Proceedings of the 17th World Congress
The International Federation of Automatic Control
Seoul, Korea, July 6-11, 2008

Velocity Control for a Variable Displacement Hydraulic Servo System Using Adaptive Fuzzy Sliding-Mode Control

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Abstract: The variable displacement hydraulic servo system performs specific characteristics on non-linearity and time-varying. An exact model-based controller is difficult to be realized. In this study, the design method and experimental implementation of an adaptive fuzzy sliding-mode controller (AFSMC) are presented, which has on-line learning ability for dealing with the system time-varying and non-linear uncertainty behaviors for adjusting the control rule parameters. The tuning algorithms are derived in the sense of the Lyapunov stability theorem; thus, the stability of the system can be guaranteed. The experimental results show that the AFSMC can perform excellent velocity control for the variable displacement hydraulic servo system.

1. INTRODUCTION

Hydraulic servo systems are widely used in the industry due to their capability of providing large driving forces or torques, higher speed of response with fast motions and possible speed reversals, and continuous full-power operation. However, oil viscosity, friction forces between cylinder and piston, variable loading cause servo hydraulic control systems to suffer from highly nonlinear time variant dynamics, load sensitivity and parameter uncertainty (Merrit 1976). Generally, it is difficult to establish or identify an accurate dynamic model of a complicated hydraulic servo system for designing optimal controller. Fuzzy control (FC) law can be designed based on some knowledge or without any knowledge about the control system. In addition, an appropriate fuzzy controller can overcome the environmental variation during operation processes. Therefore, it has been employed in the field of hydraulic servo system. In Zhao *et al.* (1993) and Zhao *et al.* (1994) developed a fuzzy state controller for a hydraulic position servo system with unknown load. A fuzzy controller has been used for a class of hydraulically actuated industrial robots (Corbet *et al.* 1996). An intelligent position control for electro-hydraulic drive has been proposed by the hybrid FC structure (Deticek 1999). In Rahbari *et al.* (2000), a PD type fuzzy controller has been used for hydraulic system. However, the design of a traditional fuzzy controller depends fully on an expert or the experience of an operator to establish the fuzzy rules bank. There is no guide rule for designing the fuzzy rules bank and parameters. The time-consuming adjusting process is required to achieve the specified control performance. Thereafter, self-tuning algorithms were introduced into fuzzy controller to adjust fuzzy parameters and improve the control performance based upon a specified performance index (Maeds *et al.* 1992). However, a complicated learning mechanism or a specific performance decision table designed by trial-and-error is required. Thus, its application still presents certain difficulty, and the large amount of the fuzzy rules also makes the analysis complex. For reducing the fuzzy rules in the fuzzy controller, some researchers have proposed FC design methods based on the sliding mode control (SMC) scheme. The self-organizing fuzzy sliding mode control is developed in the parallel control of velocity control and energy-saving for a hydraulic valve-controlled cylinder system (Chiang *et al.* 2003). In Chiang *et al.* (2004), the concurrent implementation of high velocity control performance and high energy efficiency for hydraulic injection moulding machines has been proposed. These approaches are referred to as fuzzy sliding mode control (FSMC) design methods (Kim Corbet *et al.* 1995 and Choi *et al.* 1999). Since only one variable is defined as the fuzzy input variable, the main advantage of the FSMC lies in less fuzzy rules than FC. Moreover, the FSMC has more robustness against parameter variation (Choi *et al.* 1999). Although FC and FSMC are both effective methods, their major drawback is that the fuzzy rules should be previously tuned by time-consuming trial-and-error procedures. In order to tackle this problem, adaptive fuzzy control (AFC) based on the Lyapunov synthesis approach has been extensively studied (Wang 1994 and Lee *et al.* 2001). With this approach, the fuzzy rules can be automatically adjusted to achieve satisfactory system response by an adaptive law.

The objective of this study is to propose an adaptive fuzzy sliding-mode controller (AFSMC) design method for the hydraulic servo control system altered by a variable displacement pump. This approach can automatically adjust the fuzzy rules, similar to the AFC, and can reduce the fuzzy rules, similar to the FSMC. All control parameters in AFSMC are tuned in the Lyapunov sense, thus the stability of the system can be guaranteed. Finally, the proposed control strategy is verified experimentally on the experimental setup of the variable displacement hydraulic servo control system.

2. FORMULATION OF VARIABLE DISPLACEMENT HYDRAULIC SERVO SYSTEM

The variable displacement hydraulic servo control system is designed and set up for investigating the dynamic performance and control effect. The test rig layout of the variable displacement hydraulic servo control system is shown in Fig.1. The servo system primary consists of a variable displacement axial piston pump and a controlled hydraulic cylinder, a position sensor… etc. The swash plate control unit of the variable displacement axial piston pump mainly contains an electro-hydraulic servo valve and an adjusting cylinder. The mathematics models of the main elements in this system are discussed as below:

(a) Servo amplifier

The servo amplifier proportionally transfers the input voltage into the input current driving the servo valve, and can be described as

$$i = K_a u \tag{1}$$

where i indicates the input current, K_a is the gain of servo amplifier and u is the input voltage.

(b) Servo valve

The models of the servo valve contain the valve spool dynamics and the volume flow equation. The valve spool dynamics describes the relations between the input current i and the valve spool displacement y, and can be considered as a model of zero order, 1st or even 2nd order. It depends on the relative comparison between the natural frequency of the servo valve and that of the hydraulic cylinder. In this paper, the servo valve has more than triple natural frequency of the hydraulic cylinder. Thus, it can be regarded as a zero order model for swash plate control unit of the variable displacement pump

$$y = V_v i \tag{2}$$

were V_v is the gain of servo valve.

(c) Variable displacement mechanism

The small adjusting hydraulic cylinder serves to control the swash plate angle of axial piston pump. Its natural frequency is about 200Hz, so that its dynamic characteristic can be regarded as a zero order model, and can be described as

$$\phi = K_q y \tag{3}$$

where ϕ is the plate angle, K_q is the gain of variable displacement mechanism.

(d) Model of the swash plate axial piston pump

The relationship between the volumetric displacement of pump and load flow is described in the following

$$Q_l = D_p N_p - C_l P_l = k_p \phi N_p - C_l P_l \tag{4}$$

where D_p is the volumetric displacement of pump, N_p is the pump speed (assumed constant), C_l is the leakage coefficient of the pump, P_l is the load pressure difference, k_p is displacement gradient of pump. The continuity equation to the cylinder chamber is given by

$$Q_l = A_p \dot{x}_p + C_t P_l + \left(V_t / 4\beta_e\right) \dot{P}_l \tag{5}$$

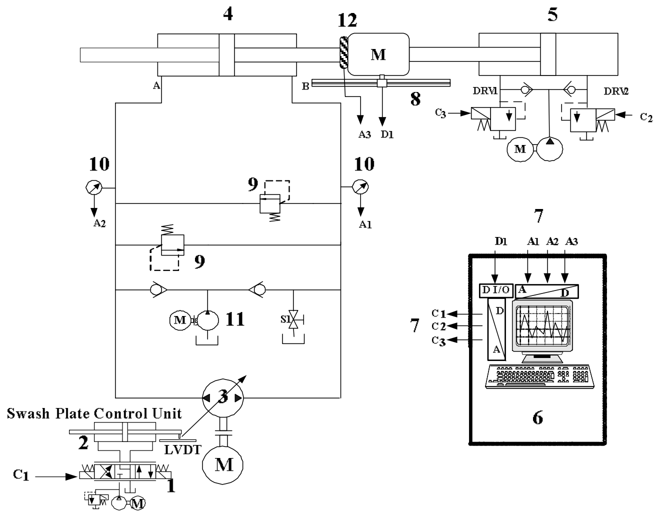
where A_p is the area of the piston, x_p is the position of the piston rod, C_t is the total load leakage coefficient of the cylinder, V_t is the total compressed volume and β_e is the effective bulk modulus of the system.

According to the Newton's 2nd law, the motion equation of controlled hydraulic cylinder and loading can be derived as

$$A_p P_l = M\ddot{x}_p + B\dot{x}_p + f_l \tag{6}$$

where M is the total mass of piston and load referred to piston, B is the viscous damping coefficient of the piston and f_l is the arbitrary load force on piston. Substituting (4) into (5) gives

$$k_p \phi N_p = A_p \dot{x}_p + \left(C_l + C_t\right) P_l + \left(V_t / 4\beta_e\right) \dot{P}_l \tag{7}$$



1. hydraulic servo valve	2. adjusting cylinder
3. variable displacement pump	4. hydraulic cylinder
5. disturbance cylinder	6. PC-Based controller
7. I/O interface card	8. position sensor
9. relief valve	10. pressure sensor
11. auxiliary pump	12. load cell

Fig. 1: Layout of the variable displacement hydraulic servo control system

By combining (1) - (7), the state equations of the variable displacement hydraulic servo control system model can be achieved as follows

$$\dot{x}_1 = x_2(t)$$
$$\dot{x}_2 = -\sum_{1}^{2} a_i x_i(t) + g(\underline{x})u + d(\underline{x}) = f(\underline{x}) + g(\underline{x})u + d(\underline{x}) \quad (8)$$

where $\underline{x}(t) = \left[x_1(t) \quad x_2(t)\right]^T = \left[\dot{x}_p(t) \quad \ddot{x}_p(t)\right]^T$,
$a_1(t) = 4\beta_e A_p^2 / MV_t + 4\beta_e \left(C_l + C_t\right) B / MV_t$,
$a_2(t) = B/M + 4\beta_e \left(C_l + C_t\right)/V_t$,
$g(\underline{x}) = 4A_p \beta_e K_p N_p K_q V_v K_a / MV_t$,
$d(\underline{x}) = -(4\beta_e \left(C_l + C_t\right) f_l / V_t M + \dot{f}_l / M)$ in which $g(\underline{x})$ is a constant with positive value.

3. ADAPTIVE FUZZY SLIDING MODE CONTROL

Designing a SMC needs to know the system models and to find the inverse form of inertia term in system dynamics. However, the accurate mathematical models are always difficult to formulate or even not available. To solve these problems, an AFSMC shown in Fig.2 is proposed to control the variable displacement hydraulic servo control system.

3.1 Fuzzy Control

Assume that there are n rules in a fuzzy base and each of them has the following form:

$$R^i: \text{ IF } S \text{ is } F^i \text{ THEN } u \text{ is } \alpha_i \quad (9)$$

where S is the input variable of the fuzzy system; u is the output variable of the fuzzy system; F^i are the triangular-type membership functions; and α_i are the singleton control actions for $i = 1, 2, ..., n$. The defuzzification of the FC output is accomplished by the method of center-of-gravity (Lee 1990)

$$u = \sum_{i=1}^{n} \alpha_i \times w_i / \sum_{i=1}^{n} w_i = \underline{\alpha}^T \xi \quad (10)$$

where w_i is the firing weight of the ith rule, $\underline{\alpha} = [\alpha_1, \alpha_2, ..., \alpha_n]^T$ is the parameter vector and $\underline{\xi} = [\xi_1, \xi_2, ..., \xi_n]^T$ is the vector of fuzzy basis functions

$$\xi_i = w_i / \sum_{i=1}^{n} w_i \quad (11)$$

For the conventional FC, the control actions α_i should be previously assigned through a lot of trails to achieve satisfactory control performance. In the following, the adaptive algorithm will be proposed to tune these control actions on-line.

3.2 Fuzzy Sliding Mode Control

The methods to design the fuzzy sliding-mode controller for a non-linear system with 2nd order where the error and the error change rate were used to synthesize fuzzy reasoning rules was proposed (Palm 1994 and Hwang *et al.* 1992). However, the rule number was larger and did not give the mathematical expression. Thus, it is difficult to analyze the properties of the control system. To overcome this problem, we adopt the sliding surface $S = 0$ of SMC as a variable to compress all the information into one type, extend the sliding surface $S = 0$ to the fuzzy sliding surface $\tilde{S} = \tilde{0}$, and make S be a linguistic description of $\tilde{S}$. In this paper the two triangular-typed functions are used to define the membership functions of IF-part and THEN-part, which are depicted in Figs.3 (a) and 3(b) respectively. The fuzzy rules are given in the following form

$$R^l: \text{IF } S \text{ is } \tilde{F}_s^l \text{ THEN } u_{fs} \text{ is } \tilde{F}_u^{8-l},\ l = 1, ..., 7. \quad (12)$$

According to the sup-min compositional rule of inference and the defuzzification of the control output accomplished by the method of center-of-area, the mathematical expression can be derived as

$$u_{fs} = \begin{cases} 1 & ,if \quad z < -1 \\ (7.5z^2 + 13.5z + 5)/(9z^2 + 15z + 5) & ,if \quad -1 \le z < -2/3 \\ (9z^2 + 11z + 2)/(18z^2 + 18z + 2) & ,if \quad -2/3 \le z < -1/3 \\ (1.5z^2 + 1.5z)/(9z^2 + 3z - 1) & ,if \quad -1/3 \le z < 0 \\ (-1.5z^2 + 1.5z)/(9z^2 - 3z - 1) & ,if \quad 0 \le z < 1/3 \\ (-9z^2 + 11z - 2)/(18z^2 - 18z + 2) & ,if \quad 1/3 \le z < 2/3 \\ (-7.5z^2 + 13.5z - 5)/(9z^2 - 15z + 5) & ,if \quad 2/3 \le z < 1 \\ -1 & ,if \quad z \ge 1 \end{cases} \quad (13)$$

where $z = S/\Phi$ and $\Phi > 0$ is a constant which describes the width of a boundary layer. As $|S| \ge \Phi$, it is easy to check $u_{fs} = -\text{sgn}(S)$.

3.3 Adaptive Fuzzy Sliding Mode Control System

The control objective is to find a control law so that the hydraulic actuator can track the desired velocity $\dot{x}_d(t)$. Define the tracking error $e(t)$ as

$$e(t) = \dot{x}_d(t) - \dot{x}_p(t) \quad (14)$$

where $\dot{x}_p(t)$ is the control output and $\dot{x}_d(t)$ is the desired velocity. Then define a sliding surface as

$$S(t) = \dot{e}(t) + k_1 e(t) \quad (15)$$

where k_1 is non-zero positive constants. Assume that parameters of the system in (8) are well known and the external load disturbance is measurable, then we can take the control law as

$$u^* = g^{-1}(\underline{x})[\eta S_\Delta(t) - f(\underline{x}) - d(\underline{x}) + \ddot{x}_d + k_1 \dot{e}(t)] \quad (16)$$

where $S_\Delta(t) = S(t) - \Phi sat(S(t)/\Phi)$. The function S_Δ has several properties as below that are useful in the design of adaptive law (Sanner *et al.* 1992).

Property1: When $|S| > \Phi$, $|S_\Delta| = |S| - \Phi$ and $\dot{S}_\Delta = \dot{S}$.

Property2: When $|S| \le \Phi$, $S_\Delta = \dot{S}_\Delta = 0$.

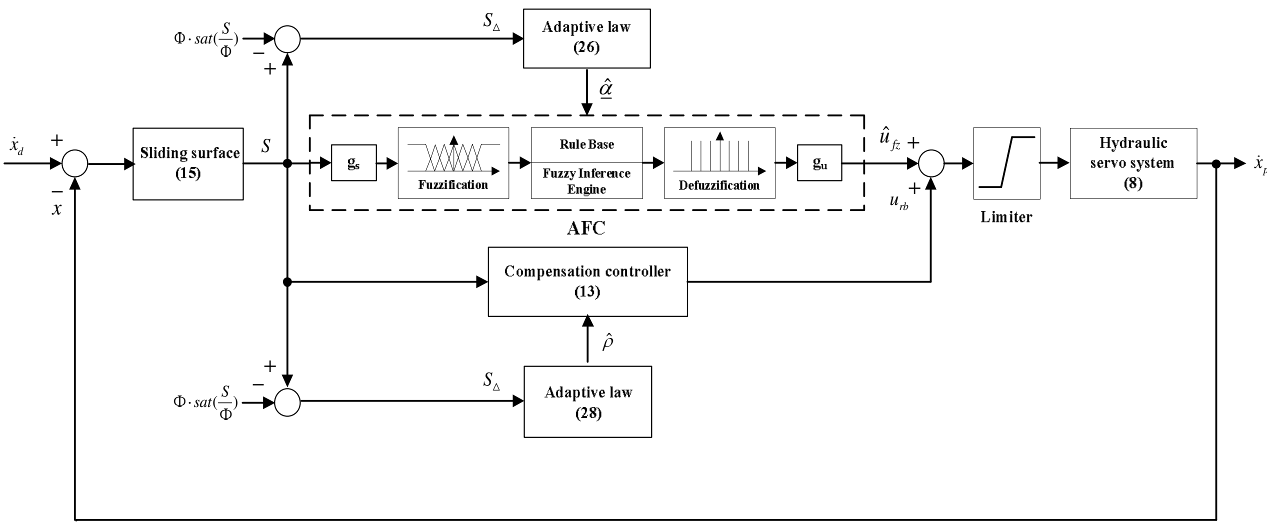


Fig.2. AFSMC for Velocity Control of a Variable Displacement Hydraulic Servo System

The above properties of the boundary layer concept are to be exploited, in the design of AFSMC, our goal being to cease adaptation as soon as the boundary layer is reached. This approach aims to avoid the possibility of unbounded growth. Differentiating (15) along the system trajectories (8), we have

$$\dot{S}(t) = -f(\underline{x}) - g(\underline{x})u - d(\underline{x}) + \ddot{x}_d + k_1\dot{e}(t) \quad (17)$$

Substituting (16) into (17) gives

$$\dot{S}(t) + \eta S_\Delta(t) = 0, \qquad \eta > 0. \quad (18)$$

Equation (18) shows that $\underline{e}(t)$ will converge to the neighbour of zero as $t \to \infty$ and the value of the neighbourhood are relative to the value of Φ (Slotine *et al.* 1896). However, the system parameters may be unknown or perturbed; the controller u^* cannot be precisely implemented. Therefore, by the universal approximation theorem (Lee *et al.* 2001), an optimal fuzzy control $\hat{u}_{fz}(S, \underline{\alpha}^*)$ in the form of (10) exists such that the approximation error of fuzzy controller can be defined as

$$\left|u^* - \hat{u}_{fz}(S, \underline{\alpha}^*)\right| = \rho^* \quad (19)$$

where ρ^* is the inherent approximation error and is assumed to be bounded by $\left|\rho^*\right| \le M$. Employing a fuzzy controller $\hat{u}_{fz}(S, \hat{\underline{\alpha}})$ to approximate u^* as

$$\hat{u}_{fz}(S, \hat{\underline{\alpha}}) = \hat{\underline{\alpha}}^T \underline{\xi} \quad (20)$$

where $\hat{\underline{\alpha}}$ is the estimated values of $\underline{\alpha}^*$. The control law for the developed AFSMC is assumed to take the following form:

$$u = \hat{u}_{fz}(S, \hat{\underline{\alpha}}) + u_{rb}(S) \quad (21)$$

where the fuzzy controller $\hat{u}_{fz}$ is designed to approximate the control u^* and the robust controller $\hat{u}_{rb}$ is designed to compensate the difference between the controller u^* and fuzzy controller $\hat{u}_{fz}(S, \hat{\underline{\alpha}})$. Through (16), (17) and (21) the dynamical equation as follow can be derived:

$$\dot{S}(t) + \eta S_\Delta(t) = g[u^* - \hat{u}_{fz}(S, \hat{\underline{\alpha}}) - u_{rb}(S)] \quad (22)$$

In order to derive the adaptive laws that ensure convergence to the boundary layer, a candidate Lyapunov function is defined as:

$$V(S_\Delta, \tilde{\underline{\alpha}}, \tilde{\rho}) = \frac{1}{2}\frac{S_\Delta^2}{g} + \frac{1}{2\eta_1}\tilde{\underline{\alpha}}^T\tilde{\underline{\alpha}} + \frac{1}{2\eta_2}\tilde{\rho}^2 \quad (23)$$

Where $\tilde{\underline{\alpha}}^{\mathrm{T}} = \underline{\alpha}^{*\mathrm{T}} - \hat{\underline{\alpha}}^{\mathrm{T}}$ and $\tilde{\rho} = \rho^* - \hat{\rho}$ are the approximation error of the parameter vectors $\underline{\alpha}^{*\mathrm{T}}$ and ρ^* respectively. In addition, η_1 and η_2 are positive constants. Differentiate (23) with respect to time as

$$\dot{V}(S_\Delta, \tilde{\underline{\alpha}}, \tilde{\rho}) = S_\Delta \dot{S}_\Delta / g + \tilde{\underline{\alpha}}^T \dot{\tilde{\underline{\alpha}}} / \eta_1 + \tilde{\rho}\dot{\tilde{\rho}} / \eta_2. \quad (24)$$

Thus, if $\left|S\right| \le \Phi$, then $S_\Delta = 0$, it follows $\dot{V}(S_\Delta, \tilde{\underline{\alpha}}, \tilde{\rho}) = 0$. If $\left|S\right| > \Phi$, then $\dot{S}_\Delta = \dot{S}$. By substituting (22) into (24), (25) can be obtained

$$\begin{aligned}\dot{V}(S_\Delta,\tilde{\underline{\alpha}},\tilde{\rho}) &= -\eta S_\Delta^2/g + S_\Delta[u^* - \hat{u}_{fz}(S,\hat{\underline{\alpha}})] - S_\Delta u_{rb}(S) + \tilde{\underline{\alpha}}^T\dot{\tilde{\underline{\alpha}}}/\eta_1 + \tilde{\rho}\,\dot{\tilde{\rho}}/\eta_2 \\ &= -\eta S_\Delta^2/g + S_\Delta[u^* - \hat{u}_{fz}(S,\underline{\alpha}^*) + \hat{u}_{fz}(S,\underline{\alpha}^*) - u_{fz}(S,\hat{\underline{\alpha}})] - S_\Delta u_{rb}(S) + \tilde{\underline{\alpha}}^T\dot{\tilde{\underline{\alpha}}}/\eta_1 + \tilde{\rho}\,\dot{\tilde{\rho}}/\eta_2 \\ &\le -\eta S_\Delta^2/g + \left|S_\Delta\right|\left|u^* - \hat{u}_{fz}(S,\underline{\alpha}^*)\right| + S_\Delta[\hat{u}_{fz}(S,\underline{\alpha}^*) - \hat{u}_{fz}(S,\hat{\underline{\alpha}})] - S_\Delta u_{rb}(S) + \tilde{\underline{\alpha}}^T\dot{\tilde{\underline{\alpha}}}/\eta_1 + \tilde{\rho}\,\dot{\tilde{\rho}}/\eta_2 \\ &= -\eta S_\Delta^2/g + [\left|S_\Delta\right|\rho^* - S_\Delta u_{rb}(S) - \tilde{\rho}\dot{\hat{\rho}}/\eta_2] + \tilde{\underline{\alpha}}^T(S_\Delta\xi(S) - \dot{\hat{\underline{\alpha}}}/\eta_1)\end{aligned}$$

(25)

For achieving $\dot{V}(S_\Delta, \tilde{\underline{\alpha}}, \tilde{\rho}) < 0$, the adaptive laws of AFSMC are chosen as

$$\dot{\hat{\underline{\alpha}}} = \eta_1 S_\Delta \xi(S) \quad (26)$$

$$u_{rb}(S) = -\hat{\rho} u_{fs} \quad (27)$$

$$\dot{\hat{\rho}} = \eta_2 \left|S_\Delta\right| \quad (28)$$

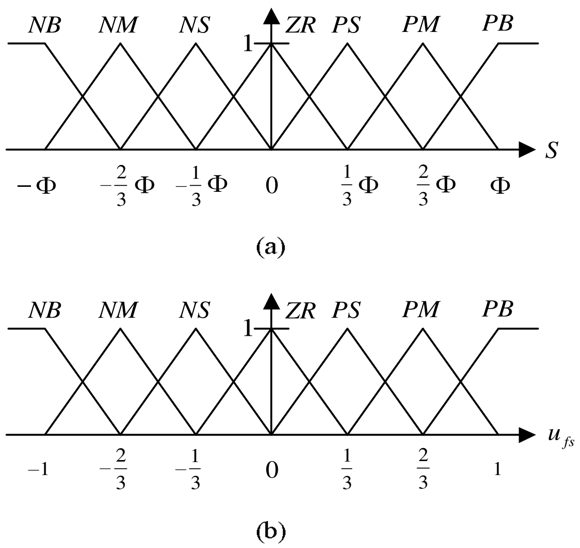
Then (25) can be rewritten as

$$\dot{V}(S_\Delta, \tilde{\underline{\alpha}}, \tilde{\rho}) \le -\eta S_\Delta^2 / g \tag{29}$$

Equations (26)-(29) only guarantee that $S_\Delta \in L_\infty$, but do not guarantee convergence. Integrating both sides of (29) and some derivations yields

$$\int_0^\infty S_\Delta^2 dt \le \frac{V(S_\Delta(\infty), \tilde{\underline{\alpha}}, \tilde{\rho}) - \dot{V}(S_\Delta(0), \tilde{\underline{\alpha}}, \tilde{\rho})}{\eta / g} \tag{30}$$

Since the right side of (30) is bounded, $S_\Delta \in L_2$. Using Barbalat's lemma (Slotine *et al.* 1991) it can be shown that $\lim_{t\to\infty} S_\Delta = 0$. This means that inequality $|S| \le \Phi$ is obtained asymptotically. Thus, the tracking error $e(t)$ converges to a neighbourhood of zero. In summary, the AFSMC system is shown in (21), where $\hat{u}_{fz}$ is given in (20) with the parameters $\hat{\underline{\alpha}}$ adjusted by (26) ; u_{rb} is given in (27) with the parameter $\hat{\rho}$ adjusted by (28). By applying this estimation law, the AFSMC system can be guaranteed to be stable in the Lyapunov sense.



Figs.3 Fuzzy partitions and membership functions of S and u_{fs} in the respective universe of discourse

4. EXPERIMENT

Figure 4 indicates the test results of the velocity control with V_{set}=20, 50 and 90 mm/s under the disturbance force of 10 kN, generated by the disturbance cylinder, in the variable displacement hydraulic servo system. Figures 4(a) and 4(b) show the velocity control response and control input of AFSMC. From about 0.2 sec to 0.3 sec the velocity control response has larger velocity change rate that results from the non-linearity of the swash plate axial piston pump. The velocity control error shown in Fig4(c) clarifies the excellent control performance of the AFSMC for the variable displacement hydraulic servo system.

For verifying the robustness of the AFSMC controller, the disturbance force set by the disturbance cylinder is changed from 10 kN to 50 kN. Figure 5 indicates the experimental results of the velocity control with V_{set}=20, 50 and 90 mm/s under the disturbance force of 50 kN in the variable displacement hydraulic servo system. In comparison with Fig.4 and Fig.5, the robustness of the proposed controller can be confirmed in this experiment clearly.

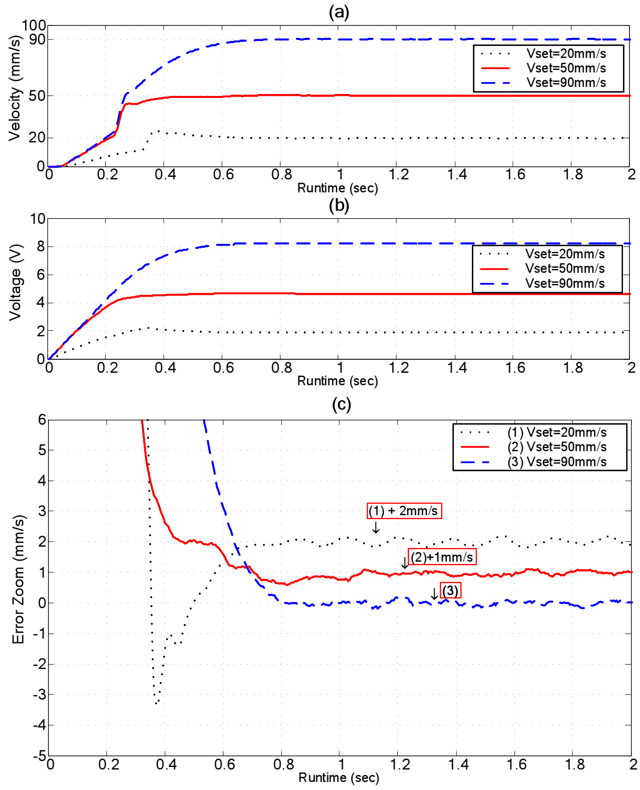


Fig.4 Experimental results of velocity control V_{set}=20, 50 and 90 mm/s in the variable displacement hydraulic system with loading force 10kN: (a) velocity control response, (b) control input, (c) velocity control error

5. CONCLUSIONS

This study has experimentally demonstrated the effectiveness of the proposed AFSMC for the velocity control of the variable displacement hydraulic servo system. Good robustness and excellent self-adaptability as well as superior dynamic performance with regard to various external disturbance forces are verified. The AFSMC is comprised of the SISO adaptive fuzzy controller for approximating the equivalent controller and the adaptive fuzzy sliding mode controller for compensating the approximate error of the equivalent controller and the external disturbance. All parameters of the fuzzy basis function can be tuned based on the Lyapunov stability theorem. Thus, the stability of the control system can be guaranteed. The effectiveness of the proposed AFSMC has been verified successfully through the various experimental results.

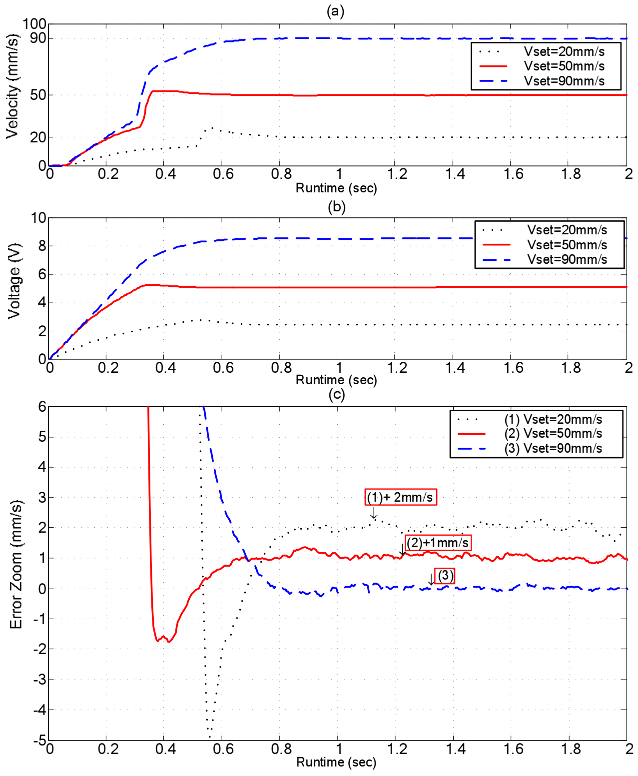


Fig.5 Experimental results of velocity control V_{set}=20, 50 and 90 mm/s in the variable displacement hydraulic system with loading force 50kN: (a) velocity control response, (b) control input, (c) velocity control error

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