Modeling and Analysis of the Pressure Shock Characteristics of a Load-sensitive Hydraulic System Jointly Controlled by Pump and Valve

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Abstract. By establishing the mathematical model of a Load-sensitive Hydraulic System Jointly Controlled by Pump and Valve, the systematic simulation model based on AMESim is built. The simulation results show that the a Load-sensitive Hydraulic System Jointly Controlled by Pump and Valve has different pressure shock characteristics under different load conditions, which has certain reference value for understanding, using and designing the load sensitive pump composite control system.

Keywords: load sensitive; pressure shock; simulation analysis; dynamic characteristics.

1. Introduction
Load-sensitive pump not only can output flow which meet the flow demand of the actuator, but also the outlet pressure of the pump changes with the load pressure, which has the advantages of good energy saving effect and high working efficiency, and is widely used in high-power construction machinery [1-2]. Pump valve compound control system considering the advantages and disadvantages of the valve control system and the pump control system, take the load sensitive pump as hydraulic source of work, work cylinder still use proportional servo valve, its control characteristic is better than that of pump control unit or general of constant pressure valve control system, at the same time, it can avoid the oil supply shortage of air-pocket, is conducive to the control of impact vibration, and can effectively restrain the system vibration noise because of the air pockets.

Many scholars at home and abroad focus on the dynamic characteristics of load sensitive pumps to carry out a lot of research work. Literature [3] analyzed the influence of spring stiffness, opening shape and additional damping on dynamic characteristics of load sensitive pump based on AMESim simulation. Literature [4] analyzed the influence of parameter matching of damping hole and capacity chamber in pump control system on output pressure stability and pressure deviation of pump through virtual prototype of load sensitive pump. Literature [5] studied and analyzed the influence of load sensitive pump under pressure control and flow control on pulsation characteristic and transient response. Literature [6] builds a simulation model to analyze the flow control characteristics and high-pressure cut-off characteristics of the pump based on the mathematical model of load-sensitive pump and load-sensitive multipath valve. Literature [7] studies the influence law of main parameters of load sensitive control system on response speed, and proposes a structural optimization scheme.
Based on the analysis of the working principle of the pump valve compound control system, the corresponding pressure shock mathematical model is deduced, the corresponding simulation model in the graphical simulation environment AMESim is established, and its pressure impact characteristics are analyzed.

2. Working principle
The schematic diagram of pump valve compound control system is shown in Figure 1.

![Diagram of pump valve compound control system](image)

1. Pressure flow compensation valve; 2. High pressure compensating valve; 3. Large chamber of control piston; 4. Small chamber of control piston; 5. Working cylinder; 6. Servo valve.

**Fig. 1** Schematic diagram of pump valve compound control system

The control piston (Part 3) of the variable displacement pump is controlled by the pressure flow compensation valve (Part 1) and the high pressure compensation valve (Part 2). The flow compensation valve (Part 1) senses the pressure of the pump outlet and the high-pressure chamber of the working cylinder, and the high-pressure compensation valve (Part 2) only senses the pressure at the pump outlet. The spring pressure of the high-pressure compensation valve (Part 2) is set at about 25MPa, which means that when the perceived load pressure reaches 25MPa, it will work to the left position, so as to control the piston to connect high-pressure oil and make the swash plate angle return to zero. When the load pressure is less than 25MPa, the high pressure compensation valve (Part 2) does not work, and the control piston is only controlled by the pressure flow compensation valve (Part 1). The set pressure of the spring of the pressure flow compensation valve (Part 1) is about 1.5MPa, which means that when the system flow demand is less than the full flow of the pump, the pressure difference between the pump outlet and the high-pressure chamber of the working cylinder is stable at about 1.5MPa, so it can pass through under this condition. The flow rate of the servo valve is approximately proportional to its opening, which is basically not controlled by the pressure difference between its two ends. The pressure at the pump outlet automatically follows the load pressure. The output pressure of the load sensing pump is always larger than the high pressure end of the two chambers of the working cylinder.
3. Mathematical modeling of a Load-sensitive Hydraulic System Jointly Controlled by Pump and Valve

The response time of the load sensitive pump is $t$, the inner diameter of the pipeline is $D$, the effective area of the oil cylinder is $A$, the equivalent mass of the load is $M$, the fluid velocity of the pipeline is $v_1$, and the length of the pipeline between the pump and the oil cylinder is $L$. It is assumed that the acceleration time of the piston is equal to the response time $t$ of the load sensitive pump.

The maximum value of pressure impact at the cylinder is $\Delta p_{_{m1}}$ (without considering the fluid inertia in the oil cylinder)

$$\Delta p_{_{m1}} = \frac{M \times \pi D^2 \times v_1}{4A \times t} \quad (1)$$

The maximum value of pressure shock at pump outlet is $\Delta p_{m}$:

$$\Delta p_{m} = \Delta p_{_{m1}} + \Delta p_{_{m2}} \quad (2)$$

$$\Delta p_{_{m2}} = \frac{\rho \times L \times v_1}{t} \quad (3)$$

Propagation velocity of pressure wave is $a$:

$$a = \frac{1}{\sqrt{\rho \left( \frac{1}{K} + \frac{D}{E \cdot e} \right)}} \quad (4)$$

Where, $K$ is the elastic coefficient of the oil; $E$ is the elastic coefficient of the pipeline; $D$ is the inner diameter of the pipe; $e$ is the wall thickness of the pipe; $\rho$ is the oil density; $a$ is the propagation velocity of the pressure wave;

Time for pressure wave to and fro in pipeline is $T$:

$$T = \frac{2 \times L}{a} \quad (5)$$

If $t<T$, it is called complete impact, at this time, the pressure impact of pump outlet meet the equation $\Delta p = \Delta p_{m}$;

If $t>T$, the pressure wave returns to the position of the wave source and forms an inverse wave with the pressure wave, which weakens the effect of the pressure wave, which is called incomplete shock. At this time the pressure impact meet the equation $\Delta p = \Delta p_{m} \times \frac{T}{t}$.

When the load changes direction suddenly and the servo valve port opens reversely, the swash plate of load sensitive pump may vibrate and stabilize at a certain angle. The oil pushes the piston to reverse action. At this time, the causes of hydraulic impact include the first type of impact caused by the action of hydraulic control elements, such as the hydraulic impact caused by the reversing or rapid closing of the valve port of the servo valve, and the second type of hydraulic impact caused by the speed change of external load, such as the pressure impact when the cylinder piston starts and stops. The reverse acceleration time of piston is $t_2$, the fluid velocity of pipeline is $v_2$, and the length of pipeline between oil cylinder and servo valve is $L_1$.

The maximum value of pressure impact at the cylinder is $\Delta p_{_{m3}}$ (without considering the fluid inertia in the oil cylinder).
\[
\Delta p_{a1} = \frac{M \times \pi D^2 \times (v_i + v_s)}{4A't_i} + \frac{\rho \times L_1 \times (v_i + v_s)}{t_1}
\]  
(6)

The maximum value of the pressure shock at the pump outlet is (at the moment when the servo valve is closed):

\[
\Delta p_{a4} = \rho \times a \times v_i
\]  
(7)

When the load mechanism is in place, the servo valve is closed, the swash plate of load sensitive pump returns to zero, and the cylinder piston decelerates and stops. At this time, there are both the first and the second causes of hydraulic impact. Suppose that the piston deceleration time is \( t_3 \), the pipeline fluid velocity is \( v_3 \), and the pipeline length between the oil cylinder and the servo valve is \( L_1 \).

The maximum value of pressure impact at the cylinder is \( \Delta p_{a5} \) (without considering the fluid inertia in the oil cylinder):

\[
\Delta p_{a5} = \frac{M \times \pi D^2 \times v_i}{4A't_i} + \frac{\rho \times L_1 \times v_i}{t_1}
\]  
(8)

The maximum value of the pressure shock at the pump outlet is \( \Delta p_{a6} \) (at the moment when the servo valve is closed)

\[
\Delta p_{a6} = \rho \times a \times v_3
\]  
(9)

4. Simulation Analysis

4.1. Pressure shock simulation model

The pressure impact simulation model of the pump valve composite control system is shown in Figure 2. The model encapsulates the load sensitive pump, and the sub model of the load sensitive pump is shown in Figure 3.

![Fig. 2 Pressure shock simulation model of a Load-sensitive Hydraulic System Jointly Controlled by Pump and Valve](image-url)
Some parameters of load sensitive pump are as follows:
(1) The number of plunger is 9, the diameter is $\phi 20\text{mm}$, the distribution circle radius is 36mm;
(2) The moment of inertia of the drive shaft is $0.0083\text{kg} \cdot \text{m}^2$;
(3) Rated speed 1500r / min;
(4) The gap value of the inner radius of the plunger cavity is 0.01mm;

4.2. Simulation analysis of pressure impact
(1) No load condition
The load response and pressure of servo valve A, B and P are shown in Fig. 4 (a) - (d).

![Fig. 3 load sensitive pump model](image)
The Load-sensitive Hydraulic System Jointly Controlled by Pump and Valve has the characteristics of restraining pressure impact. Only the pressure at port A of servo valve has pressure impact at the beginning of hydraulic cylinder action, and its value is far lower than that of general valve control system, as shown in Fig. 4 (b).

**Fig. 4** No load condition simulation results
The pressure drop of three pressure ports of servo valve exists at the beginning of hydraulic cylinder working. The reason is: before the start of the hydraulic cylinder action, the servo valve is at zero position, because there is no load, then the pressure of port A and port B are equal, which is half of the pressure of port P; when the hydraulic cylinder starts to work, the servo valve is in full open position, then port A and port P are connected, port B and port T are connected (so the pressure of port B drops suddenly, as shown in Figure 4 (c)), the cylinder piston can be accelerated to start; however, due to the flow response of the servo valve of the load sensitive pump is slower than that of the servo valve, so there is a pressure drop at port A and port P (as shown in Fig. 4 (d)).

(2) 10MPa load

The load response and pressure of servo valve A, B and P are shown in Fig. 5 (a) - (d).

![Load response curve](image1)

(a) Load response curve

![Pressure curve of port A](image2)

(b) Pressure curve of port A
When the load mechanism moves from the limit position to the zero position, it is a negative load, and when the zero position moves to the limit movement, it is a positive load. When the load is loaded, the pressure drop of the valve is much greater than that of the positive load. Therefore, the action speed under the load is obviously faster than that under the positive load, which is the broken line reflected in the load response curve, as shown in Fig. 5 (a).

At the beginning of the action of the hydraulic cylinder, there are pressure impact and pressure drop. The reasons are the same as before.

When the load mechanism moves in place, there is no significant pressure impact, but there is pressure overshoot, which is caused by the back pressure of port B.

At the beginning of the loading mechanism action (12s), the pressure of port A appears "ladder". The reason is that before the operation of the hydraulic cylinder, the load mechanism is at -30° position \( Q_s \gg Q_s \), \( P_i \approx 0 \), \( P_s \approx P_l \) (PL is the load pressure), \( P_s \approx P_s + \Delta P \); after the hydraulic cylinder starts to work, when it reaches the steady state \( P_r = P_s + \Delta P \), \( P_r - P_s = P_s \), \( P_s - P_a = P_s \), it is shown in Fig. 5 (b) (the 13s ~ 15s). However, the pressure of port A is zero from 12s to 13s, which is caused by the fact that the output flow of the pump can not reach the theoretical flow under the pressure difference, which means cavitation.

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**Fig. 5** Simulation results of 10MPa load condition

![Graph](image-url)
5. Conclusion
The mathematical model of hydraulic shock of load sensitive pump is derived, and AMESim model of a Load-sensitive Hydraulic System Jointly Controlled by Pump and Valve is established. Through the analysis of simulation results, the hydraulic shock characteristics of the compound control system of load sensitive pump under different load conditions are obtained.

1. Under the condition of no-load, the compound control system of sensitive pump has small hydraulic impact characteristics;
2. The load will aggravate the hydraulic impact characteristics of the sensitive pump compound control system, but has little effect on the overall operation of the system.
3. The simulation results are basically consistent with the theoretical analysis, which has great reference value for understanding, using and designing a Load-sensitive Hydraulic System Jointly Controlled by Pump and Valve.

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