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Research Article

Valve-Pump Parallel Variable Mode Control for Complex Speed Regulation Processes

Haigang Ding, Henan Song, Jiyun Zhao, and Chaowen Lin

School of Mechatronic Engineering, China University of Mining and Technology, Xuzhou, Jiangsu 221106, China

Correspondence should be addressed to Haigang Ding; haierdhg@126.com

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To improve comprehensive performances of hydraulic systems with complex speed regulation processes, this paper proposes a new control scheme, valve-pump parallel variable mode control, which can change control modes according to control requirements and adjust the proportion of valve control and pump control in the speed regulation process. In this paper, we design a valve-pump parallel variable mode control system, explain its working principle, establish its mathematical model, analyse the influences of valve control on the system parameters, and at last, build an experimental system to carry out an experimental research. The experimental results show that during the speed regulation process, control modes could vary with control requirements, the switch between different control modes is smooth and meets expectations, and the proposed control approach can achieve excellent comprehensive performances for complex speed regulation, such as low-speed stability, fast response to load disturbance, and high efficiency. The valve-pump parallel variable mode control makes hydraulic control systems more flexible and suitable and enrich the current control schemes of hydraulic speed regulation systems.

1. Introduction

Hydraulic control has the characteristics of large power to mass ratio, high speed stiffness, and easy regulation and is widely used in aerospace, engineering machinery, and machine tools [1, 2]. The hydraulic hoists are the main equipment to lift materials, equipment, and personnel in high gas mines. Hydraulic hoists adopts hydraulic transmission with complex speed regulation process, since the process includes five stages of start, acceleration (acc.), constant speed, deceleration (dec.), and stop, and requires different control performances at different operating phase [3, 4]. The traditional hydraulic control has two basic forms: valve control and pump control. The dynamic response of the valve control is fast, but the efficiency is low [5, 6]; the efficiency of the pump control is high, but the dynamic response is slow [7–9]. Therefore, the independent pump control or valve control cannot solve the conflict between efficiency and response, and is not suitable to the complex speed regulation.

Recently, the valve-pump combined control has become the development direction of hydraulic control, which integrates the advantages of valve control and pump control, and could strike a balance between energy efficiency and dynamic response. The valve-pump combined control is a complex electrohydraulic control with nonlinearity [10–12] and includes two types, series control and parallel control. Manasek [13] proposed a valve-pump series control to fasten the pump control response. Shen et al. and Jin et al. [14, 15] added energy-regulating device to above series control to improve the response at acceleration stage. Xu et al. [16] proposed a pump/valve coordinate control of the independent metering system for mobile machinery, which is also valve-pump series control and could work at work in multi-modes. However, valve-pump series control is not suitable for applications with high power, because the control valve connected in series on the main circuit limits the maximum system flow.

The valve-pump parallel control could avoid this disadvantage, for the control valve is mounted in parallel on the main circuit. Qi et al. [17], Cochoy et al. [18], and Rongjie et al. [19] applied the parallel control to electrohydraulic actuators (EHA) to improve the flight's dynamic characteristics, which are position servo systems. At present, the valve-pump parallel control is mainly used in the position control system with a single control model and has not yet been used to speed regulation systems. In order to achieve excellent control performances for speed regulation systems, such as low-speed stability, fast response, and high efficiency, the valve-pump parallel control should change its control model during the adjustment process, but it is a challenging work, and nobody has studied it so far.

To improve comprehensive performances for complex speed regulation process, this paper puts forward a new kind of hydraulic control scheme: valve-pump parallel variable mode control, which can change control mode according to control requirements and can also adjust the proportion of valve control and pump control in the speed regulation process. In this paper, we design a valve-pump parallel variable mode control system, clarify its working principle, establish its mathematical model, and set up an experimental system to study control performances. The valve-pump parallel variable mode control could enrich the current hydraulic control method and make the hydraulic control system more flexible and adaptable. It has a wide application value in engineering with complex speed regulation process.

2. System Design and Working Principle

The valve-pump parallel variable mode control system mainly consists of a proportional variable pump (PVP), a proportional directional valve (PDV), a hydraulic motor, and a variable model controller, as shown in Figure 1. In the new system, the PDV is connected in parallel with the PVP outlet, the control valve and pump work together to regulate the rotating speed of hydraulic motor. Figure 2 shows the control principle of the proposed system; it is obvious that the control system is a multiple input and single output control system, and the variable model controller outputs two control signal, one to the PVP to regulate its displacement, another to the PDV to control its opening.

In order to show the regulation role of the valve control circuit and pump control circuit in the combined regulation [20], we propose a concept of valve-pump weight ratio: $k_{vp} = k_v : k_p$, where k_v is the weight of valve control circuit, and k_p is the weight of pump control circuit.

If $k_{vp} \geq 1$, it indicates that the function of valve control is greater than the pump control, so the system is mainly controlled by PDV.

If $k_{vp} < 1$, it means that the function of pump control is greater than the valve control, so the system is mainly controlled by PVP.

In the new system, the PDV can work in two states: leaking and replenishing, so the parallel control system could work at following three control models in a working cycle, as shown in Figure 3.

(1) Under leaking parallel valve control (LPVC) mode at the start and stop stages. Under LPVC mode, the PDV works at the leaking state, and the oil from the hydraulic motor high pressure cavity leaks into the tank through PDV, and the flow supply into the hydraulic motor is equal to the pump flow minus the valve flow, meanwhile, $k_{vp} \geq 1$, so the system is mainly controlled by PDV to enhance low-speed stability of hydraulic motor.

(2) Under replenishing parallel valve control (RPVC) mode at the constant speed stage. Under RPVC mode, the PDV works at the replenishing state, and the oil source supplies fluid to the hydraulic motor high pressure cavity through PDV, and the flow supply into the motor is equal to the pump flow plus the valve flow, meanwhile, $k_{vp} \geq 1$, so the system is mainly controlled by PDV to improve the dynamic response to load disturbance.

(3) Under parallel pump control (PPC) mode at the acceleration and deceleration stages. Under PPC mode, the PDV could work at both leaking status and replenishing status, meanwhile, $k_{vp} < 1$, so the system is mainly controlled by PVP to keep high efficiency. The PPC mode is regarded as the transition phase between LPVC and RPVC.

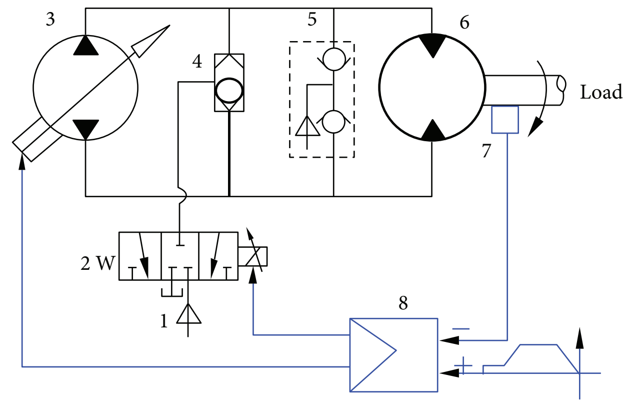


Figure 1: The schematic diagram of valve-pump parallel variable mode control. 1: oil source; 2: PDV; 3: PVP; 4: shuttle valve; 5: replenishing arrangement; 6: hydraulic motor; 7: speed encoder; 8: variable mode controller.

3. System Mathematical Modeling

The valve-pump parallel variable model control system is made up of pump control circuit, valve control circuit, and hydraulic motor circuit [21]. In the section above, three control links will be established, respectively, and combined into a control block diagram.

3.1. Pump Control Circuit. The variable mechanism of PVP is usually considered as a first-order inertia element [22], and its flow rate can be written as

$$q_p = q_{p0} - C_p P_h,$$
$$q_{p0} = \frac{K_u u_p}{(s/\omega_{bp}) + 1}, \tag{1}$$

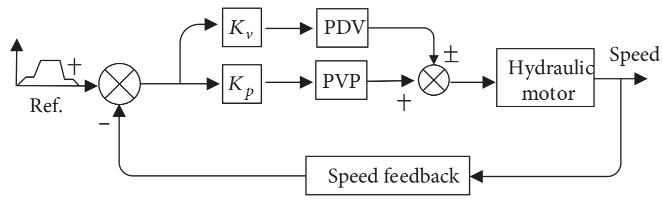


Figure 2: Control principle of valve-pump parallel variable mode control.

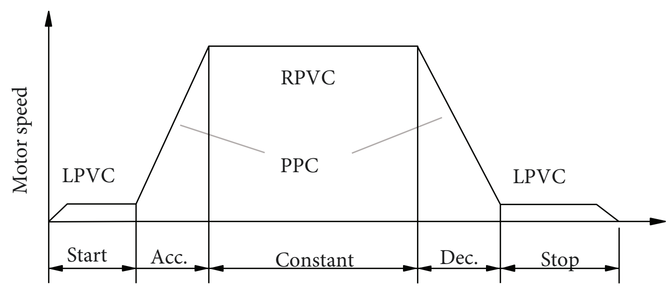


Figure 3: Speed regulation process in a working circle.

where q_{p} and q_{p0} are the pump flow rate under load and without load, respectively, C_{p} is the pump total leakage coefficient, P_h is the pump outlet pressure, K_{u} is the gain of input voltage-unload flow, ω_{bp} is the break frequency of the variable mechanism, and u_{p} is the pump control input.

3.2. Valve Control Link. The orifice flow equation of the PDV can be expressed as

$$q_v = C_{\mathrm{sv}} u_{\mathrm{v}} \sqrt{\Delta P}, \tag{2}$$

where q_v is the PDV flow rate, C_{sv} is the valve constant, and u_{v} is the control voltage, ΔP is the pressure drop across orifice, $\Delta P = P_h$ when PDV works at leaking status, $\Delta P = P_s - P_h$ when PDV works at replenishing status, in which P_s is the oil supplying pressure for PDV.

Using the Taylor series methods, the linearized orifice flow equation under LPVC mode is written as

$$\begin{aligned} q_{\mathrm{vl}} &= K_{\mathrm{ql}} u_{\mathrm{v}} + K_{\mathrm{cl}} P_h, \\ K_{\mathrm{ql}} &= \frac{\partial q_{\mathrm{vl}}}{\partial u_{\mathrm{v}}} = C_{\mathrm{sv}} \sqrt{P_h}, \\ K_{\mathrm{cl}} &= \frac{\partial q_{\mathrm{vl}}}{\partial P_h} = C_{\mathrm{sv}} \frac{u_{\mathrm{v}}}{2\sqrt{P_h}}, \end{aligned} \tag{3}$$

where q_v, K_{ql}, and K_{cl} are the flow rate, flow gain, and flow-pressure gain of PDV under LPVC mode, respectively.

Similarly, the linearized orifice flow equation under RPVC mode could be given by

$$\begin{aligned} q_{\mathrm{vr}} &= K_{\mathrm{qr}} u_{\mathrm{v}} - K_{\mathrm{cr}} P_h, \\ K_{\mathrm{qr}} &= \frac{\partial q_{\mathrm{vr}}}{\partial u_{\mathrm{v}}} = C_{\mathrm{sv}} \sqrt{P_s - P_h}, \\ K_{\mathrm{cr}} &= \frac{\partial q_{\mathrm{vr}}}{\partial P_h} = C_{\mathrm{sv}} \frac{u_{\mathrm{v}}}{2\sqrt{P_s - P_h}}, \end{aligned} \tag{4}$$

where q_{vr}, K_{qr}, and K_{cr} are the flow rate, flow gain, flow-pressure gain of PDV under RPVC mode, respectively.

The PDVC is usually considered as a second-order oscillating link [23, 24], and its unload flow rate is given by

$$q_{v0} = \frac{K_{\mathrm{qx}} u_v}{(s^2/\omega_v{}^2) + (2\xi_v/\omega_v)s + 1}, \tag{5}$$

where K_{qx} represents K_{qr} and K_{ql}, ω_v, and ξ_v are the hydraulic natural frequency and damping ratio of the PDV, respectively.

3.3. Hydraulic Motor Circuit. The continuity equation of the whole system is

$$q_p \pm q_v = C_{\mathrm{m}} P_h + D_{\mathrm{m}} \omega + V_0 \frac{s P_h}{\beta_e}, \tag{6}$$

where ω is the motor angular speed, C_{m} is a motor leakage coefficient, D_{m} is the motor displacement, and V_0 is the average volume of high-pressure chamber. In (6), "−" is used under LVPC model, and "+" is used under RVPC model.

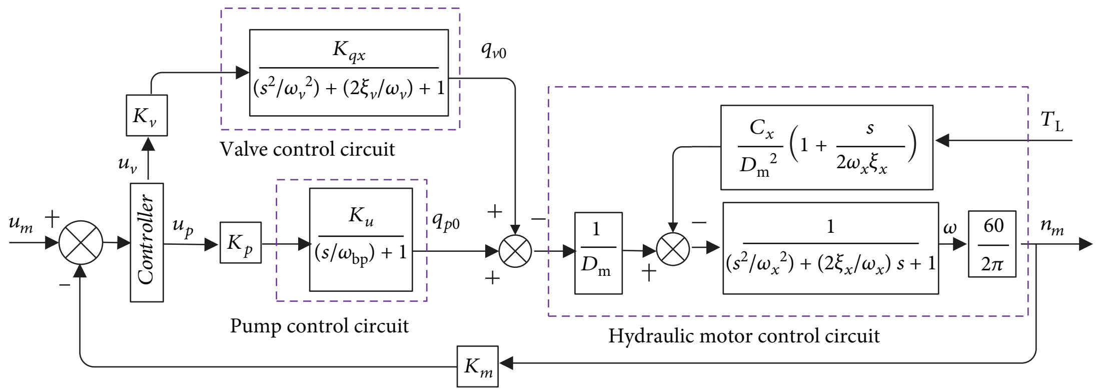


Figure 4: Block diagram of the valve-pump parallel variable mode control system.

The torque balance equation for the hydraulic motor is

$$D_m P_h = Js\omega + B_m\omega + T_L, \tag{7}$$

where J is the equivalent total inertia, B_m is the viscous damping coefficient, and T_L is the load torque produced by the loading pump.

Combining (1), (3), (4), (5), (6), and (7), the open loop dynamic equations of the parallel control system are given by

$$\omega = \frac{\left(\left(K_p q_{p0} \pm K_v q_{v0}\right)/D_m\right) - \left(C_x/D_m{}^2\right)\left(1 + (s/2\omega_x\xi_x)\right)T_L}{\left((s^2/\omega_x{}^2) + (2\xi_x/\omega_x)\right)s + 1}, \tag{8}$$

where the subscript x represents r and l, so $C_l = C_p + C_m + K_{cl}$, $\omega_l = \sqrt{\beta_e D_m{}^2/V_0 J}$, $\xi_l = (C_l/2D_m)\sqrt{\beta_e J/V_0}$, and C_l, ξ_l, and ω_l are the total leakage coefficient, damping ratio, and natural frequency under LPVC mode, respectively; $C_r = C_p + C_m + K_{cr}$, $\omega_r = \sqrt{\beta_e D_m{}^2/V_0 J}$, $\xi_r = (C_r/2D_m)\sqrt{\beta_e J/V_0}$, and C_r, ξ_r, and ω_r are the total leakage coefficient, damping ratio and natural frequency under RPVC mode, respectively.

When PDV closed, valve-pump parallel control switches to single pump control, and its open loop dynamic equation [25, 26] is

$$\omega = \frac{\left(\left(q_{p0}/D_m\right) - \left(C_t/D_m{}^2\right)\right)\left(1 + (s/2\omega_m\xi_m)\right)T_L}{(s^2/\omega_m{}^2) + \left((2\xi_m/\omega_m)s\right) + 1}, \tag{9}$$

where $C_t = C_p + C_m$, $\omega_m = \sqrt{\beta_e D_m{}^2/V_0 J}$, $\xi_m = (C_t/2D_m)\sqrt{\beta_e J/V_0}$, C_t, ξ_m, and ω_m are the total leakage coefficient, damping ratio, and hydraulic natural frequency under single pump control mode. Comparing (8) and (9), the single pump control may be considered a special case of valve-pump combined control.

The encoder used to measuring to hydraulic motor speed. It responds fast and could be simplified as a proportional element:

$$K_m = \frac{u_m}{n_m}, \tag{10}$$

where K_m is the feedback gain, u_m is the feedback voltage, and $n_m = 30\omega/\pi$ is the motor angular velocity.

3.4. Total System Mathematical Model. Combining the three links above, the block diagram of the parallel control system is obtained, as shown in Figure 4. The block diagram indicates that the valve-pump parallel system is a type-0 system, and the valve control circuit and pump control circuit are all unstable. To obtain good control performances, the integral correction is applied to control circuits. After correction, the open loop transfer functions of the two control circuits are, respectively, given by:

$$G_{vk} = \frac{K_{sx}}{s\left((s^2/\omega_v{}^2) + ((2\xi_v/\omega_v)s) + 1\right)\left((s^2/\omega_x{}^2) + ((2\xi_x/\omega_x)s) + 1\right)},$$

$$G_{pk} = \frac{K_{sp}}{s\left((s/\omega_{bp}) + 1\right)\left((s^2/\omega_x{}^2) + ((2\xi_x/\omega_x)s) + 1\right)}, \tag{11}$$

where $K_{sx} = 30K_m K_{vI} K_{qx}/(\pi D_m)$, and K_{sv} and K_{vI} are the open loop gain and integral gain of the corrected valve control circuit, respectively; $K_{sp} = 30K_m K_{pI} K_u D_p/(\pi D_m)$, and K_{sp} and K_{pI} are the open loop gain and integral gain of the corrected pump control circuit, respectively.

Therefore, the synthetical open loop gain under LPVC mode is

$$K_{sl} = K_v K_{sl} + K_p K_{sp}. \tag{12}$$

Similarly, the synthetical open loop gain under RPVC mode is

$$K_{sr} = K_v K_{sr} + K_p K_{sp}. \tag{13}$$

4. System Parameters Analysis

Comparing with the single pump control system, the system parameters in valve-pump parallel variable model control have the following characteristics:

(1) *The same hydraulic natural frequency*. Equations (8) and (9) indicate that the introduction of valve control

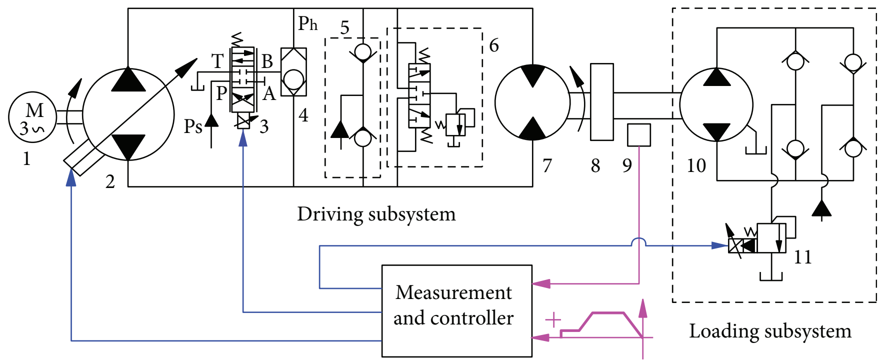


FIGURE 5: Schematic framework of the experimental system. 1: electric motor; 2: fixed displacement pump; 3: PDV; 5: shuttle valve; 6: replenishing arrangement; 7: hydraulic motor; 8: inertia; 9: encoder; 10: loading motor; 11: PRV.

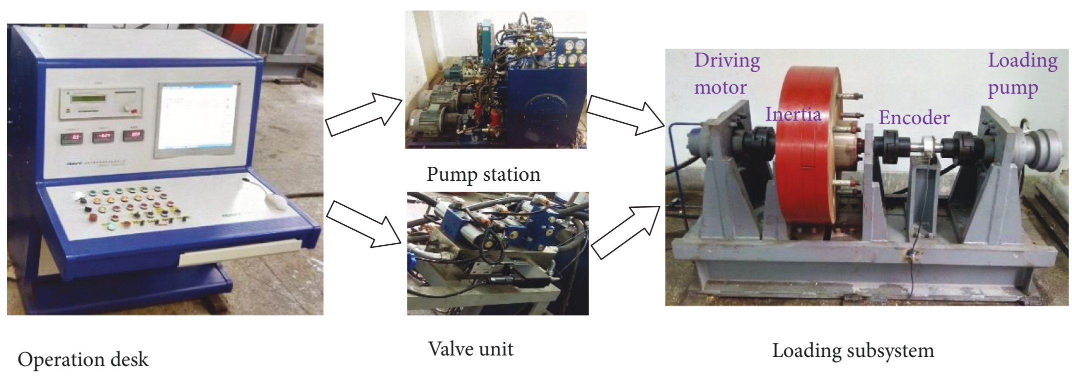


FIGURE 6: Experimental system in valve-pump parallel variable mode control.

does not change the system's hydraulic natural frequency, and the system both under LPVC mode and RPVC mode has the same hydraulic natural frequency as the single pump control system.

(2) *Larger and variable total leakage coefficients*. After the introducing valve control to a pump control system, the flow-pressure gain K_{cl} and K_{cr} are added into the total leakage coefficient, and there are $C_l = C_t + K_{\mathrm{cl}}$ under LPVC mode and $C_r = C_t + K_{\mathrm{cr}}$ under RPVC mode. Generally, C_t is small and stable, but K_{cl} and K_{cr} are greater than C_t and vary widely with operating points, such as pressure P_h and control input u_v, according to $K_{\mathrm{cl}} = C_{\mathrm{sv}} u_{\mathrm{v}}/2\sqrt{P_h}$ and $K_{\mathrm{cr}} = C_{\mathrm{sv}} u_{\mathrm{v}}/2\sqrt{P_s - P_h}$. Therefore, C_l and C_r are larger and vary widely with operating points.

(3) *Greater and variable damping ratios*. Since the damping ratio is proportional to the total leakage coefficient, damping ratios in the valve-pump parallel control system are much greater than that of single pump control systems, and there are $\xi_r \gg \xi_m$ and $\xi_l \gg \xi_m$. Moreover, both ξ_r and ξ_l also vary widely with operating points. In general, greater damping ratios will contribute to system stability, but great variation of damping ratios might cause difficulty of parameter prediction.

(4) *Faster dynamic response*. As we know, increasing the open loop gain will speed up system response. As long as the weighting factors K_v and K_p are set reasonably, the open loop gain of the valve-pump parallel control systems will be always greater than that of single pump control systems, that is, $K_{\mathrm{sl}} > K_{\mathrm{sp}}$ and $K_{\mathrm{sr}} > K_{\mathrm{sp}}$. Therefore, the valve-pump parallel control systems will respond faster than the single pump control system.

(5) *RPVC mode more flexible*. Increasing the flow gain will increase the open-loop gain and then fasten system response. Comparing the expressions of flow gain under RPVC mode and LPVC mode, $K_{\mathrm{ql}} = C_{\mathrm{sv}}\sqrt{P_h}$ and $K_{\mathrm{qr}} = C_{\mathrm{sv}}\sqrt{P_s - P_h}$, and K_{qr} can be further increased by increasing the supplying pressure P_s, and if $P_s > 2P_h$, there is $K_{\mathrm{qr}} > K_{\mathrm{ql}}$, and RPVC will respond faster than LPVC.

5. Experiment and Analysis

5.1. Experimental System. According to the schematic diagram in Figure 1, we build a valve-pump parallel variable mode control experimental system, as shown in Figures 5 and 6. The experimental system mainly consists of the driving subsystem, load subsystem, and measurement and

Table 1: System configuration.

Components	Specification
Electric motor	Power 11 kW, rated speed 1500 r/min
PVP	Axial piston pump, geometric displacement 40 mL/r, electrohydraulic control with proportional solenoid, 0–10 V input
PDV	4/3 version, direct operated, size 6, rated flow 9 L/min at $\Delta p = 10$ bar, frequency 60 Hz, damping ratio 0.7, 0–10 V input
PRV	Pilot operated, pressure rating 25 MPa, maximum flow 200 L/min, 0–10 V input
Hydraulic motor	Speed range 0–90 r/min, displacement 468 mL/r, rated pressure 40 MPa
Rotational inertia	48 kgm^2, modular design
Encoder	2000 P/r, TTL output

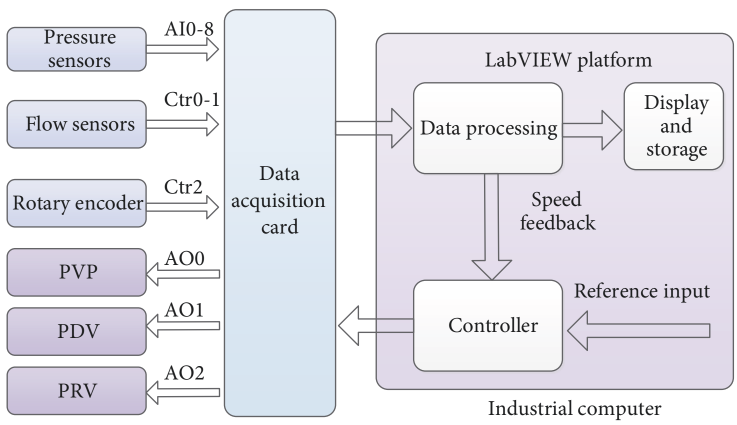


Figure 7: Measurement and control subsystem.

control subsystem. The driving subsystem is a closed hydraulic circuit, and the PVP and PDV work together under different control modes to drive a hydraulic motor. The load subsystem shown in Figure 6 mainly includes a loading motor and a proportional relief valve (PRV), the loading pump has the same displacement with the driving motor, and its loading pressure is regulated by the PRV. The main system configuration is listed in Table 1. The nominal pressure of the system is 25 MPa, total power 11 kW, system flow 50 L/min (including pump control flow 40 L/min and valve control flow 10 L/min), hydraulic motor displacement 468 mL/r, maximum speed 90 r/min, and rotational inertia 48 kg·m^2, replenishing pressure 0.8 MPa, flushing pressure 0.5 MPa.

The measurement and control subsystem is developed on the LabVIEW platform, as shown in Figure 7. Various types of sensors are used to measure the system pressures, flow of pump and valve, and hydraulic motor speed. These signals are captured by the acquisition card 6229 and inputted into the IPC for data processing, display, and storage. The motor speed is fed back to the variable mode controller and compared with the command. After the calculation, the controller output control signals to the PVP, PDV, and PRV via the acquisition card to realize the variable mode control. By using the variable mode controller, the proposed control system could work under LPVC mode, RPVC mode, and PPC mode.

5.2. Experiments and Analysis. From Section 3, we know that the pump control circuit and valve control circuit is unstable, so before the valve-pump parallel variable model control, the two control circuit should be compensated. The PI compensation is applied to the two control circuits, and its transfer function is given by

$$G_c = K_c\left(1 + \left(\frac{1}{T_{\mathrm{i}}s}\right)\right) = K_c + \frac{K_{\mathrm{I}}}{s}, \tag{14}$$

where K_c is the proportional gain, T_{i} is the internal time, K_{I} is the internal gain and $K_{\mathrm{I}} = K_c/T_{\mathrm{i}}$.

The optimal PI parameters are obtained through step response experiments. As shown in Table 2, there is always a relationship, $K_I \gg K_c$ in the two control circuits, so the PI compensation can be considered as an internal compensation, as discussed in Section 3.4.

Then, the valve-pump parallel variable mode control is applied to a trapezoidal speed regulation process, which is a complexed process and consists of the stages of start-up, acceleration, constant speed, deceleration, and stop. The LPVC mode is used to the star-up and stop stages with speed below 10 r/min, the PPC mode is used to the acceleration and deceleration stages, and the RPVC mode is used to the constant speed stage with speed of 60 r/min, and the setting of valve-pump weight ratio in each speed regulation stage is shown in Table 3. Figure 8 shows the dynamic response of valve-pump parallel variable mode control in a working cycle, where a 2 MPa step pressure is applied to PRV to produce step load disturbance at

Table 2: PI parameter setting of each control circuit.

Control circuit	K_c	T_i	K_I	Adaptation speed
Pump control	0.3	0.18	1.67	All
Valve control for oil leaking	0.6	0.12	5	0~15 r/min
Valve control for oil replenishing	0.7	0.12	5.8	55~70 r/min

Table 3: k_{vp} at different speed regulation stages.

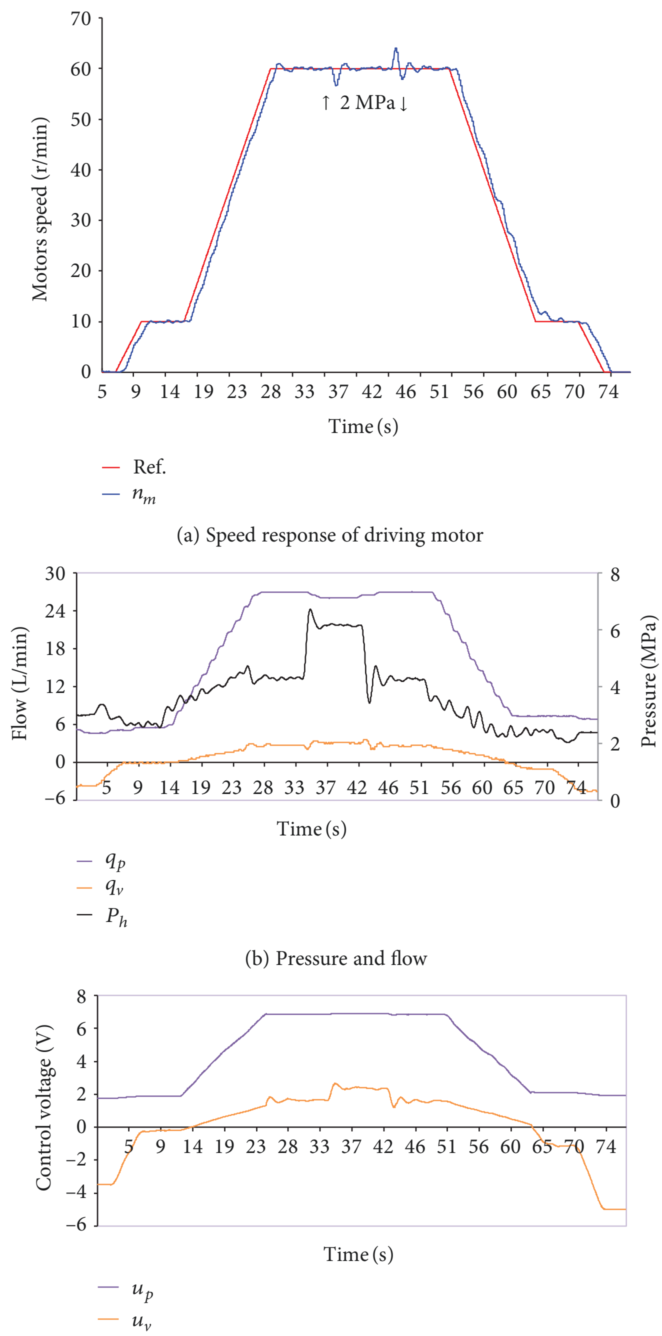
Speed regulation stage	k_v	k_p	k_{vp}	Control Model
Start-up and stop	1	0.1	10	LPVC
Acc. and dec.	0.1	1	0.1	PPC
Constant speed	1	0.2	5	RPVC

the uniform speed stage. Here we can obtain the following results from the experiment results.

(1) The hydraulic speed below 10 r/min keeps stable by LVPC, so that LPVC improves the low-speed stability at the start-up and stop stages; the motor speed quickly returns to 60 r/min by RPVC when subjected to load disturbance, so RPVC contributes to realize the fast response to load disturbance at the constant speed stage; the PPC achieves the fast tracking of speed at the acceleration and deceleration stages. Therefore, during the complex speed regulation process, the valve-pump parallel variable mode control obtains excellent comprehensive characteristics.

(2) During the speed adjustment process, the working state of control valve and control pump is in line with expectations, and the switch between different control modes is smooth and continuous. In particular, the valve control voltage changes from a negative value, about −3.8 V to a positive value, about 1.5 V, and then to a negative value, about −5 V, and the conversion between the oil leaking status and oil replenishing status is achieved.

(3) During the whole speed adjustment process, the variable pump provides most of the flow, about 90% of total system flow, while the control valve only works at a small flow state, about 10% of total system flow, so the proposed system could work efficiently as pump control systems [27].



(a) Speed response of driving motor

(b) Pressure and flow

(c) Controlling voltage of pump and valve

Figure 8: Dynamic response in a speed regulation cycle.

6. Conclusion

To improve the comprehensive performances for a complex speed regulation process, we propose a new hydraulic control scheme, that is, valve-pump parallel variable mode control, and its control mode could vary with control requirements during the process of speed regulation. At the start-up and stop stages, the LPVC mode is used to improve the low-speed stability; at the constant speed stage, the RPVC mode is applied to realize the fast response to load disturbance; at the acceleration and deceleration stages, the PPC mode is used to achieve the fast tracking of motor speed.

The valve-pump parallel variable mode control establishes a flexible control mechanism by using double channels of valve control and pump control. Therefore, the new control scheme will enrich the control form of the current hydraulic system, make the electro-hydraulic control system more flexible and adaptive to achieve excellent control performances, and has wide value in future application.

During the control process, the system parameters, such as leakage coefficient and damping ratio, change greatly with the operation points, which will increase the difficulty of system parameter prediction and control. To demonstrate the control characteristics of the proposed scheme itself, this work only uses the traditional PID control. In the future, advanced control strategies will be used to adapt to the parameters variation to further improve comprehensive performances.

Data Availability

The data used to support the findings of this study are included within the article.

Conflicts of Interest

The authors declare that there are no conflicts of interest regarding the publication of this article.

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