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A New Hydraulic Speed Regulation Scheme: Valve-Pump Parallel Variable Mode Control

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ABSTRACT To improve the comprehensive performances of hydraulic speed regulation systems, this paper proposes and develops a new control scheme, valve–pump parallel variable mode control, which can adopt different control modes in different speed regulation stages and can also adjust the weight ratio between pump control and valve control in the control process. In this paper, we design a hydraulic speed regulation in valve–pump parallel variable mode control, explain its principle, establish the system mathematical model, analyze the system parameters, and build a test system to verify regulation performances. The experimental results show that during the speed adjustment process, the switching between different control modes is smooth, the change rule of proportional valve and variable pump is in accordance with the expectation, and the ratio of valve to pump is reasonable, and the proposed scheme can improve the comprehensive performance of speed governing systems. The valve–pump parallel variable mode control could make full use of advantages of valve control and pump control, and will make hydraulic control systems more flexible and suitable and enrich the current control schemes of hydraulic speed regulation systems.

INDEX TERMS Hydraulic speed regulation systems, variable mode control, valve control, pump control, valve-pump weight ratio.

I. INTRODUCTION

The traditional hydraulic speed control system has two basic forms: valve control and pump control [1]–[3]. The dynamic response of the valve control system is fast, but the efficiency is low [4], [5]; the efficiency of the pump control system is high, but the dynamic response of the system is slow [6], [7]. The Valve-pump parallel control system combines the advantages of the valve control system and the pump control system, which is mainly used to electrohydraulic actuators (EHA) in flights to achieve excellent control performances [8], [9]. But at present, the control mode of valve-pump parallel control is too single, and can't change the control mode according to the speed regulation requirements, so it is hard to apply the applications with obvious speed regulation process. In view of the above problems, this paper puts forward a valve-pump parallel variable mode hydraulic speed control system, which can adopt different control modes in different speed regulating stages, and can also adjust the weight ratio of pump control and valve control in the process of speed regulation. In this paper, we design the valve-pump parallel variable mode hydraulic speed control system firstly and clarify its working principle. Then, we set up the experimental system and carry out an experimental study of the speed regulation period. The valve-pump parallel variable mode control can enrich the speed regulating mode of the current hydraulic system, making the hydraulic speed regulating system more flexible and adaptable. It also has a wide application value for engineering with complexed speed regulations, such as hydraulic hoists [10], [11], which usually go through five speed adjustment processes in a working cycle: low speed start, acceleration, uniform speed, slow down and low speed shutdown.

II. SYSTEM DESIGN

In this section, a hydraulic speed regulation system in valve-pump parallel variable mode control is designed (as shown in Figure 1) and its operating principle is explained as follows. In this system, a proportional variable pump (PVP) is parallel with a proportional directional valve (PDV) to regulate the motor speed together. The PDV can work in two states: oil drain and oil replenishing. When the PDV works at the state of oil replenishing, the valve control source adds the oil into

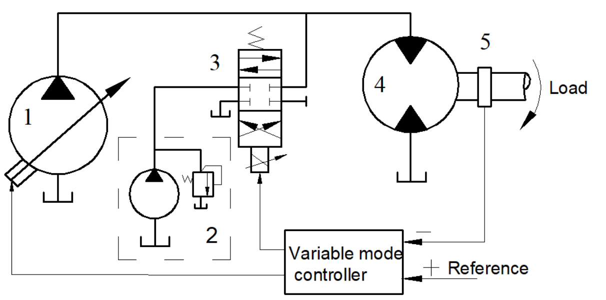


FIGURE 1. **The schematic diagram of hydraulic speed regulation systems in valve-pump parallel variable mode control. 1- PVP; 2- valve oil control source; 3- PDV; 4- hydraulic motor; 5- speed encoder.**

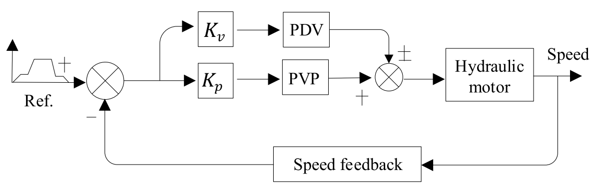


FIGURE 2. **Control principle of valve-pump parallel variable mode control.**

the high pressure cavity of the hydraulic motor through the PDV, and the flow that adds into the motor is equal to the sum of the pump control flow and the valve control flow. At this time the system is working at the oil replenishing valve-pump parallel control (RVPC) mode. When the PDV is working at the state of oil drain, the oil liquid from the motor high pressure cavity leaks into the tank through the PDV, and the flow that adds into the motor is equal to the difference between the control flow of the pump and the control flow of the valve. At this time the system is working at the oil leaking valve-pump parallel control (LVPC) mode.

The valve-pump variable model control system is a closed loop control system with multiple input and single output, includes two control loop, valve control loop and pump control loop as shown in Figure 2. In order to show the role of valve control and pump control in the combined speed regulation, we propose a new concept of valve-pump weight ratio: $k_{vp} = k_v : k_p$, where k_v is the weight of valve control links, and k_p is the weight of pump control links. k_v and k_p are important parameters for the valve-pump parallel variable mode control, and we can regulate the role of valve control or pump control in the speed regulation process by changing the ratio between k_v and k_p, that is k_{vp},. It should be noted that excessive k_v and k_p will affect the stability of the system.

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

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

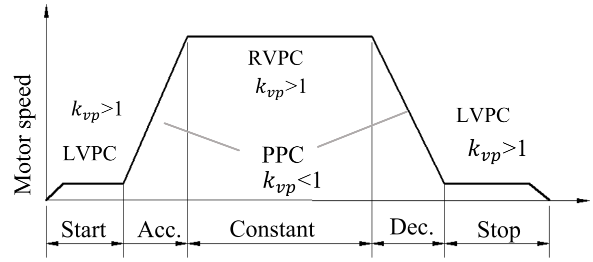


FIGURE 3. **Speed regulation process in a duty circle.**

A complex speed regulation process, including start, acceleration, constant speed, deceleration, and stop, is used to hydraulic hoists. Different control performances are required at different stages. The valve-pump parallel variable mode control is applied to the speed regulation process, and the control mode can vary with the control requirements at different stages. Fig. 3 shows the speed regulation process in a working cycle.

(1) At the start-up and stop stages, to improve low speed stability, the system is under the RVPC mode keeping $k_{vp} > 1$.

(2) At the acceleration (Acc.) and deceleration (Dec.) stages, to save energy, the system is mainly controlled by the variable pump, so $k_{vp} < 1$, and the system is under parallel pump control (PPC) mode, and. At this stage, the control valve can stay in the state of oil drain and oil replenishing.

(3)At constant speed stage, to achieve fast repose to load disturbance, the system is under RVPC mode keeping $k_{vp} > 1$.

III. SYSTEM MATHEMATICAL MODELING AND PARAMETER ANALYSIS

A. SYSTEM MATHEMATICAL MODELING

The mathematical model of valve-pump parallel control system consists of three loops: pump control loop, valve control loop and hydraulic motor loop [12]. We connect these three links and construct the control block diagram [13], as shown in Fig. 4.

The open loop dynamic equation under LVPC mode is:

$$\omega = \frac{\frac{K_p q_{p0} - K_v q_{v0}}{D_\mathrm{m}} - \frac{C_l}{D_m^2}\left(1 + \frac{s}{2\omega_l \xi_l}\right) T_L}{\frac{s^2}{\omega_l^2} + \frac{2\xi_l}{\omega_l} s + 1} \tag{1}$$

where ω is the motor angular speed, D_m is the motor displacement, q_{p0} and q_{v0} are the pump flow and valve flow without load, respectively, T_L is the load torque, C_l, ξ_l, and ω_l are the total leakage coefficient, damping ratio and natural frequency under LVPC mode, respectively, and $C_l = C_t + K_{cl}$, $\omega_l = \sqrt{\frac{\beta_e D_m^2}{V_0 J}}$, $\xi_l = \frac{C_l}{2D_m}\sqrt{\frac{\beta_e J}{V_0}}$, and C_t is the leakage coefficient of pump and motor, K_{cl} is the valve flow-pressure coefficient at leaking status.

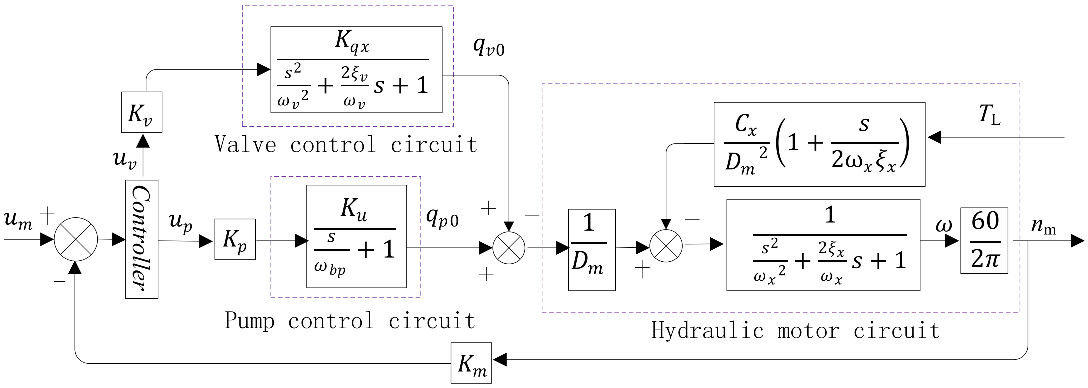


FIGURE 4. System control block diagram ("x" means "l" and "r").

The open loop dynamic equation under RVPC mode is:

$$\omega=\frac{\frac{K_pq_{p0}+K_vq_{v0}}{D_m}-\frac{C_r}{D_m^2}\left(1+\frac{s}{2\omega_r\xi_r}\right)T_L}{\frac{s^2}{\omega_r^2}+\frac{2\xi_r}{\omega_r}s+1} \tag{2}$$

where C_r, ξ_r, and ω_r are the total leakage coefficient, damping ratio and natural frequency under RVPC mode, respectively, and $C_r = C_t + K_{cr}$, $\omega_r = \sqrt{\frac{\beta_e D_m^2}{V_0 J}}$, $\xi_r = \frac{C_r}{2D_m}\sqrt{\frac{\beta_e J}{V_0}}$, K_{cr} is the valve flow-pressure coefficient at replenishing status.

When the control valve is closed, the valve-pump combined system will become a single pump control system, and its open loop dynamic equation is given by

$$\omega=\frac{\frac{q_{p0}}{D_m}-\frac{C_m}{D_m^2}\left(1+\frac{s}{2\omega_m\xi_m}\right)T_L}{\frac{s^2}{\omega_m^2}+\frac{2\xi_m}{\omega_m}s+1} \tag{3}$$

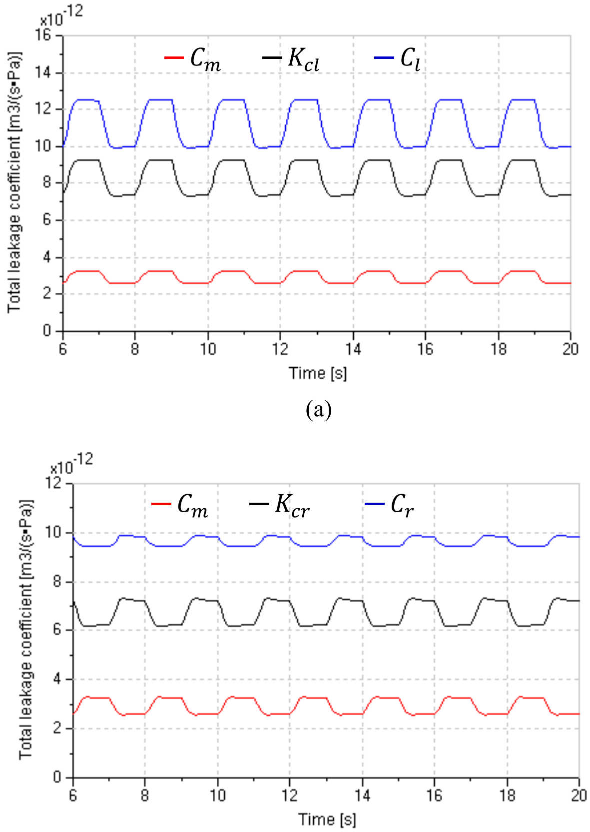
where C_m, ξ_m, and ω_m are the total leakage coefficient, damping ratio and natural frequency in single pump control, respectively, and $C_m = C_t$, $\omega_m = \sqrt{\frac{\beta_e D_m^2}{V_0 J}}$, $\xi_m = \frac{C_m}{2D_m}\sqrt{\frac{\beta_e J}{V_0}}$.

B. PARAMETER ANALYSIS

Comparing with the traditional pump control system, the parameters of the valve-pump parallel system have the characteristics as follow:

(1) The natural frequency of the system remains constant. Comparing the expressions of natural frequency in different control modes, there is $\omega_m = \omega_l = \omega_r$, which implies that the introduction of valve control does not change the hydraulic natural frequency, so the natural frequency is equal between the valve-pump parallel system and the single pump control system.

(2) The total leakage coefficient is increased and the change is significant. Fig.4 shows the simulation results of total leakage coefficient under different control modes, and simulation is carried out under the square wave load. It is obvious that there are $C_m \ll C_l$ and $C_m \ll C_r$. That is because $C_l = C_t + K_{cl}$ and $C_r = C_t + K_{cr}$, where C_m is small



(a)

(b)

FIGURE 5. Total leakage coefficient under different control modes. (a) under LVPC mode; (b) under RVPC mode.

and stable, but the valve flow-pressure coefficient K_{cr} and K_{cl} are much larger than C_m, and vary with the working point, such as valve input signal and system pressure [14], [15]. The Fig.4 also indicates that C_r is more stable than C_l under variable loads. That is because the change trend of C_l and C_m is the same, but the trend of C_r and C_m is opposite.

(3) The damping ratio increases and changes significantly with the working points. Fig. 6 shows the simulation results

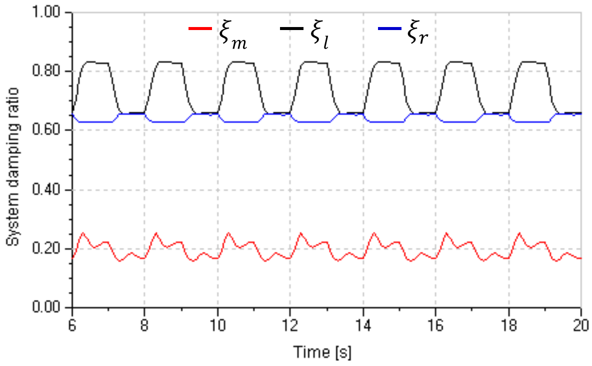


FIGURE 6. Damping ratios under different control modes.

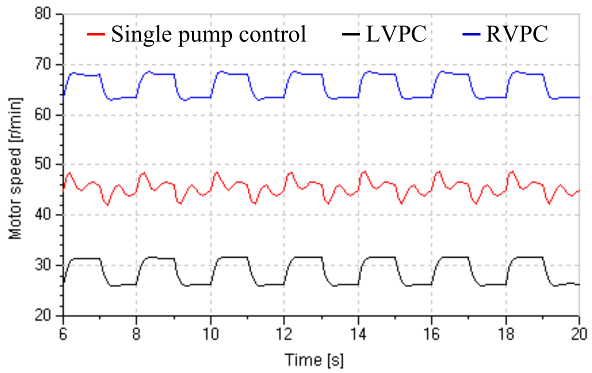


FIGURE 7. Hydraulic motor speed responses under different control modes.

ofdamping ratios under different control modes and variable load. It is obvious that $\xi_m \ll \xi_r < \xi_l$, and ξ_r is much more stable than ξ_l. That is because the damping ratio is proportional to the total leakage coefficient that changes with the working point significantly.

(4) The speed stiffness dents. Fig. 7 shows the simulation results of hydraulic motor speed responses under different control modes, which is also carried out under the square wave load. It is obvious that compared with single pump control, the both hydraulic motor speed in LVPC and RVPC is much more susceptible to load interference, so the LVPC and RVPC are characteristic with weak velocity stiffness. That is because the velocity stiffness is inversely proportional to the total leakage coefficient, and the total leakage coefficient of the valve-pump parallel system is much greater than that of the pump control system.

(5) Response faster. Assuming that K_{sv} is the open-loop gain of valve control loop, and K_{sp} is the open-loop gain of pump control loop, K_s is the integrated open-loop gain of the valve-pump combined system, there is

$$K_s = K_v K_{sv} + K_p K_{sp} \tag{4}$$

As long as the K_p and K_v are set up reasonably, there is always $K_s > K_{sv}$ and $K_s > K_{sp}$, so the valve-pump combined system respond faster than the single pump control and single valve control system.

TABLE 1. PI parameter setting of different control circuits.

Control circuit	K_c	T_i	K_I	Speed rang [r/min]
Pump control	0.3	0.18	1.67	All
Leaking valve control	0.6	0.12	5	0 ~ 15
Replenishing valve control	0.7	0.12	5.8	55 ~ 70

IV. EXPERIMENT AND ANALYSIS

A. TEST SYSTEM

A test system in valve-pump parallel variable mode control is established as shown in Fig. 8. The test system consists of a pump station, a valve control unit, a loading system and a measurement and control system. The loading system uses the hydraulic motor to load, and its loading pressure regulates by a proportional relief valve. The measurement and control system is developed by the NI LabView platform, integrating multiple functions, such as data acquisition, control, display, and could change control modes according to control requirements The main parameters of the test system are as follows: rated pressure 20MPa, rated flow rate 50L/min (including pump control flow 40L/min and valve control flow 10L/min), maximum motor speed 90r/min, and rotational inertia 48kg·m^2.

From Equations (1) and (2), we know valve-pump parallel variable model control system is unstable before compensation[16], so PI compensation is used to correct the pump control circuit and pump circuit, and the its transfer function is given by

$$G_c = K_c\left(1 + \frac{1}{T_i s}\right) = K_c + \frac{K_I}{s} \tag{5}$$

where K_c is the proportional gain, T_i is the internal time, K_I is the internal gain and $K_I = K_c/T_i$. The PI parameters for different control circuits are shown in Table 1.

B. VARIABLE MODE CONTROL

In this section, the experiments in variable mode control are carried out. A variable mode controller is developed, and its control mode could switch according to control requirements. In a working cycle, the LVPC mode is applied to the start-up and stop stages with low speed, the RVPC mode is applied to the constant speed stages with high speed, and the PPC mode is applied to acceleration and deceleration stages.

Table 2 shows the weight setting in each speed regulation stage, and Fig.9 presents the system dynamic response in valve-pump parallel variable mode control, where 2MPa step load is applied to the uniform speed stage.

Here we can obtain the following results from the experiments.

(1) Low speed stability can be enhanced by using LVPC control, and the rapid adjustment of load disturbance can be realize by RVPC in the constant speed stage.

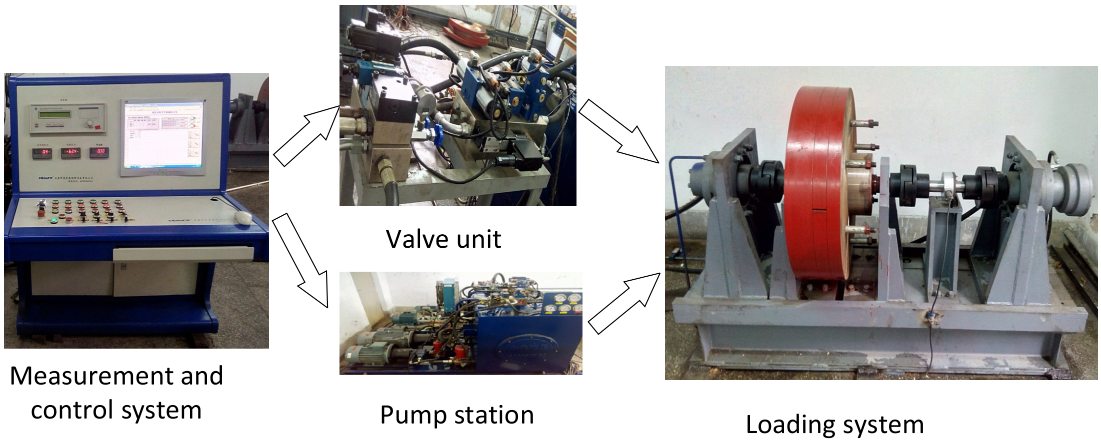


FIGURE 8. Experimental System.

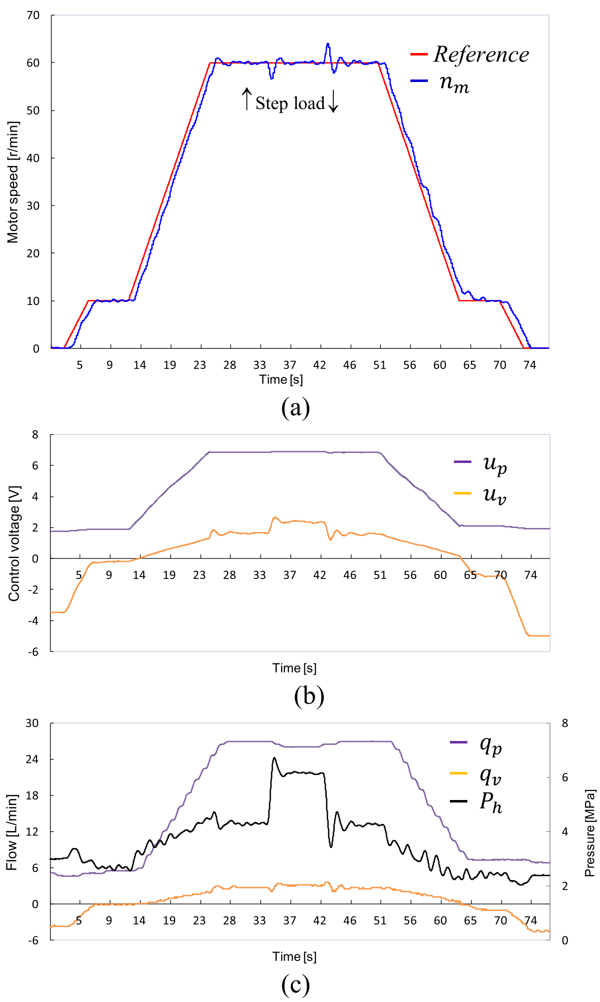


FIGURE 9. Dynamic response in a speed regulation cycle. (a) speed response of driving motor;(b) controlling voltage of pump and valve; (c) pressure and flow.

(2) During the whole speed regulation process, the speed tracking characteristics is good and the steady-state accuracy is well.

TABLE 2. Weight setting in different speed regulation stages.

Regulation stages	k_v	k_p	k_{vp}	Control model
Start-up and stop	1	0.1	10	LVPC
Acc. and dec.	0.1	1	0.1	PPC
Constant speed	1	0.2	5	RVPC

(3) The pump voltage is always positive and varies with speed. The valve voltage changes from negative voltage to positive voltage, and then to negative voltage, this means that the control valve works from leaking oil status to replenishing oil status, and then to leaking oil status. So the working state of the control valve and variable pump is in line with expectations, and the control mode switches smoothly.

(4) During the whole speed adjustment process, the variable pump provides most of the flow while the proportional directional valve is in a small flow state, so the system efficiency is relatively high.

V. SYSTEM EFFICIENCY ANALYSIS

In this section, we will discuss the energy efficiency in valve-pump parallel control. As we known, the hydraulic power is the product of flow and pressure. In the hydraulic system, the pressure depends on load without artificial intervention, and only the fluid flow could be controlled. Therefore, the energy control for a hydraulic system is the flow control essentially.

In the valve-pump parallel control system, by ignoring system leakage, the input flow of hydraulic motor q_m is

$$q_m = q_p \pm q_v \tag{6}$$

where q_p is the flow rate in pump control, and q_v is the flow rate in valve control, and there are $q_m = q_p + q_v$ under RVPC mode, and $q_m = q_p - q_v$under LVPC mode.

In the valve-pump parallel control system, energy loss is mainly caused by unavoidable throttling loss from the valve control. By ignoring mechanical efficiency and volumetric efficiency of pump and motor, the total system efficiency is described as follows

$$\begin{cases} \eta = \dfrac{N_m}{N_m + N_v} \\ N_m = q_m \times P_h \\ N_v = q_v \times \Delta P \\ \gamma = q_v / q_p \end{cases} \tag{7}$$

where η is the total system efficiency, γ is the flow ratio of pump to valve, there is $\gamma = 1/k_{vp}$, $N_m$is output power of hydraulic motor, N_v is the throttling loss from valve control, and ΔP is the pressure drop at valve orifice because of throttling, P_h is the system pressure varying with loads.

Combining Equations (6) and (7), the total system efficiency could be written as

$$\eta = \frac{(\gamma \pm 1)P_h}{\Delta P + (\gamma \pm 1)P_h} \tag{8}$$

where "+" is used to RVPC mode, and "−" is applied to LVPC mode. As shown in Fig. 9(c), in the combined system, the valve only works at small flow status and the pump provides majority of flow, so there is $\gamma \gg 1$, and Equation (8) can be simplified as

$$\eta = \frac{\gamma P_h}{\Delta P + \gamma P_h} \tag{9}$$

So far, a unified efficiency expression of valve-pump parallel variable mode control system is obtained as shown in Equation (9). It indicates that total system efficiency of is characterized with the flow ratio γ or the valve-pump weight ratio k_{vp}, and the greater γ or the smaller k_{vp}, the higher system efficiency.

VI. CONCLUSION

In this paper, we develop a new hydraulic control scheme called value-pump parallel variable mode control system to achieve excellent comprehensive performances for complex speed regulation system. This hydraulic control scheme can vary the control models with the control requirements to apply the different regulation stages. The LVPC mode is applied to the start-up and stop stages to improve the low speed stability; the RVPC is used to the constant speed stage to achieve the fast regulation to load disturbance; the PPC mode is applied to the acceleration and deceleration stages to save system energy.

By combining the valve control, the leakage coefficient, damping ratio of the valve-pump parallel variable mode hydraulic system changes with the change of the valve opening and the system pressure, which will contribute to system stability, but also increase the parameter prediction and control difficulty of the system. So in the future work, the advance control strategies should be considered to improve the characteristics of the valve-pump parallel system.

Valve-pump parallel variable mode control establishes a flexible control mechanism by using double channels of valve control and pump control and changing the weight ratio of valve and pump control. The proposed control scheme will enrich control methods of current hydraulic systems, and will contribute to improve the comprehensive performance of hydraulic speed control system with high power, such as low speed stability, fast adjustment, and high efficiency.

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CONFLICT OF INTEREST

The authors declare that there is no conflict of interest regarding the publication of this paper.

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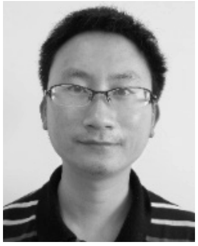

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