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An Electro-Hydraulic-Load-Sensitive System on the Basis of Torque Open-Loop Control

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Abstract: Facing the development trend of electrification of construction machinery, in view of the drawbacks of the existing electro-hydraulic-load-sensitive system in terms of dynamic characteristics and usage of energy, based on the drive source of a servo motor-driven quantitative pump, an electro-hydraulic-load-sensitive system on the basis of torque open-loop control was proposed. Firstly, the working principle of the system was introduced and the system's operating characteristics and energy consumption characteristics were theoretically analyzed. Secondly, in order to balance the system's energy usage and maneuverability, a control strategy with a variable pressure margin was designed. Meanwhile, in order to solve the problem that the hydraulic pump's mechanical efficiency causes system pressure control deviation, a torque compensation method based on offline data and speed prediction was proposed. Finally, simulation and testing were used to confirm the viability of the control strategy. The test results show that: the system could realize stable pressure margin control, and the response rise time was within 0.7 s under a variety of flow circumstances; the system could follow the control instruction to change linearly and the flow rate changed smoothly in the adjustable pressure gap control; after using the compensation method, the deviation of the pressure gap control was within 2%.

Keywords: torque control; pressure control; electric construction machinery; load sensitive

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1. Introduction

Through the predetermined pressure gap, the load-sensitive system keeps the system pressure consistently higher than the load pressure by a given constant amount. Additionally, the system self-adaptively realizes the flow supply and demand matching, and the load flow is only related to the directional control valve's spool displacement. As a result, the load-sensitive system, which is extensively employed in construction machines [1–4] and marine machines [5,6], has strong energy-saving and control properties.

Through the shuttle valve and the hydraulic pipeline, the typical machine-hydraulic-load-sensitive system applies the maximum load pressure to the load-sensitive pump, and realizes the follow-up control of the hydraulic pump's output pressure through the mechanical force balance of the spool of the load-sensitive valve maintained by a pressure compensating valve to achieve load-independent flow [7]. Although the hydraulic-load-sensitive system has achieved good power matching of the system, the system feedbacks pressure through the hydraulic line and controls the pressure through the variable mechanism, and the rapidity and stability of the system response are poor [8,9].

To be able to solve the shortcomings of the traditional mechanical-hydraulic-load-sensitive system in terms of response and stability, relevant scholars have conducted research on the electro-hydraulic-load-sensitive system [10–12]. Xu et al. [13] collected the load pressure through the sensor, replaced the load-sensitive pump with the electric proportional pump. Additionally, for controlling the pump output pressure, the proportional pump's displacement was adjusted using closed-loop pressure, which solved the problem of the lagging feedback pressure transmission of the mechanical-hydraulic-load-sensitive system. Yang et al. [14] introduced an electro-hydraulic flow matching system with pre-valve compensation on excavators, which calculated the system demand flow according to the proportional valve opening signal and synchronously controlled the electric proportional pump displacement to achieve flow supply-demand alignment. The problem of pump lagging valve control in machine-hydraulic-load-sensitive system was solved. Cheng et al. [15] proposed an electro-hydraulic-load-sensitive system to regulate the proportional pump displacement through a combination of flow feed-forward and pressure feedback, which was characterized by high response and damping, and could achieve accurate flow matching. The above related research has led to a major breakthrough in the dynamic response and stability of load-sensitive techniques. However, the system adopts the form of a fixed-speed variable-displacement power source, which always maintains the rated speed, the prime mover's efficiency is lower when there is no load, and the variable pump's efficiency is likewise lower when the displacement is minimal [16].

With the popularization of the global concept of resource conservation and environmental protection and the increasingly strict emission policy of construction machinery, pure electric drive technology has become an inevitable trend for future development [17]. Pure electric drive construction machinery to battery and servo motor instead of fuel engine, with no pollution, high efficiency and good control performance and other characteristics [18–20]. Based on the power source form of a servo motor-driven quantitative pump and combined with the load-sensitive system to realize the machine's electro-hydraulic control system transition is one of the research hot spots [21–23]. Yuan et al. [24] proposed an electro-hydraulic flow matching system with variable speed based on valve front compensation on a truck crane, and designed an anti-flow saturation controller to realize the enhancement of the machine's overall handling and energy efficiency. Liang et al. [25] proposed a pump-valve dual-source cooperative drive multi-actuator system, which solved the problem of extra power loss caused by the load difference of multi-actuators. Guo et al. [26] used a variable speed flow matching system in the steering system of an electric loader to establish the correspondence between the servo motor's speed and the angle at which the steering wheel rotates to realize the steering system flow's supply and demand matching. Fu [27] proposed an electro-hydraulic-load-sensitive system with pressure feedback closed-loop control of servo motor speed in an electric excavator, and paired with an adjustable divergent pressure control approach to increase the overall machine's agility and energy economy. Li [28] proposed a variable speed load-sensitive import and export independent control system based on energy control and position control mode switching, and its control performance and energy saving effect were higher than that of the traditional pump control and valve control system. Liu et al. [29] aimed at the problem of mutual coupling of pump and valve control in electro-hydraulic-load-sensitive systems, designed position and pressure self-immunity controllers, respectively, and achieved separation of the pump control and valve control subsystems.

From the previous study, it can be seen that the type of electro-hydraulic-load-sensitive system is mainly divided into two types: a flow matching type and a pressure feedback type. The flow matching type load-sensitive system, according to the directional control valve opening signal to calculate the system demand flow, the open-loop matching control servo motor speed, the system response is fast, and has high stability; but because of the need for a pressure compensation valve and a constant directional valve before and after the pressure difference, the valve structure is more complex, there is excess throttling loss and, due to the influence of the system's nonlinear factors, it is difficult to realize the flow rate of the accurate matching [30]. The electro-hydraulic load sensing system uses pressure feedback by controlling the servo motor speed in a closed-loop manner using pressure feedback. The system uses electronic pressure compensation, without an additional pressure compensation valve, can eliminate some of the throttling loss and reduce the corresponding cost but, due to the need to introduce closed-loop feedback control, in the complex and variable load conditions, it is difficult to take into account the system's stability and dynamic response [31].

Aiming at the shortcomings of the existing electro-hydraulic-load-sensitive system, founded on the power source form of a servo motor-driven quantitative pump, an electro-hydraulic-load-sensitive system on the basis of variable torque management is proposed. By adjusting the servo motor's torque, the system realizes open-loop control of the system pressure and achieves change control of the pressure margin according to the load condition, so as to increase the dynamic characteristics and energy efficiency of the system under complicated and changeable load situations. At the same time, considering the nonlinear relationship between servo motor torque and hydraulic pump outlet pressure, a torque compensation approach based on offline data and speed prediction is proposed to provide accurate control of the open loop of system pressure.

2. System Operation Principle

Figure 1 shows the structure of the electro-hydraulic-load-sensitive system based on torque open-loop control. The system is mostly made up of servo motors, quantitative hydraulic pump, proportional reversing valves, pressure compensation valves, solenoid switching valves, hydraulic cylinders and pressure sensors and other components. The system operates on the following principle: The controller transforms and outputs the servo motor torque control signal in accordance with the highest load pressure and the predetermined pressure gap after receiving the system pressure and the proportional directional valve opening signal from the pressure sensor. Additionally, then the hydraulic pump's outlet pressure is under open-loop control, which realizes the purpose of system load sensitivity by maintaining a constant pressure margin between the outlet pressure of the hydraulic pump and the maximum load pressure. The opening signal of the proportional reversing valve determines the setting value of the system pressure margin. The controller detects the proportional reversing valve's opening signal, and according to the correlation between the predetermined pressure margin and the proportional reversing valve's opening signal input into the controller beforehand, the system outputs the pressure margin value in real time under the current operating conditions. As a way to avoid the mutual impact of the load pressure changes in the hydraulic cylinders of the heavy-loaded and light-loaded circuits during the compound action, a pressure compensation valve is used in the light-loaded circuit to keep the pressure differential between before and after the proportional reversing valve constant. The electromagnetic switching valve controls the pressure compensation valve's pilot oil port simultaneously. When the light-loaded circuit carries out the composite action, the electromagnetic switching valve loses power, and the pressure compensation valve works normally; when the light-loaded circuit carries out the separate action, the electromagnetic switching valve gains power, and the pressure compensation valve does not have the compensating effect, which reduces the throttling loss of the pressure compensation valve under the full working condition, and improves the system's working energy efficiency.

The system proposed in this research has the following primary properties, which can be seen by the analysis just mentioned: (1) The outlet pressure of the hydraulic pump is controlled by open-loop control of the torque of the servo motor. Compared with the pressure feedback servo motor speed control method, this control method does not require a pressure sensor, simple structure, high stability, while the motor torque control is an inner loop, the response is also more rapid. (2) Proportional directional valve for heavy-duty circuit controls the difference in pressure between the front and back through electronic compensation, and proportional directional valves for light-duty circuits choose electronic compensation or mechanical compensation of pressure compensation valves according to the working conditions, which reduces the throttling loss of pressure compensation valves under full working conditions and increases the system's energy efficiency. (3) According to the proportional directional valve opening signal, there is a change in the system's pressure margin. The proportional directional valve opening is small, the system is in small flow conditions, and this reduces the pressure margin to reduce the system throttling loss; the proportional directional valve opening is large, the system is in large flow conditions, and this increase the pressure margin to meet the system's large flow of fast response requirements.

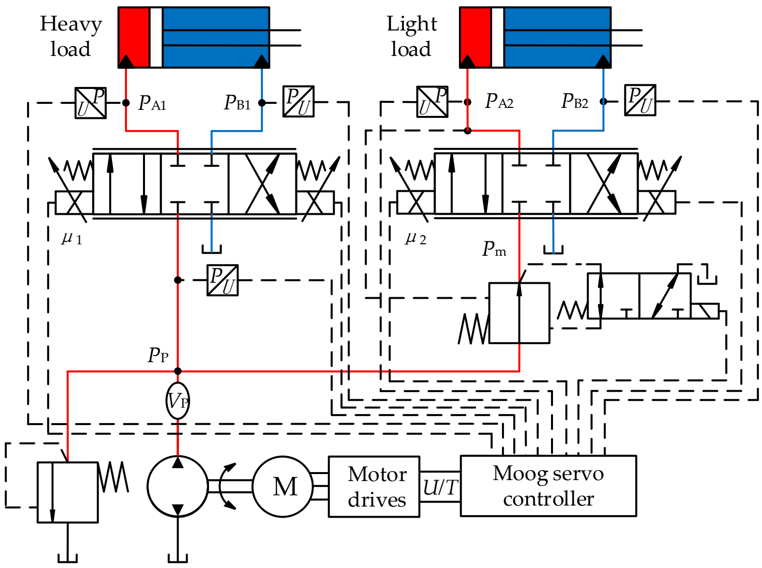

Figure 1. Structure of electro-hydraulic-load-sensitive system based on torque open-loop control.

3. System Characterization

3.1. Operating Characteristics

3.1.1. System Flow Characteristics

The hydraulic cylinder's flow continuity equation is shown below:

$$q_L = A_p \frac{dx_p}{dt} + C_{tp} p_L + \frac{V_t}{4\beta_e} \frac{dp_L}{dt} \tag{1}$$

where C_{tp} is the hydraulic cylinder's leakage coefficient, V_t is the cylinder's overall compressed volume, and p_L is the differential pressure between the rodless and rodded cavities.

It can be seen from Equation (1) that part of the flow entering the hydraulic cylinder is used to push the piston, part of it leaks and part of it is compressed. The proportional reversing valve controls the load flow of the hydraulic cylinder. The flow pressure drop equation of the proportional reversing valve is expressed as:

$$q_L = C_d x_{vi} W \sqrt{2\Delta p_i / \rho} \tag{2}$$

where x_{vi} is the spool displacement of proportional valve i (i = 1, 2), W is the area gradient, and Δp_i is the pressure difference before and after proportional valve i.

The proportional directional valve's frequency response can be regarded as a proportionate link since its frequency width is substantially larger than the hydraulic intrinsic frequency, at this point:

$$x_{vi} = k_v \cdot u_i \tag{3}$$

where u_i is the electrical signal of the proportional valve i and k_v is the gain coefficient.

The equilibrium equations for the load force and output force on the hydraulic cylinder are expressed as:

$$A_1 p_A - A_2 p_B = m_t \frac{d^2 x_p}{dt} + B_p \frac{dx_p}{dt} + K x_p + F_L \tag{4}$$

where A_1 and A_2 are the area of action of rodless and rodded cavities, m_t is the action areas of the rodless cavity and the rod cavity, B_p is the coefficient of viscous damping, F_L is the outside force and K is the load spring's stiffness.

From Equations (1), (2) and (4), it is clear that the pressure differential between the front and back and the displacement of the proportional valve spool are the key factors influencing the load flow produced by the proportional directional control valve. When the external load force changes, the proportional directional valve's pressure differential between its front and back changes, and the load flow output by the proportional directional valve changes, that is, the movement speed of the hydraulic cylinder changes. The displacement of the spool and the output flow of the proportional directional control valve will have an almost linear connection without being impacted by variations in load as long as the pressure differential before and after the proportional directional control valve is constant.

Considering the influence of the system pressure on the flow leakage of the hydraulic pump, the total output flow of the hydraulic pump can be expressed as:

$$q_p = C_d x_1 W \sqrt{2\Delta p_1/\rho} + C_d x_2 W \sqrt{2\Delta p_2/\rho} + k_{lp} p_p \tag{5}$$

where k_{lp} is the hydraulic pump's leakage coefficient.

At this point, the speed of the servo motor can be expressed as:

$$n_p = q_p / D_p \tag{6}$$

where D_p is the hydraulic pump's displacement.

3.1.2. Pressure Control Characteristics of the System

Proportional directional control valves can realize mechanical differential pressure compensation through differential pressure reducing valves. The compensation principle is to maintain a consistent pressure difference through the valve port through the mechanical balance of the internal spool. The force balance equation can be expressed as:

$$p_m \cdot A_a = p_{A2} \cdot A_a + F_a \tag{7}$$

where A_a is the compensation valve's spool area and F_a is the preset spring force.

The pressure compensation valve will cause redundant throttling loss. In the electro-hydraulic load sensing system, a pressure sensor can be combined to realize the load sensing function by controlling the speed or torque of the servo motor. The link between motor speed and hydraulic pump outlet pressure can be represented as follows:

$$\frac{V_p}{\beta_e}\frac{dp}{dt} = D_P n_p - q_L - q_{lp} \tag{8}$$

where Vp is the volume of the hydraulic pump's exit cavity.

It can be seen from Equation (8) that when the speed of the servo motor is adjusted, the hydraulic pump's outlet pressure changes correspondingly. However, due to unknown parameters such as q_L and q_{lp}, adjusting the servo motor's speed in an open loop to precisely control the hydraulic pump's outlet pressure is challenging. When pressure closed-loop feedback control is introduced, it is difficult for the system to balance dynamic response and stability under complex and variable load conditions.

By controlling the servo motor's torque, the pressure at the outlet of the hydraulic pump can also be controlled. The torque equation for the hydraulic pump and the servo motor's motion equation are each expressed as:

$$\begin{cases} T_e = T_p + J\frac{d\omega}{dt} + B_m\omega \\ T_p = \frac{(p_P - p_T)D_P}{2\pi\eta_m} \end{cases} \tag{9}$$

where J is the sum of the rotating inertias of the pump and the motor, T_{p} is the output torque of the motor, T_{e} is the motor's electromagnetic torque, ω is the motor's angular velocity, p_{T} is the pressure of the tank, p_{p} is the hydraulic pump's output pressure, and η_{m} is the hydraulic pump's mechanical efficiency.

The torque that comes out of the servo motor is proportional to the hydraulic pump's output pressure, as can be observed from Equation (9) if the mechanical efficiency of the hydraulic pump is not taken into account. Direct control of the pump output pressure is possible by adjusting the servo motor's torque. This pressure control method does not need to set pressure closed-loop feedback control, the system stability is high, and the servo motor's torque control, which is an inner loop component, responds rather quickly.

In the actual hydraulic system, the hydraulic pump's mechanical-hydraulic efficiency cannot be ignored. The mechanical-hydraulic efficiency of the swash plate axial piston pump can be expressed as [32]:

$$\eta_{\mathrm{m}} = \frac{1}{1 + \frac{\mu n_{\mathrm{p}} C_{\mathrm{s}}}{k C_{\mathrm{v}} (p_{\mathrm{p}} - p_{\mathrm{T}})} + C_{\mathrm{f}} + \frac{2\pi T_{\mathrm{s}}}{(p_{\mathrm{p}} - p_{\mathrm{T}}) n_{\mathrm{p}} D_{\mathrm{p}}}} \tag{10}$$

where T_{s} is the torque loss that is unaffected by pressure and speed, C_{f} is the coefficient of mechanical resistance, C_{v} is the coefficient of flow resistance, C_{s} is the coefficient of leakage, and μ is the viscosity of kinematics.

Equation (10), which shows that the mechanical efficiency of the axial piston pump is mostly influenced by speed and the difference in pressure between the intake and output, and is also affected by factors such as the oil temperature and the structural parameters of the hydraulic pump. It is difficult to obtain accurate mechanical efficiency values of hydraulic pumps through theoretical calculations. This paper intends to calibrate the output pressure of hydraulic pumps at different speeds and torques through experiments to compensate for the deviation of hydraulic pump mechanical efficiency from system pressure control.

3.2. System Characteristics of Energy Consumption

Ignoring the pressure loss of the hydraulic pipeline of the system, the system's output pressure can be written as:

$$p_{\mathrm{p}} = p_{\mathrm{A1}} + \Delta p_1 = p_{\mathrm{A2}} + \Delta p_2 + \Delta p_{\mathrm{m}} \tag{11}$$

where p_{A1} is the rodless cavity pressure of the heavy load circuit cylinder, Δp_1 is the system's predetermined pressure margin, p_{A2} is the rodless cavity pressure of the light load circuit cylinder, Δp_2 is the difference in pressure compensated by the pressure valve, and Δp_{m} is the pressure compensation valve's pressure loss.

The output flow of the system can be expressed as:

$$q_{\mathrm{p}} = q_{\mathrm{L1}} + q_{\mathrm{L2}} + q_{\mathrm{lp}} \tag{12}$$

where q_{L1} is the heavy load circuit load flow, q_{L2} is the light load circuit load flow, and q_{lp} is the system leakage flow.

The power consumed by the system can be expressed as:

$$P_{\mathrm{loss}} = \Delta p_1 q_{\mathrm{L1}} + (\Delta p_2 + \Delta p_{\mathrm{m}}) q_{\mathrm{L2}} + p_{\mathrm{p}} q_{\mathrm{lp}} \tag{13}$$

It is evident from Equations (11)–(13) that the system can match supply and demand for flow rate, and that its main sources of power loss are the pressure compensation valve and the proportional reversing valve. Under the same flow demand, the greater the preset pressure margin of the system, the greater the power loss of the system. Compared with the electro-hydraulic flow matching system, the electro-hydraulic-load-sensitive system on the basis of changeable torque control, and can realize the control of the system's changeable pressure margin in accordance with the system working conditions, so it is more energy-saving.

The correlation between the pressure of the rodless cavity and the pressure loss of the pressure compensation valve can be expressed as:

$$\Delta p_{\mathrm{m}} = (p_{\mathrm{A1}} - p_{\mathrm{A2}}) + (\Delta p_1 - \Delta p_2) \tag{14}$$

Equation (14) shows that the power loss of the system increases with the load pressure differential between the heavy load and the light load. In the hydraulic system of construction machinery with multiple actuators, how to reduce the extra power loss caused by load difference is also one of the research hotspots.

4. Control Strategy

When the controller simultaneously sends out the current control signals of proportional reversing valves 1 and 2, the system enters the control mode of compound action. At this time, the heavy load circuit adopts the control mode of electronic pressure compensation. The light load circuit magnetic switch valve loses power, the pressure compensation valve works normally, and by using machine-hydraulic pressure difference compensation, the proportional reversing valve's pressure differential between the front and back is controlled.

The control strategy of electro-hydraulic-load-sensitive system on the basis of changeable torque control is shown in Figure 2. The controller identifies the current signal μ_1 of the proportional directional valve of the heavy load circuit, and outputs the pressure margin command Δp_1 of the system under this working condition according to the relationship curve between the proportional directional valve current signal μ_1 calibrated in advance and the system pressure margin Δp_1, at the same time, according to the highest load pressure p_{A1} of the heavy load circuit hydraulic cylinder collected by the pressure sensor, output the control command p_{p}* of the hydraulic pump's outlet pressure in real time, and convert and output the torque control command T_{cmd} of the servo motor through the torque command module. The method controls the hydraulic pump's outlet pressure by directly controlling the torque of the servo motor, thereby controlling the pressure margin of the system according to the change in the working condition, realizing the function of load sensitivity, and adaptively matching the flow required by the output system.

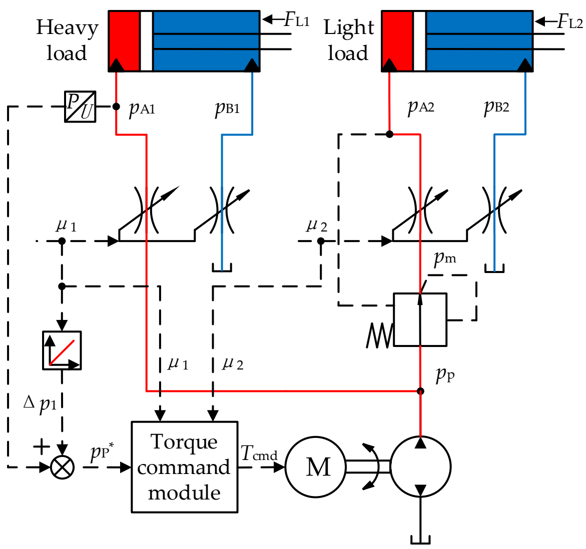

Figure 2. Control strategy of electro-hydraulic-load-sensitive system on the basis of changeable torque control.

It can be seen from Equation (9) that due to the influence of the hydraulic pump's mechanical-hydraulic efficiency, the control command of the hydraulic pump outlet pressure cannot be simply converted into the control command of the servo motor torque through theoretical calculations. Therefore, this paper suggests an approach based on offline data and speed prediction. The torque compensation method. The general idea is as follows: calibrate the hydraulic pump's outlet pressure at different speeds and torques by means of tests, and predict the motor speed when the system is working in real time, combined with the system preset pressure margin, output the torque control command of the servo motor online, to compensate the deviation of the hydraulic pump's mechanical-hydraulic efficiency to the system pressure control, and its specific control strategy is shown in Figure 3.

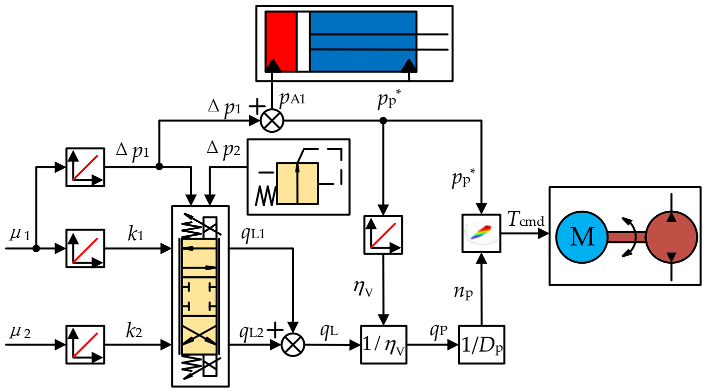

Figure 3. Control strategy of torque instruction module.

The controller identifies the current signals μ_1 and μ_2 of the proportional valve, and predicts the output load flows q_{L1} and q_{L2} of the proportional directional valve according to the system predetermined pressure gap Δp_1 and the preset pressure difference Δp_2 of the proportional directional valve 2 of the light load circuit.

The amount of the input current signal and the valve's pressure drop have an impact on the actual flow via the proportional directional valve. The following expression can be used to determine the proportional reversing valve's load flow rate under a specific pressure drop:

$$Q = k \cdot Q_N \sqrt{\frac{\Delta p}{\Delta p_N}} \tag{15}$$

where k is the actual opening of the proportional valve, Δp_N is the rated pressure drop, and Q_N is the rated flow rate.

Equation (12) demonstrates that in addition to precisely forecasting the servo motor's speed, one must also forecast the hydraulic pump's leakage flow in real time. The hydraulic pump's volumetric efficiency is mainly related to the system pressure. The relationship between the hydraulic pump's volumetric efficiency η_V and the system pressure p_p is calibrated in advance, as shown in Figure 4. According to the maximum load pressure p_{A1} collected by the pressure sensor and combined with the preset pressure margin Δp_1, the leakage flow q_{lp} of the hydraulic pump can be predicted. Given the known system load flow q_L and leakage flow q_{lp}, the servo motor speed n_p can be obtained. Through the calibrated relationship between the hydraulic pump torque, rotational speed and output pressure, the accurate control command of the servo motor torque under the outlet pressure p_p* of the pre-controlled hydraulic pump can be obtained. The correlation between hydraulic pump torque, speed and output pressure is shown in Figure 5.

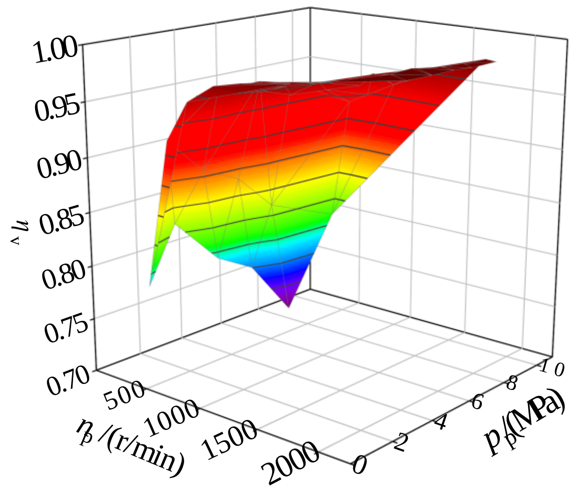

Figure 4. Test curve of volume efficiency of hydraulic pump at different speeds and pressures.

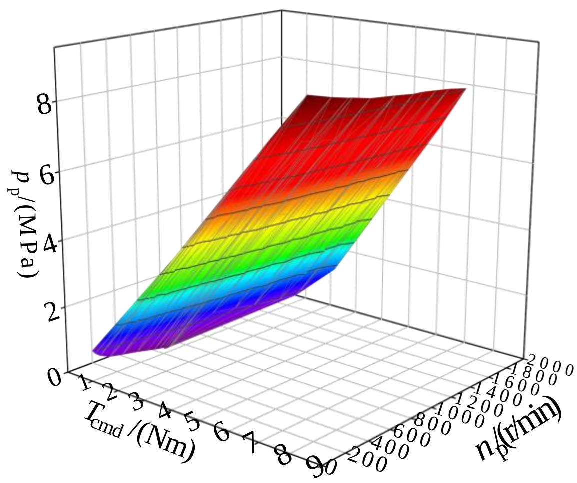

Figure 5. Test curve of torque, speed and output pressure of hydraulic pump.

5. Simulation Modeling

5.1. Construction of Simulation Model

A simulation model was created and examined using the hydraulic simulation program AMESim in order to investigate the operational characteristics of the electro-hydraulic-load-sensitive system on the basis of changing torque control. Figure 6 displays the simulation model for the system.

Table 1 displays the AMESim simulation model's setting parameters.

Table 1. Simulation parameters.

Parameter	Units	Value
Displacement of fixed displacement pump	mL/r	6.3
Maximum output torque	N·m	14.5
Setting pressure of safety valve	MPa	15
Hydraulic cylinder diameter/rod diameter	mm	50/35
Rated pressure drop of proportional valve	MPa	7
Rated flow rate of proportional valve	L/min	19

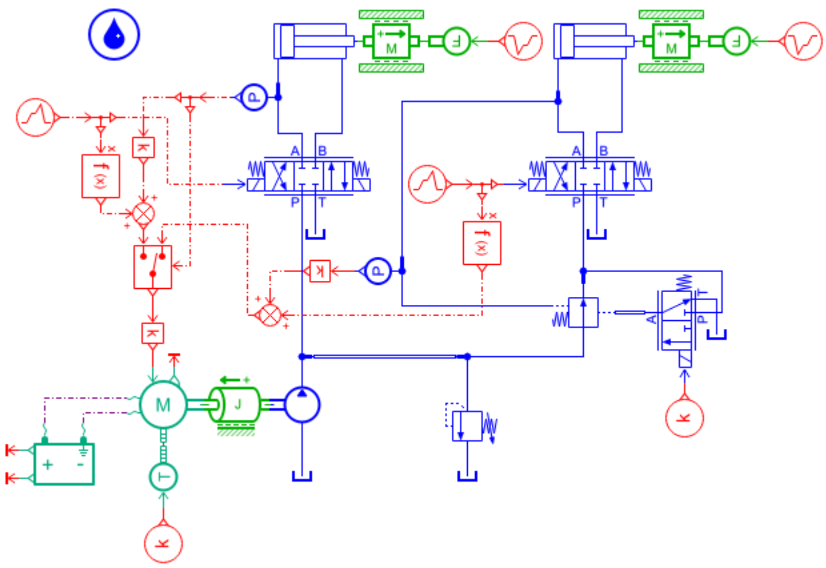

Figure 6. AMESim simulation model.

5.2. Simulation Analysis

In this paper, the torque of the servo motor is directly controlled to accomplish open-loop control of the hydraulic pump's outlet pressure, and combined with the highest load pressure collected by the pressure sensor, the function of load sensitivity is realized. At the same time, the system's pressure margin can be changed according to the requirements of system flow, energy consumption and dynamic response. The control command is given automatically by the correlation curve between the predetermined proportional directional valve current signal and the system pressure margin.

Firstly, the flow and pressure response of the system when the load is suddenly changed is studied through simulation. Figure 7 depicts the simulation results.

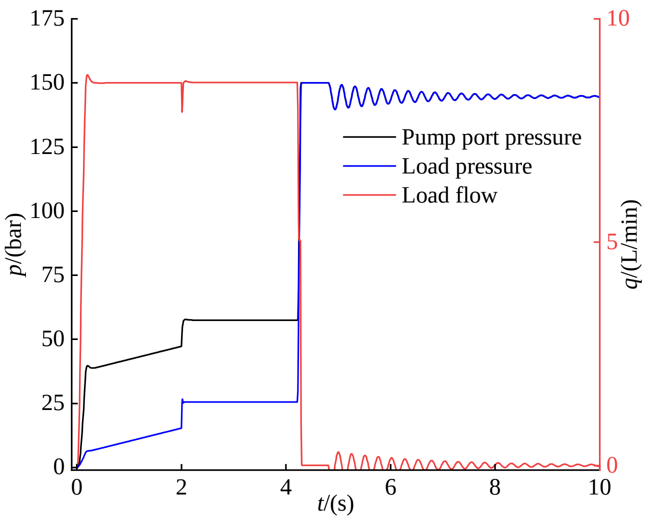

Figure 7. System pressure and flow curves under variable load.

In Figure 7, when the load pressure increases linearly or stepwise, the difference between the hydraulic pump's outlet pressure and the maximum load pressure remains constant at 3.2 MPa, realizing the load-sensitive function. Although the load pressure is constantly changing, the proportional directional valve's output flow remains unchanged at 8.6 L/min. At this time, the change in load pressure has no effect on the load flow, which is exclusively connected to the current signal of the proportional directional valve. Once the system pressure exceeds the preset safety pressure, the servo motor maintains the maximum torque output, the hydraulic pump adapts to no flow output, and the system has no overflow loss. However, when the servo motor operates at low speed, the system pressure tends to oscillate.

Keeping the load constant, the simulation investigates the flow and pressure conditions of the system when the pressure margin is changed in response to the proportional directional valve's current signal. The results are shown in Figure 8.

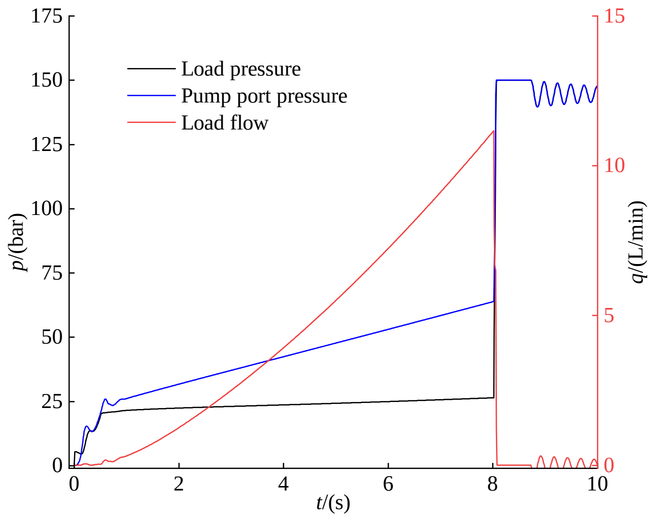

Figure 8. Simulation curve of system pressure and flow under variable pressure margin control.

In Figure 8, when the current signal of the proportional reversing valve increases gradually, the system pressure margin can increase linearly from 0 MPa to 4 MPa corresponding to the calibration correlation. The system output flow is comparatively stable during the system changeable pressure margin control operation. However, when the opening of the proportional reversing valve is small, there will be certain pressure fluctuations in the system.

Figure 9 depicts a comparison of the system's consumption of energy under variable pressure gap management and constant pressure difference control.

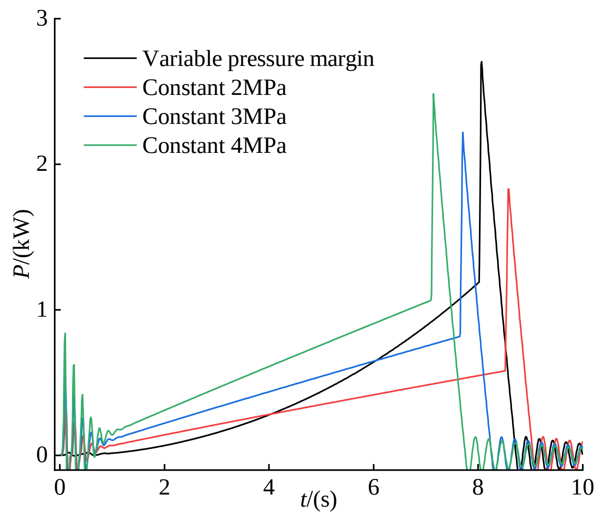

Figure 9. System energy consumption curves under different pressure control modes.

In Figure 9, the energy consumption of the system under variable pressure margin control and constant 2, 3, 4 MPa differential pressure control are compared, respectively. When the opening of the proportional reversing valve is small, the system consumes less energy under variable pressure margin control. When the proportional reversing valve's opening is large, the energy consumption of the system under variable pressure margin control is greater than that of constant 2 MPa and 3 MPa differential pressure control, but less than that of constant 4 MPa differential pressure control. Although the system's usage of energy is low when the valve opening is large under a constant small pressure difference control, the system's output flow is small at this time, and it is difficult to meet the demand of rapid change in large load flow. The use of variable pressure margin control can take into account the reduction in energy usage of the system during minor flow conditions as well as the dynamic responsiveness of the system under big flow conditions.

The pressure of the system under compound action conditions is shown in Figure 10, and the flow rate is shown in Figure 11.

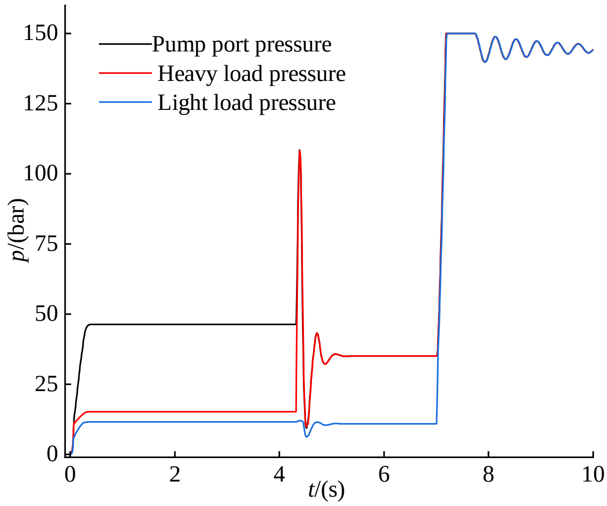

Figure 10. System pressure curve during compound action.

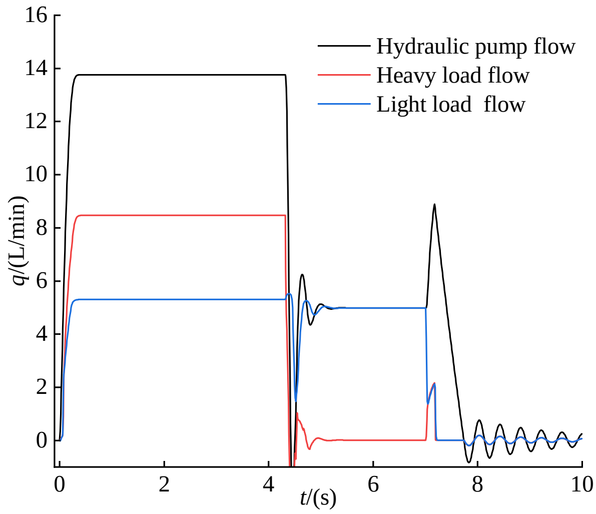

Figure 11. System flow curve during composite action.

In Figures 10 and 11, when the system performs compound action, the heavy load circuit proportional directional valve outputs a constant flow of 8.5 L/min, the light load circuit proportional directional valve outputs a constant flow of 5.3 L/min, and the hydraulic pump automatically matches the output of 13.8 L/min flow rate indicates that the system can achieve accurate flow matching during compound actions. When the heavy load circuit hydraulic cylinder first reaches the limit stroke and the system pressure approaches the safety pressure, switch to the light load circuit single control strategy, the system pressure is reduced, and the system's pressure loss under this working condition is reduced. When both the heavy and light load circuit hydraulic cylinders reach the limit stroke, the hydraulic pump's output flow first rises and then falls within 1 s, which is generated by the short-term opening of the relief valve due to the pressure shock.

Theoretical analysis demonstrates that the mechanical-hydraulic efficiency of the hydraulic pump, which is directly controlled by the torque of the servo motor, affects the pressure at the hydraulic pump's outlet. The pressure output of the hydraulic pump at different mechanical efficiencies are shown in Figure 12.

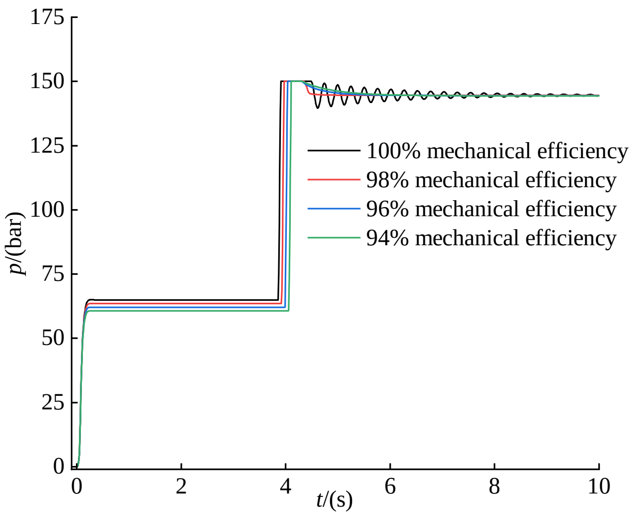

Figure 12. Output pressure curves of hydraulic pumps with different mechanical efficiencies.

In Figure 12, under the same pressure control instruction, the greater the hydraulic pump's mechanical-hydraulic efficiency, the greater the output pressure of the hydraulic pump. When the system works at low speed, the lower the hydraulic pump's mechanical-hydraulic efficiency, the more stable the output pressure. Since it is difficult to truly simulate the hydraulic pump's mechanical-hydraulic efficiency during simulation, the feasibility of the torque compensation method based on offline data and speed prediction in compensating the hydraulic pump outlet pressure control deviation will be verified in the experimental part.

6. Experimental Research

6.1. Test Platform

As a way to confirm the effectiveness of the above system theoretical analysis and simulation results, according to the hydraulic principle shown in Figure 13 the test system shown in Figure 14 is built. The test system is composed of a servo motor, a quantitative pump, a proportional reversing valve, a throttle valve, a pressure sensor, a flow sensor, a MOOG controller, a servo motor driver, etc. In the actual test, the main control program is compiled by the MOOG axis control software MACS 3.4 and imported into the MOOG controller. The upper computer controls the MOOG controller to output analog signals, synchronously controls the torque of the proportional reversing valve and the servo motor, and the throttle valve is used to replace the simulated load of the hydraulic cylinder.

In the test system, the servo motor model is HP11318-G202A, with a rated speed of 2000 r/min and a rated torque of 20 N·m from Ningbo Haitian Drive Systems; the model of the quantitative pump is TFH-630, with a displacement of 6.3 mL/r, and the manufacturer is Ningbo Haitian Drive Systems; The servo valve model is G761-3003, with a rated pressure of 7 MPa and a rated flow rate of 19 L/min from MOOG. The controller is the servo controller M300 of MOOG; The relief valve is Sun's RDBALCN with a maximum relief pressure of 35 MPa; The throttle valve is sun's NFBCLCCN, with a rated flow rate of 20 L/min; The range of the flow sensor is 0.2–30 L/min, and the measurement accuracy is ±0.3%; The pressure sensor has a range of 0–16 MPa and a measurement accuracy of ±0.2%.

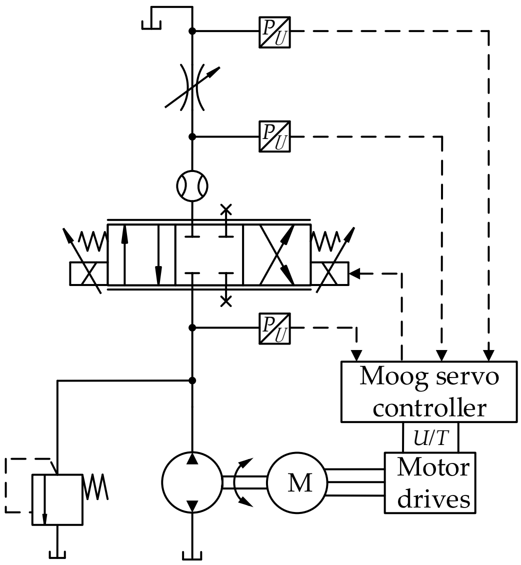

Figure 13. Principle of test system.

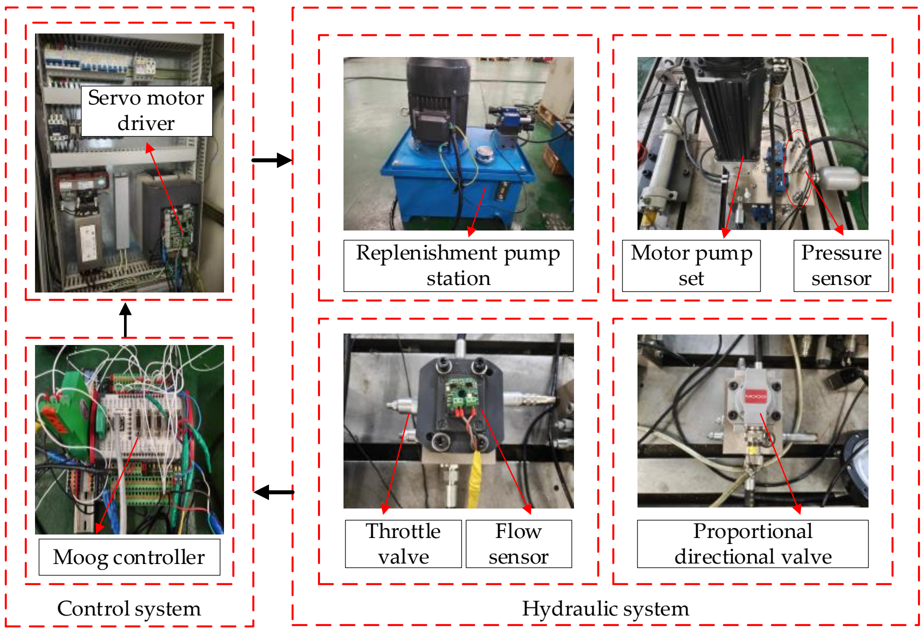

Figure 14. Composition of the test system.

6.2. Experimental Analysis

In the experiment, the operating characteristics of the electro-hydraulic-load-sensitive system on the basis of changeable torque control and the possibility under variable pressure margin control are analyzed first. In addition, according to the aforementioned theory and simulation results, it is clear that the mechanical-hydraulic efficiency of the hydraulic pump will lead to the deviation of the system pressure control. Additionally, the test will explicitly analyze how the system controls pressure when operating with various mechanical hydraulic pump efficiency.

The pressure margin control command of 4 MPa is given to the system, and then the control command of the proportional reversing valve opening step from 70% to 50% and

100% is given, respectively, to study the load sensitivity characteristics of the system when the pressure changes. The test results are shown in Figures 15 and 16.

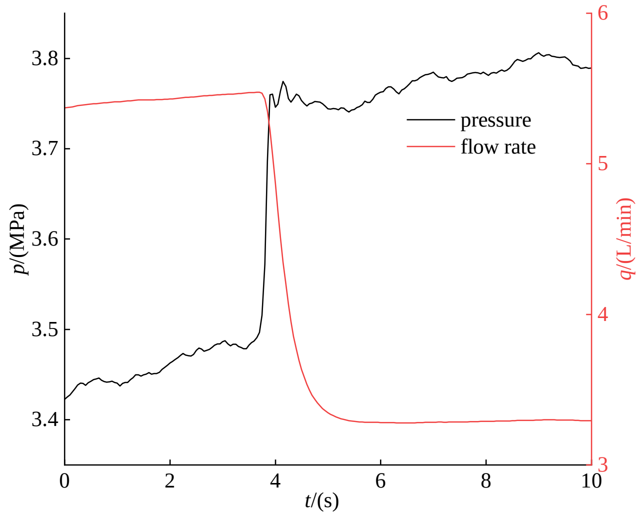

Figure 15. Flow rate and pressure curve with a step increase in valve port.

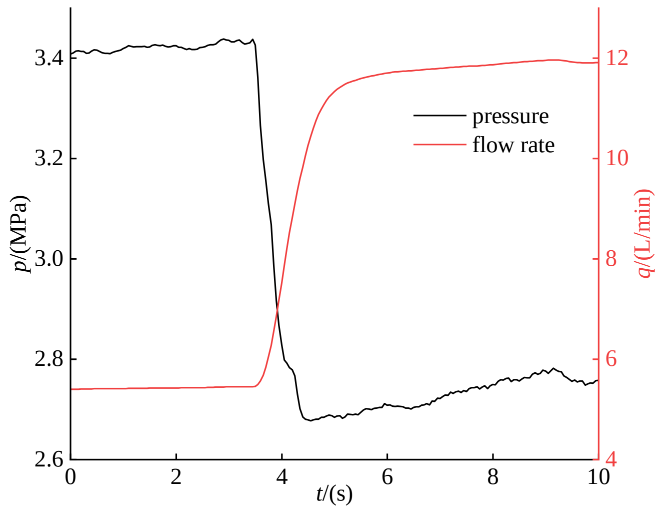

Figure 16. Flow and pressure curve of valve port step reduction.

In Figures 15 and 16, given the system pressure gap command of 4 MPa, the pressure margin of the system is basically stable before and after the step of the proportional valve reversing valve opening, indicating that the system can realize the load-sensitive function. However, affected by the hydraulic pump's mechanical-hydraulic efficiency, the pressure margin control of the system is not accurate, especially when the proportional reversing valve step increases or decreases, the pressure margin of the system will also decrease or increase accordingly. If this part of the deviation is not compensated, the system will have poor load sensitivity performance when the load flow changes suddenly. When the proportional reversing valve opening increases and decreases step by step, the flow changes are relatively stable without overshoot.

Calibrate the correlation between the proportional directional valve's opening and the system pressure margin in advance, so that when the proportional directional valve's opening changes linearly from 0% to 50%, the pressure margin control command output by the system increases linearly from 0 MPa to 5 MPa. When the valve opening changes linearly from 50% to 100%, the system keeps outputting a pressure margin control command of 5 MPa. In the test, a ramp signal with a gradually increasing opening of the proportional reversing valve is given to study the actual pressure margin of the system and the change in load flow. The test results are shown in Figure 17.

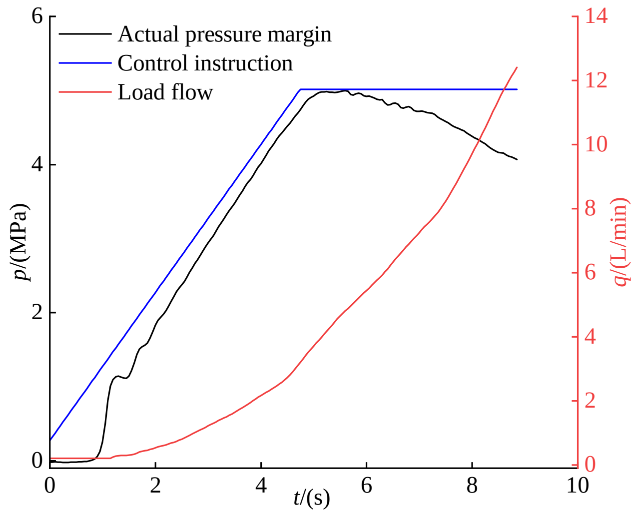

Figure 17. Test curve of system pressure and flow under variable pressure margin control.

In Figure 17, the actual pressure margin of the system can increase linearly following the control instructions, but the pressure margin of the system is poorly controllable under the condition of minimal flow. The larger the system pressure margin, the larger the load flow, and it is relatively stable during the change process without oscillation. This shows that the electro-hydraulic-load-sensitive system on the basis of changeable torque control is feasible under changeable pressure gap control. In addition, it can be seen that there is a deviation between the actual pressure margin of the system and the control command, which is also the deviation caused by the hydraulic pump's mechanical-hydraulic efficiency.

According to theoretical analysis, the hydraulic pump's mechanical-hydraulic efficiency is mainly related to the speed of the hydraulic pump and the difference in pressure between the hydraulic pump's input and exit. In the test, the inlet pressure of the hydraulic pump remains constant at 0.22 MPa. Therefore, the two main elements contributing to the deviation of the system pressure management are the hydraulic pump's outlet pressure and speed.

Given a pressure margin control command of 4 MPa, the proportionate directional valve's opening is set to 15%, 30%, 65%, and 100%, respectively, respectively. The pressure response of the system under different load flows is shown in Figure 18.

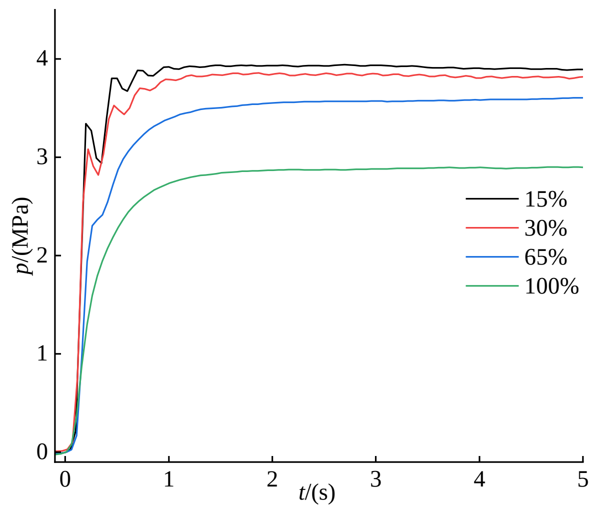

Figure 18. Pressure response curves under different load flow rates.

In Figure 18, the openings of the proportional reversing valves are 15%, 30%, 65%, and 100%, respectively, the actual pressure margins of the system are 3.92 MPa, 3.82 MPa, 3.66 MPa, and 2.86 MPa, respectively, and the deviation values are 0.08 MPa, 0.18 MPa, 0.34 MPa, and 1.14 MPa. It is evident that the deviation of the system pressure control increases with load flow rate, or with the speed of the hydraulic pump. In addition, it is evident from the illustration that the greater the load flow, the smaller the fluctuation of the system pressure but the slower the response.

Keep the 65% opening of the proportional reversing valve constant and the throttle valve fully open. At this time, the pressure gap of the system is approximately the difference in pressure between the hydraulic pump's input and exit. Control instructions with a pressure margin of 2 MPa, 3 MPa, 4 MPa, and 5 MPa are given, respectively. The pressure response of the system is shown in Figure 19.

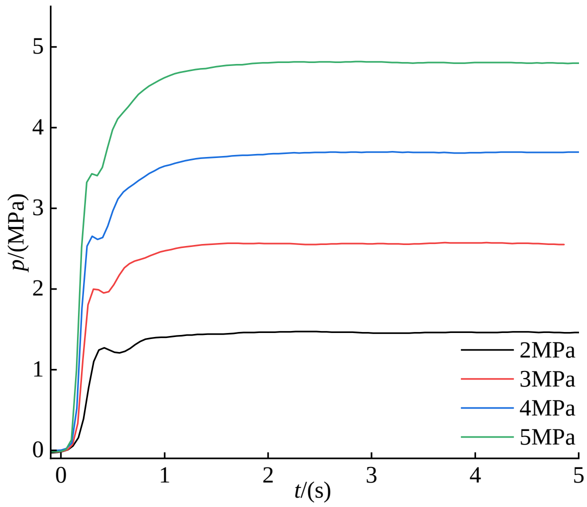

Figure 19. System pressure response curve.

In Figure 19, the pressure margin control commands of 2 MPa, 3 MPa, 4 MPa, and 5 MPa are, respectively, given, and the actual pressure margins of the system are 1.44 MPa, 2.56 MPa, 3.67 MPa, and 4.81 MPa, respectively, and the deviation values are 0.56 MPa, 0.44 MPa, 0.33 MPa, and 0.19 MPa. It can be seen that the greater the difference in pressure between the hydraulic pump's input and exit, the smaller the deviation of the system pressure control. In addition, it is evident from the illustration that the larger the pressure margin, the smaller the pressure fluctuation of the system and the faster the pressure response. This also indirectly verifies the feasibility of the aforementioned system to increase the pressure margin to meet the flow response requirements of the system under large flow conditions.

6.3. Experimental Verification

The accuracy of the system pressure margin control directly influences how well the hydraulic system of the whole machine performs, as can be observed from the analysis of the above-mentioned test results. The hydraulic pump's mechanical-hydraulic efficiency will lead to some deviation in the pressure control of the system. Due to the impact of different nonlinear components, it is difficult to obtain the mechanical efficiency of the hydraulic pump through simple theoretical calculation. For this reason, this paper proposes a torque compensation method based on off-line data and speed prediction to reduce the deviation of the system in pressure control. In the experiment, the control strategy shown in Figure 3 is programmed and introduced into the controller to study the pressure control of the system after adopting the compensation strategy.

Firstly, after adopting the compensation strategy, the impact of the speed of the hydraulic pump on the control of the system pressure margin is studied. Additionally, set the opening of the proportional reversing valve to 15%, 30%, 65%, and 100%, respectively, and given a pressure margin control command of 2 MPa, the pressure response of the system under different load flow rates is shown in Figure 20.

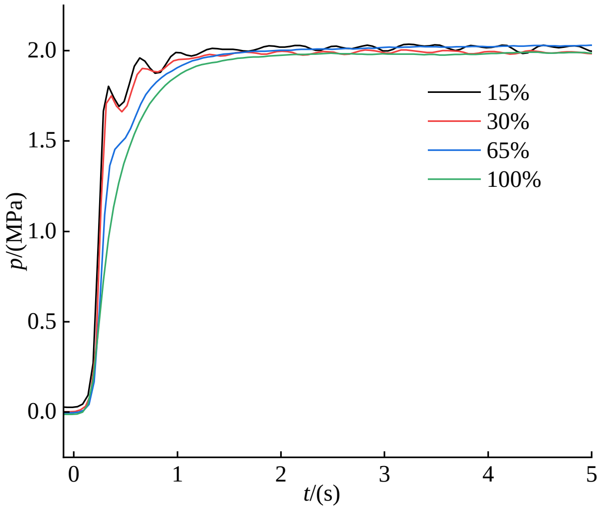

Figure 20. Pressure response under different load flow rates after compensation.

In Figure 20, the proportional directional valve openings are 15%, 30%, 65%, and 100%, respectively. After adopting the compensation strategy, the actual pressure margins of the system are 2.01 MPa, 2.01 MPa, 1.99 MPa, and 1.98 MPa, respectively. The control deviation is less than 2%. Under each flow condition, the rise time of pressure response is 173 ms, 391 ms, 557 ms, and 665 ms, respectively.

After adopting the compensation strategy, keep the 65% opening of the proportional reversing valve constant, and give control commands with pressure margins of 2 MPa, 3 MPa, 4 MPa, and 5 MPa, respectively. The pressure response of the system is shown in Figure 21.

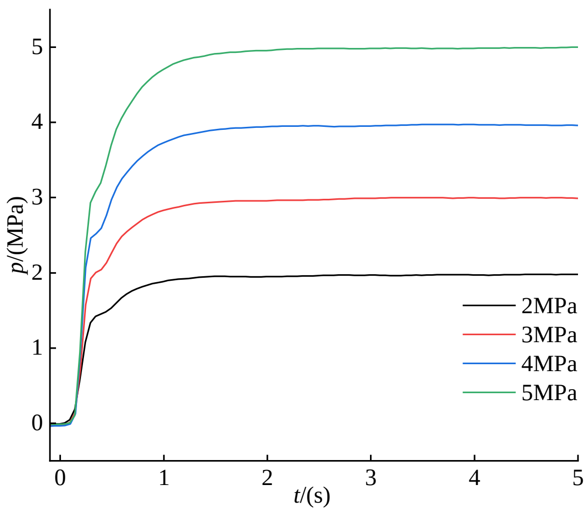

Figure 21. System pressure response curve after compensation.

In Figure 21, given control commands with pressure margins of 2 MPa, 3 MPa, 4 MPa, and 5 MPa, the actual pressure margins of the system are 1.97 MPa, 2.99 MPa, 3.97 MPa, and 4.98 MPa, respectively, and the pressure control deviation is less than 2%. The rise time of each pressure response is 607 ms, 613 ms, 659 ms, and 638 ms, respectively.

As a way to analyze the system's pressure and flow characteristics when the load flow changes suddenly after the compensation strategy is adopted, a pressure margin control command of 2 MPa is given for the system, and then the opening of the proportional reversing valve is given step by step from 70% to 50% and 100%. control instructions, the test results are shown in Figures 22 and 23.

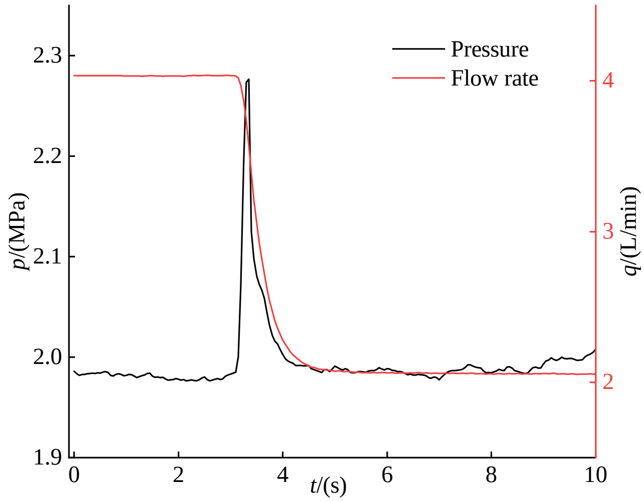

Figure 22. Pressure and flow curves when the valve port decreases step by step.

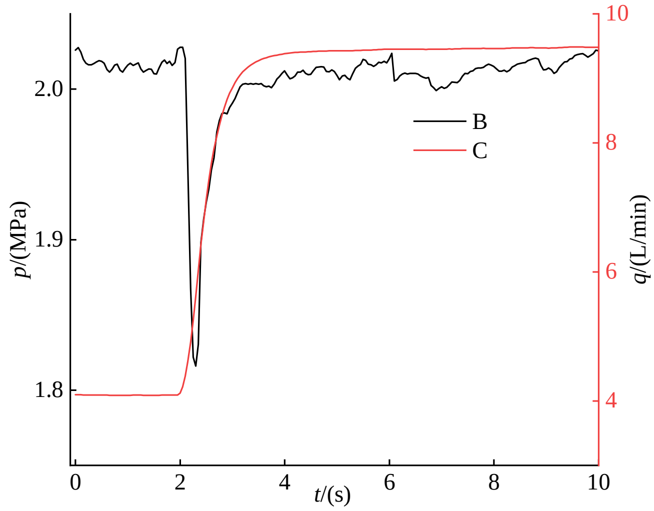

Figure 23. Pressure and flow curves when the valve port increases step by step.

In Figures 22 and 23, given the system pressure margin control command of 2 MPa, when the proportional reversing valve opening increases or decreases step by step, the actual pressure margin of the system can basically stabilize at approximately 2 MPa within a short time. The system pressure has no overshoot, and the flow rate changes smoothly, indicating that the system has better load-sensitive performance after adopting the compensation strategy.

7. Conclusions

Facing the development trend of electrification of construction machinery, based on the power source of a servo motor driving quantitative pump, an electro-hydraulic-load-sensitive system on the basis of torque open-loop control and a control method using a variable pressure margin were proposed. At the same time, in addition to address the problem of deviation brought on by the hydraulic pump's mechanical-hydraulic efficiency to the system pressure control, a torque compensation method based on offline data and speed prediction was proposed. The findings indicate that:

1. The suggested system controls the output pressure of the hydraulic pump in an open loop by directly controlling the torque of the servo motor. The system has a simple structure, can basically achieve stable pressure margin control, and has good load sensitivity characteristics. Under a wide range of various flow conditions, the response rise time of the system pressure margin is within 0.7 s. However, when the system pressure is small or the load flow is small, there will be certain pressure fluctuations.
2. The proposed system can realize variable pressure margin control according to the current signal of the proportional directional valve. The real pressure margin of the system can change linearly in response to the control command during the control procedure, and the load flow changes smoothly, while taking into account the energy usage in low-flow situations and the response requirements in high-flow situations.
3. The deviation in system pressure control increases with hydraulic pump speed or with a reduction in the pressure differential between intake and output. By adopting the torque compensation method based on off-line data and speed prediction, the deviation of system pressure margin can be controlled within 2%.

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