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The Force Control of a Novel Variable Rotational Speed Hydraulic Pump-Controlled System Using Adaptive Fuzzy Controller with Self-tuning Fuzzy Sliding-mode Compensation

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Abstract: The conventional hydraulic valve-controlled systems have high response but low energy-efficiency. Hydraulic pump-controlled servo systems have high energy efficiency. However, the conventional pump-controlled systems, which are altered by displacement via variable displacement pumps, have lower response. This paper aims to investigate the force control performance of the high response electro-hydraulic pump-controlled systems 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 research. Thus, the new hydraulic pump-controlled system with an AC motor servo and a constant displacement axial piston pump is investigated for force control of hydraulic servo machines. For that, a novel adaptive fuzzy controller with self-tuning fuzzy sliding-mode compensation (AFC- STFSMC) is proposed for force control in the variable rotational speed pump-controlled system (VRSPCS). Thus, the developed high response variable rotational speed pump-controlled systems controlled by AFC-STFSMC are implemented and verified experimentally for force control in different force targets. The experimental results of the force control in the VRSPCS show that the proposed system and control method can achieve excellent control performance and robustness with regard to parameter variations and external disturbance.

Keywords: force control, variable rotational speed pump- controlled system, AC servo motor, constant displacement axial piston pump.

1. INTRODUCTION

Hydraulic servo systems are widely used in the industry due to the advantages of high power-weight ratio, large driving forces and high robustness. However, the present hydraulic servo systems are requested for both high response and high energy-efficiency in competition with the full electrical motor driving systems. In view of the hydraulic circuits, two different hydraulic systems are classified, such as hydraulic valve-controlled system and hydraulic displacement-controlled system (Murrenhoff, 1998). The conventional hydraulic valve-controlled systems, which actuators are controlled by hydraulic servo valves, have high response but low energy-efficiency. Some researches have focused on the improvement of energy-efficiency of the hydraulic-valve-controlled systems. The energy-efficiency of the hydraulic valve-controlled system can be improved by the energy-saving control systems; however, it is still lower than that of the hydraulic pump-controlled system due to the orifice effect of the servo valve (Helduser, 1999; Habibi, 1999; Helbig, 2002).

Hydraulic pump-controlled systems have high energy efficiency. However, the conventional pump-controlled systems that are altered by displacement via variable displacement pumps or constant displacement pumps driven by AC inducted motors with variable rotational speed have lower response. Recently, high response pump-controlled systems driven by AC servo motors are introduced (Esders, 1994). Helduser (1995) 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 (1997) 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 50μm, is still unsatisfactory. Kazmeier and Feldmann (1998) 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. Bildstein (1998) compared the performance of pump-controlled systems with variable rotational speed and that with variable displacement applying to flight control system of Air Bus A321. Helduser (1999) developed an electric-hydrostatic drive using AC servo motor and constant displacement internal gear pump for energy-saving power and motion control system. Habibi and Goldenberg (1999) discussed the design problem of the electro-hydraulic actuator using gear pump and electromotor. Helbig (2002) achieved high efficiency and high response characteristics in velocity and pressure control for injection moulding machine with electric-hydrostatic drives. From surveying the references, AC servo motors combined with internal gear pumps were use mostly. The position control

can reach 50μm accuracy. Besides, the investigations on high response and high efficiency pump-controlled systems are still in progress. New applications of the high response and high energy-efficiency in different hydraulic servo machines are still being developed.

This paper aims to investigate the force control performance of the high response electro-hydraulic pump-controlled systems driven by an AC servo motor with variable rotational speed. Instead of internal gear pumps discussed in the references, a constant displacement axial piston pump, which has better performance and efficiency than the internal gear pump, is used in this research. Thus, the new hydraulic pump-controlled system with an AC motor servo and a constant displacement axial piston pump is investigated for force control of hydraulic servo machines. For that, a novel adaptive fuzzy controller with self-tuning fuzzy sliding-mode compensation (AFC-STFSMC) is proposed for force control in the variable rotational speed pump-controlled system (VRSPCS). Thus, the developed high response variable rotational speed pump-controlled systems controlled by AFC-STFSMC are implemented and verified experimentally for force control in different force targets and sine wave force tracking.

2. THE LAYOUT OF EXPERIMENTAL SYSTEM

The test rig layout of the variable rotational speed pump-controlled system (VRSPCS) shown in Fig.1 is set up for experimentally investigating the dynamic behaviours of the control system in this paper. The test rig can be divided into four subsystems, including the hydraulic servo cylinder system, the hydraulic power supply system, the disturbance system and the PC-based control system. The specifications of the main components are listed in Table 1.

The hydraulic servo cylinder system 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, adjusts the supply volume flow by the rotational speed controlled by the AC servo motor. In the force control of the VRSPCS, the motion of the controlled cylinder is regulated directly by the volume flow of the constant displacement pump, i.e. the motion of the controlled cylinder is controlled directly by the rotational speed of the AC servo motor. Thus, the control input signals of the AC servo motor are given from the PC-based controller with the sampling time of 5 ms via a D/A converter and enlarged by a servo amplifier. The force signal is measured by the load cell and fed back to the PC-based controller. Therefore, the overall system contains an electro-hydraulic pump-controlled system driven by the variable rotational speed AC servo motor. The overall electrical power supplied to the electro-hydraulic pump-controlled system is measured by the power quality recorder for energy efficiency analysis and comparison. Besides, the disturbance system, including a disturbance cylinder, two relief valves and a gear pump, is used here to generate external disturbance forces, which can be determined by setting the pressure of the relief valves DRV1 and DRV2, for the different loading conditions of experiments.

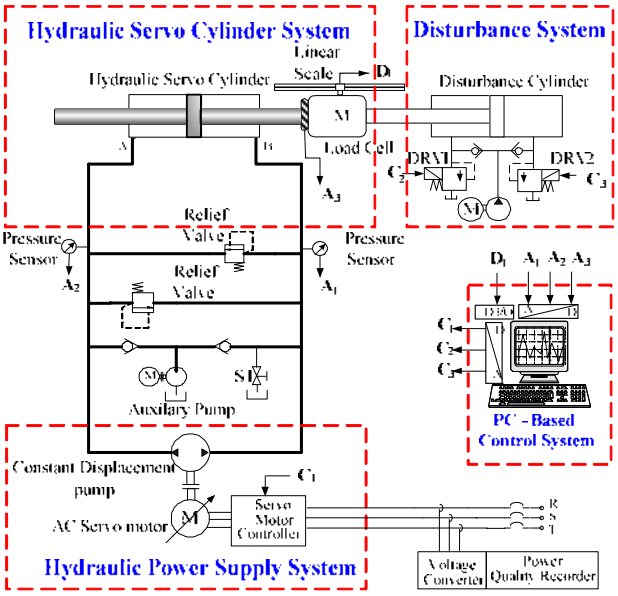

Fig. 1. The layout of test rig

Table 1: Main components' specifications of the test rig

Components	Specifications
AC servo motor	Input : 3φ 220V Rated rotational speed: 2000 rpm Rated output: 7.0kW
Servo motor driver	Input voltage: 200~230V FLC: 46A Output freq.:0~ 200Hz Output power: 7.5 kW
Hydraulic pump	Swash plate axial piston pump Max. pressure: 40 MPa Fixed displacement: 12 ml/rev Max. rotational speed: 8000 rpm Direction of flow: reversible
Hydraulic servo cylinder	Max. pressure: 21 MPa Max. stroke: 400mm Piston diameter: 80mm Rod diameter: 45mm
Disturance cylinder system	Single rod double acting cylinder Max. stroke: 400mm Piston diameter: 80mm Rod diameter:45mm Proportional relief valve: max.20.7MPa
Optical encoder	Range: 500 mm Decoder: PCL-833 Resolution: 0.1 μm
Load cell	Range: 111kN/10V
PC-based controller	AMD K6-2 450 CPU 12 bit A/D; 12 bit D/A; D/I、D/O

3. FORMULATION OF VARIABLE ROTATIONAL SPEED PUMP-CONTROLLED SYSTEM

In accordance with the test rig in Fig. 1, the mathematic models of the main elements in the VRSPCS, including the servo motor control unit, the swash plate axial piston pump with fixed displacement and the controlled cylinder, are derived for the controller design. The state equations of the VRSPCS model can be achieved as follows

$$\begin{aligned} x_1 &= \int P_l\,dt \\ \dot{x}_1 &= x_2(t) = P_l \\ \dot{x}_2 &= x_3(t) = \dot{P}_l = -\sum_1^2 a_i x_i(t) + g(\underline{x})u + d(\underline{x}) = f(\underline{x}) + g(\underline{x})u + d(\underline{x}) \end{aligned} \tag{1}$$

where $\underline{x} = [\, x_1(t) \quad x_2(t)\,]^T = \left[\, \int P_l\,dt \quad P_l \right]^T$,

$a_1(t) = \dfrac{4\beta_e A_P^{\,2}}{MV_t}$, $a_2(t) = \dfrac{4\beta_e\,(C_l + C_t)}{V_t}$,

$g(\underline{x}) = \dfrac{4\beta_e D_l K_a}{V_t}$, $d(\underline{x}) = \dfrac{4\beta_e A_P}{MV_t}(B \cdot x_P + \int f_l dt)$

in which $g(\underline{x})$ is a constant with positive value.

4. CONTROLLER DESIGN

The adaptive fuzzy controller with self-tuning fuzzy sliding-mode compensation (AFC-STFSMC) is proposed in this paper for realizing the force control of the VRSPCS.

The control objective is to find a control law that can make the hydraulic actuator of the VRSPCS tracking the desired force $A_P P_l(t)$. Define the force error $e(t)$ as

$$e(t) = f_t(t) - f_{out}(t) = f_t(t) - A_P P_l(t) \tag{2}$$

where $f_{out}(t) = A_P P_l(t)$ indicates the control output and $f_t(t)$ is the desired force target. Define a sliding surface as

$$S(t) = \dot{e}(t) + k_1 e(t) \tag{3}$$

where k_1 is non-zero positive constant. Assume that parameters of the system in Eq. (1) are well known and the external disturbance is measurable, then the control law can be derived as

$$u^* = g^{-1}(\underline{x})[\frac{\eta}{k_1 A_P} S_\Delta(t) - f(\underline{x}) - d(\underline{x}) + \frac{\dot{f}_t(t)}{A_P} + \frac{\ddot{e}(t)}{k_1 A_P}] \tag{4}$$

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

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

$$\text{As } |S| \le \Phi, \Rightarrow S_\Delta = \dot{S}_\Delta = 0.$$

The above properties of the boundary layer concept are to be 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. Differentiate Eq. (3) as

$$\dot{S}(t) = -k_1 A_P[f(\underline{x}) + g(\underline{x})u + d(\underline{x})] + k_1 \dot{f}_t(t) + \ddot{e}(t) \tag{5}$$

Substituting Eq. (4) into Eq. (5) gives

$$\dot{S}(t) + \eta S_\Delta(t) = 0, \qquad \eta > 0. \tag{6}$$

Eq. (6) shows that $\underline{e}(t) = [e(t), \dot{e}(t)]$ will converge to the neighbourhood of zero as $t \to \infty$, and the value of the neighbourhood depends on the value of Φ. However, the system parameters may be unknown or perturbed; the controller u^* cannot be precisely implemented. Therefore, by the universal approximation theorem], for any $\rho^* > 0$, an optimal fuzzy control $\hat{u}_{fz}(S, \underline{\alpha}^*)$ exists and satisfies

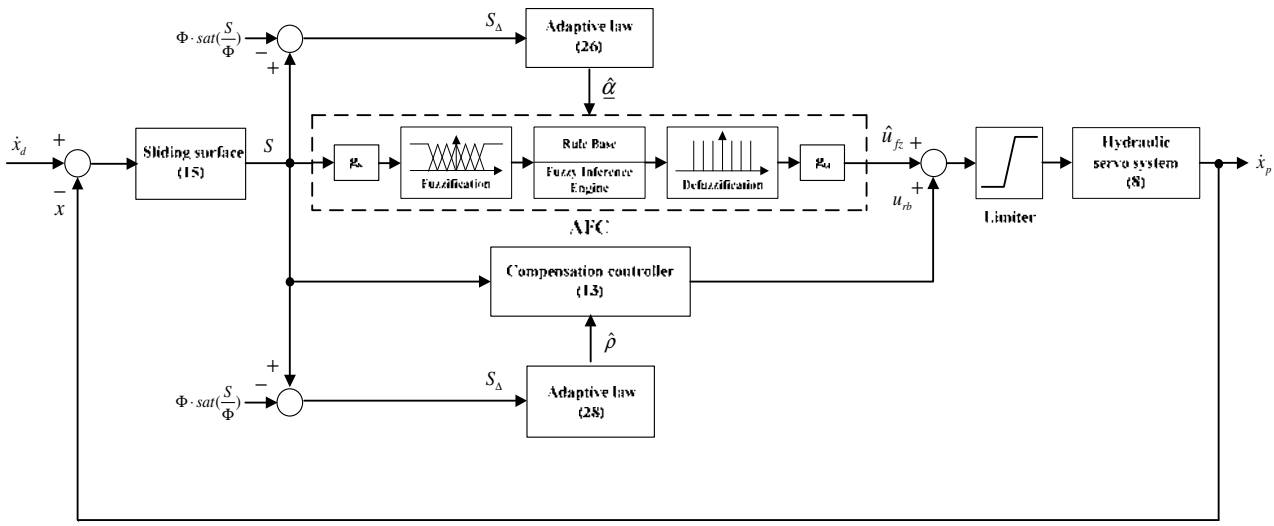

Fig.2. AFSMC for Velocity Control of a Variable Displacement Hydraulic Servo System

$$\left|u^* - \hat{u}_{fz}(S,\underline{\hat{\alpha}}^*)\right| \le \rho^* \tag{7}$$

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

$$\hat{u}_{fz}(S,\underline{\hat{\alpha}}) = \underline{\hat{\alpha}}^T \underline{\xi} \tag{8}$$

where $\underline{\hat{\alpha}}$ is the estimated values of $\underline{\alpha}^*$. The control law for the developed adaptive fuzzy controller with self-tuning fuzzy sliding-mode compensation (AFC-STFSMC) is defined as the following form:

$$u = \hat{u}_{fz}(S,\underline{\hat{\alpha}}) + u_{comp}(S) \tag{9}$$

where the fuzzy controller $\hat{u}_{fz}$ is designed to approximate the controller u^* and the fuzzy sliding-mode compensation u_{comp} is proposed to compensate the difference between the controller u^* and fuzzy controller $\hat{u}_{fz}(S,\underline{\hat{\alpha}})$. Through Eqs. (4), (5) and (9), the dynamic equation can be derived as:

$$\dot{S}(t) + \eta S_\Delta(t) = g[u^* - \hat{u}_{fz}(\mathrm{S},\underline{\hat{\alpha}}) - u_{comp}(S)] \tag{10}$$

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

$$V(S_\Delta,\underline{\tilde{\alpha}},\tilde{\rho}) = \frac{1}{2}\frac{S_\Delta^2}{g} + \frac{1}{2\eta_1}\underline{\tilde{\alpha}}^T\underline{\tilde{\alpha}} + \frac{1}{2\eta_2}\tilde{\rho}^2 \tag{11}$$

where $\underline{\tilde{\alpha}}^{\mathrm{T}} = \underline{\hat{\alpha}}^{*\mathrm{T}} - \underline{\hat{\alpha}}^{\mathrm{T}}$ and $\tilde{\rho} = \rho^* - \hat{\rho}$ are the approximation error of the parameter vectors $\underline{\hat{\alpha}}^{*\mathrm{T}}$ and ρ^* respectively. In addition, η_1 and η_2 are positive constants. Differentiate Eq. (11) with respect to time as

$$\dot{V}(S_\Delta,\underline{\tilde{\alpha}},\tilde{\rho}) = \frac{S_\Delta \dot{S}_\Delta}{g} + \frac{1}{\eta_1}\underline{\tilde{\alpha}}^T\underline{\dot{\tilde{\alpha}}} + \frac{1}{\eta_2}\tilde{\rho}\dot{\tilde{\rho}}. \tag{12}$$

Thus, if $|S| \le \Phi$, then $\dot{V}(S_\Delta,\underline{\tilde{\alpha}},\tilde{\rho}) = 0$, if $|S| > \Phi$, then $\dot{S}_\Delta = \dot{S}$. By substituting Eq. (10) into Eq. (12), Eq. (13) can be obtained

$$\begin{aligned}
\dot{V}(S_\Delta,\underline{\tilde{\alpha}},\tilde{\rho}) &= -\eta\frac{S_\Delta^2}{g} + S_\Delta[u^* - \hat{u}_{fz}(S,\underline{\hat{\alpha}})] - S_\Delta u_{comp}(S) + \frac{1}{\eta_1}\underline{\tilde{\alpha}}^T\underline{\dot{\tilde{\alpha}}} + \frac{1}{\eta_2}\tilde{\rho}\,\dot{\tilde{\rho}} \\
&= -\eta\frac{S_\Delta^2}{g} + S_\Delta[u^* - \hat{u}_{fz}(S,\underline{\hat{\alpha}}^*) + \hat{u}_{fz}(S,\underline{\hat{\alpha}}^*) - \hat{u}_{fz}(S,\underline{\hat{\alpha}})] - S_\Delta u_{comp}(S) + \frac{1}{\eta_1}\underline{\tilde{\alpha}}^T\underline{\dot{\hat{\alpha}}} + \frac{1}{\eta_2}\tilde{\rho}\,\dot{\hat{\rho}} \\
&\le -\eta\frac{S_\Delta^2}{g} + |S_\Delta|\left|u^* - \hat{u}_{fz}(S,\underline{\hat{\alpha}}^*)\right| + S_\Delta[\hat{u}_{fz}(S,\underline{\hat{\alpha}}^*) - \hat{u}_{fz}(S,\underline{\hat{\alpha}})] - S_\Delta u_{comp}(S) + \frac{1}{\eta_1}\underline{\tilde{\alpha}}^T\underline{\dot{\hat{\alpha}}} + \frac{1}{\eta_2}\tilde{\rho}\,\dot{\hat{\rho}} \\
&\le -\eta\frac{S_\Delta^2}{g} + [|S_\Delta|\rho^* - S_\Delta u_{comp}(S) - \frac{1}{\eta_2}\tilde{\rho}\dot{\hat{\rho}}] + \underline{\tilde{\alpha}}^T(S_\Delta\underline{\xi}(S) - \frac{1}{\eta_1}\underline{\dot{\hat{\alpha}}})
\end{aligned} \tag{13}$$

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

$$\underline{\dot{\hat{\alpha}}} = \eta_1 \cdot S_\Delta \cdot \underline{\xi}(S) \tag{14}$$

$$u_{comp}(S) = -\hat{\rho} \cdot u_{fs} \tag{15}$$

$$\dot{\hat{\rho}} = \eta_2 \cdot |S_\Delta| \tag{16}$$

Thus, Eq. (11) can be rewritten as

$$\dot{V}(S_\Delta,\underline{\tilde{\alpha}},\tilde{\rho}) \le -\eta \cdot \frac{S_\Delta^2}{g} \le 0 \tag{17}$$

In summary, the AFC-STFSMC is presented in Eq. (9), where $\hat{u}_{fz}$ is given in Eq. (8) with the parameters $\underline{\hat{\alpha}}$ adjusted by Eq. (14) and u_{comp} is given in Eq. (13) with the parameter $\hat{\rho}$ adjusted by Eq. (16). By applying these adaptive laws, the controller of Eq. (9) can be guaranteed to be stable.

5. EXPERIMENTS

The force control experiments are achieved for different force targets such as 30kN, 50kN and 65kN. Figure 2 shows the response comparison in the experiment. Figure 2(a) shows that the rising time of the various force control outputs are below 0.71 seconds, and the settling times are controlled within 0.89 seconds for different step force inputs. Thus, the high response performance of the VRSPCS is verified. Figiure 2(b) schematically depicts the zoom in of the steady-state errors that can be controlled within 1.0 kN so that the performance of the VRSPCS on force control accuracy can be clarified.

The power consumption in the experiment of force control is discussed. The overall electrical power P_{in} supplied to the VRSPCS is directly measured by the power quality recorder. The output power P_{out} of the controlled cylinder can be described as

$$P_{out} = F \cdot \dot{x} \tag{18}$$

where F and $\dot{x}$ are the output force and velocity of the controlled cylinder respectively. The output force is directly measured by the load cell.

Figure 3(a) shows the rotational speed variations of AC servo motor in the force control processes that are proportional to the control inputs from the force controller. The supply pressure to the cylinder p_s and the velocity of the cylinder $\dot{x}$ in the experiment of force control are shown in Fig.3(b) and Fig.3(c) respectively. They verify that the VRSPCS requires adequate pressure supplied but very low output rotational speed while the system state entering the steady state. Fig. 2(a) indicates the variations of the cylinder's output force F in the force control. Thus, the supply power P_{in} measured by the power quality recorder and the output power P_{out} calculated by Eq.(2) for the force control of 30kN, 50kN and 65kN respectively are shown in Fig.4, which indicate that the VRSPCS requires adequate power supplied while steady state. Therefore, it is evident that the VRSPCS realizes the force control performance of high response and low power consumption.

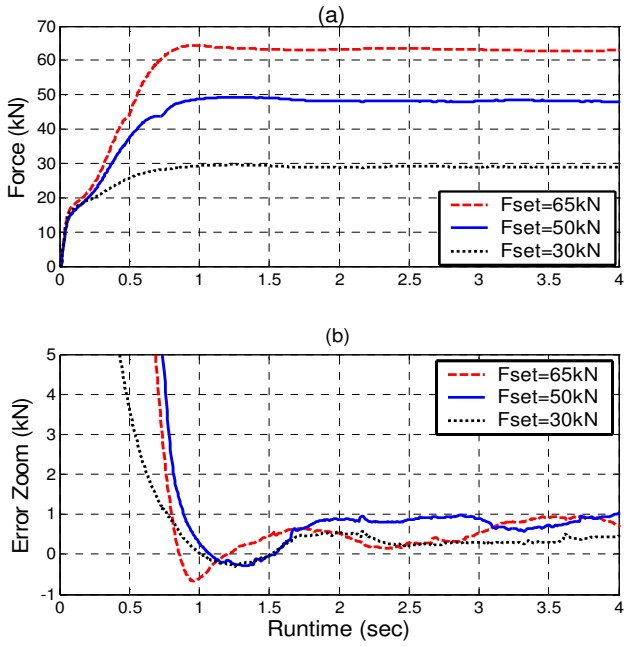

Fig.2. Experimental results of force control on different step force inputs in variable rotational speed pump-controlled system: (a) force control response (b) Zoom in of control error

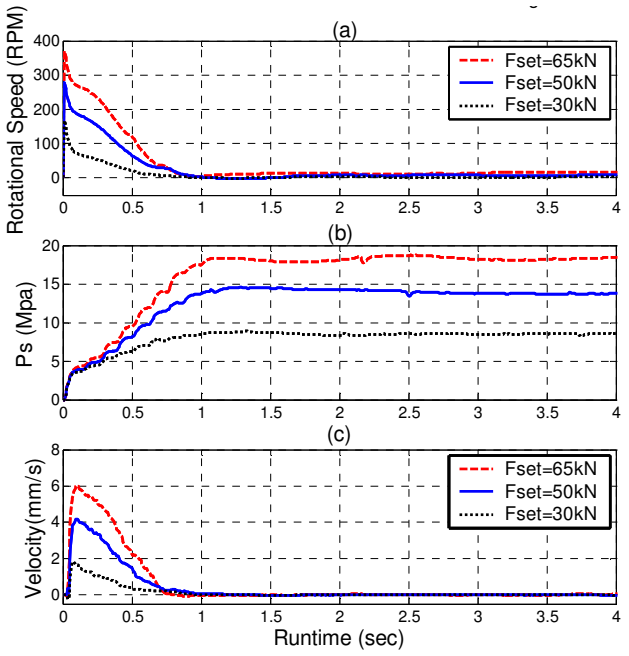

Fig.3. (a) Rotational speeds of AC servo motor for different force targets (b) Supply pressure variations for different force targets (c) Velocity of hydraulic cylinder for different force target

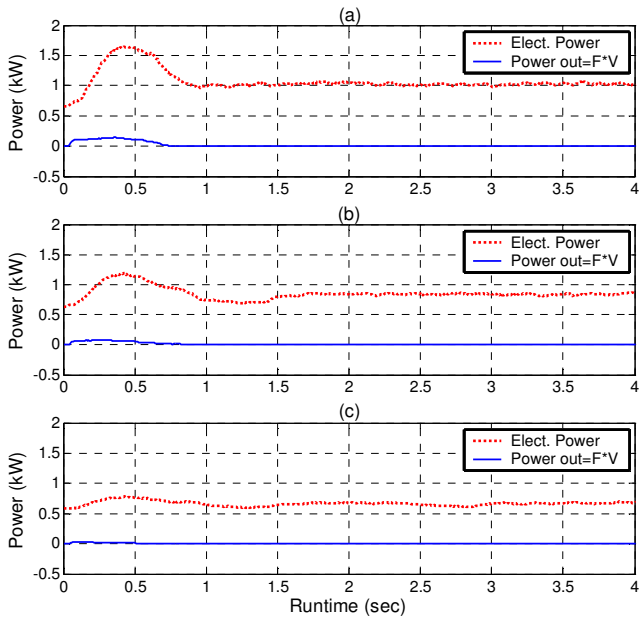

Fig.4. Supply power and output power in force control for different force targets: (a) 65kN (b)50kN (c)30kN

6. CONCLUSIONS

This study developed a new variable rotational speed electro-hydraulic pump-controlled system driven by the AC servo motor for realizing force control with both high response and high energy-efficiency, instead of the integration control concept of the hydraulic valve-controlled system that are complicated and have lower energy-efficiency.

The experiments of force control are implemented for different force targets, such as 30kN, 50kN and 65 kN. The experimental results show that the rising times of the various force control outputs are below 0.71 seconds, and the settling times are controlled within 0.89 seconds for different step force inputs. Thus, the high response performance of the VRSPCS is verified.

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