo system is constituted by linear Eq. (1) of the control element, the flow Eq. (2) of the hydraulic cylinder, load force balance Eq. (3) of the hydraulic cylinder:

$$Q_L = K_Q x_v - K_C p_l \tag{1}$$

$$Q_l = A_p \frac{dx_p}{dt} + C_{tp} P_l + \frac{V_t}{4\beta_e}\frac{dP_l}{dt} \tag{2}$$

$$A_p P_l = m_t \frac{d^2 x_p}{dt^2} + B_p \frac{dx_p}{dt} + K x_p + F_L \tag{3}$$

$$x_v = K_v K_{sv} u \tag{4}$$

The function of the piston displacement x_p and control voltage signal u is expressed as:

$$\frac{x_p}{u} = \frac{\dfrac{4K_v K_{sv} K_Q \beta_e A_p}{m_t V_t}}{s^3 + \left(\dfrac{4K_{ce}\beta_e}{V_t} + \dfrac{B_p}{m_t}\right)s^2 + \left(\dfrac{4\beta_e A_p^2}{m_t V_t} + \dfrac{4B_p K_{ce}\beta_e}{m_t V_t} + \dfrac{K}{m_t}\right)s + \dfrac{4KK_{ce}\beta_e}{m_t V_t}}$$

where, Q_L is the load flow, K_Q is the flow gain of slide valve, x_v is the displacement of slide valve spool relatively neutral position, K_C is the amplification coefficient of the slide valve flow and pressure, p_l is the load pressure, A_p is the piston area of the hydraulic cylinder, x_p is the piston displacement of the hydraulic cylinder, V_t is the total compression volume of the hydraulic cylinder, β_e is the effective volume modulus, m_t is the total piston mass converted by the piston and the load. B_p is the viscous damping coefficient of the piston and load, K is the load spring stiffness, F_L is the any external load force on the piston, K_v is the gain of the controller, K_{sv} is the gain of the servo valve, K_{ce} is the coefficient of total flow and pressure control and u is the control voltage:

$$\begin{cases} \dot{e}_1 = e_2 \\ \dot{e}_2 = e_3 \\ \dot{e}_3 = -\sum_{i=0}^{2} a_i e_{i+1} - b_0(t)u(t) + F(t) \end{cases} \tag{5}$$

The deviation of the system e is taken as the state variable. The variable r is used as given value. So the deviation of the system e is equal to r-x_p. Suppose there are some relationships, such as $e = e_1$, $e_2 = \dot{e}_1$ and $e_3 = \dot{e}_2$, deviation matrix E is expressed with $E = (e_1\ e_2\ e_3)^T$.

Among there is:

$$a_0 = \frac{4KK_{ce}\beta_e}{m_t V_t}, a_1 = \frac{4\beta_e A_p^2}{m_t V_t} + \frac{4B_p K_{ce}\beta_e}{m_t V_t} + \frac{K}{m_t}, a_2 = \frac{4K_{ce}\beta_e}{V_t} + \frac{B_p}{m_t},$$
$$b_0 = \frac{4K_v K_{sv} K_Q \beta_e A_p}{m_t V_t}, F(t) = \frac{d^3 r}{dt^3} + a_2 \frac{d^2 r}{dt^2} + a_1 \frac{dr}{dt} + a_0 r$$

Above state space model based on bias included all the main parameters of the system, does not consider external leakage of the system, the state parameters can be measured and have limit.

SLIDING MODE CONTROLLER OF SELF-TUNING SWITCHING GAIN IN HYDRAULIC EPC SYSTEM

The mathematical equations of proportion switching sliding mode controller with main feedback: Taking into account the design needs of the actual control system, the main feedback proportion switching control is more suitable for the closed-loop control of the electro hydraulic servo system. For the aforementioned state-space model of electro hydraulic servo system with uncertainty, the deviation is taken as the state feedback vector and then the desirable control strategy is as follow:

$$u = \psi_1 e_1 \tag{6}$$

where, ψ_1 is:

$$\psi_1 = \begin{cases} \alpha_1 & e_1 s > 0 \\ \beta_1 & e_1 s < 0 \end{cases}$$

s is the sliding mode switching function and is expressed with $s = c_1e_1+c_2e_2+e_3$, then there is $e_3 = s - c_1 e_1 - c_2 e_2$.

Therefore, the motion equations of the system are for:

$$\begin{cases} \frac{d}{dt}e_1 = e_2 \\ \frac{d}{dt}e_2 = e_3 \\ \frac{d}{dt}e_3 = -a_0 e_1 - a_1 e_2 - a_2 e_3 - b_0\psi_1 e_1 + F(t) \end{cases} \tag{7}$$

Then $\dot{s}$ is expressed for:

$$\dot{s} = c_1 e_2 + c_2 e_3 - a_0 e_1 - a_1 e_2 - a_2 e_3 - b_0\psi_1 e_1 + F(t) =$$
$$(c_2 - a_2)s + (-c_1 c_2 + c_1 a_2 - a_0 - b_0\psi_1)e_1 +$$
$$(-c_2^2 + a_2 c_2 - a_1 + c_1)e_2 + F(t)$$

when s is equal 0, $e_3 = -c_1e_1 - c_2e_2$ and then:

$$\lim_{s\to 0}\frac{d}{dt}s = (-c_1 c_2 + c_1 a_2 - a_0 - b_0\psi_1)e_1 +$$
$$(-c_2^2 + a_2 c_2 - a_1 + c_1)e_2 + F(t)$$

For the fixed value control systems, the formula $F(t) = a_0 r$ is found. To ensure the sliding mode existence conditions that $\lim_{s\to 0} s\frac{ds}{dt} < 0$, that satisfies the Lyapunov stability conditions, as follow the formulas are must established:

$$c_1 - a_1 + c_2 a_2 - c_2^2 = 0 \tag{8}$$

$$\alpha_1 > \left(-a_0 + c_1 a_2 - c_1 c_2\right) b_0^{-1} \tag{9}$$

$$\beta_1 < \left(-a_0 + c_1 a_2 - c_1 c_2\right) b_0^{-1} \tag{10}$$

At the same time the following conditions (11) is satisfied to ensure the sliding mode exists:

$$|e_1| > \frac{|a_0 r|}{|-c_1 c_2 + c_1 a_2 - a_0 - b_0 \psi_1|} \tag{11}$$

Under the above constraints, the equivalent control of sliding motion is:

$$u^* = \left(-a_0 + c_1 a_2 - c_1 c_2\right) b_0^{-1} e_1 \tag{12}$$

Thus the corresponding motion equation is:

$$\begin{cases} \frac{d}{dt} e_1 = e_2 \\ \frac{d}{dt} e_2 = e_3 \\ \frac{d}{dt} e_3 = -a_0 e_1 - a_1 e_2 - a_2 e_3 - \left(-a_0 + c_1 a_2 - c_1 c_2\right) e_1 \end{cases} \qquad s = c_1 e_1 + c_2 e_2 + e_3 = 0$$

The sliding motion equation is:

$$\begin{cases} \frac{d}{dt} e_1 = e_2 \\ \frac{d}{dt} e_2 = e_3 = -c_1 e_1 - c_2 e_2 \end{cases}$$

If the Eq. (8), (9) and (10) are satisfied and the Eq. (11) is not satisfied, the system will appear self-sustaining oscillations or static error. Therefore the following control strategy is used:

$$u = \psi_1 e_1 + H,\ H = a_0 r b_0^{-1}$$

Because the following relationship exits:

$$\begin{aligned} &\lim_{s\to 0} \frac{d}{dt} s = \left(-c_1 c_2 + c_1 a_2 - a_0\right) e_1 - b_0 u + a_0 r = \\ &\left(-c_1 c_2 + c_1 a_2 - a_0\right) e_1 - b_0\left(\psi_1 e_1 + H\right) + a_0 r = \\ &\left(-c_1 c_2 + c_1 a_2 - a_0 - b_0 \psi_1\right) e_1 - b_0\left(a_0 r b_0^{-1}\right) + a_0 r \\ &= \left(-c_1 c_2 + c_1 a_2 - a_0 - b_0 \psi_1\right) e_1 \end{aligned}$$

So the above generalized sliding condition is still valid, so the primary feedback proportional switching sliding mode control system with the steady state error may be constituted.

The control strategy of the fuzzy self-tuning switching gain: Let the expression is built:

$$u = \psi_1 e_1 + u_s$$

where the variables are expressed:

$$\psi_1 = \begin{cases} \alpha_1 & e_1 s > 0 \\ \beta_1 & e_1 s < 0 \end{cases},\ u_s = k\,\mathrm{sgn}(s)$$

among
$$\begin{aligned} &\alpha_1 > \sup\left[\left(-a_0 + c_1 a_2 - c_1 c_2\right) b_0^{-1}\right] \\ &\beta_1 < \inf\left[\left(-a_0 + c_1 a_2 - c_1 c_2\right) b_0^{-1}\right] \\ &c_2 < \inf\left(a_2\right),\ k > |H|_{\max} \end{aligned}$$

Among above expression sup[] is for the supremum function and inf[] is for the infimum function. When the sliding mode switching function is designed, two-dimensional fuzzy controller is used. The input variables are respectively defined for s and $\dot{s}$ and they respectively are expressed for the relative distance of system movement point from the sliding surface and the relative speed moving to sliding surface. The output ΔU of the fuzzy controller is the fuzzy variable of the control variable quantity Δu. The control variable u is adjusted by the fuzzy control rules to meet the expression: $s\dot{s} < 0$.

The fuzzy sets is defined by using 'Positive Big' (PB), 'Positive Middle' (PM), 'Zero' (ZO), 'Negative Middle' (NM), 'Negative Big' (NB). In this study the fuzzy control is introduced in conventional proportion switching sliding mode. The derivative of switching function has very large effects on chattering and too quick speed will result in a substantial chattering. Therefore the control gain should be reduced when close to the sliding, to lower the speed of the state point crossing sliding. But the control gain need to be a large gain when in the more distance from sliding surface. With regards to this, the product of switching function s and its derivative $\dot{s}$ is used for the input of the fuzzy controller and the switching gain (it is expressed with k) is used for the output. The shape and distribution of membership function are shown in Fig. 2.

The following rules are established: When s and $\dot{s}$ are all for PB and then $s\dot{s}$ is for PB. That a large positive change of the input needs to be control, so that $s\dot{s}$ is made to quickly decrease. When $s\dot{s}$ is less than 0, then it for the desired working condition and the control variable is equal to 0. When s and $\dot{s}$ all are for NB and then $s\dot{s}$ is for PB. That a large positive change of the input needs to be control, so that $s\dot{s}$ is made to quickly decrease.

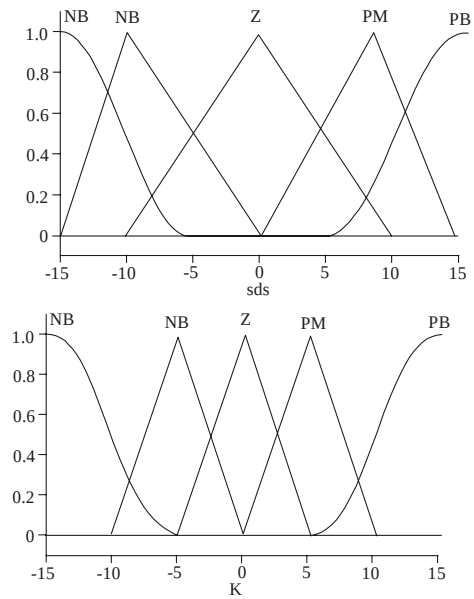


Fig. 2: Membership functions of input and output variables

Because the design of the fuzzy system meets s $\dot{s}$ <0, the fuzzy sliding mode system is stable. Finally the fuzzy output is accurately quantified by ambiguity resolution. Fuzzy rules are as follows:

R1: if s $\dot{s}$ is PB, then k is PB
R2: if s $\dot{s}$ is PM, then k is PM
R3: if s $\dot{s}$ is ZO, then k is ZO
R4: if s $\dot{s}$ is NM, then k is NM
R5: if s $\dot{s}$ is NB, then k is NB

Co-simulation: In the servo valve of the EPC hydraulic system there are the nature nonlinear, parameter uncertainty, external interference and other factors and many parameters change slowly with working state and temperature. So in the EPC electro-hydraulic system the main parameters may be is changing or uncertain during the system running and its value can be obtained by deriving directly or indirectly nominal parameters. When there is a lot of uncertainty in the system parameters, the design goal of the control system is that the system output *y* tracks the desired output signal *r*.

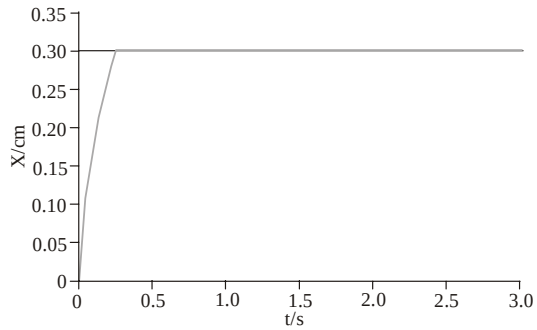
In order to analyze the chattering in variable structure algorithm having impact on the system, the model of valve controlled asymmetric cylinder EPC system is established under considering fixed institutional flexibility. According to the system features, the softness of connecting mechanism and sensor feedback mechanism is not considered.

System parameters: Cylinder diameter *D* (0.16 m); piston rod diameter *d* (0.063 m); hydraulic cylinder effective area A_p (1.48×10^{-2} m^2); piston displacement *x* (0.15 m); maximum total mass m_t (35000 Kg); hydraulic cylinder two-chamber volume V_t (0.222×10^{-2} m^3); effective volume elastic modulus β_e 7×10^8 N/m^2; fluid dynamic viscosity μ (1.4×10^{-2} $Pa\cdot s$); total flow pressure coefficient K_{ce} (8.25×10^{-12} m^3/$s\cdot Pa$); servo valve no load flow Q_L (40 *L*/min); the rated current of servo valve I (300 mA); actual oil pressure P_s (7 MPa); fixed mechanism stiffness (2×10^8 N/m); fixed mechanism damping (20000 $N\cdot s/m$); dead volume (50 cm^3); pipe diameter (22 m); pipe length (3 m).

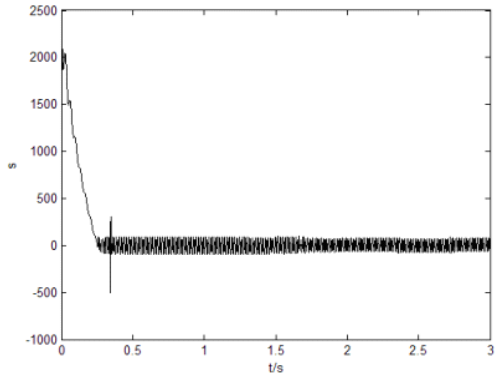
In system design position feedback is used. After the feedback signal is compared with the given signal, their difference is sent to the S-function of the controller after twice differentiated and is taken as the input of the main feedback proportion switching control. By limiting link the output signal is used to limit the maximum output current of the servo amplifier and by adding a low-pass filter it can suppress high frequency interference and play a smooth function on high-frequency switching of the sliding, to meet the requirements of high-precision control. Because the system is time-varying loads of large inertia, including friction, tension fluctuations and system inherently uncertainty, disturbances will be produced under the role of control force. According to the actual situation of the project, the system adopts asymmetric hydraulic cylinder and consider the impact of the dead volume and pipeline parameters on the system. By using AMEsim super component function a group of fluid elements are selected and then these elements are compressed and packaged into cylinder icon and can be connected with the rigidity institutions, finally the mechanical model is established under considering the softness of the fixed mechanism. In the model parameters can be arbitrarily modified during simulation analysis. It provides a convenience for the study of system characteristics.

According to the range of model parameters and their uncertainty and according to the existence and accessibility of sliding, basic proportional sliding switching parameters are respectively taken as $c_1 = 7561$, $c_2 = 18$. According to the relationship $\beta_1<u^*<\alpha_1$ and the system calculation, considering uncertainty and without loss of generality, then sup (α_1) is equal to 80 and inf (β_1) is equal to -80. Switching gain k is initially determined according to the model parameters and its range. In order to obtain mixture simulation parameters to be close actual system, they will be finally determined by considering uncertainty and by simulating. At first the invariant parameters are taken to analyze: in parameter type the parameter k equal 50 is taken; in parameter type the parameter k equal 10 is taken; in parameter type fuzzy self-tuning gain proportion sliding is taken. System simulation characteristic curves are shown in Fig. 3, 4 and 5, where the Fig. (a) shows the 3 mm step response characteristics; the Fig. (b) shows the switching function curve; the Fig. (c) shows the curve of the control variable.

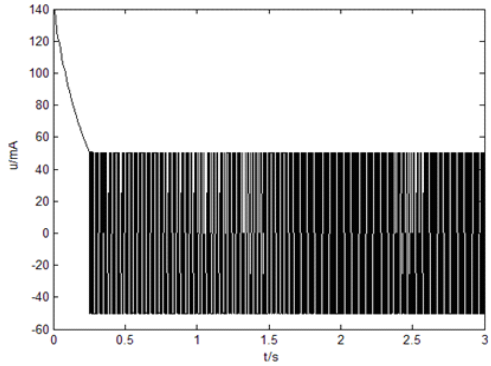
As can be seen from Fig. 3, when using the parameters of type, due to the sliding gain selected is



(a) The 3 mm step response characteristics



(b) The switching function curve

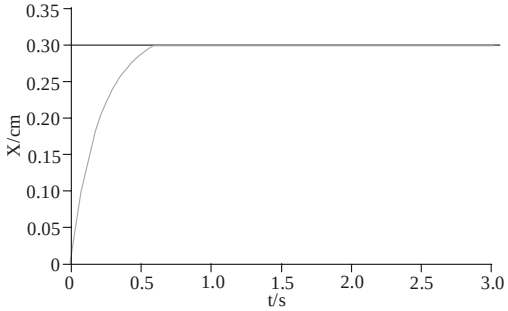


(c) Thecurve of the control variable

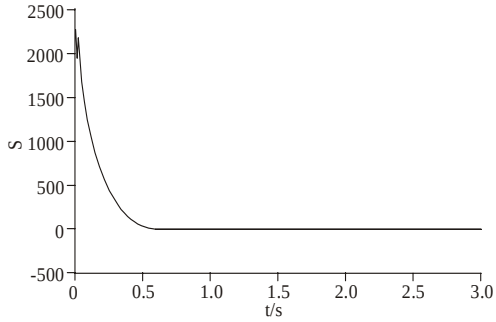
Fig. 3: System characteristic curves of parameter type

larger, the system response is faster and the system tracks completely 3 mm deviation in the 0.4s. But along with the larger chattering and the oscillation, therefore it is not conducive to improve the steady-state accuracy.

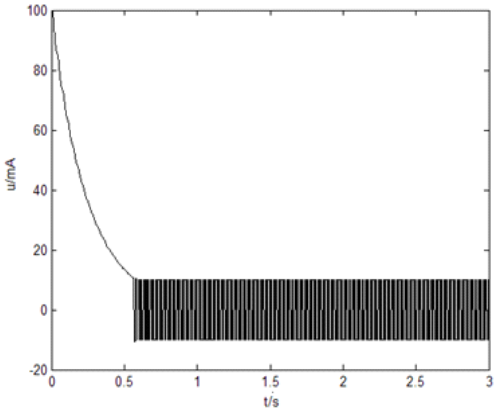
As can be seen from Fig. 4,When using parameter type, due to the sliding gain selected is relatively smaller, the system response becomes slow and the system tracks 3 mm deviation in the 0.6s that affecting the rapidity. But the chattering is smaller and steady-state accuracy is higher.



(a) The 3 mm step response characteristics



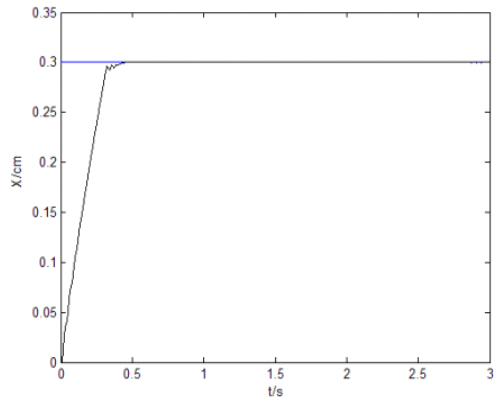
(b) The switching function curve



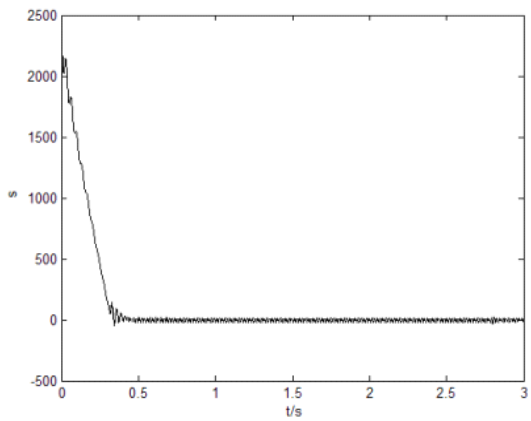
(c) The curve of the control variable

Fig. 4: System characteristic curves of parameter type ll

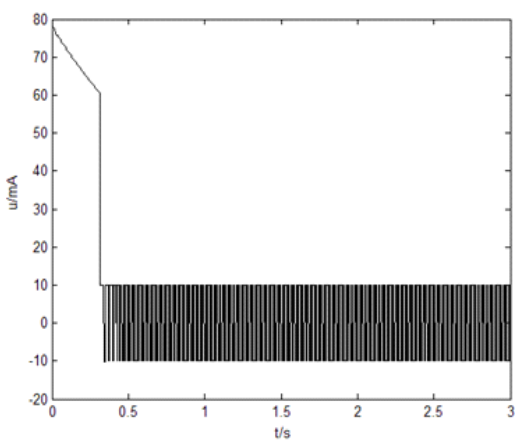
As can be seen from Fig. 5,When using parameter type, due to the fuzzy sliding switching gain is selected, the system response speed is quicker and the system tracks completely 3 mm deviation in the 0.5s and the system chattering is small that will be favorable to improve the system steady state accuracy. Therefore compared with the traditional proportion sliding, the fuzzy self-tuning proportion sliding ensures high response speed and overcomes the major problems of large chattering and



(a) The 3 mm step response characteristics



(b) The switching function curve



(c) The curve of the control variable

Fig. 5: System characteristic curves of parameter type lll

steady-state error caused by the traditional large proportion sliding mode switching gain when rapid response is required.

CONCLUSION

Traditional proportional control has the poor ability to inhibit parameter changes and external disturbance. The proportion sliding mode control has the stronger robustness to the uncertainties caused by system parameters changes and external disturbances. The robustness is stronger when the gain is greater and then the system can be adapted to the conditions changes. But the larger gain leads to a strong chattering.

After the fuzzy self-tuning switching gain sliding control is used, the system response is fast, the output is smooth, the steady-state accuracy is higher and the chattering is greatly weakened. For considering un-modeled dynamics and mechanical resonance of the system, the more excellent features can be obtained. Therefore it has strong practical value.

REFERENCES

Bai, H., C. Guan and P.E. Feng, 2009. Adaptive fuzzy sliding-mode control with variable universe for electro-hydraulic proportional system. Elect. Mach. Contr., 13(5): 728-733.

Fang, Y.M., Z.J. Wang and Y.P. Xie, 2010. Sliding mode variable structure control of multi-model switching for rolling mill hydraulic servo position system. Elect. Mach. Contr., 14(5): 91-96.

Gao, Q., L. Wang and L.F. Qian, 2007. Application of adaptive fuzzy sliding mode control to rocket launcher electro-hydraulic position servo system. Acta Armamentarii, 28(9): 1148-1152.

Guan, C. and S.A. Zhu, 2005. Backstepping-based multiple cascade adaptive sliding mode control of an electro-hydraulic servo system. Chin. J. Sci. Instr., 26(6): 569-573.

Jin, H.Z., Y.M. Luo and Z. Xiao, 2007. Investigation of a novel method of saturation function for chattering reduction of sliding mode control. J. Harbin Eng. U., 28(3): 288-291.

Liu, T., Q.H. Liu and J.H. Jiang, 2010. Adaptive fuzzy sliding mode control for hydrostatic transmission system. Trans. Chinese Soc. Agric. Mach., 41(1): 29-33.

Luo, X.Y., Z.H. Zhu and X.P. Guan, 2009. Chattering reduction adaptive sliding-mode control for nonlinear time-delay systems. Contr. Decis., 24(9): 1429-1435.

Su, W.H. and J.H. Jiang and Q.H. Liu, 2010. PD controller based on simulated annealing ant colony optimization algorithm in direct drive electro-hydraulic servo rotary vane steering gear. Elect. Mach. Cont., 14(1): 102-106.

Wang, Y.M., Y. Feng and Q.L. Lu, 2008. Design of free-chattering sliding mode control systems for permanent magnet synchronous motor. Elect. Mach. Cont., 12(5): 514-519.

Zhang, X.Z. and Y.N. Wang, 2009. Sliding mode control with parameters direct self-tuning for electro-hydraulic servo system. China Mech. Eng., 20(11): 1285-1289.

Zhong, W.F. and X.X. He, 2008. Fuzzy neural network control of electro-hydraulic position servo system. Elect. Mach. Contr., 12(4): 478-482.