Parallel Control of Velocity Control and Energy-Saving Control for a Hydraulic Valve-Controlled Cylinder System Using Self-Organizing Fuzzy Sliding Mode Control*

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Conventional hydraulic valve-controlled systems that incorporate positive displacement pumps and relief valves have a problem of low energy efficiency. The objective of the research is to implement parallel control of energy-saving control in an electro-hydraulic load-sensing system and velocity control in a hydraulic valve-controlled cylinder system to achieve both high velocity control accuracy and low input power simultaneously. The overall control system is a two-input two-output system. For that, the control strategy of self-organizing fuzzy sliding mode control (SOFSMC) is developed in this study to reduce the fuzzy rule number and to self-organize on-line the fuzzy rules. To compare the energy-saving performance, the velocity control is implemented under three different energy-saving control systems, such as load-sensing control system, constant supply pressure control system and conventional hydraulic system. The parallel control of the velocity control and energy-saving control by the SOFSMC is implemented experimentally.

Key Words: Electro-hydraulic Valve-Controlled System, Energy-Saving Control, Load-Sensing Control, Velocity Control, Parallel Control, Self-Organizing Fuzzy Sliding Mode Control

1. Introduction

Conventional hydraulic valve-controlled systems, which maximum supply pressure is set by positive displacement pumps and relief valves, have a problem of low energy efficiency. An electro-hydraulic load-sensing system (ELS) is developed to improve this problem. The load-sensing pressure of a hydraulic cylinder is changed according to external loads. The load-sensing pump, which is a swash plate axial piston pump with variable displacement, can automatically adjust the supply pressure to maintain constant difference between the supply pressure and the load-sensing pressure⁽²⁾⁻⁽¹⁰⁾.

Research in the field of load-sensing control has been published⁽²⁾⁻⁽¹⁰⁾. The publications^{(2),(3)} concen-

trated on the parallel control of motion control and energy-saving control on a hydraulic valve-controlled system. Rotational speed control and load-sensing control on a hydraulic valve-controlled motor system were integrated using PID control⁽²⁾. The path control integrated with load-sensing control was discussed⁽³⁾. Therefore, the integration of servo control and energy-saving control becomes a complex MIMO control problem due to their coupling interaction. As the ELS regulates the supply pressure, the dynamic response of servo control of hydraulic actuators is influenced. In contrast, the switch of the servo valve for the servo control also affects the stability of the supply pressure control of the ELS.

This research aims to implement the parallel control of energy-saving control, including load-sensing control and constant supply pressure control, in the ELS and velocity control in a hydraulic valve-controlled cylinder system using modern control theory. Because the control system is a complex two-input two-output (TITO) system, self-organizing

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fuzzy sliding mode control (SOFSMC) is developed in this study as controllers' solutions. The SOFSMC combines self-organizing fuzzy control^{(15)–(18)} and fuzzy sliding mode control^{(12)–(14)} so that the number of fuzzy rules can be reduced by a fuzzy sliding surface and the fuzzy rules can be on-line self-tuning by a self-organizing modifier. The complex control problem of the parallel control of the two control systems, i.e. velocity control and energy-saving control, is solved by the SOFSMC experimentally.

2. Layout of Experimental System

The test rig in this work can be divided into a hydraulic valve-controlled cylinder system for velocity control and an electro-hydraulic load-sensing system (ELS) for energy-saving control, as shown in Fig. 1. The specifications of the main components are listed in Table 1. Therefore, the overall system, including velocity control system and energy-saving control system, becomes a two-input two-output (TITO) control system.

2.1 Velocity control system

The velocity control is implemented in the hydraulic valve-controlled cylinder system that incorporates a load-sensing cylinder, a load-sensing valve (proportional valve), a position sensor, a force sensor, pressure sensors and a disturbance system. The velocity signals are generated by the digital differential of position signals. The disturbance system, including a disturbance cylinder, two relief valves and a gear pump, is used here to generate external disturbance force. The pressure variations in the two chambers of the load-sensing cylinder are measured, but only the pressure from the load-sensing

valve, so-called load-sensing pressure p_{LS} , is fed back for the load-sensing control, and the supply pressure is controlled by the ELS.

2.2 Energy-saving control system

The electro-hydraulic load-sensing system (ELS) constitutes a load-sensing pump, pressure sensors and a swash plate control unit. The load-sensing pump is a swash plate axial piston pump with variable displacement, which swash plate's angle is controlled by the swash plate control unit to alter the pump's displacement. The swash plate control unit contains a small hydraulic valve-controlled cylinder system with position control. The supply pressure p_s is fed back. The object of the load-sensing control is to retain constant difference Δp between the supply pressure p_s from the load-sensing pump and the load-sensing pressure p_{LS} of the load-sensing cylinder, namely $\Delta p = p_s - p_{LS} = \text{constant}$, so that the supply pressure can be varied with the load-sensing pressure automatically.

3. Self-Organizing Fuzzy Sliding Mode Control Concept

Conventional fuzzy control theory contains fuzzification, fuzzy rule base, fuzzy inference and defuzzification. The fuzzy rule base that is established from experts' experience, however, may be unsuitable due to wrong experience^{(12)–(14)}. The self-organizing fuzzy sliding mode control (SOFSMC) which block diagram is shown in Fig. 2 is developed in this work to improve this problem. In many fuzzy control systems, the fuzzy rule base depends on control error e and control error rate ec that complicate the fuzzy inference rules and the membership functions. Instead of that, fuzzy sliding mode control (FSMC) introduces a sliding surface function σ to

Fig. 1 Layout of test rig

Table 1 Main components' specifications of the test rig

Component	Specifications
Load-sensing pump	variable displacement axial piston pump $q_{\max} = 98,3 \text{ ml/rev}$ $\alpha_{\max} = 21,5^\circ$ $\eta_{\text{vol}} = 98\% \text{ at } 1760 \text{ rpm}$
Swash plate control unit	servo valve $\pm 10 \text{ volt input}$ hydraulic cylinder $\pm 1,27 \text{ mm}$ with LVDT
Load-sensing valve	proportional directional valve $Q_{\text{rated}} = 150 \text{ l/min at } \Delta p = 7 \text{ MPa}$ $\omega_{-3\text{dB}} = 110 \text{ Hz at } \pm 5\% \text{ stroke}$
Load-sensing cylinder	$\phi 100/65, 500 \text{ mm}$, with LVDT load cell 30000 lbs (0-5 V, 30mA)
Disturbance cylinder unit	$\phi 100/65, 500 \text{ mm}$ relief valves max. pressure 20.7 MPa
PC-based control unit	PC with AMD K6-2 450 CPU $6 \times 12 \text{ bits D/A}$, $16 \times 12 \text{ bits A/D}$

Fig. 2 Block diagram of self-organizing fuzzy sliding mode control (SOFSMC) system

reduce the number of input variables and fuzzy inference rules^{(12)–(14)}. Besides, self-organizing fuzzy control can adapt on-line the fuzzy control rules in accordance with the environment variations^{(15)–(18)}. Therefore, self-organizing fuzzy sliding mode control (SOFSMC) is developed here to combine the advantages of fuzzy sliding mode control and self-organizing fuzzy control so that both the simplified fuzzy rule base and the capability of on-line self-organizing fuzzy rules can be achieved.

3.1 Fuzzy sliding mode control (FSMC)

Consider the state vector of a second order system as

$$\mathbf{x} = [x, \dot{x}]^T \quad (1)$$

The tracking error of the state vector is defined as

$$\mathbf{e} = \mathbf{x}_d - \mathbf{x} = [e, \dot{e}]^T = [x_d - x, \dot{x}_d - \dot{x}]^T \quad (2)$$

where x_d and $\dot{x}_d$ indicate the desired output and the desired output change rate. The fuzzy sliding surface is described as

$$\sigma = (e\dot{c} + \alpha \cdot e) = \text{ZERO} \quad (3)$$

where α is a positive constant. The fuzzy sliding surface $\sigma = \text{ZERO}$ is a straight line with slope α in the phase plane^{(12)–(15)}, as shown in Fig. 3 (a). The sliding surface can be divided into 7 sections by the membership function set of $M(\tilde{\sigma}) = \{NB, NM, NS, ZR, PS, PM, PB\}$, where NB, NM, NS, ZR, PS, PM and PB are negative big, negative medium, negative small, zero, positive small, positive medium and positive big respectively. The membership function set for the control input u is defined as $M(\tilde{u}) = \{NB, NM, NS, ZR, PS, PM, PB\}$. Therefore, instead of 7×7 fuzzy rules with control error e and error rate $\dot{e}$ in the conventional fuzzy control, the fuzzy sliding mode control can reduce the fuzzy rules into 7 rules via the fuzzy sliding surface σ , shown in Fig. 3. The Mamdani method is used in the fuzzy inference and the centroid method is used for defuzzification.

3.2 Self-organizing modifier

The self-organizing modifier of the SOFSMC is designed by self-organizing fuzzy control theory^{(14)–(16)} for on-line adapting the fuzzy rules according to the

Fig. 3 Fuzzy sliding surface and membership function of SOFSMC. (a) fuzzy sliding surface σ , (b) rule base of SOFSMC and membership function of fuzzy sliding surface $\tilde{\sigma}$, (c) the modified output $x'(nT)$ between $x(nT)$ and x_d

variations of working points and external disturbance. The system input $u(nT)$ results in the system output $x(nT)$, and the modified system input $u'(nT)$ has the modified system output $x'(nT)$. Define the modified value of the system input Δu and the system output change Δx as

$$\Delta u = u'(nT) - u(nT) \quad (4)$$

$$\Delta x = x'(nT) - x(nT) \quad (5)$$

In order to make the system output x approaching the desired output x_d and the output change rate $\dot{x}$ approaching the desired output change rate $\dot{x}_d$ simultaneously, the modified input value Δu must incorporate the two compensations, Δu_e and Δu_{ec} , for the change of system output Δx and that of system output change rate $\Delta \dot{x}$, which can be described as^{(15)–(18)}

$$\Delta u_e = \frac{1}{M} \Delta x \quad (6)$$

$$\Delta u_{ec} = \frac{T}{M} \Delta \dot{x} \quad (7)$$

$$\Delta u = (1 - \zeta) \cdot \Delta u_e + \zeta \cdot \Delta u_{ec} \\ = (1 - \zeta) \cdot \frac{1}{M} \Delta x + \zeta \cdot \frac{T}{M} \Delta \dot{x}, \quad 0 \leq \zeta \leq 1 \quad (8)$$

where M is a constant; T is the sampling period and ζ indicates a weighting factor for Δu_e and Δu_{ec} . Furthermore, in order to make $x(nT)$ approaching x_d progressively, the modified output $x'(nT)$ has to be located between the output $x(nT)$ and the desired output x_d , as shown in Fig. 3 (c),

$$x'(nT) = (1 - \gamma) \cdot x(nT) + \gamma \cdot x_d, \quad 0 \leq \gamma \leq 1 \quad (9)$$

where γ indicates a weighting factor for $x(nT)$ and x_d . Similarly,

$$\dot{x}'(nT) = (1 - \gamma) \cdot \dot{x}(nT) + \gamma \cdot \dot{x}_d, \quad 0 \leq \gamma \leq 1 \quad (10)$$

Therefore, $\Delta x(nT)$ and $\Delta \dot{x}(nT)$ can be derived from Eqs. (5), (9) and (10)

$$\begin{aligned}
\Delta x(nT) &= x'(nT) - x(nT) \\
&= (1-\gamma) \cdot x(nT) + \gamma \cdot x_d - x(nT) \\
&= \gamma(x_d - x(nT)) \\
&= \gamma \cdot e(nT)
\end{aligned} \quad (11)$$

$$\begin{aligned}
\Delta xc(nT) &= xc'(nT) - xc(nT) \\
&= (1-\gamma) \cdot xc(nT) + \gamma \cdot xc_d - xc(nT) \\
&= \gamma(xc_d - xc(nT)) \\
&= \gamma \cdot ec(nT)
\end{aligned} \quad (12)$$

Therefore, in accordance with Eqs.(8), (11) and (12), the modified input value Δu becomes

$$\Delta u = \frac{\gamma}{M} [(1-\zeta) \cdot e(nT) + \zeta \cdot T \cdot ec(nT)] \quad (13)$$

In this investigation, the SOFSMC combines fuzzy sliding mode control with self-organizing modifier. That is, the self-organizing modifier depends on the fuzzy sliding surface σ . Thus, a relation function between the weighting factor ζ of the self-organizing modifier and the fuzzy sliding surface σ is introduced

$$\zeta = \frac{1}{(1+\alpha \cdot T)}, \quad 0 \leq \zeta \leq 1 \quad (14)$$

Equation (14) describes that the weighting factor ζ can be decided by the fuzzy sliding surface σ , and ζ can keep $0 \leq \zeta \leq 1$ if the slope of sliding surface satisfy $\alpha \geq 0$. In contrast, Eq.(14) is also helpful for determining a suitable α in the controller design of SOFSMC. Therefore, the modifying value Δu of the control input can be derived from Eqs.(13) and (14) as

$$\begin{aligned}
\Delta u &= \frac{\gamma}{M} [(1-\zeta) \cdot e(nT) + \zeta \cdot T \cdot ec(nT)] \\
&= \frac{\gamma}{M} \left[\frac{\alpha \cdot T}{1+\alpha \cdot T} \cdot e(nT) + \frac{1}{1+\alpha \cdot T} \cdot T \cdot ec(nT) \right] \\
&= \frac{\gamma \cdot T}{M(1+\alpha \cdot T)} [\alpha \cdot e(nT) + ec(nT)] \\
&= \frac{\gamma_s}{M_s} \cdot \sigma(nT)
\end{aligned} \quad (15)$$

where $\gamma_s = \gamma \cdot T$ indicates the learning rate of the modifier and $M_s = M \cdot (1+\alpha \cdot T)$ is a constant ratio between Δu and $\sigma(nT)$. Furthermore, the i th rule can be modified by the modifying value Δu through an exciting intensity w_i

$$\begin{aligned}
u_i(nT+T) &= u_i(nT) + w_i \cdot \Delta u \\
&= u_i(nT) + w_i \frac{\gamma_s}{M_s} \sigma(nT), \quad i=1, \dots, 7
\end{aligned} \quad (16)$$

where the exciting intensity w_i is the grade of membership, as shown in Fig.3(b). According to the normalized fuzzy sliding surface $\tilde{\sigma}$, two rules, the i th and $(i+1)$ th rules, are excited and modified at the same time. Therefore, the fuzzy rules of the SOFSMC can be described as

$$\text{RULES}_i(nT+T) = \text{RULES}_i(nT) + w_i \frac{\gamma_s}{M_s} \sigma(nT) \quad (17)$$

4. Controller Design

In this study, the parallel control of the energy-saving control in the ELS and the velocity control in the hydraulic valve-controlled cylinder system is implemented, so the overall control system contains two hydraulic servo control systems and becomes a complex MIMO system. Figure 4 shows the block diagram of the overall system integrated by velocity control and load-sensing control. Herein the two control systems are regarded as two single-input single-output (SISO) systems to simplify the controller design problem. In this parallel control system, the supply pressure controlled by the ELS influences the dynamic response of the hydraulic cylinder's velocity control system. In contrast, the servo valve's switching of the velocity control system affects the stability of the supply pressure control in the ELS. These interactions and coupling relations are considered as disturbance. The SOFSMC is used here to design the controllers of the two control systems, namely energy-saving controller and velocity controller, which control parameters are shown in Table 2 respectively.

5. Experiment

The parallel control of the energy-saving control on the ELS and the velocity control on the hydraulic valve-controlled cylinder system is implemented using

Table 2 Control parameters of SOFSMC for energy-saving controller and velocity controller

Energy-Saving Controller	Velocity Controller
$\sigma_{ES} = 3e_p + \dot{e}_p$	$\sigma_v = 4e_v + \dot{e}_v$
$G\sigma_{ES} = 0.035$	$G\sigma_v = 0.2$
$G u_{ES} = 0.001$	$G u_v = 0.01$
$\gamma_{ES} / M_{ES} = 0.0005$	$\gamma_v / M_v = 0.001$

Fig. 4 Block diagram of the overall system integrated by velocity control and load-sensing control.

SOFSMC experimentally. Two concepts of the energy-saving control, namely load-sensing control and constant supply pressure control, are realized here by means of a variable displacement pump with a swash plate control unit. Moreover, in order to compare the energy-saving efficiency, the velocity control is implemented under three different energy-saving control conditions, such as (a) load-sensing control, (b) constant supply pressure control and (c) conventional hydraulic system without energy-saving control.

5.1 Parallel control of velocity control and load-sensing control

The block diagram of the overall control systems of the parallel control of the velocity control and load-sensing control is shown in Fig. 4. In the load-sensing control system, the load-sensing pressure p_{LS} of the hydraulic cylinder and the supply pressure p_s of the load-sensing pump are fed back simultaneously so as to keep their difference constant, $\Delta p = p_s - p_{LS} = \text{constant}$, by means of the load-sensing controller. The load-sensing pump controls the supply pressure via the swash plate control unit. In the velocity control system, the velocity of the load-sensing cylinder is fed back and controlled by the velocity controller. Both of the two controller are designed by the SOFSMC respectively, which control parameters are shown in Table 2, to achieve both high velocity control accuracy and lower energy consumption at the same time. For the purpose of comparing the control performance, the performance index of integral of absolute error (IAE) is defined

$$\text{IAE} = \int |e| dt \quad (18)$$

Figure 5 shows experimental results of the parallel control of the velocity control with set velocity of $V_{set} = 50 \text{ mm/s}$ and the load-sensing control with a set pressure difference of $\Delta p_{set} = 3 \text{ MPa}$ using FSMC and

Fig. 5 Comparison of the control performance of FSMC and SOFSMC in the experiment results of parallel control of velocity control and load-sensing control, $\Delta p_{set} = 3 \text{ MPa}$, $V_{set} = 50 \text{ mm/sec}$

SOFSMC. The effect of the self-organizing modifier of the SOFSMC is verified here through the experimental comparison. From the 1st run to the 3rd run of the experiment, the fuzzy rules of the velocity controller and the load-sensing controller that are listed respectively in Table 3 can be modified to diminish the control error e and the IAE performance index gradually.

Moreover, in order to ensure the robustness of the controllers, a step disturbance command from 4 sec to

Table 3 Modification of the SOFSMC's rule bases for velocity control and load-sensing control

	Velocity control						
	NB	NM	NS	ZR	PS	PM	PB
Initial	-1.00	-0.66	-0.33	0.00	0.33	0.66	1.00
After 1 st run	-1.00	-0.67	-0.34	0.00	0.34	0.69	1.03
After 3 rd run	-1.03	-0.72	-0.36	-0.01	0.36	0.81	1.14
	Load-sensing control						
	NB	NM	NS	ZR	PS	PM	PB
Initial	-1.50	-1.00	-0.50	0.00	0.50	1.00	1.50
After 1 st run	-1.51	-1.24	-0.65	-0.11	0.65	1.30	1.59
After 3 rd run	-1.98	-1.96	-1.01	-0.35	1.13	1.95	1.96

Fig. 6 Experiment results of parallel control, with external disturbance. (a) velocity control, $V_{set} = 50 \text{ mm/s}$, (b) load-sensing control, $\Delta p_{set} = 2 \text{ MPa}$ and $p_{s \max} = 16 \text{ MPa}$, (c) disturbance command from 4 sec to 6 sec and force measured by load cell

6 sec is given to the relief valve DRV2 of the disturbance system (see Fig. 1) for setting the maximum value of the chamber pressure p_{DRV2} and generating a disturbance force from the disturbance cylinder, $F_d = p_{DRV2} \cdot A_d$, as shown in Fig. 6(c). Thus, a time delay between the disturbance command and the disturbance force measured by the load cell is caused and the disturbance force is not constant until p_{DRV2} reaches the set maximum chamber pressure. Therefore, the load-sensing pressure p_{LS} of the load-sensing cylinder is changed and the supply pressure p_s is adjusted instantly through the load-sensing controller to follow the load-sensing pressure to maintain $\Delta p = p_s - p_{LS} = 2$ MPa. The velocity control is also influenced by this disturbance but still can keep the control accuracy.

5.2 Energy-saving comparison

Besides the load-sensing control system in this work, the velocity control is also implemented here under the constant supply pressure control system and the conventional hydraulic system without energy-saving control for comparing the energy-saving ability.

In the constant supply pressure control system, as shown in Fig. 7, the supply pressure is fed back directly to maintain constant supply pressure, $p_s = \text{constant}$, via the variable displacement pump, the swash plate control unit and the constant supply pressure controller, which is designed by the SOFSMC and has the same control parameters as that of the load-sensing controller in Table 2.

In the conventional hydraulic valve-controlled system, a relief valve and a positive displacement pump set the maximum supply pressure without closed-loop supply pressure control.

Figure 8 shows the experimental comparison of the parallel control for the velocity control with set velocity of $V_{set} = 50$ mm/s under three different

Fig. 7 Block diagram of the overall system to integrate velocity control and constant supply pressure control

energy-saving control systems, namely

(a) the load-sensing control with set pressure difference of $\Delta p_{set} = 2$ MPa and maximum supply pressure $p_{s\max} = 16$ MPa,

(b) the constant supply pressure control with set supply pressure of $p_{set} = 16$ MPa and

(c) the conventional hydraulic system with maximum supply pressure $p_{s\max} = 16$ MPa set by the relief valve and the positive displacement pump which displacement is the same as the maximum displacement in (a).

The velocity control accuracy in the three energy-saving control systems is almost the same, but the supply pressure responses are quite different due to the different energy-saving control concepts. The supply pressure of the load-sensing control is varied with the load-sensing pressure. In the constant supply pressure control, the supply pressure is controlled to keep constant. The supply pressure in the conventional hydraulic system is without closed-loop control.

In order to compare the pump's input power of the velocity control under the three supply pressure control systems, the input power P of the pump can be calculated by the following equation⁽¹⁾:

$$P = \frac{1}{\eta_{vol}} (p_s \cdot Q) = \frac{1}{\eta_{vol}} \left(p_s \cdot n \cdot q_{max} \cdot \frac{x_p}{x_{p,max}} \right) \quad (15)$$

p_s : supply pressure

Q : volume flow of pump

q_{max} : maximum displacement of pump

n : rotational speed of pump

$x_{p,max}$: maximum swash plate position of pump

x_p : swash plate position of pump

Fig. 8 Comparison of parallel control of velocity control with $V_{set} = 50$ mm/s under three different conditions: (a) load-sensing control $\Delta p_{set} = 2$ MPa, (b) constant supply pressure control $p_{set} = 16$ MPa and (c) conventional hydraulic system without energy-saving control $p_{s\max} = 16$ MPa

Fig. 9 Comparison of pump's input power for velocity control $V_{set}=50$ mm/s under three different conditions: (a) load-sensing control $\Delta p_{set}=2$ MPa, (b) constant supply pressure control $p_{s set}=16$ MPa and (c) conventional hydraulic system without energy-saving control $p_{s max}=16$ MPa

η_{vol} : volumetric efficiency of pump

The supply pressure p_s and the swash plate position x_p can be measured directly by pressure sensors and a LVDT (linear variable differential transformer) position sensor. The constant rotational speed is $n=1760$ rpm. The volumetric efficiency of the pump is $\eta_{vol}=98\%$ at 1760 rpm according to the test data of the pump's manufacturer.

Figure 9 shows the pump's input power of the velocity control under three different energy-saving control systems as in Fig. 8. The load-sensing control system consumes the lowest power, and the power in the constant supply pressure control system is also less than that of the conventional hydraulic system. The effects of the energy-saving control are confirmed here experimentally. Furthermore, the input energy of this case can be calculated by the integral of the input power curves in Fig. 9 as 5.23×10^4 J for the load-sensing control, 1.18×10^5 J for the constant supply pressure control, and 1.35×10^5 J for the conventional hydraulic system. In other words, the pump's input energy of the velocity control with the constant supply pressure control is 87.4% of that with the conventional hydraulic system, and the load-sensing control system have even only 38.8% of the input energy of the conventional system to achieve the same velocity control accuracy.

From the above experimental comparison and input energy analysis, the feasibility of the parallel control of the velocity control and the energy-saving control, including the load-sensing control and the constant supply pressure control, is proved.

6. Conclusions

The controllers with the self-organizing fuzzy sliding mode control concept for the complex TITO control systems, including the velocity control and the energy-saving control, are developed to simplify the fuzzy rule base by selecting the fuzzy sliding surface and to self-organize on-line the fuzzy rules via the self-organizing modifier. It is shown that not only the velocity control but also the energy-saving control, inclusive of the load-sensing control and the constant supply pressure control, perform well by the self-organizing fuzzy sliding mode controllers.

In this study, the parallel control of the velocity control in the hydraulic valve-controlled cylinder system and the energy-saving control, including load-sensing control and constant supply pressure control, in the electro-hydraulic load-sensing system is realized. In order to compare the energy-saving effects, the velocity control is implemented under three different energy-saving control systems, such as the load-sensing control, the constant supply pressure control and the conventional hydraulic valve-controlled system without closed-loop supply pressure control. The experiment results show that the load-sensing control system and the constant supply pressure control system can perform the same velocity control accuracy as the conventional hydraulic valve-controlled system and save the pump's input energy at the same time. Under the experimental conditions of this study, the load-sensing control can save about 61.2% of input energy and the constant supply pressure control system can also save about 12.6% of input energy in comparison with the conventional hydraulic system. The implementation of the parallel control of the velocity control in the hydraulic valve-controlled cylinder system and the energy-saving control in the electro-hydraulic load-sensing system is verified experimentally.

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