

The Positioning control of an Electro-Hydraulic Variable Rotational Speed Pump-Controlled System Using Adaptive Fuzzy Controller with Self-tuning Fuzzy Sliding Mode Compensation

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Abstract—The electro-hydraulic variable rotational speed pump-controlled system (EHVRSPCS) 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 controller with on-line self-tuning fuzzy sliding-mode compensation (AFC-STFSMC) are proposed to deal with the system time-varying and non-linear uncertain behaviours by adjusting the control rule parameters. This control strategy employs the adaptive fuzzy approximation technique to design the equivalent controller of the conventional sliding-mode control (SMC). Furthermore, the fuzzy sliding-mode control scheme with self-tuning ability is introduced to compensate the approximation error of the equivalent controller for improving the control performance. 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 AFC-STFSMC can perform excellent positioning control for the EHVRSPCS.

Keywords: *electro-hydraulic pump-controlled system, AC servo motor, fuzzy sliding mode control, adaptive fuzzy, sliding-mode control*

1. INTRODUCTION

Hydraulic servo systems are widely used in the industry due to the advantages of high power-weight ratio, large driving forces, high response and good conduction of heat out of components by pressure fluid. However, oil viscosity, friction force, and leakage cause highly nonlinear time-variant dynamics, load sensitivity and parameter uncertainty in the hydraulic servo control systems result in the control complexity and difficulty [1-2]. In view of the hydraulic circuits, two different hydraulic systems are classified as the high-response but low-energy efficiency hydraulic valve-controlled system and the high-energy efficiency but lower response hydraulic pump-controlled system. The hydraulic valve-controlled

systems have been applied widely in the industrial servo control. And, recently, the investigations on high response pump-controlled systems driven by AC servo motors or switch reluctance motor are introduced [3]. Helduser [3] first presented the concept of hydraulic pump-controlled system, driven by gear pumps and AC servo motors with variable rotational speed for injection moulding machines in 1995. Ruhlicke [4] studied the position control of asymmetrical cylinder with double pump-controlled system driven by AC servo motor with variable rotational speed, which the position accuracy, about 50 μ m, is still unsatisfactory. Kazmeier and Feldmann [5] used Fuzzy control to study the pump-controlled system with variable rotational speed for positioning control with small power, max. 330W, and small stroke 0.5mm. Helduser [6] developed an electric-hydrostatic drive using AC servo motor and constant displacement internal gear pump for energy-saving power and motion control system. In recent years, AC servo motors combined with internal gear pumps were used mostly, and they can reach 50 μ m accuracy in position control. New applications of the high response in different hydraulic servo machines are still being developed. Therefore, this paper aims to investigate the position control performance of the high response pump-controlled system driven by an AC servo motor with variable rotational speed. Instead of internal gear pumps, a constant displacement axial piston pump is used in this paper for positioning control of hydraulic servo machines.

The classical SMC [7-9] have been proposed for solving the hydraulic servo control problems during the past decades. However, the drawbacks of classical SMC, such as chattering, large control effort and difficulty for deciding the equivalent control, limit its practical applicability. Moreover, the boundary of the uncertainty has to be estimated in order to guarantee the stability of the SMC systems. Thus, the SMC systems are still difficulty in real-world applications.

The fuzzy control, which doesn't require mathematical models of the system, has been used in several industrial applications. Zhao and Vivalo [10-11] developed a fuzzy state controller for a hydraulic position servo system with unknown load. Deticek [12] proposed an intelligent position control of electro-hydraulic drive using hybrid FC structure. The design of a traditional fuzzy controller mostly depends on expert experience for establishing the fuzzy rule base. However, for complex systems there are no guide rules for designing the fuzzy rule base and parameters. Thus, self-tuning algorithms were introduced in FC for adjusting fuzzy parameters and improving the control performance based upon a specified performance index [13-14]. 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 FC, FC design methods based on SMC scheme, namely fuzzy sliding-mode control (FSMC), have proposed [15-16]. Since only one variable is defined as the fuzzy input variable, the main advantage of FSMC lies in less fuzzy rules than FC. Although FC and FSMC are both effective design 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 [17-18]. With this approach, the fuzzy rules can be automatically adjusted to achieve system response by an adaptive law.

The objective of this paper is forcing on experimental implementation of the adaptive fuzzy controller with self-tuning fuzzy sliding-mode compensation (AFC-STFSMC) for the positioning control with high response in the EHVRSPCS. The AFC-STFSMC combines the adaptive fuzzy control and the self-tuning fuzzy sliding-mode control scheme. The adaptive fuzzy approximation method is to establish the unknown equivalent controller for releasing the model-based requirement. Furthermore, the fuzzy sliding-mode control scheme with self-tuning ability is introduced to compensate the function approximation error of the equivalent controller for improving the control performance and reducing the implementation difficulty. The AFC-STFSMC approach comprises the advantages of which the fuzzy rules can be automatically adjusted, similar to AFC, and can reduce the fuzzy rules, like FSMC. All controller parameters of this scheme are tuned in the Lyapunov sense, thus the stability of the system can be guaranteed.

II. FORMULATION OF ELECTRO-HYDRAULIC VARIABLE ROTATIONAL SPEED PUMP-CONTROLLED SYSTEM

The EHVRSPCS is designed and set up for investigating the dynamic performance and control effect. The test rig layout of the EHVRSPCS is shown in Fig.1. The hydraulic servo cylinder system of EHVRSPCS contains a double-rod symmetrical hydraulic cylinder fitted with a linear encoder with the resolution of 0.1 μm . The hydraulic power supply system,

which consists of a swash plate axial piston pump with constant displacement of 12 ml/rev and is driven by an AC servo motor of 7.0 kW, adjusts the supply volume flow by the rotational speed controlled by the AC servo motor. The mathematics models of the main elements in this system are discussed as below:

(a) Servo motor control unit

The servo motor control unit proportionally transfers the input voltage into the rotational speed of the servo motor, and can be described as

$$\omega = K_a \cdot u \quad (1)$$

where ω indicates the rotational speed of the servo motor, K_a is the gain of servo control unit and u is the input voltage.

(b) Swash plate axial piston pump with constant displacement

The pump volumetric displacement of the swash plate axial piston pump D_l is supposed to be constant. The load volume flow Q_l from the pump to the controlled hydraulic cylinder can be described as

$$Q_l = D_l \cdot \omega - C_l \cdot P_l \quad (2)$$

where ω indicates the pump rotational speed, C_l is the leakage coefficient of the pump, P_l is the load pressure.

(c) The controlled cylinder

The continuity equation of the controlled cylinder chamber is given by

$$Q_l = A_p \dot{x}_p + C_l P_l + \left(\frac{V_t}{4\beta_e} \right) \dot{P}_l \quad (3)$$

where A_p is the effective area of piston, x_p is the position of piston rod, C_l is the leakage coefficient of cylinder, V_t is the total compressed volume and β_e is the effective bulk modulus of hydraulic oil. Substituting Eq. (1) and Eq. (2) into Eq. (3), we have

$$D_l \cdot K_a \cdot u = A_p \dot{x}_p + (C_l + C_t) P_l + \left(\frac{V_t}{4\beta_e} \right) \dot{P}_l \quad (4)$$

According to the Newton's 2nd law, the motion equation of controlled cylinder on position control can be derived as

$$A_p P_l - f_l = M \ddot{x}_p + B \dot{x}_p + K x_p \quad (5)$$

where M indicates the total mass of piston, rod and load, B is the viscous damping coefficient, K is the spring constant and f_l is the arbitrary load force on piston. By combining Eqs. (1)-(5), the state equations of the EHVRSPCS model can be achieved as follows

$$\begin{aligned} \dot{x}_1 &= x_2(t) \\ \dot{x}_2 &= x_3(t) \\ \dot{x}_3 &= -\sum_{i=1}^3 a_i x_i(t) + g(\underline{x})u + d(\underline{x}) = f(\underline{x}) + g(\underline{x})u + d(\underline{x}) \end{aligned} \quad (6)$$

where

$$\underline{x}(t) = [x_1(t) \ x_2(t) \ x_3(t)]^T = [x_p(t) \ \dot{x}_p(t) \ \ddot{x}_p(t)]^T$$

$$a_1(t) = \frac{4\beta_e(C_l + C_t)K}{MV_t} \quad a_2(t) = \frac{K}{M} + \frac{4\beta_e A_p^2}{MV_t} + \frac{4\beta_e(C_l + C_t)B}{MV_t}$$

$$a_3(t) = \frac{B}{M} + \frac{4\beta_e(C_l + C_t)}{V_t}, \quad g(x) = \frac{4A_p\beta_e D_l K_a}{MV_t}$$

$$d(x) = -\left(\frac{4\beta_e(C_l + C_t)}{V_t M} f_l + \frac{1}{M} \dot{f}_l\right)$$

in which $g(x)$ is a constant with positive value.

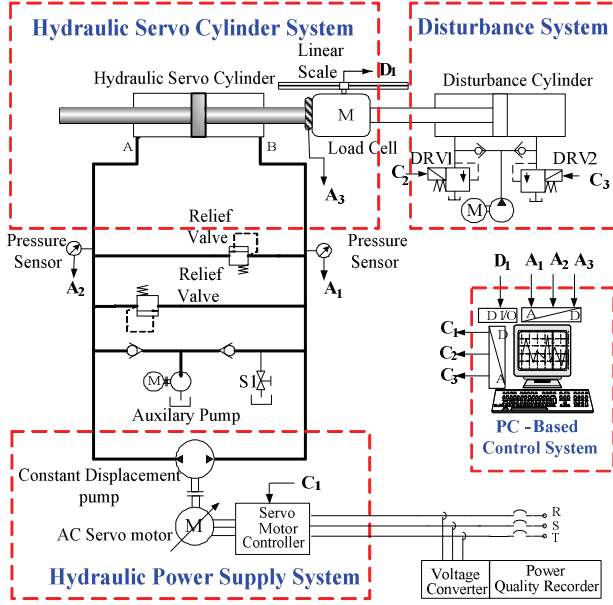


Fig. 1: Layout of the electro-hydraulic variable rotational speed pump-controlled system

III. CONTROLLER DESIGN AND STABILITY ANALYSIS

The design of a sliding mode controller requires the system models and has to find the inverse form of the inertia term in system dynamics. However, the accurate mathematical models are sometimes difficult to be formulated or even not available. To solve these problems, an AFC-STFSMC is proposed in this paper for realizing the positioning control in EHVRSPCS. The block diagram of the AFC-STFSMC control system for EHVRSPCS is shown in Fig.2 in the last page.

A. Fuzzy control

According to the definitions [12], the basic configuration of fuzzy logic systems consists of four components: a fuzzifier, a fuzzy rule base, a fuzzy inference engine and a defuzzifier. In this paper single-input single-output rules will be used in formulation of the control law. Assume that there are n rules in a fuzzy base and each of them has the following form:

$$R^i: \text{IF } S \text{ is } F^i \text{ THEN } u \text{ is } \alpha_i \quad (7)$$

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 $i=1,2,\dots,n$. The defuzzification of the FC output is accomplished by the method of center-of-gravity [12]

$$u = \frac{\sum_{i=1}^n \alpha_i \times w_i}{\sum_{i=1}^n w_i} = \underline{\alpha}^T \underline{\xi} \quad (8)$$

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

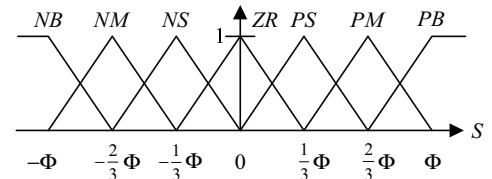
$$\xi_i = \frac{w_i}{\sum_{i=1}^n w_i} \quad (9)$$

In conventional FC, the control actions α_i should be previously assigned through a lot of trails for achieving satisfactory control performance. In this paper, an adaptive algorithm will be proposed to tune these control actions on-line.

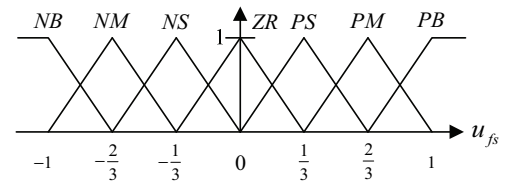
B. Fuzzy Sliding-Mode Control

Hwang *et al.* [13] proposed 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. 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, this paper adopts the sliding surface $S = 0$ of SMC and extends it to the fuzzy sliding surface $\tilde{S} = \tilde{0}$, and make S be a linguistic description of $\tilde{S}$. In this study 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

$$R^l: \text{IF } S \text{ is } \tilde{F}_s^l \text{ THEN } u_{fs} \text{ is } \tilde{F}_u^{8-l}, \quad l=1,\dots,7. \quad (10)$$



(a)



(b)

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

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

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

where $z = \frac{S}{\Phi}$, and as $|S| \geq \Phi$, it is easy to check $u_{fs} = -\text{sgn}(S)$.

C. Adaptive Fuzzy Controller with Self-Tuning Fuzzy Sliding-Mode Compensation

The control objective is to find a control law that can make the hydraulic actuator of EHVRSPCS tracking the desired target $x_d(t)$. Define the tracking error $e(t)$ as

$$e(t) = x_d(t) - x(t) \quad (12)$$

where $x(t)$ indicates the control output and $x_d(t)$ is the desired target. Define a sliding surface as

$$S(t) = \ddot{e}(t) + k_1 \dot{e}(t) + k_2 e(t) \quad (13)$$

where k_1 and k_2 are non-zero positive constants. Assume that parameters of the system in Eq. (6) are well known and the external load disturbance is measurable, then the control law can be defined as

$$u^* = g^{-1}(\underline{x})[\eta S_\Delta(t) - f(\underline{x}) - d(\underline{x}) + \ddot{x}_d + k_1 \dot{e}(t) + k_2 e(t)] \quad (14)$$

where $S_\Delta(t) = S(t) - \Phi \text{sat}(S(t)/\Phi)$ and Φ is the boundary layer width of the sliding surface S . The properties of the function S_Δ that are useful in the design of adaptive law [14] are described as below.

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

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

The above properties of the boundary layer concept are exploited in the design of controller to cease adaptation as soon as the boundary layer is reached and to avoid the possibility of unbounded growth. Differentiating Eq. (13), we have

$$\dot{S}(t) = -f(\underline{x}) - g(\underline{x})u - d(\underline{x}) + \ddot{x}_d + k_1 \dot{e}(t) + k_2 e(t) \quad (15)$$

Substituting Eq. (16) into Eq.(17) gives

$$\dot{S}(t) + \eta S_\Delta(t) = 0, \quad \eta > 0. \quad (16)$$

Equation (16) shows that $\underline{e}(t) = [e(t), \dot{e}(t), \ddot{e}(t)]$ will converge to the neighborhood of zero as $t \rightarrow \infty$, and the value of the neighborhood depends on the value of Φ [19]. However, the system parameters may be unknown or perturbed; the controller u^* cannot be precisely implemented. Therefore, in accordance with the universal approximation theorem [11], for any $\rho^* > 0$, an optimal fuzzy control $\hat{u}_{fz}(S, \underline{\alpha}^*)$ exists in the form of Eq. (10) that satisfies

$$|u^* - \hat{u}_{fz}(S, \underline{\alpha}^*)| = \rho^* \quad (17)$$

where ρ^* is assumed to be bounded by $|\rho^*| \leq M$. Utilize a fuzzy controller $\hat{u}_{fz}(S, \underline{\alpha})$ to approximate u^* as

$$\hat{u}_{fz}(S, \underline{\alpha}) = \hat{\underline{\alpha}}^T \underline{\xi} \quad (18)$$

where $\hat{\underline{\alpha}}$ is the estimated values of $\underline{\alpha}^*$. The control law for the developed AFC-STFSMC is defined as the following form:

$$u = \hat{u}_{fz}(S, \underline{\alpha}) + u_{comp}(S) \quad (19)$$

where the fuzzy controller $\hat{u}_{fz}$ is designed to approximate the control u^* and the fuzzy sliding-mode compensation u_{comp} is proposed to compensate the difference between the controller u^* and fuzzy controller $\hat{u}_{fz}(S, \underline{\alpha})$. Through Eqs. (14), (15) and (19), the dynamical equation can be derived as:

$$\dot{S}(t) + \eta S_\Delta(t) = g[u^* - \hat{u}_{fz}(S, \underline{\alpha}) - u_{rb}(S)] \quad (20)$$

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

$$V(S_\Delta, \underline{\tilde{\alpha}}, \tilde{\rho}) = \frac{1}{2} \frac{S_\Delta^2}{g} + \frac{1}{2\eta_1} \underline{\tilde{\alpha}}^T \underline{\tilde{\alpha}} + \frac{1}{2\eta_2} \tilde{\rho}^2 \quad (21)$$

where $\underline{\tilde{\alpha}}^T = \hat{\underline{\alpha}}^{*T} - \hat{\underline{\alpha}}^T$ and $\tilde{\rho} = \rho^* - \hat{\rho}$ are the approximation error of the parameter vectors $\hat{\underline{\alpha}}^{*T}$ and ρ^* respectively. In addition, η_1 and η_2 are positive constants. Differentiate Eq. (21) with respect to time as

$$\dot{V}(S_\Delta, \underline{\tilde{\alpha}}, \tilde{\rho}) = \frac{S_\Delta \dot{S}_\Delta}{g} + \frac{1}{\eta_1} \underline{\tilde{\alpha}}^T \dot{\underline{\tilde{\alpha}}} + \frac{1}{\eta_2} \tilde{\rho} \dot{\tilde{\rho}}. \quad (22)$$

Thus, if $|S| \leq \Phi$, then $S_\Delta = 0$, it follows $\dot{V}(S_\Delta, \underline{\tilde{\alpha}}, \tilde{\rho}) = 0$. If $|S| > \Phi$, then $\dot{S}_\Delta = \dot{S}$. By substituting Eq. (20) into Eq. (22), Eq. (23) can be obtained

$$\begin{aligned}
\dot{V}(S_\Delta, \tilde{\alpha}, \tilde{\rho}) &= -\eta \frac{S_\Delta^2}{g} + S_\Delta [u^* - \hat{u}_f(S, \hat{\alpha})] - S_\Delta u_{rb}(S) + \frac{1}{\eta_h} \tilde{\alpha}^T \dot{\tilde{\alpha}} + \frac{1}{\eta_2} \tilde{\rho} \dot{\tilde{\rho}} \\
&= -\eta \frac{S_\Delta^2}{g} + S_\Delta [u^* - \hat{u}_f(S, \hat{\alpha}) + \hat{u}_f(S, \hat{\alpha}^*) - u_{fb}(S, \hat{\alpha})] - S_\Delta u_{rb}(S) + \frac{1}{\eta_h} \tilde{\alpha}^T \dot{\tilde{\alpha}} + \frac{1}{\eta_2} \tilde{\rho} \dot{\tilde{\rho}} \\
&\leq -\eta \frac{S_\Delta^2}{g} + |S_\Delta| |u^* - \hat{u}_f(S, \hat{\alpha}^*)| + S_\Delta [\hat{u}_f(S, \hat{\alpha}^*) - \hat{u}_f(S, \hat{\alpha})] - S_\Delta u_{rb}(S) + \frac{1}{\eta_h} \tilde{\alpha}^T \dot{\tilde{\alpha}} + \frac{1}{\eta_2} \tilde{\rho} \dot{\tilde{\rho}} \\
&= -\eta \frac{S_\Delta^2}{g} + |S_\Delta| |\rho^* - S_\Delta u_{rb}(S)| - \frac{1}{\eta_2} \tilde{\rho} \dot{\tilde{\rho}} + \tilde{\alpha}^T (S_\Delta \xi(S) - \frac{1}{\eta_h} \dot{\tilde{\alpha}})
\end{aligned} \tag{23}$$

For achieving $\dot{V}(S_\Delta, \tilde{\alpha}, \tilde{\rho}) \leq 0$, the adaptive laws of AFC-STFSMC are chosen as

$$\dot{\tilde{\alpha}} = \eta_1 \cdot S_\Delta \cdot \xi(S) \tag{24}$$

$$u_{comp}(S) = -\hat{\rho} \cdot u_{fs} \tag{25}$$

$$\dot{\tilde{\rho}} = \eta_2 \cdot |S_\Delta| \tag{26}$$

Thus Eq. (23) can be rewritten as

$$\dot{V}(S_\Delta, \tilde{\alpha}, \tilde{\rho}) \leq -\eta \cdot \frac{S_\Delta^2}{g} \leq 0 \tag{27}$$

Equations (24)-(27) guarantee only $S_\Delta \in L_\infty$ but don't guarantee convergence. Integrating both sides of (27) and using some derivation, we have

$$\int_0^\infty S_\Delta^2 dt \leq \frac{V(S_\Delta(\infty), \tilde{\alpha}, \tilde{\rho}) - V(S_\Delta(0), \tilde{\alpha}, \tilde{\rho})}{\frac{\eta}{g}}. \tag{28}$$

Since the right side of Eq. (28) is bounded, $S_\Delta \in L_2$. Using Barbalat's lemma [19], $\lim_{t \rightarrow \infty} S_\Delta = 0$ can be shown. That means that the inequality $|S| \leq \Phi$ is obtained asymptotically, so the tracking error $e(t)$ converges to neighborhood of zero. In summary, AFC-STFSMC is derived in Eq. (19), where $\hat{u}_f$ is given in Eq. (18) with the parameters $\hat{\alpha}$ adjusted by Eq. (24) and u_{comp} is given in Eq. (25) with the parameter $\hat{\rho}$ adjusted by Eq. (26). By applying these adaptive laws, the controller of Eq. (19) can be guaranteed the EHVRSPCS to be stable.

IV. EXPERIMENTS

For verifying the control performance of the proposed AFC-STFSMC controller, the positioning control experiments in EHVRSPCS are implemented under different strokes and different external loadings.

(a) Positioning control at different strokes with external loading force

The experiment is respectively achieved for different stroke set points such as 100, 200 and 300 mm with external loading force 30kN. Fig.4 shows the response for different strokes. The experimental results show that the piston of cylinder can, at different strokes with loading force 30kN, reach the set point within a little overshoot 18 μ m and remain the steady state error within 10 μ m.

(b) Positioning control under different loading conditions

For confirming the control performance under different external loading conditions, the experiments of position control under three constant external loading conditions, such as 10kN, 30kN and 50kN, are investigated. Fig.5 indicates the step responses of the position control of 300 mm stroke with different loadings. The experimental results show that the piston of cylinder can, under 10kN and 30kN loading conditions, reach the set point within a little overshoot 14 μ m and remain the steady state error within 3.6 μ m. Even under 50kN loading condition, the piston can reach the set point within 38 μ m overshoot and remain the steady state error within 17 μ m.

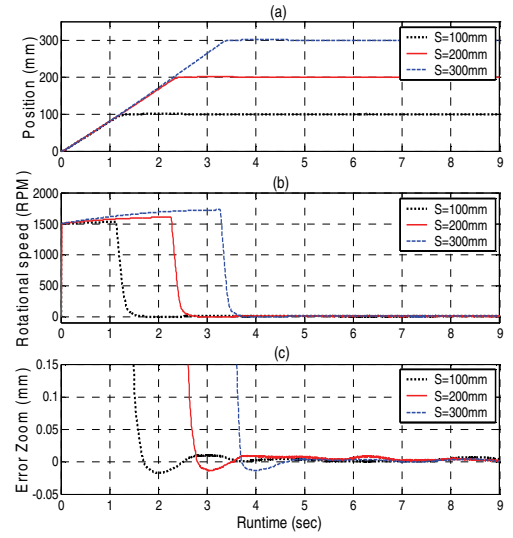


Fig.4: Experimental results of positioning control for different strokes with loading force 30kN in variable rotational speed pump-controlled system: (a) position control response (b) controller input (c) control error

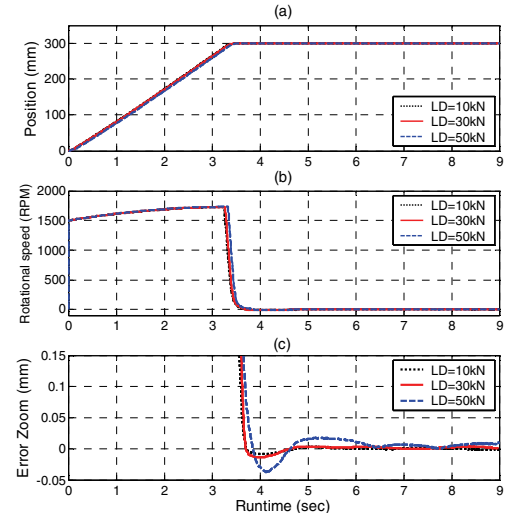


Fig.5: Experimental results of positioning control of 300mm stroke with different loadings in variable rotational speed pump-controlled system: (a) position control response (b) controller input (c) control error

V. CONCLUSIONS

This paper develops a new EHVRSPCS driven by an AC servo motor for realizing motion control of hydraulic servo machines with high response. The swash plate axial piston pump with constant displacement combines with the AC servo motor due to the better response, instead of the internal gear pump used in [3-6]. The positioning control in different loading conditions are implemented and verified experimentally for EHVRSPCS in this paper. For achieving better control performance, an AFC-STFSMC is proposed for positioning control of the EHVRSPCS. The AFC-STFSMC strategy combining with the adaptive fuzzy control and the self-tuning fuzzy sliding-mode control scheme is comprised of the adaptive fuzzy approximation technique to establish the equivalent controller of the traditional SMC and the FSMC scheme with self-tuning ability for compensating the approximation error of equivalent controller and external disturbance. Excellent self-adaptability with regard to various strokes and uncertain external disturbance force are verified experimentally.

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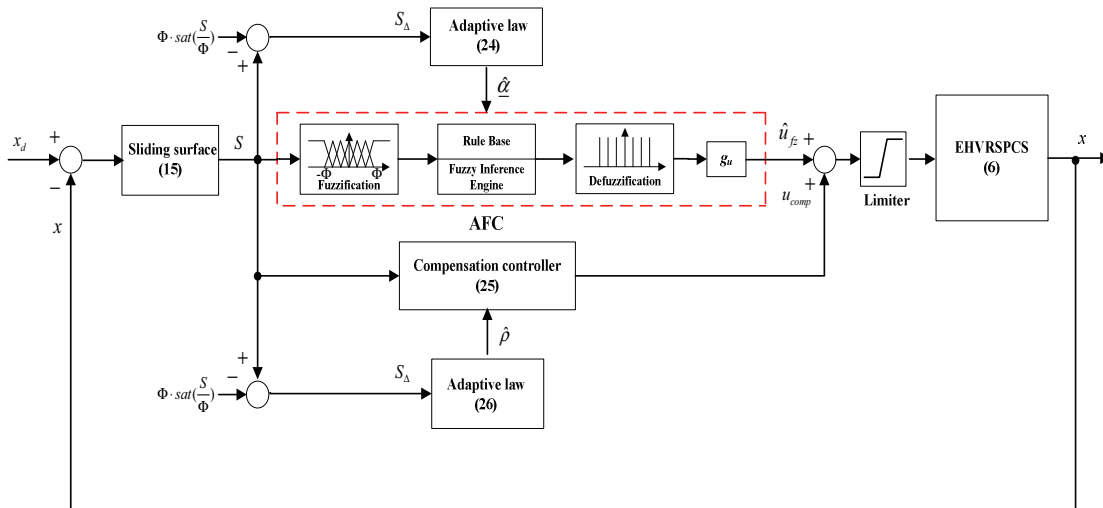


Fig.2. Adaptive Fuzzy Controller with Self-Tuning Fuzzy Sliding-Mode Compensation for Electro-Hydraulic Variable Rotational Speed Pump-Controlled System