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The Influence of Valve-Pump Weight Ratios on the Dynamic Response of Leaking Valve-Pump Parallel Control Hydraulic Systems

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Abstract: A new leaking valve-pump parallel control (LVPC) oil hydraulic system is proposed to improve the performance of dynamic response of present variable speed pump control (VSPC) system, which is an oil hydraulic control system with saving energy. In the LVPC, a control valve is operating at leaking status, together with a variable speed pump, to regulate the system flow of hydraulic oil simultaneously. Therefore, the degree of valve control and pump control can be adjusted by regulating the valve-pump weight ratio. The LVPC system design, mathematical model development, system parameter and control performance analysis are carried out systematically followed by an experimental for validation process. Results have shown that after introducing the valve control, the total leakage coefficient increases significantly over a wide range with the operating point and this further increases damping ratios and reduces the velocity stiffness. As the valve-pump weight ratio determines the flow distribution between the valve and the pump and the weight factors of the valve and/or the pump controls determines the response speed of the LVPC system, thus if the weight factors are constrained properly, the LVPC system will eventually have a large synthetic open-loop gain and it will respond faster than the VSPC system. The LVPC will enrich the control schemes of oil hydraulic system and has potential value in application requiring of fast response.

Keywords: rapid response; leaking valve-pump parallel control; variable speed pump control; parallel valve control; valve-pump weight ratio

1. Introduction

Traditional hydraulic control systems are of two basic types: pump control and valve control [1]. Valve control systems respond quickly to valve and load inputs but are less efficient due to throttling and overflow losses [2,3]. Pump control systems work efficiently since both system pressure and flow are closely matched with load requirements [4,5] but have a critical drawback: slow response. The variable speed pump control (VSPC) system is a new class of pump control system, using a variable speed motor to drive a fixed displacement pump, which is applicable to hydraulic elevators [6], shield tunneling machine [7] and water distribution systems [8,9]. Compared with traditional pump control systems, the VSPC system has advantages of lower cost, higher reliability and higher energy-efficiency. However, response time is poor, due to the high inertia of the electric motor and the low overload capacity of the inverter driving it, preventing their use in applications requiring rapid response.

Two classes of combined valve-pump control may be identified: serial control and parallel control, balancing the respective advantages of each, with respect to dynamic response and energy efficiency [10]. To accelerate the response of VSPC systems, Manasek [11] proposes a valve-pump series control system, in which a flow control valve is connected in series to the outlet of the pump. Shen et al. propose a variable speed hydraulic control system based on an energy regulation strategy [12,13], in which an energy regulation device releases energy to improve dynamic response serial valve-pump control systems are not suitable for high power systems, due to the control valve installed in the main circuit limiting maximum flow of the system. However, valve-pump parallel control systems can avoid this disadvantage, as the parallel control valve in the main circuit does not limit the system flow. Recent research on parallel control systems mainly focus on electrohydraulic actuators in a hybrid primary flight control system [14,15]. The current parallel control system is of a replenishing type, requiring an additional oil supply device and current research does not explain the role of pump control and valve control in the combined system and its corresponding influences on dynamic response.

To improve the response of VSPC systems, this paper develops a leaking valve-pump parallel control (LVPC) system, in which a leaking control valve is placed in parallel to a variable speed pump to collectively regulate system flow. In the paper, the LVPC system is designed and its working principles explained. Mathematical models are constructed, an experimental system set up and the influence of different valve-pump weights on system response are discussed, demonstrating the role of pump and valve control in the combined system. Thus, the LVPC provides a new method for improving dynamic response of pump control systems and will enrich the control schemes of hydraulic systems. LVPC might have potential value in application requiring of fast response. Beyond the particular application on a hydraulic test rig for volumetric machines, the same method could be maybe applied on a turbomachinery test rig to regulate for example the head of the testing model, which are classically also equipped either with a variable speed pump.

2. Methods

2.1. Principle of Proposed System

The schematic diagram of the LVPC is shown in Figure 1, its main circuit is a variable speed pump control motor circuit and a proportional throttle valve (PTV) is connected in parallel to the main circuit through a shuttle valve and forms a bypass leakage path. In the parallel control system, the variable speed pump (VSP) supplies a basic flow, which is varied via the output frequency of the inverter and the individual replenishing device is connected to the return chamber to compensate for leakage and maintain a constant chamber pressure. The flushing device is used to exchange hot fluid in the return chamber and to cool the system.

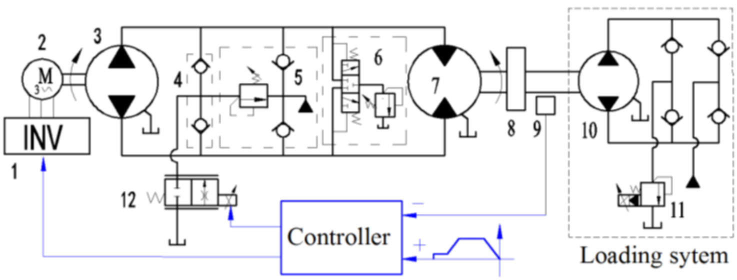


Figure 1. Schematic framework of LVPC (leaking valve-pump parallel control) system. 1—Inverter; 2—electric motor; 3—Fixed displacement pump; 4—Shuttle valve; 5—Oil replenishing device; 6—Flushing device; 7—Hydraulic motor; 8—Inertia; 9—Encoder; 10—Loading pump; 11—Proportional relief valve; 12—PTV (proportional throttle valve).

The control principle of the LVPC is shown in Figure 2. The hydraulic motor speed is controlled by the VSP and PTV, the flow into the hydraulic motor is equal to the output flow of the pump minus the bypass flow back to the tank though the control valve. In order to distinguish the role of valve control and pump control in the combined system, the valve-pump weight ratio K_{vp} is given as

$$K_{vp} = K_v : K_p \tag{1}$$

where K_v is the weight factor of the valve control link and K_p is the weight factor of the pump control link.

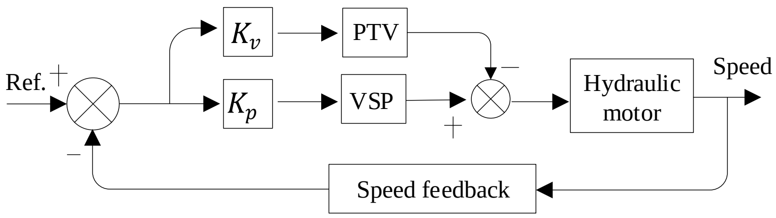


Figure 2. Control principle of leaking valve-pump parallel control.

The LVPC system is a multi-input and output control system, is composed of valve control and pump control and each control link is in closed-loop control. The encoder detects the actual speed of the motor and feeds back to the controller. The actual speed is compared with the reference speed to obtain the speed error. This error is assigned to the valve control channel and the pump control channel according to the weight ratio and under the common adjustment of valve control and pump control, the motor reaches a certain desired speed.

If $K_{vp} > 1$, the role of the valve control is greater than that of the pump control and the system is mainly controlled by the PTV. When $K_v : K_p$ = 1:0, the combined system becomes a conventional valve control system, totally controlled by the valve.

Conversely, if $K_{vp} < 1$, the role of the pump control is greater than that of the valve control and the system is mainly controlled by the VSP, a ratio of $K_v : K_p$ = 0:1 implying a conventional VSPC system.

2.2. Mathematical Formulations

2.2.1. Inverter-Electric Motor Link

The inverter-electric motor link may be regarded as a first-order inertia element

$$n_p = \frac{K_u u_p - K_h P_h}{\dfrac{s}{\omega_{bp}} + 1} \tag{2}$$

where n_p is the electric motor speed, u_p is the input voltage of the inverter, K_u is the coefficient of voltage, P_h is the high-pressure chamber pressure of the system, K_h is the coefficient of pressure and ω_{bp} is the break frequency, which is inversely proportional to the pump inertia.

Equation (2) indicates that n_p increases with increasing u_p and decreasing P_h and the actual flow of the pump q_p is given by

$$q_p = q_{p0} - C_p P_h \tag{3}$$

where $q_{p0} = D_p n_p$, is the unload flow of the pump and when the system runs under no load, $P_h = 0$, thus $q_{p0} = \frac{D_p K_u u_p}{\frac{s}{\omega_{bp}} + 1}$, in which D_p is the pump displacement and C_p is the total leakage coefficient of the pump.

2.2.2. Parallel Valve Control Circuit

The orifice flow equation of the PTV can be expressed as

$$q_v = C_{sv} u_v \sqrt{P_h} \tag{4}$$

where C_{sv} is the valve constant and u_v is the control voltage corresponding to spool displacement.

The orifice flow equation is nonlinear, using the Taylor series method for linearizing Equation (4) at the null operation point and the linearized flow equation is given by

$$\begin{cases} q_v = K_q u_v + K_c P_h \\ K_q = \dfrac{\partial q_v}{\partial u_v} = C_{sv}\sqrt{P_h} \\ K_c = \dfrac{\partial q_v}{\partial P_h} = C_{sv}\dfrac{u_v}{2\sqrt{P_h}} \end{cases} \tag{5}$$

where K_q is the flow gain, K_c is the pressure-pressure-flow coefficient. The coefficients K_q and K_c are known as the valve coefficients and are extremely important for determine the valve's stability, frequency response and other dynamic characteristics.

In generally, proportional valves are considered as a second-order oscillating link in the hydraulic control system and the control valve's unload flow is given by

$$q_{v0} = \frac{K_q u_v}{\dfrac{s^2}{{\omega_v}^2} + \dfrac{2\xi_v}{\omega_v}s + 1} \tag{6}$$

where q_{v0}, ω_v and ξ_v is its unload flow, hydraulic natural frequency and damping ratio and its unload flow is promotional to the control input u_v, which corresponds to the opening of the control valve.

2.2.3. Valve-Pump Parallel Control Motor Link

The continuity equation for the whole system is

$$q_p - q_v = C_{\mathrm{m}} P_h + D_{\mathrm{m}}\omega + V_0 s P_h/\beta_e \tag{7}$$

where ω is the motor angular speed, C_{m} is the motor leakage coefficient, D_{m} is the motor displacement and V_0 is the average volume of high-pressure chamber.

The torque balance equation for the hydraulic motor is

$$D_{\mathrm{m}} P_h = Js\omega + B_m\omega + T_L \tag{8}$$

where J is the equivalent total inertia, B_m is the viscous damping coefficient, $T_L = D_L P_h$ is the load torque produced by the loading pump (ignoring the low-pressure chamber pressure) and $D_L = D_m$ is the displacement of the loading pump.

Combining (3), (5), (6), (7) and (8), the open loop dynamic equation for the LVPC [16] is given by:

$$\omega = \frac{\frac{K_p q_{p0} - K_v q_{v0}}{D_{\mathrm{m}}} - \dfrac{C_l}{{D_m}^2}\left(1 + \dfrac{s}{2\omega_l \xi_l}\right)T_L}{\dfrac{s^2}{{\omega_l}^2} + \dfrac{2\xi_l}{\omega_l}s + 1} \tag{9}$$

where $C_l = C_p + C_m + K_c$, $\omega_l = \sqrt{\frac{\beta_e {D_m}^2}{V_0\ J}}$, $\xi_l = \frac{C_l}{2D_m}\sqrt{\frac{\beta_e\ J}{V_0}}$, C_l, ξ_l and ω_l are the total leakage coefficient, damping ratio and natural frequency of the LVPC system, respectively.

When the PTV is closed and $K_v = 0$, $K_p = 1$, such that $K_{vp} = 0$ *and* q_{v0} and K_c will be zero in (9), the system control mode will change from LVPC to VSPC and its open loop dynamic equation [17] is

$$\omega = \frac{\frac{q_{p0}}{D_m} - \frac{C_t}{D_m{}^2}\left(1 + \frac{s}{2\omega_m \xi_m}\right)T_L}{\frac{s^2}{\omega_m{}^2} + \frac{2\xi_m}{\omega_m}s + 1} \tag{10}$$

where $C_t = C_p + C_m$, $\omega_l = \sqrt{\frac{\beta_e D_m{}^2}{V_0\, J}}$, $\xi_m = \frac{C_t}{2D_m}\sqrt{\frac{\beta_e\, J}{V_0}}$, C_m, ξ_m and ω_m are the total leakage coefficient, damping ratio and hydraulic natural frequency of the VSPC system respectively. Equations (9) and (10) indicate that a VSPC may be considered a special case of an LVPC. Notice that the linearized Equations 9 and 10 are valid around a specific operating point, for the real behavior is strongly nonlinear. Moreover, in these equations, the natural frequency can be computed with fair confidence but the damping ratio especially ξ_l is much more variable and nebulous.

The rotary component is simplified as a proportional element due to its fast response:

$$K_m = \frac{u_m}{n_m} \tag{11}$$

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

2.2.4. Total System Mathematical Model

The block diagram of the parallel control system is obtained by combining the three links above, as shown in Figure 3. The block diagram indicates that the parallel valve-pump system is a type-0 system and the valve and pump control links are all unstable, requiring the two links to be corrected in order to obtain sound control performance. The simplest method, integral correction, is used here.

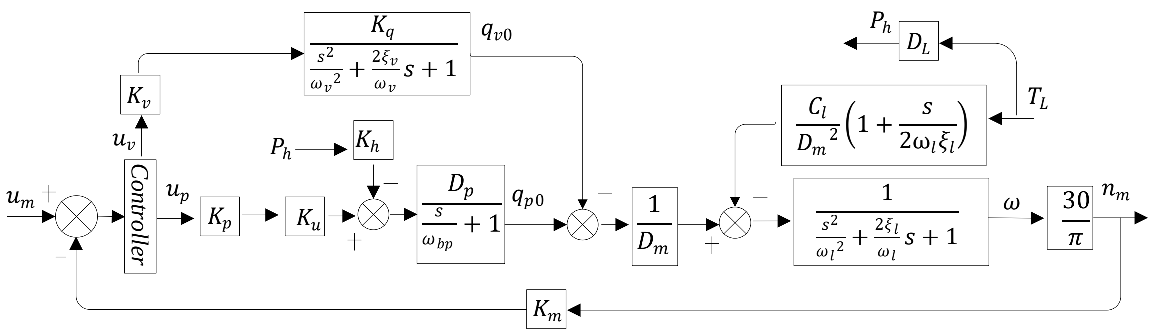


Figure 3. Block diagram of LVPC system.

After correction, the open loop transfer functions of the valve control link and pump control link are respectively given by:

$$G_{vk} = \frac{K_{sv}}{s(\frac{s^2}{\omega_v{}^2} + \frac{2\xi_v}{\omega_v}s + 1)(\frac{s^2}{\omega_l{}^2} + \frac{2\xi_l}{\omega_l}s + 1)} \tag{12}$$

$$G_{pk} = \frac{K_{sp}}{s(\frac{s}{\omega_{bp}} + 1)(\frac{s^2}{\omega_l{}^2} + \frac{2\xi_l}{\omega_l}s + 1)} \tag{13}$$

where K_{sv} *is* the open loop gain of the corrected valve control link, *and* $K_{sv} = 30\ K_m K_{vI} K_q/(\pi D_m)$, in which K_{vI} is the integral gain of the valve control link, K_{sp} *is* the open loop gain of the corrected pump control link, *and* $K_{sp} = 30\ K_m K_{pI} K_u D_p / (\pi D_m)$, in which K_{pI} is the integral gain of the pump control link.

The LVPC system is a multiple input single output (MISO) system. To facilitate the study, the system could be assumed a linear system that satisfies the superposition principle, that is the total output produced by multiple inputs is equal to the sum of the outputs produced by a single input. Specific to this system, LVPC can be viewed as a superposition of a pump controlled motor and a valve controlled motor, so the o integrated pen-loop gain of the combined system K_{sl} is equal to the sum of the valve-controlled open-loop gain K_{sv} and the pump-controlled open-loop gain K_{sp}.

$$K_{sl} = K_v K_{sv} + K_p K_{sp} \tag{14}$$

In the equation, K_v and K_p are used to adjust the proportion of pump control and valve control. The velocity stiffness of the LVPC system is given by

$$\left|\frac{T_L}{n_m}\right|_l = \frac{K_{sl} {D_m}^2}{C_l} \tag{15}$$

2.3. System Parameter Analysis

After adding a valve control to a VSPC system, the parameters of LVPC systems will have the following characteristics:

(1) Identical hydraulic natural frequency. Equations (9) and (10) indicate that the introduction of valve control does not change the system's hydraulic natural frequency and the LVPC system has the same hydraulic natural frequency as the VSPC system.

(2) Larger and variable total leakage coefficients. In general, total leakage coefficients of pump control systems C_t is small and stable. However, after introducing valve control, the total leakage coefficient of LVPC systems becomes larger and varies widely with system pressure P_h and control inputs u_v. This is due to the pressure-flow gain K_c being much greater compared to C_t and changing with P_h and u_v, according to $K_c = C_{sv} u_v / \left(2\sqrt{P_h}\right)$.

(3) Greater and variable damping ratios. Because the damping ratio is proportional to the total leakage coefficient, the damping ratio of LVPC systems is much greater than that of VSPC systems and varies widely with P_h and u_v. Greater damping ratios will benefit system stability but values will cause difficulty with parameter prediction and system control.

(4) Lower velocity stiffness. Equation (15) indicates that velocity stiffness decreases with the increase of total leakage coefficients and $C_t \ll C_l$. Thus, compared with VSPC systems, LVPC systems have lower velocity stiffness and will be more susceptible to load disturbance.

(5) Faster dynamic response. It is well known that increasing open loop gain will speed system response. As long as the weight factors K_v *and* K_p are appropriate, the open loop gain of LVPC systems will always be greater than that of VSPC systems (i.e. $K_{sl} > K_{sp}$) and therefore the LVPC system will respond faster than the VSPC system.

3. Case Studies

3.1. Experiment

An experimental LVPC system based on the principle of Figure 1 is set up as shown in Figure 4 and its main configuration as listed in Table 1. The rated pressure of the system is 25 MPa, flow 35 L/min, maximum speed of the hydraulic motor 90 r/min. The flushing device includes a hydraulically controlled directional control valve and a relief valve. A portion of the hot oil from the motor low pressure side flows out of the device to cool the system. The pressure of the relief valve is called the flushing pressure and is generally lower than the replenishing pressure by 0.2–0.3 MPa. Here it is set to 0.5 MPa. In Figure 4b, the left is a driving motor and the right is a loading pump, they have the same displacement and the loading pressure is regulated by a proportional relief valve. The rotary inertia block is installed between the two hydraulic motors, its total weight being 900 kg and its total inertia 72 kg·m^2. In order to study dynamic response under different inertias, the inertia block is designed to be modular, with one basic and four small blocks.

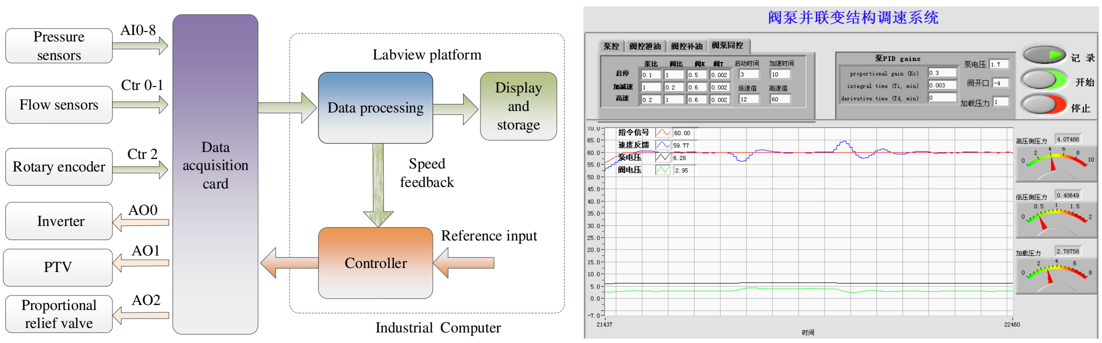
(**a**) (**b**)

Figure 4. Experimental system. (**a**) A view of the experimental system; (**b**) Loading system.

Table 1. System configuration.

Components	Specification
Electric motor	power:7.5 Kw; rated speed:3000 r/min; control voltage: 0–10 V
Inverter	power:11 kW with vector control; frequency range:0.1–100 Hz;
Fixed pump	displacement:12.6 mL/r; speed range:500–3000 r/min
Hydraulic motor	speed range: 0–90 r/min; displacement:468 mL/r; rated pressure: 40 MPa
PTV	rate flow: 9 L/min at 1.5 MPa per notch; frequency: 60 Hz; damping ratio: 0.7
Proportional relief valve	pressure rang: 0.7–31.5 MPa; rate flow:200 L/min
Rotary inertia	72 kg·m^2

A virtual control system is developed using the LabVIEW platform as shown in Figure 5. The system pressures, pump flow rates, valve flow and hydraulic motor's rotary speed are measured by corresponding sensors and are fed back to an industrial computer via a data acquisition card. After data processing, the control system outputs control signals to the inverter, PTV and PRV. The control system integrates the functions of data acquisition, control, display and data storage and can work under LVPC and VSPC modes.



(**a**) (**b**)

Figure 5. Measurement and control system. (**a**) Control principle; (**b**) Interface.

From Section 2.2, we know that the valve-pump parallel system is a type-0 system and is unstable before compensation, so the PI compensation is applied to the pump control circuit and valve control circuit. The PI transfer function is

$$G_c = K_p\left(1 + \frac{1}{T_i s}\right) = K_p + \frac{K_I}{s} \tag{16}$$

where K_p is the proportional gain, T_i is the internal time, K_I is the internal gain and $K_I = K_p/T_i$. Thus K_I is defined by K_p and T_i.

Step response experiments of the valve control and the pump control circuits are carried out as shown in Figure 6, and optimal PI parameters are obtained through repeated experiments, as shown in Table 2. A relationship, $K_I \gg K_p$ is maintained in the two control circuits, so PI compensation can be considered as an internal compensation mechanism, as discussed in Section 2.2.4. The experimental results show that the selected PI parameters are reasonable and the valve control responds faster than the pump control.

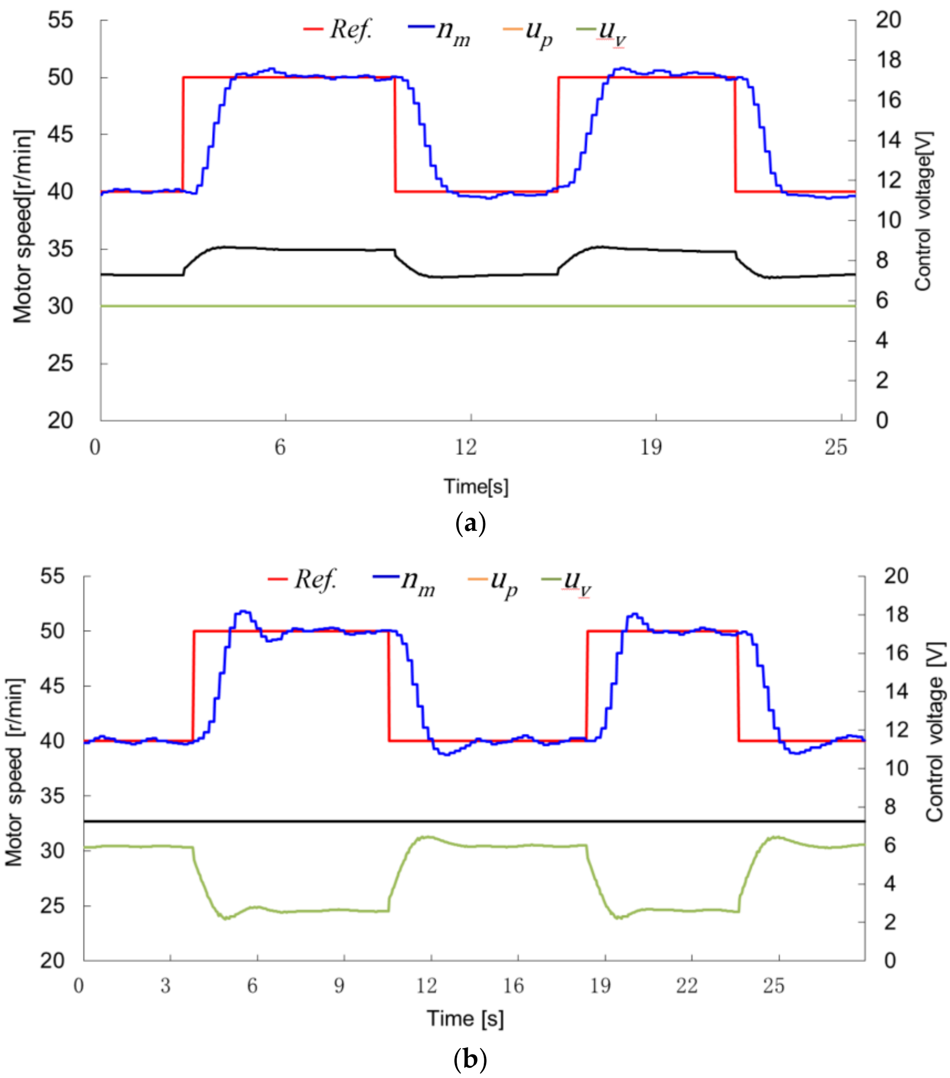


Figure 6. Step response of pump control and valve control. (**a**) Pump control circuit; (**b**) Valve control circuit.

Table 2. Coefficients of PI under different mode.

Control Mode	K_p	T_i	K_I
Pump control	0.3	0.18	1.67
Valve control	0.6	0.12	5

3.2. Results and Discussion

3.2.1. Step Responses to Reference Inputs

In order to investigate the dynamic response characteristics of LVPC systems, experiments in step responses to reference inputs for different weight ratios K_{vp} are carried out, as shown in Figure 7. The experimental condition is described as follows: the initial voltage of the pump u_{p0} = 7.2 V, the initial voltage of the valve $u_{v0} = -5.8$ V, the system pressure P_h = 4.2–5 MPa and PI parameters of valve control circuit and pump control circuit are set as in Table 2. Taking $K_{vp} = 1$ as the axis of symmetry, on the left side of the axis, there is $K_{vp} > 1$ and the combined system is mainly dominated by the PTV and when $K_v : K_p$ = 1:0, the combined system is totally controlled by the valve; on the right of the axis, there is $K_{vp} < 1$ and the combined system is mainly dominated by the VSP and when $K_v : K_p$ = 0:1, the combined system is totally controlled by the pump. The following rules can be obtained:

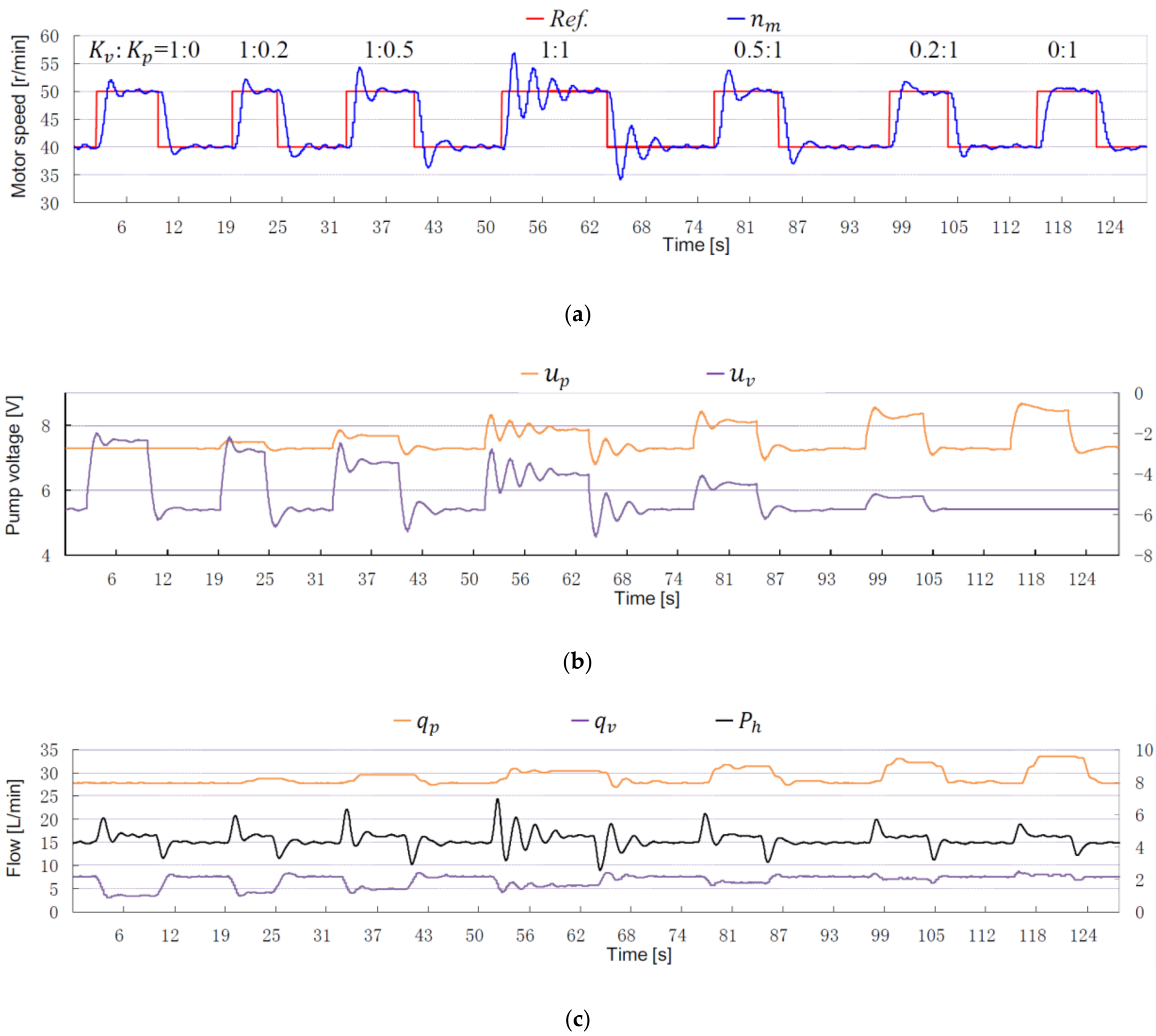


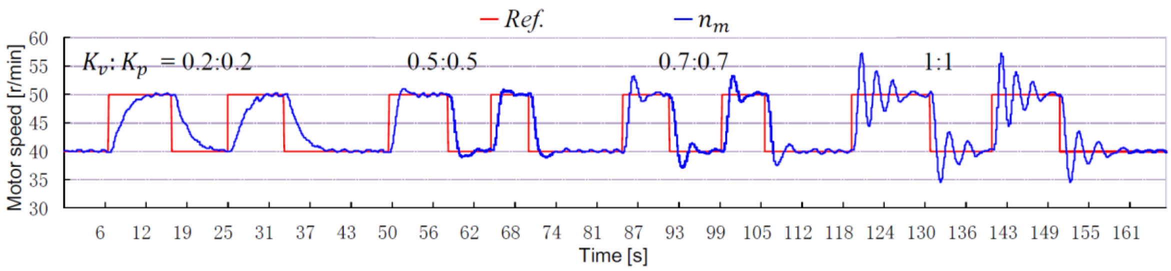
Figure 7. Step response to reference inputs under different weight ratios K_{vp}. (**a**) Motor speed response; (**b**) Pump and valve voltage variation; (**c**) System pressure and flow response.

(1) The step response to reference inputs are of pyramid form, both ends respond slowly, the middle responds fast and LVPC responds faster than the single valve control and the single pump control. That is due to the weight factor K_p or K_v being equal to 1 in the experiment and $K_{sl} > K_{sv}$ *or* $K_{sl} > K_{sp}$ is maintained in Equation (14), when the bigger K_p+K_v, the larger the K_{sl}, the faster the LVPC system responds.

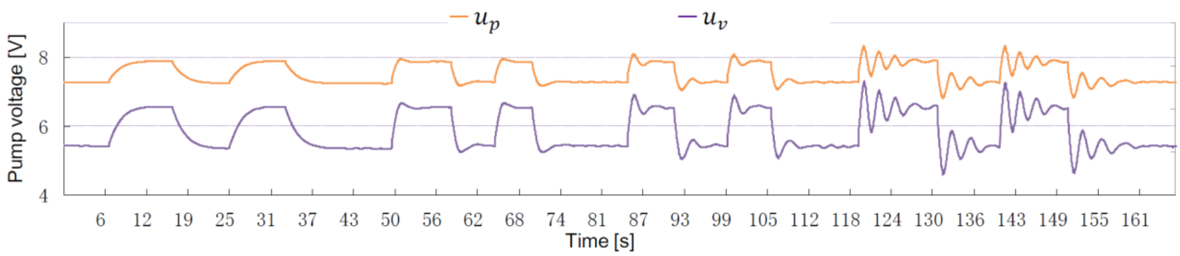
(2) For the same sum of weight factors, the response mainly based on valve control is greater than that based on pump control. That is because $K_{sv} > K_{sp}$, for the same of sum of weight factor $(K_v + K_p)$ and the synthetic open-loop gain of the symmetrical axis on the left side (mainly based on valve control) is larger than that on the right side (mainly by pump control).

(3) The valve-pump weight ratio determines the flow distribution relationship of the valve and the pump. This can be verified from Figure 7b,c, from $K_v : K_p$ = 1:0 to $K_v : K_p$ = 0:1, K_{vp} gradually decreases, the valve control effect decreases as pump control effect increases, so valve flow gradually decreases and pump flow gradually increases.

The step response experiments under the same weight ratio K_{vp} are also carried out, as shown in Figure 8 and the following rules can be obtained.



(**a**)



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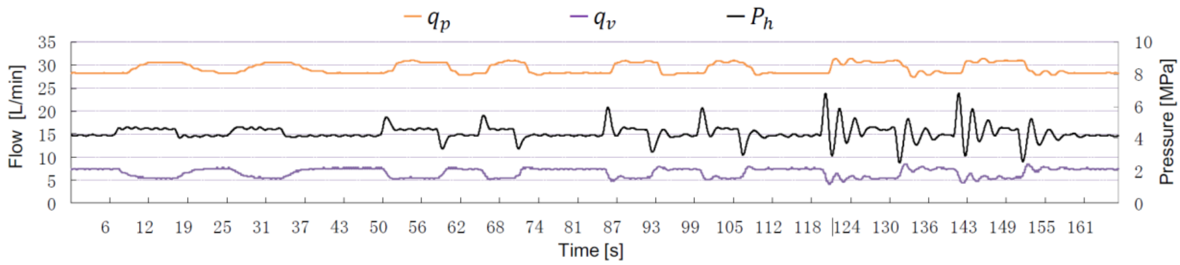


(**c**)

Figure 8. Step response to reference inputs when $K_{vp} = 1$. (**a**) Motor speed response; (**b**) Pump and valve voltage variation; (**c**) System pressure and flow response.

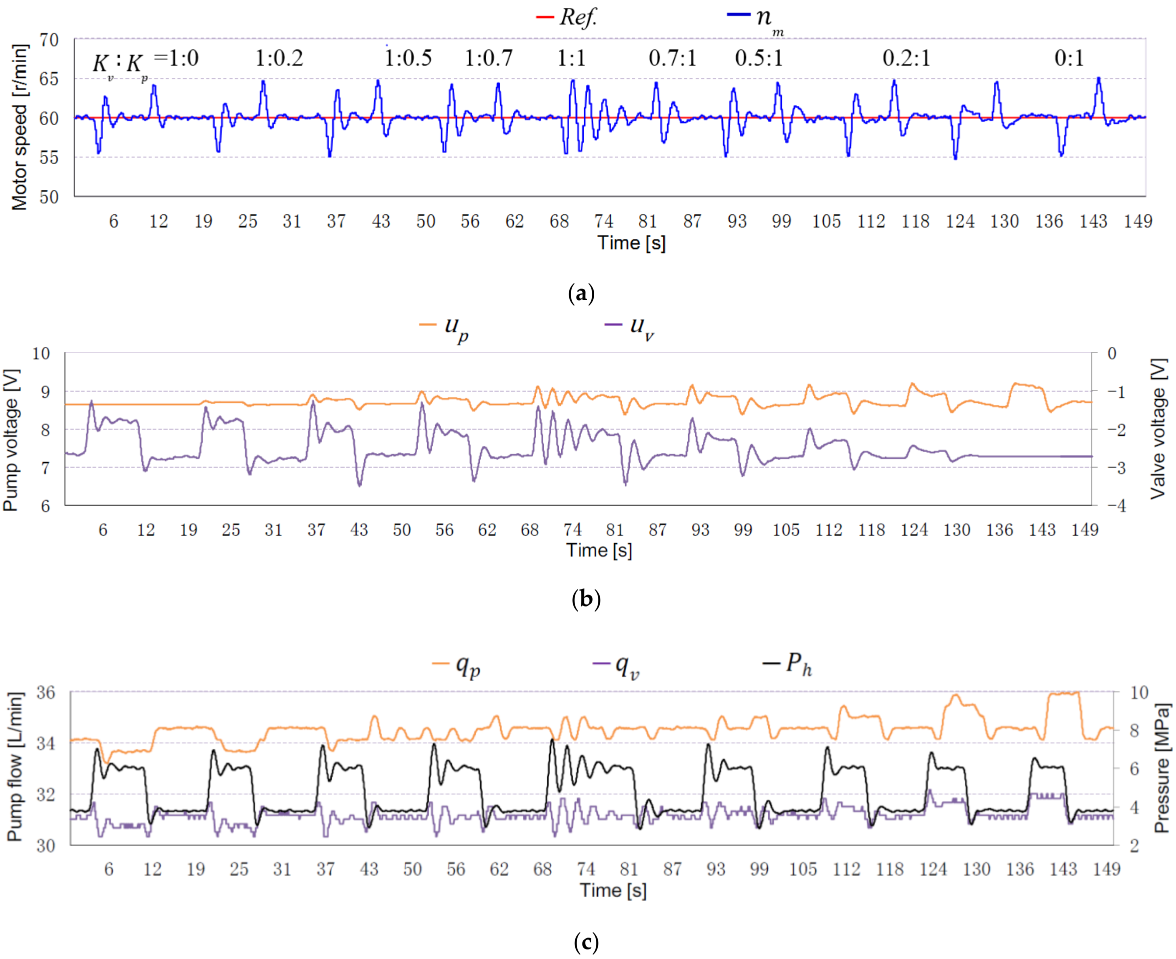
(1) Under the same weight ratio, the response speed of the LVPC system is proportional to the weight factors of the valve control and the pump control. For example, when $K_v : K_p$ = 0.2:0.2, the LVPC system responds slowest and when $K_v : K_p$ = 1:1, the LVPC system responds fastest but the overshoot will be larger and stability is degraded when K_v and K_p are too great. That is because, according to equation (14), K_v and K_p determine K_{sl} *and* K_{sl} determines the response speed of the combined system.

(2) Under the same weight ratio, the weight factor does not affect the valve flow and pump flow in the stable state and the ratio between valve flow and pump flow remains constant but great

weight factor will cause a high-pressure response. For example, from $K_v : K_p$ = 0.2:0.2 to $K_v : K_p$ = 1:1, the stable flow in pump control is always 28 L/min and stable leakage flow in the valve control is always 7.5 L/min.

3.2.2. Step Responses to Loads

After the experiments in step responses to reference inputs, step responses to loads under different and equal vales of K_{vp} are carried out, as shown in Figures 9 and 10. Experimental condition are as follows: initial pump voltage u_{p0} = 8.6 V, initial valve voltage $u_{v0} = -2.8$ V and PI parameters of the control circuit are set as in Table 2. The reference speed of the hydraulic motor is set to 60 r/min, pressure disturbance is generated by the PRV and the high-pressure side of the system produces step pressures from 4 MPa to 6 MPa and 6 MPa to 4 MPa.



(**a**)

(**b**)

(**c**)

Figure 9. Step response to loads under different weight ratios K_{vp}. (**a**) Motor speed response; (**b**) Pump and valve voltage variation; (**c**) System pressure and flow response.

The following conclusions can be made:

(1) The response speed to load disturbance in the LVPC is faster than that in single pump control and in valve control. That is because K_{sl} determines response speed and $K_{sl} > K_{sv}$ *or* $K_{sl} > K_{sp}$ is always true in the experiment.

(2) The velocity stiffness to load disturbance in the LVPC is lower than that in single pump control. Because velocity stiffness is inversely proportional to the leakage coefficients, after adding the valve control to VSPC systems, the total leakage coefficient of LVPC systems increase significantly and

$C_l \gg C_t$, so, compared with VSPC systems, LVPC systems have a lower velocity stiffness and will be more susceptible to load disturbance.

(3) The greater the weight factors K_v and K_p, the faster the LVPC system response and the weight ratio K_{vp} determines the flow distribution relationship between the valve and the pump.

(4) In terms of response speed and flow distribution, the response to loads has a similar a relationship to the response to reference inputs, which could be verified by comparing Figures 7 and 9, Figures 8 and 10, respectively.

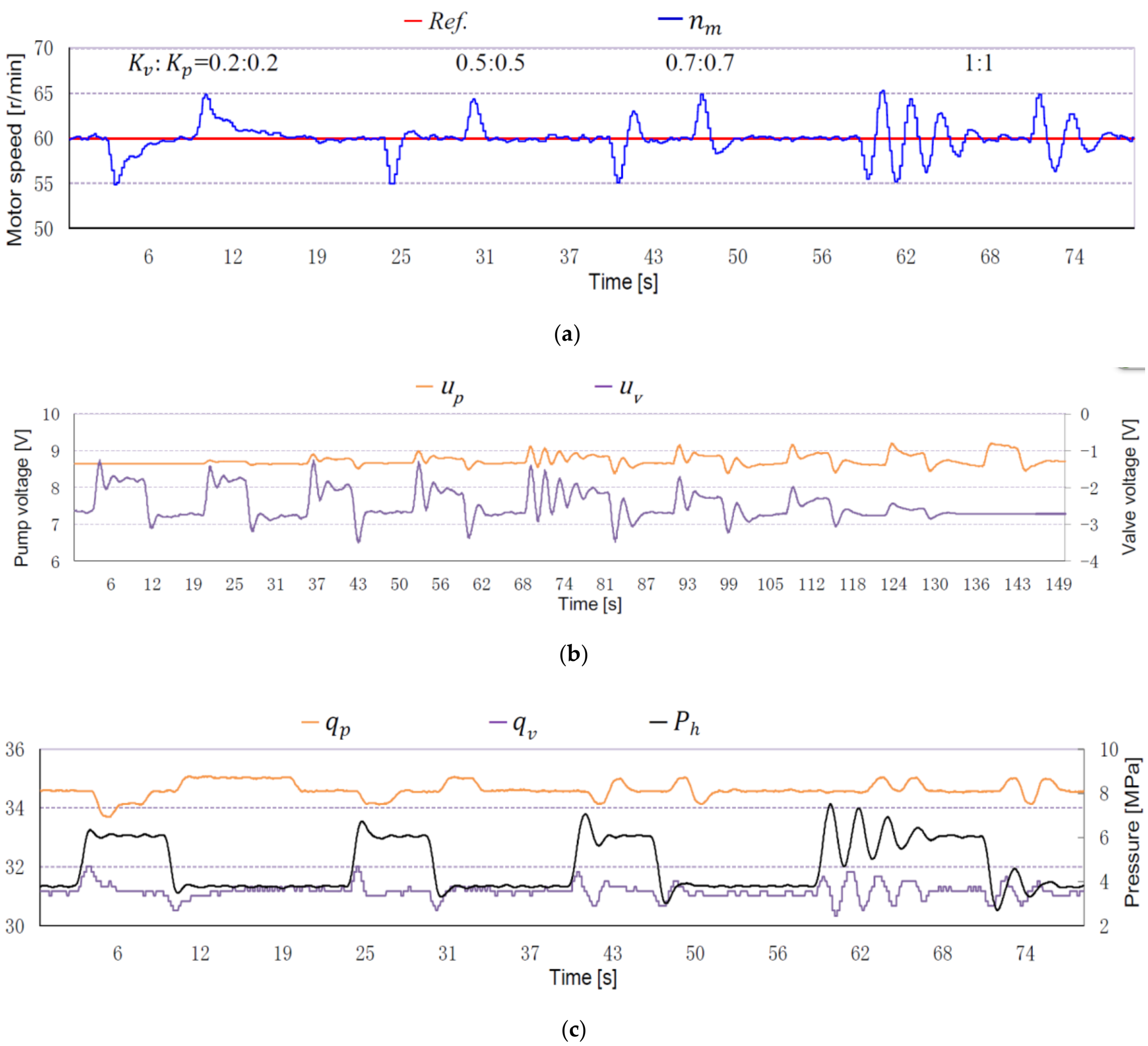


Figure 10. Step response to loads when $K_{vp} = 1$. (**a**) Motor speed response; (**b**) Pump and valve voltage variation; (**c**) System pressure and flow response.

4. Conclusions

In order to improve the dynamic response of VSPC systems, the new LVPC system is developed by the insertion of a control valve in parallel to the main circuit, the control valve works together with the variable speed pump to rapidly regulate overall system flow. Based on theoretical analysis, experimental measurement and results analysis, the following conclusions can be drawn:

(1) After the introduction of valve control total leakage coefficients become greater and vary widely with system pressures and openings of the control valve. Therefore, compared to VSPC system, the LVPC system has greater but unstable damping ratios, which reduce oscillations but increase

the difficulty of parameter prediction. The LVPC system also has lower velocity stiffness, as velocity stiffness is inversely proportional to leakage coefficients.

(2) The role of valve control and pump control in the LVPC system can be adjusted by changing the valve-pump weight ratio, which determines the flow distributions between the valve and the pump.

(3) In the LVPC system, the valve control and the pump control work together on the hydraulic motor. As long as the weight factors are set properly, LVPC will have a greater open-loop gain than VSPC and a faster response speed. However, it should be noted that if the weight is set too large, the system may respond too quickly and there will be a large overshoot.

(4) It is well known that pump control efficiency is high and valve control efficiency is low because of throttling losses. LVPC is a composite control of valve control and pump control and the greater the valve-pump weight ratio, the greater the valve control effect, the lower the system efficiency. Therefore, in order to make the system have higher response while keeping high efficiency, the valve-pump weight ratio should be reasonably set, so that the pump provides a larger flow rate and the valve only works in a small flow state.

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Nomenclatures Index

LPVC	leaking valve-pump parallel control
VSPC	variable speed pump control
PTV	proportional throttle valve
VSP	variable speed pump
K_{vp}	valve-pump weighting ratio, $K_{vp} = K_v : K_p$
K_v	weighting factor of the valve control link
K_p	weighting factor of the pump control link
K_q	flow gain of PTV
K_c	pressure- flow coefficient of PTV
q_v	valve control flow
q_p	pump control flow
C_l	total leakage coefficient of the LVPC system
ξ_l	damping ratio of the LVPC system
ω_l	hydraulic natural frequency of the LVPC system
C_m	total leakage coefficient of the VSPC system
ξ_m	damping ratio of the VSPC system
ω_m	hydraulic natural frequency of the VSPC system

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