Optimization design and analysis of the proportional solenoid valve for the power shift transmission of cotton picker using NSGA-II algorithm

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Abstract: In order to further improve the dynamic response performance of the proportional solenoid valve of power shift transmission of cotton picker, the structure and working principle of the proportional solenoid valve were analyzed, and the simulation model of electro-hydraulic shift system of the proportional solenoid valve was built based on Amesim software. In addition, the dynamic response curve of oil pressure output of shift valve was obtained and verified by bench. In Isight software, the optimal Latin hypercube test design method was used to analyze the key parameters affecting the oil pressure response characteristics of the proportional solenoid valve, and the key parameters were identified. Taking the oil pressure step response time and oil pressure response overshoot of the proportional solenoid valve as the optimization objectives, the NSGA-II algorithm was used to optimize the key parameters of the proportional solenoid valve. The results show that after optimization, the oil step response time of the proportional solenoid valve is shortened by 4.55%, the oil response overshoot is reduced by 68.28%, and the flow response time is shortened by 4.76%, which improves the dynamic response characteristics of the proportional solenoid valve. The optimization results have certain significance for improving the shift quality of the cotton picker power shift system.

Keywords: power shift transmission, proportional solenoid valve, multi-objective optimization, cotton picker, NSGA-II
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1 Introduction

The development and progress of power shift technology is largely due to the development of electro-hydraulic control technology[1,2]. By using an electronic control unit and hydraulic control system, the precise control of the shifting process of the power shift transmission can be realized. The dynamic response characteristics of the proportional solenoid valve are very important to realize automatic shift and improve the shift quality of the power shift system of the cotton picker. The selection of structural parameters of the proportional solenoid valve has a decisive influence on its dynamic response and on the oil pressure of the shifting clutch and shifting brake[3-5], so the reasonable design of structural parameters of the proportional solenoid valve is the key to improving its response speed and control accuracy[6].

In recent years, many scholars have carried out extensive and in-depth research on shift solenoid valves and made many achievements, mainly focusing on three aspects: dynamic modeling of solenoid valves, multi-objective optimization, and intelligent control algorithms. In the field of dynamic modeling, scholars have deeply explored the coupling effect between the solenoid valve and hydraulic system through decoupling analysis and multi-domain coupling simulation. For example, Yang et al.[7] conducted the fluid-solid-thermal coupling simulation analysis of the electro-hydraulic proportional directional valve, which provided a certain reference for the thermal deformation resistance design of the valve. Meng et al.[8,9] established an exact mathematical model for a proportional solenoid valve, realized the linear control of input signal and output pressure/flow, and verified the effectiveness of the model through simulation and experiment. Salloom et al.[10], aiming at the insufficient bidirectional control performance of the proportional directional valve, proposed to add a bidirectional valve to make it only throttle in one direction. Simulation and experiment verified that the improved scheme was effective, which provided a reference for developing new high-performance valves. Mo et al.[11] constructed two models of empirical formula and CFD-

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electromagnet joint calibration for a pilot proportional pressure reducing valve. The experiment shows that the calibration model is in better agreement with the experimental results, can accurately predict the static and dynamic performance, and provides a reliable basis for valve optimization. In the aspect of multi-objective optimization, researchers have improved the performance of solenoid valves through advanced algorithms. For example, Jian et al.[12] proposed an optimal design method of a pressure control valve considering stability. The influence of solenoid valve parameters on stability boundary and output pressure response time was analyzed by experimental design method, and the response characteristics of the solenoid valve were improved. Yu et al.[13] studied the multi-objective optimization of a high-frequency solenoid valve's current drive strategy using BPNN and NSGA-II genetic algorithms. In the aspect of intelligent control algorithm, Zou et al.[14] took the self-developed 7-speed automatic transmission as the research object, discussed the relationship between proportional solenoid valve current, clutch pressure, torque, and speed in the shift control process, and proposed two adaptive control strategies of torque phase and inertia phase. Yang et al.[15] used an LSTM model for data mining and feedforward compensation to solve the nonlinear hysteresis problem of proportional solenoid valve pressure-current in a dual-clutch transmission. The tests show that the method can effectively predict the pressure in the lifting section and improve the control accuracy and driving smoothness of the vehicle. Zhong et al.[16], aiming at the contradiction between fast response and low power consumption of high-speed solenoid valves, based on multi-physics coupling analysis and PECA algorithm, significantly improved the switching efficiency of solenoid valves and reduced power loss. Ouyang et al.[17] proposed a shift strategy based on a robust tracking control algorithm, which significantly improved the shift quality of heavy vehicles and provided an effective solution for the shift control of automatic transmission of heavy vehicles.

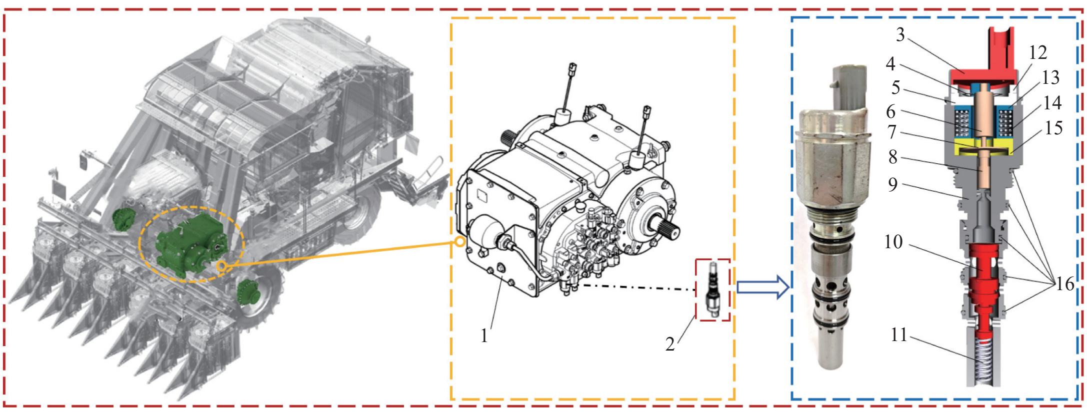
The existing research on solenoid valve optimization mostly focuses on traditional fields such as automobiles and heavy vehicles. The working environment of cotton pickers is complex, and there are load fluctuations during field picking and transportation. Its picking load has instantaneous impact characteristics with different cotton plant density, which will cause the picking ingot to break the cotton boll or knock down the cotton, which will greatly affect the cotton recovery rate and impurity rate. Therefore, higher requirements for the response speed and stability of the shift solenoid valve are put forward. However, there are currently relatively few systematic studies on specific application scenarios of cotton picker shift solenoid valves. Although some scholars use DOE and other methods to analyze the influence of solenoid valve structural parameters, they mostly focus on single or a few parameters, lacking a systematic analysis of the system response characteristics under the influence of multiple parameters.

This paper takes the shift proportional solenoid valve of the power shift transmission of the cotton picker as the research object. It firstly analyzes its structure and working principle, constructs the mathematical model of the shift valve, and then builds the simulation model of the electro-hydraulic shift system based on Amesim. The optimal Latin hypercube design method is used to screen the key parameters affecting the response characteristics of the solenoid valve. Taking the hydraulic step response time and overshoot as the optimization objectives, the key parameters are optimized by the NSGA-II algorithm, which improves the hydraulic response characteristics of the solenoid valve in the shift process and provides a reference for improving the shift quality of the cotton picker.

2 Analysis of proportional solenoid valve system and modeling of shift valve

2.1 Structural analysis of proportional solenoid valve

The proportional solenoid valve of the cotton picker studied in this paper is a two-stage valve system, which consists of a solenoid valve and a shift valve. This combination ensures the precise control and high efficiency of the system. The position and structure diagram of the proportional solenoid valve for the power shift transmission of the cotton picker is shown in Figure 1.



1. Power shift transmission 2. Proportional solenoid valve 3. Plastic end cap 4. Disc spring 5. Rear yoke 6. Armature 7. Magnetic separator 8. Putter 9. Housing 10. Valve core 11. Spring 12. Retaining ring 13. Coil skeleton 14. Coil 15. Front yoke 16. O-ring

Figure 1 Position and structure diagram of the proportional solenoid valve for the power shift transmission of the cotton picker

2.2 Analyzing proportional solenoid valve operation and creating a shift valve mathematical model

The proportional solenoid valve of the power shift transmission of the cotton picker studied belongs to the normally closed pilot-operated shift solenoid valve. When the solenoid valve is in the normal position, as shown in Figure 2, the solenoid valve core push rod remains in the left position under the action of the left spring. In this state, the main oil pressure inlet P_m of the shift valve is disconnected from the working oil port A, while the return oil port T_m is connected to the working oil port A. At this time, there is no oil flowing into the clutch piston chamber, and the main oil pressure of the shift valve flows from P_m to T_s for oil leakage. When the power shift transmission control unit (TCU) receives a shift command and sends a corresponding control signal, the solenoid valve is energized, as shown in Figure 3. At this time, the T_s port is closed, and the oil leakage has ended. The oil flows through the pilot oil hole P_s to the left chamber of the shift valve. As the oil continues to flow in, the pressure in the left chamber of the shift valve gradually increases. When the pressure in the left chamber of the shift valve is greater than the friction force, spring force, and feedback chamber pressure of the shift valve, the shift valve core is pushed, so that the P_m port and A port are connected, and the T_m port and A port are disconnected. The pressure in the clutch piston chamber gradually increases, pushing the clutch steel plate and friction plate, combining to achieve power transmission.

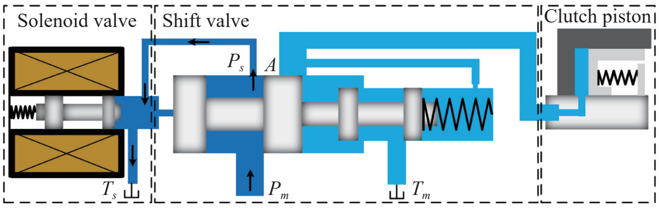


Figure 2 Powered off state of the proportional solenoid valve

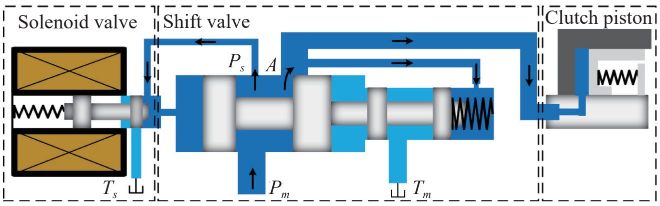


Figure 3 Proportional solenoid valve energized state

When the solenoid valve is energized, the shift valve core is mainly affected by the hydraulic pressure in the left chamber of the valve core, the hydraulic pressure in the feedback chamber, the hydraulic force, the viscous friction force, the inertia force, and the right return spring force. Assuming that the oil source is an ideal oil source and the oil passage leakage is ignored, the kinematics equation of the shift valve core is:

$$P_1A_1 - P_2A_2 - K_{spr}(x_v + x_0) = m\frac{d^2x_v}{dt^2} + C_v\frac{dx_v}{dt} + F_s + F_t \tag{1}$$

where, P_1 is the control oil pressure at the left end of the shift valve core, bar; A_1 is the effective area of the left end face of the shift valve core, m²; P_2 is the oil pressure in the feedback chamber, bar; A_2 is the effective area of feedback cavity, m²; K_{spr} is the stiffness coefficient of shift valve spring, N/m; x_v is the displacement of the shift valve core, m; x_0 is the initial compression of the shift valve spring, m; m is the mass of the shift valve core, kg; C_v is the viscous damping coefficient of the shift valve core, N·s/m; F_s is the steady-state hydrodynamic force, N; F_t is the transient hydrodynamic force, N.

For the shift valve slide valve structure, the calculation equation of the steady-state hydrodynamic force is:

$$F_s = C_d\omega x_v\sqrt{2\rho\Delta P}\cos\theta \tag{2}$$

where, C_d is the flow coefficient; ω is the valve port area gradient, m; x_v is the displacement of the valve core, m; ρ is the liquid density, kg/m³; ΔP is the pressure difference before and after the valve port, bar; θ is the jet angle, °.

The calculation equation of the transient hydrodynamic force is:

$$F_t = C_d\omega L\dot{x}_v\sqrt{2\rho\Delta P} \tag{3}$$

where, L is the actual flow of hydraulic oil in the valve body cavity, m; $\dot{x}_v$ is the spool speed, m/s.

The Laplace transform is performed on both sides of Equation (1):

$$\mathcal{L}\left[P_1A_1 - P_2A_2 - K_{spr}(x_v + x_0)\right] = \mathcal{L}\left[m\frac{d^2x_v}{dt^2} + C_v\frac{dx_v}{dt} + F_s + F_t\right] \tag{4}$$

The transformed equation is:

$$P_1A_1 - P_2A_2 - K_{spr}(X_v(s) + x_0) = ms^2X_v(s) + C_vsX_v(s) + F_s(s) + F_t(s) \tag{5}$$

Equation (2) is obtained by Laplace transformation:

$$F_s(s) = C_d\omega\sqrt{2\rho\Delta P}\cos\theta\cdot X_v(s) \tag{6}$$

The Laplace transform of Equation (3) is obtained:

$$F_t(s) = C_d\omega L\sqrt{2\rho\Delta P}\cdot sX_v(s) \tag{7}$$

By substituting Equations (6) and (7) into Equation (5), we can get:

$$\begin{aligned}P_1A_1 - P_2A_2 - K_{spr}X_v(s) - K_{spr}x_0 = ms^2X_v(s) + C_vsX_v(s) + \\ C_d\omega\sqrt{2\rho\Delta P}\cos\theta\cdot X_v(s) + C_d\omega L\sqrt{2\rho\Delta P}\cdot sX_v(s)\end{aligned} \tag{8}$$

The shift valve is divided into two states: oil filling and oil draining. In the oil filling state, the flow rate of the feedback chamber is:

$$Q_f = K_2(P_A - P_2) = \frac{V_2}{B_{eff}}sP_2 - S_2sX_v(s) \tag{9}$$

where, K_2 is the fluid conduction of the throttle port of the feedback chamber, m⁵/N·s; P_A is the pressure of the working oil port A， bar; P_2 is the pressure of the feedback chamber, bar; V_2 is the volume of the feedback chamber, m³; B_{eff} is the effective bulk modulus of the fluid, MPa; and S_2 is the area of the feedback chamber, m².

By bringing Equation (9) into Equation (8), we can get:

$$\begin{aligned}P_1A_1 - \frac{A_2K_2P_A}{K_2 + \dfrac{V_2}{B_{eff}}s} - K_{spr}x_0 = \Bigg(ms^2 + C_vs + C_d\omega\sqrt{2\rho\Delta P}\cos\theta + \\ C_d\omega L\sqrt{2\rho\Delta P}s + \frac{A_2S_2s}{K_2 + \dfrac{V_2}{B_{eff}}s} + K_{spr}\Bigg)X_v(s)\end{aligned} \tag{10}$$

The flow balance equation of the control chamber of the shift valve is:

$$K_m(P_m - P_A) + K_qX_v(s) = \frac{V_c}{B_{eff}}sP_A + Q_A + K_lP_A + K_2(P_A - P_2) \tag{11}$$

where, K_m is the fluid conductance of the main oil port of the shift valve, m⁵/N·s; K_q is the flow gain coefficient of the shift valve, m²/s; K_l is the leakage coefficient of the shift valve, m⁵/N·s; V_c is the total volume of the control pressure chamber, m³; and Q_A is the load flow of the working oil port, m³/s.

It is obtained from the solution of Equation (9):

$$P_2 = \frac{K_2P_A + S_2sX_v(s)}{K_2 + \dfrac{V_2}{B_{eff}}s} \tag{12}$$

After substituting Equation (12) into Equation (11), we can get:

$$\begin{aligned}(K_mP_m - Q_A)\left(\frac{sV_2}{B_{eff}K_2} + 1\right) + K_qX_v(s)\left[1 + \left(\frac{V_2}{B_{eff}K_2} + \frac{S_2}{K_q}\right)s\right] = \\ P_AK_{eff}\left[\left(\frac{sV_2}{B_{eff}K_2} + 1\right)\left(\frac{sV_c}{B_{eff}K_{eff}} + 1\right) + \frac{V_2}{V_c}\frac{sV_c}{B_{eff}K_{eff}}\right]\end{aligned} \tag{13}$$

When the shift valve is in the oil drain state, the movement direction of the valve core is opposite to that of the oil filling state, and the flow rate of the feedback chamber in the oil drain state is:

$$-Q_f = K_2(P_2 - P_A) = \frac{V_2}{B_{eff}} sP_2 + S_2 sX_v(s) \tag{14}$$

The flow balance equation of the drain port is:

$$K_2(P_2 - P_A) + K_q X_v(s) = \frac{V_c}{B_{eff}} sP_A - Q_A + K_l P_A + K_{T_1}(P_A - P_{T_1}) \tag{15}$$

where, K_{T1} is the fluid conduction of the oil drain port of the shift valve, m^5/N·s; P_{T1} is the oil pressure of the oil drain port of the shift valve, bar.

Collation available:

$$(K_{T_1}P_{T_1}+Q_A)\left(\frac{sV_2}{B_{eff}K_2}+1\right)+K_qX_v(s)\left[1+\left(\frac{V_2}{B_{eff}K_2}+\frac{S_2}{K_q}\right)s\right]= P_AK_{eff}\left[\left(\frac{sV_2}{B_{eff}K_2}+1\right)\left(\frac{sV_c}{B_{eff}K_{eff}}+1\right)+\frac{V_2}{V_c}\frac{sV_c}{B_{eff}K_{eff}}\right] \tag{16}$$

Therefore, the transfer function block diagram of the proportional solenoid shift valve can be obtained as shown in Figure 4.

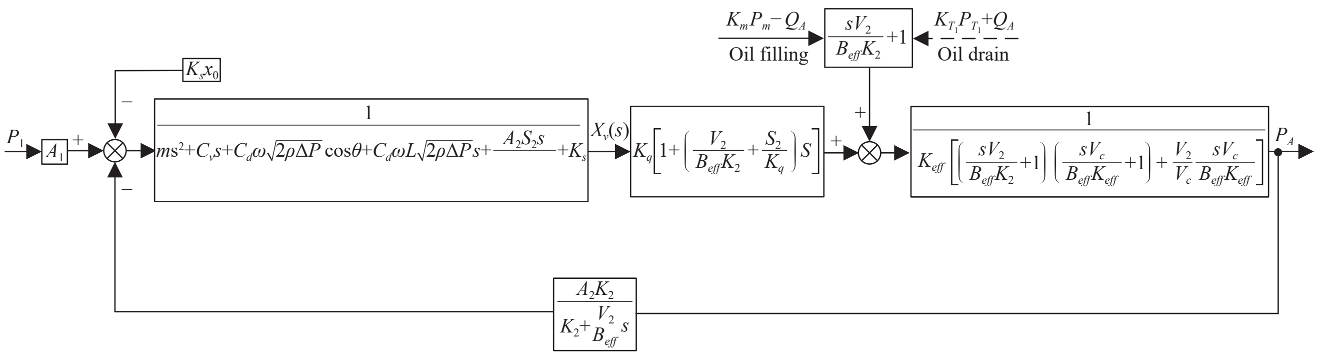


Figure 4 Transfer function block diagram of the shift valve

3 Simulation of proportional solenoid valve electro-hydraulic shift system in Amesim and bench test validation

The simulation model of the electro-hydraulic shift system of a proportional solenoid valve is constructed in Amesim based on the structure of the solenoid valve and the parameters of the shift hydraulic system of the power shift transmission[18]. This is illustrated in Figure 5 and consists of solenoid valve modules, shift valve modules, clutch piston modules, and oil supply system modules. Table 1 displays the primary parameters of the Amesim simulation model for the electro-hydraulic shift system with a proportional solenoid valve. The simulation time is set to 2 s, and the printing interval is set to 0.01 s.

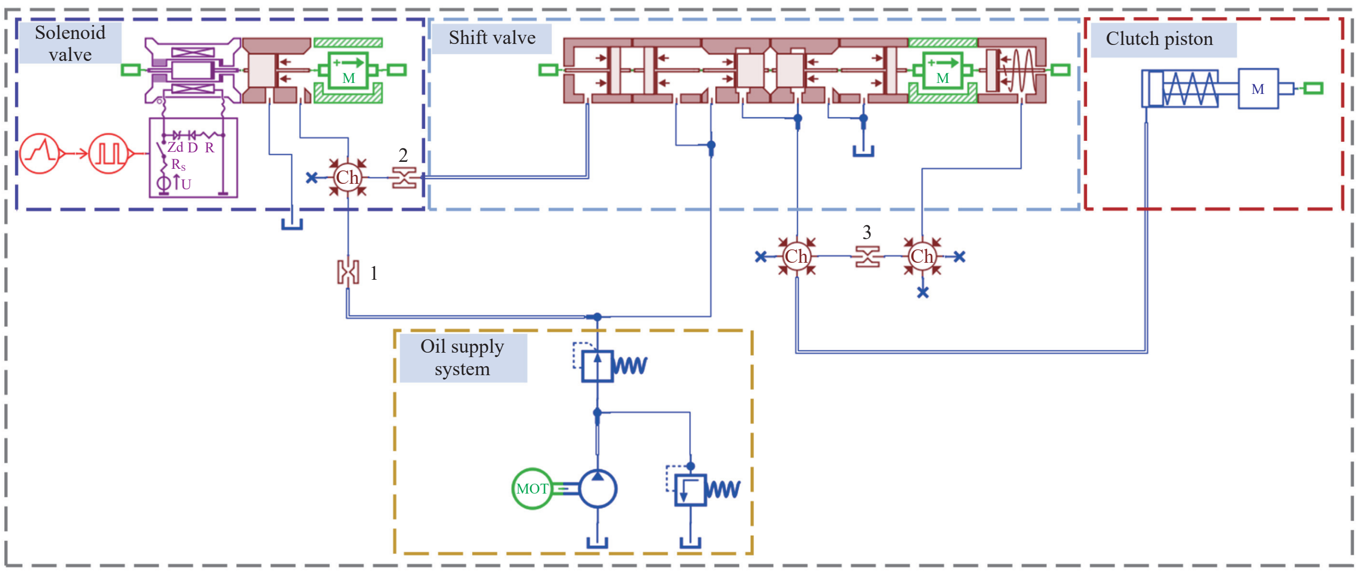


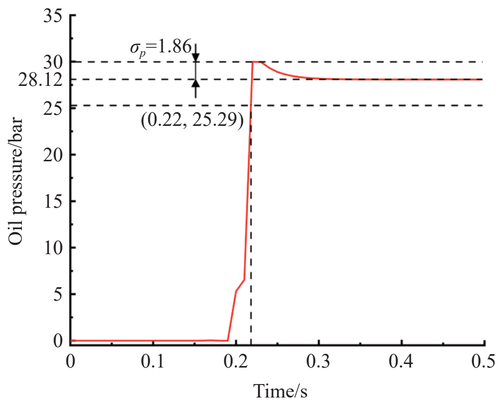
Figure 5 Proportional solenoid valve electro-hydraulic shift system Amesim simulation

Through simulation calculation, the dynamic response curve of oil pressure output of the shift valve is obtained, as shown in Figure 6. Figure 6a is the dynamic response curve of oil pressure output of the shift valve during the oil filling process. It can be seen that the dynamic response curve of oil pressure output of the shift valve during oil filling process basically conforms to the ideal oil filling pressure change curve during shift process, including rapid oil filling stage, buffer boost stage, step boost stage, and oil pressure stability stage[5], and the oil pressure of shift valve is stable at 28.12 bar. The step response time of oil pressure output and the overshoot of oil pressure response of the shift valve can characterize its rapidity and dynamic characteristics[3,4,19], so this paper selects these two indices as the performance evaluation indices of the proportional solenoid valve. It can be seen from Figure 6a that the step response time and oil pressure response overshoot of the proportional solenoid valve are 0.22 s and 1.86 bar, respectively.

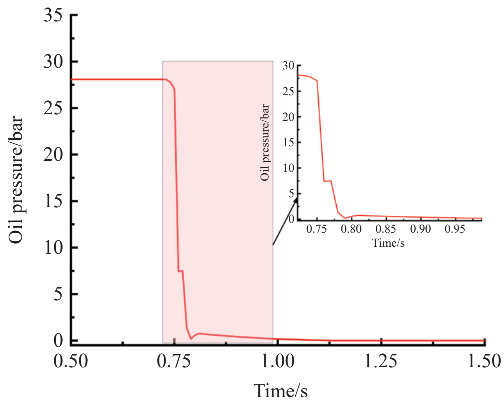
Figure 6b shows the dynamic response curve of the oil pressure output of the shift valve during the oil drainage process. The oil pressure during the oil drainage process drops rapidly and shows a decreasing trend, which reflects the efficient pressure relief characteristics of the system during the oil drainage stage, is adapted to the clutch separation action, and ensures the smooth conversion and system stability during shifting.

Table 1 Main Amesim parameters for the proportional solenoid valve electro-hydraulic shift system

Parameter	Symbols	Value	Parameter	Symbols	Value
Voltage/V	U	24	Hydraulic oil density/kg·m^{-3}	ρ	868
Resistance/Ω	R	7.5	Bulk modulus of elasticity/bar	b	17 000
Number of turns of coil/turn	N	700	Kinematic viscosity/cSt	v	77
Rated pressure/MPa	P_e	3	Main oil pressure/bar	op	30
Main pressure port diameter/mm	D_1	5.2	Orifice 1 diameter/mm	jlk1	0.9
Control oil port diameter/mm	D_2	3.1	Orifice 2 diameter/mm	jlk2	0.7
Push rod diameter of solenoid valve/mm	D_3	5.5	Orifice 3 diameter/mm	jlk3	0.4
Solenoid valve spool maximum displacement/mm	L	0.4	Control chamber diameter/mm	d_1	9.5
Oil pump speed/r·min^{-1}	n	1500	Feedback chamber diameter/mm	d_2	8
Oil pump displacement/cc·r^{-1}	V	100	Valve spool coverage/mm	zgl	−2.44
Solenoid valve spool weight/g	m_1	12.3	Spring force at zero displacement/N	sfzd	30
Shift valve spool weight/g	m_2	21	Spring compression at zero displacement/mm	scazd	3.5
Temperature/°C	T	40	Shift valve spring rate/N·mm^{-1}	sr	10



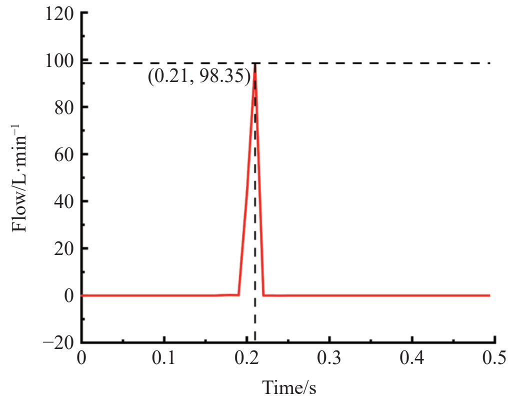
a. Oil-filled state



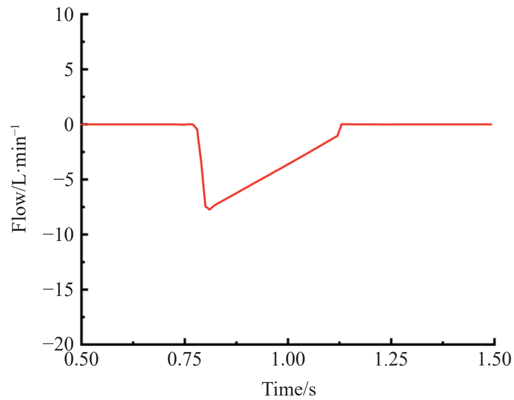
b. Oil-drained state

Figure 6 Dynamic response curve of oil pressure output of the shift valve

Figure 7 shows the dynamic response curve of the flow output of the shift valve. Figure 7a shows that the initial flow rate in the oil filling stage is 0, and it surges to the peak value of 98.35 L/min in 0.21 s, showing high transient flow characteristics, which can quickly establish the shift hydraulic condition[20]. When the clutch piston reaches the maximum displacement, the flow rate falls back to 0, and the system tends to be stable. During the oil leakage stage in Figure 7b, the flow rate fluctuates negatively for about 0.78 s (the positive or negative flow rate depends on the oil flow direction: the inflow is positive and the outflow is negative), the oil flows out of the piston chamber, the oil pressure of the shift valve decreases, and the clutch piston begins to separate.



a. Oil-filled state



b. Oil-drained state

Figure 7 Dynamic response curve of flow output of the shift valve

To verify the credibility of the simulation analysis results of the electro-hydraulic shift system of the proportional solenoid valve constructed in this paper, we actually measured the output oil pressure of the shift valve of the proportional solenoid valve in the shift process on the test bench of the power shift transmission of the cotton picker. The specific structure of the test bench is shown in Figure 8. Among them, the input end motor has a rated power of 22-24.5 kW, a rated current of 39 A, a rotational speed of 2950-3540 r/min, and a frequency of 50-60 Hz; the input speed and torque sensor has a range of 1000 N·m, an output signal with a square wave frequency of 0-12 V, a zero torque of 10 kHz, a forward full scale of 15 kHz, a reverse full scale of 5 kHz, and a load current <10 mA; the rotational speed and torque sensor at the output end has a rotational speed range of 0-4000 r/min, a temperature range of 0°C-55°C, and a rated torque of 2000 N·m; the magnetic powder brake has a rated torque of 500 N·m, an absorbed power of 110 kW, and an input speed of 3000 r/min; the rated power of the oil pump motor is 3 kW, the rated voltage is 380 V, the rated current is 6.8 A, the rated speed is 1430 r/min, and the frequency is 50 Hz; the rated pressure of the gear oil delivery pump is 2.5 MPa, and the rated flow rate is 2.1 m^3/h; the diameter of the relief valve is 10 mm, the rated flow rate is 40 L/min, and the pressure regulation range is ≤31.5 MPa; the pressure detection range of the oil pressure sensor is 0-20 MPa, the output signal is 4-20 mA, and the power supply voltage is 24 V; the temperature sensor detection range is –50-200°C, output signal is PT100, and supply voltage is 24 V.

The oil pressure output test curve of the shift valve is shown in Figure 9. It can be seen from the figure that the actual pressure change trend is consistent with the simulation curve as a whole, and the measured pressure in the stable stage is 28 bar, and the relative error between it and the simulation value is only 0.43%. This result fully confirms that the established model has high accuracy in describing the steady-state characteristics of the shift valve, which provides a reliable basis for subsequent dynamic characteristic analysis and model application. However, there are differences in details, and the test curve does not reflect the rapid oil filling process, which may be insensitive to the rapid change of oil pressure due to the long sampling period of 0.1 s of the oil pressure sensor. Overall, the consistency between the simulation and test results verifies the practical value of the simulation model in the performance analysis of the shift system.

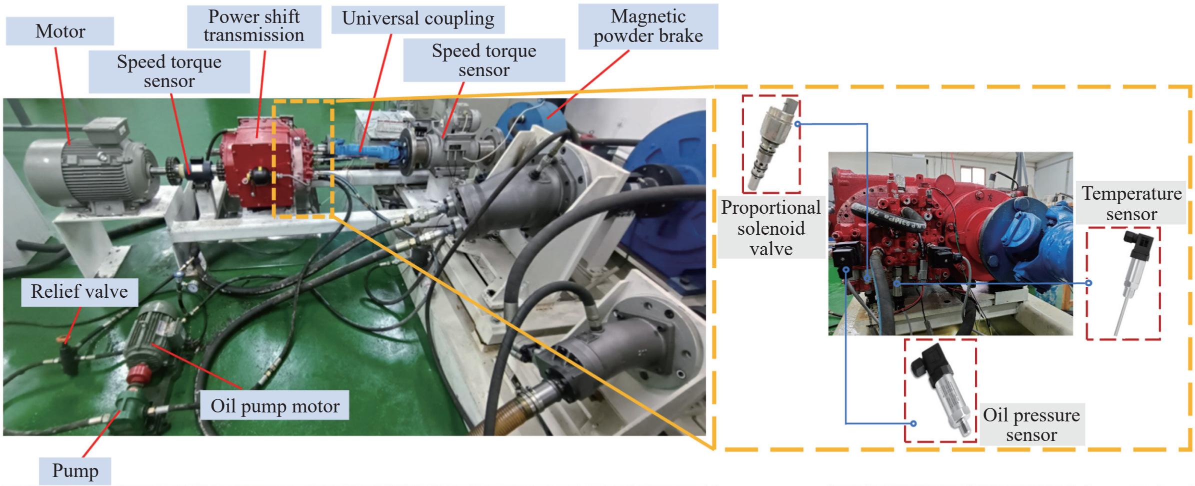
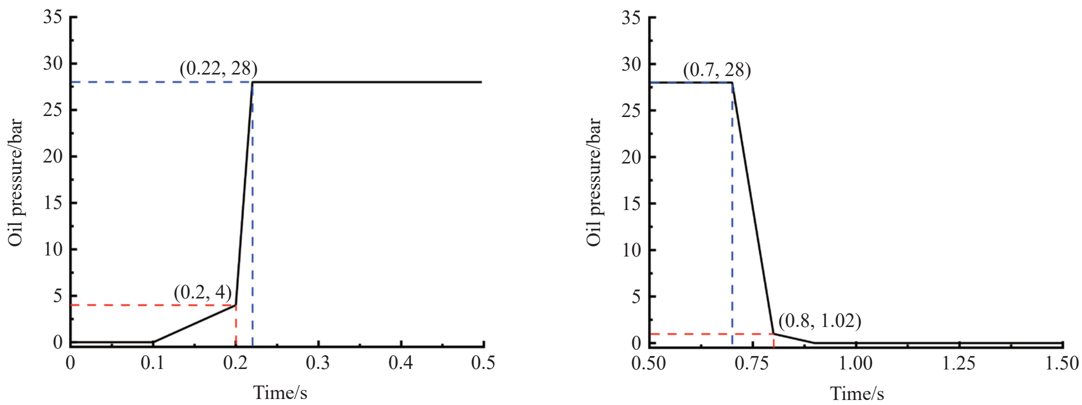


Figure 8 Power shift transmission test bench for cotton picker



a. Oil-filled state b. Oil-drained state

Figure 9 Oil pressure output test curve of the shift valve

4 Multi-objective optimization of proportional solenoid valve structural parameters

4.1 Key parameters of proportional solenoid valve structure

The optimization process of key parameters of proportional solenoid valve structure in this paper is shown in Figure 10, which mainly includes the framework construction of Isight-Amesim experimental design, the selection of experimental design method, the identification of key parameters, Isight-Amesim Optimization Design, the selection of optimization algorithm, and the obtaining of optimization results. The proportional solenoid valve has a lot of structural factors, and each one will have a big impact on the way it responds. In order to achieve optimal design, it is crucial to precisely determine the critical elements that significantly affect the proportional solenoid valve's response characteristics. DOE is a statistical method that can quickly determine the influence of different factors on the trial response[21,22]. DOE design methods include full factor design, partial factor design, optimal Latin hypercube design, etc.[23-25]. Among them, the optimal Latin hypercube test design method can obtain as accurate a response prediction as possible within a limited number of tests, and at the same time, all test points can be evenly distributed in the design space as much as possible[26,27]. Therefore, the optimal Latin hypercube design method was chosen to determine the structural parameters that have a significant impact on the response characteristics of proportional solenoid valves and to serve as a foundation for the optimization of these structural parameters. The range of design factors for the proportional solenoid valve's structural parameters is listed in Table 2.

The multi-disciplinary and multi-objective optimization analysis software Isight and Amesim software were used to realize the optimal Latin hypercube test design of proportional solenoid valve structural parameters. Three hundred test sample points of eight design variables of the proportional solenoid valve were collected to comprehensively evaluate the influence of each parameter variable on the performance of the proportional solenoid valve. Figure 11 is the distribution diagram of optimal Latin hypercube sampling points, and Figure 11a shows the structural parameter variables scazd, sfzd, and sr of the proportional solenoid valve randomly selected and presented in the form of a three-dimensional scatter diagram; Figure 11b shows the proportional solenoid valve's structural parameter variables op, scazd, and sr three-dimensional scatter plot. Figure 11 shows that the optimal Latin hypercube test design approach equally distributes sample points over the design space, demonstrating its superiority in parameter sampling.

The main purpose of the main effect plot is to visualize the extent to which each independent variable affects the dependent variable[28]. Figure 12 is the main effect diagram of different structural parameters of the proportional solenoid valve on the oil pressure output step response time t_s and oil pressure response overshoot σ_p of the shift valve. It can be seen from Figure 12 that the parameter scazd has the most significant influence on t_s, while the parameters op and jlk2 have relatively little influence on t_s. The influence of parameter jlk3 on σ_p is the most significant, and the influence of parameters sr and sfzd on σ_p is relatively small.

Pareto analysis is a problem-solving and decision-making approach that maximizes efficiency and effectiveness by identifying and focusing on the few key factors that have the greatest impact[29-31]. Figure 13 is the Pareto diagram of the structural parameters of the proportional solenoid valve to the step response time t_s and the overshoot σ_p of the oil pressure output of the shift valve. The contribution rate of the parameters scazd and sr to the step response time t_s of the oil pressure output of the shift valve is the most significant and negative. The contribution rate of parameters op, jlk2, and sfzd to the step response time t_s of oil pressure output of shift valve is relatively small, in which the contribution rate of parameters op and jlk2 is positive and that of sfzd is negative. Parameters jlk3 and op have the largest contribution rate to the hydraulic response overshoot σ_p of shift valve, in which the contribution rate of parameter jlk3 is negative and that of parameter op is positive. The contribution rate of parameters sr, sfzd, and jlk2 to the hydraulic response overshoot σ_p of shift valve is relatively small, in which the contribution rate of parameters sr and sfzd is positive and that of jlk2 is negative.

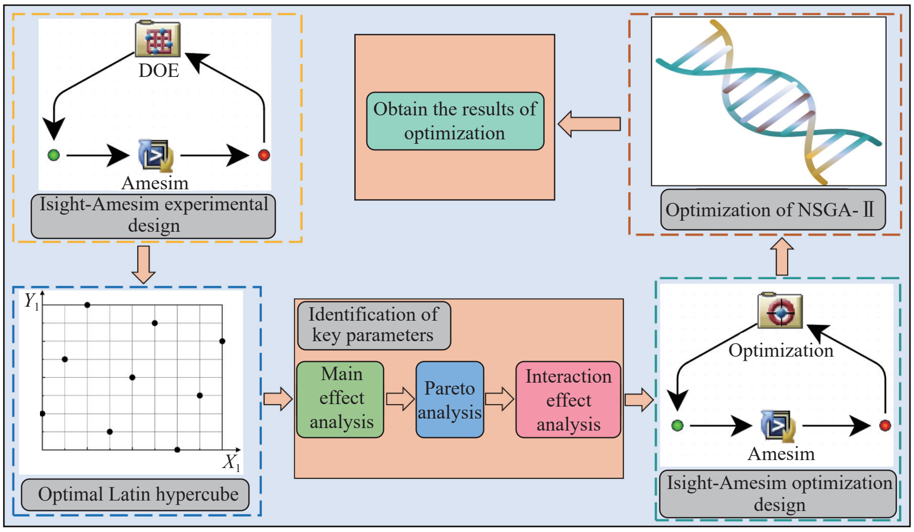


Figure 10 Optimization process of proportional solenoid valve structural parameters

According to the DOE analysis results, the multivariate quadratic regression equation of oil pressure step response time t_s can be fitted as follows:

$$\begin{aligned}t_s =\ & 0.030+0.030\mathrm{jlk1}+0.045\mathrm{jlk2}+0.094\mathrm{jlk3}-0.008\mathrm{op}+\\ & 0.002\mathrm{scazd}+8.778\mathrm{e}^{-4}\mathrm{sfzd}+9.574\mathrm{e}^{-4}\mathrm{sr}+0.012\mathrm{zgl}-0.003\mathrm{jlk1}^2-\\ & 0.003\mathrm{jlk2}^2-0.035\mathrm{jlk3}^2+1.988\mathrm{e}^{-4}\mathrm{op}^2-2.569\mathrm{e}^{-5}\mathrm{scazd}^2+\\ & 1.189\mathrm{e}^{-5}\mathrm{sfzd}^2+3.528\mathrm{e}^{-5}\mathrm{sr}^2-9.152\mathrm{e}^{-5}\mathrm{zgl}^2-0.029\mathrm{jlk1}\cdot\mathrm{jlk2}-\\ & 0.012\mathrm{jlk1}\cdot\mathrm{jlk3}+0.001\mathrm{jlk1}\cdot\mathrm{op}+0.001\mathrm{jlk1}\cdot\mathrm{scazd}-\\ & 3.881\mathrm{e}^{-4}\mathrm{jlk1}\cdot\mathrm{sfzd}-8.184\mathrm{e}^{-4}\mathrm{jlk1}\cdot\mathrm{sr}-0.004\mathrm{jlk1}\cdot\mathrm{zgl}-\\ & 0.003\mathrm{jlk2}\cdot\mathrm{jlk3}+0.002\mathrm{jlk2}\cdot\mathrm{op}-0.004\mathrm{jlk2}\cdot\mathrm{scazd}-\\ & 9.081\mathrm{e}^{-4}\mathrm{jlk2}\cdot\mathrm{sfzd}+0.001\mathrm{jlk2}\cdot\mathrm{sr}-0.003\mathrm{jlk2}\cdot\mathrm{zgl}-\\ & 7.524\mathrm{e}^{-4}\mathrm{jlk3}\cdot\mathrm{op}+0.001\mathrm{jlk3}\cdot\mathrm{scazd}+2.806\mathrm{e}^{-4}\mathrm{jlk3}\cdot\mathrm{sfzd}-\\ & 8.827\mathrm{e}^{-4}\mathrm{jlk3}\cdot\mathrm{sr}-0.001\mathrm{jlk3}\cdot\mathrm{zgl}-1.729\mathrm{e}^{-4}\mathrm{op}\cdot\mathrm{scazd}-\\ & 2.469\mathrm{e}^{-5}\mathrm{op}\cdot\mathrm{sfzd}-9.603\mathrm{e}^{-5}\mathrm{op}\cdot\mathrm{sr}-4.588\mathrm{e}^{-4}\mathrm{op}\cdot\mathrm{zgl}+\\ & 4.873\mathrm{e}^{-5}\mathrm{scazd}\cdot\mathrm{sfzd}+3.544\mathrm{e}^{-5}\mathrm{scazd}\cdot\mathrm{sr}+3.169\mathrm{e}^{-4}\mathrm{scazd}\cdot\mathrm{zgl}-\\ & 1.756\mathrm{e}^{-5}\mathrm{sfzd}\cdot\mathrm{sr}-3.951\mathrm{e}^{-5}\mathrm{sfzd}\cdot\mathrm{zgl}+2.247\mathrm{e}^{-4}\mathrm{sr}\cdot\mathrm{zgl}\end{aligned} \tag{17}$$

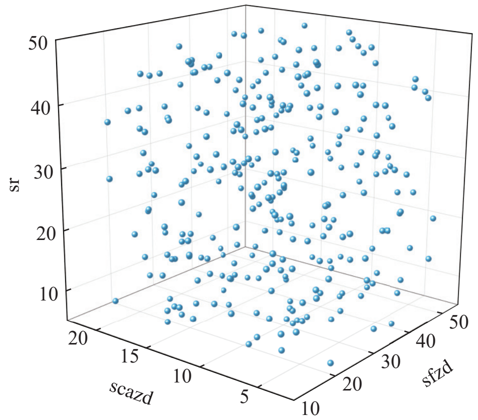
In the same way, the multivariate quadratic regression equation of the oil pressure response overshoot σ_p is fitted as:

$$\begin{aligned}\sigma_p =\ & 23.468+0.968\mathrm{jlk1}-1.865\mathrm{jlk2}+2.933\mathrm{jlk3}-1.849\mathrm{op}+\\ & 0.028\mathrm{scazd}-0.006\mathrm{sfzd}+0.079\mathrm{sr}+0.078\mathrm{zgl}+0.527\mathrm{jlk1}^2-\\ & 0.143\mathrm{jlk2}^2-0.835\mathrm{jlk3}^2+0.034\mathrm{op}^2-9.818\mathrm{e}^{-4}\mathrm{scazd}^2-\\ & 6.364\mathrm{e}^{-5}\mathrm{sfzd}^2+2.050\mathrm{e}^{-4}\mathrm{sr}^2+0.002\mathrm{zgl}^2-0.356\mathrm{jlk1}\cdot\mathrm{jlk2}-\\ & 0.752\mathrm{jlk1}\cdot\mathrm{jlk3}+0.003\mathrm{jlk1}\cdot\mathrm{op}+0.033\mathrm{jlk1}\cdot\mathrm{scazd}-\\ & 0.018\mathrm{e}^{-4}\mathrm{jlk1}\cdot\mathrm{sfzd}-0.035\mathrm{e}^{-4}\mathrm{jlk1}\cdot\mathrm{sr}-0.089\mathrm{jlk1}\cdot\mathrm{zgl}+\\ & 0.757\mathrm{jlk2}\cdot\mathrm{jlk3}+0.052\mathrm{jlk2}\cdot\mathrm{op}-0.036\mathrm{jlk2}\cdot\mathrm{scazd}+\\ & 0.003\mathrm{jlk2}\cdot\mathrm{sfzd}+0.022\mathrm{jlk2}\cdot\mathrm{sr}+0.019\mathrm{jlk2}\cdot\mathrm{zgl}-0.043\mathrm{jlk3}\cdot\mathrm{op}+\\ & 0.020\mathrm{jlk3}\cdot\mathrm{scazd}-0.001\mathrm{jlk3}\cdot\mathrm{sfzd}-0.019\mathrm{jlk3}\cdot\mathrm{sr}+\\ & 0.009\mathrm{jlk3}\cdot\mathrm{zgl}-0.002\mathrm{op}\cdot\mathrm{scazd}+0.001\mathrm{op}\cdot\mathrm{sfzd}-0.002\mathrm{op}\cdot\mathrm{sr}-\\ & 0.002\mathrm{op}\cdot\mathrm{zgl}-1.128\mathrm{e}^{-4}\mathrm{scazd}\cdot\mathrm{sfzd}+5.043\mathrm{e}^{-4}\mathrm{scazd}\cdot\mathrm{sr}+\\ & 0.003\mathrm{scazd}\cdot\mathrm{zgl}-2.079\mathrm{e}^{-4}\mathrm{sfzd}\cdot\mathrm{sr}-0.002\mathrm{sfzd}\cdot\mathrm{zgl}+0.002\mathrm{sr}\cdot\mathrm{zgl}\end{aligned} \tag{18}$$

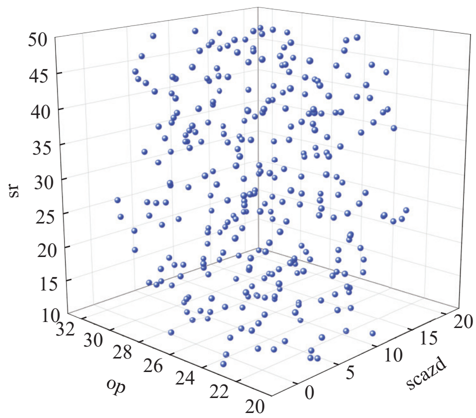
It can be seen that the analysis results of the main effect diagram of structural parameters of the proportional solenoid valve are consistent with those of the Pareto diagram, and the influence of parameter jlk2 and parameter sfzd on the step response time t_s of oil pressure output of shift valve and the overshoot σ_p of oil pressure response of shift valve is smaller than that of other structural parameters. Therefore, the influence of these two variables is not considered in the optimization process of structural parameters of the proportional solenoid valve in this paper, thus simplifying the optimization model and reducing the computational complexity. Figure 14a is the response surface diagram of the interaction between parameter scazd and parameter sr on the step response time t_s of the oil pressure output of the shift valve. When parameter scazd is a fixed value, with the increase of parameter sr, t_s shows a trend of first decrease and then rise. Figure 14b is the response surface diagram of the interaction between parameter jlk3 and parameter op on the oil pressure response overshoot σ_p of the shift valve. When parameter jlk3 is a fixed value, with the increase of parameter op, σ_p shows a trend of first decrease and then rise.

Table 2 Range of proportional solenoid valve design factors and structural parameters

Design variable name	Initial value	Range of change
op/bar	30	22-33
jlk1/mm	0.9	0.5-2
jlk2/mm	0.7	0.4-1.5
jlk3/mm	0.4	0.3-1.5
zgl/mm	–2.44	–5-6
sfzd/N	30	10-50
scazd/mm	3.5	2-20
sr/N·mm⁻¹	10	5-50

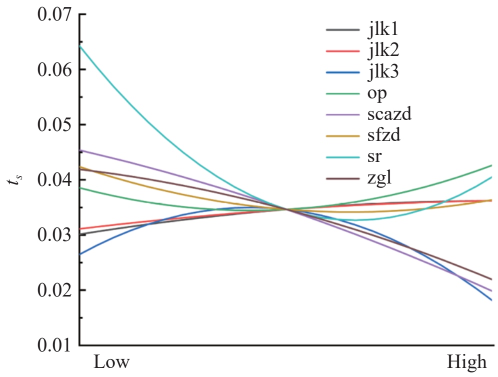


a. scazd, sfzd, sr

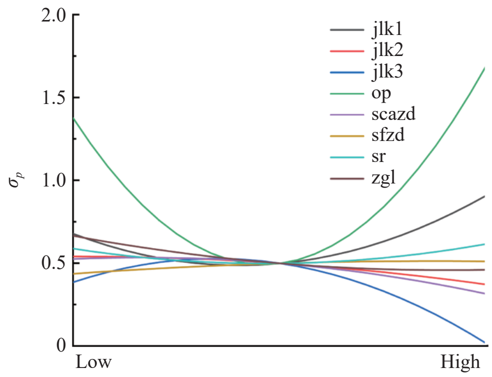


b. op, scazd, sr

Figure 11 Optimal Latin hypercube sampling 3D scatter plot

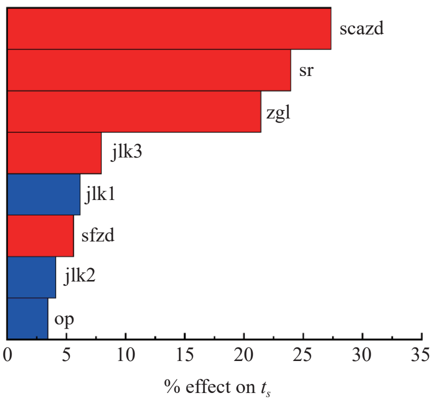


a. Step response time of oil pressure output of the shift valve

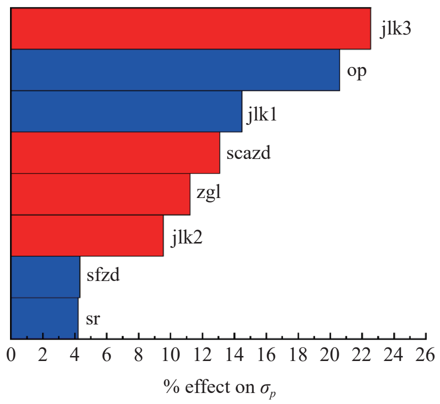


b. Hydraulic response overshoot

Figure 12 Main effect diagram of structural parameters of proportional solenoid valve

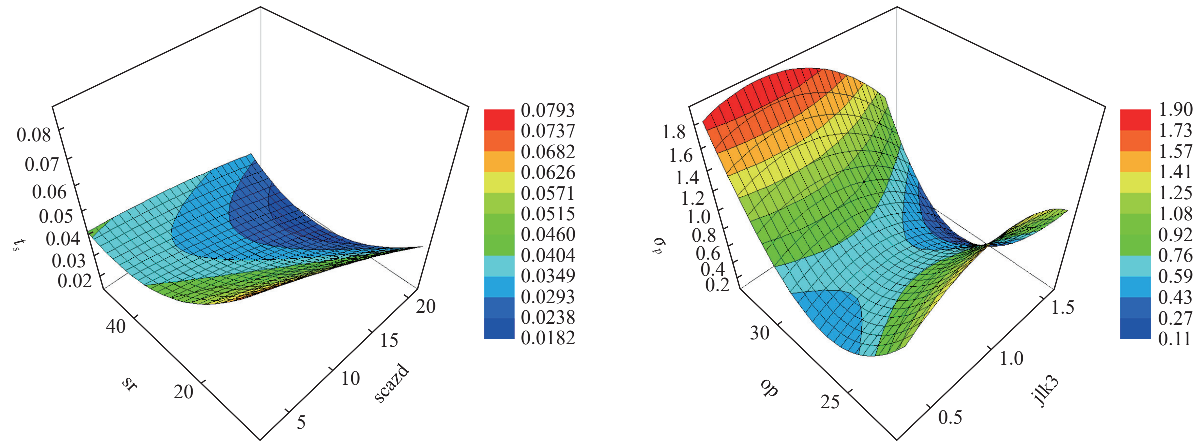


a. Pareto diagram of oil pressure step response time



b. Pareto diagram of oil pressure response overshoot

Figure 13 Pareto diagram



a. The parameters scazd and sr correspond to t_s

b. The parameters jlk3 and op correspond to σ_p

Figure 14 Response surface diagram

4.2 Optimization of structural parameters of proportional solenoid valve

1) Optimization variable selection

According to the identification and analysis results of key parameters of proportional solenoid valve structure in Section 4.1, parameter variables op, jlk1, jlk3, zgl, scazd, and sr are selected as optimization variables of proportional solenoid valve structural parameters.

2) Establishment of an optimization mathematical model

The performance evaluation index of the proportional solenoid valve is the oil pressure output step response time t_s and the oil pressure response overshoot σ_p of the shift valve, which belongs to a multi-objective optimization problem. The specific mathematical model can be expressed as:

$$\text{Objectives}: t_s = \text{minimize}, \ \sigma_p = \text{minimize} \tag{19}$$

$$\text{Constraints: } 22 \leq \text{opop} \leq 33 \tag{20}$$

$$\text{Variable:} \left\{ \begin{array}{c} 22 \leq \text{op} \leq 33 \\ 0.5 \leq \text{jlk1} \leq 2 \\ 0.3 \leq \text{jlk3} \leq 1.5 \\ -5 \leq \text{zgl} \leq 6 \\ 2 \leq \text{scazd} \leq 20 \\ 5 \leq \text{sr} \leq 50 \end{array} \right\} \tag{21}$$

where, t_s is the step response time of oil pressure output of shift valve, s; σ_p is the oil pressure response overshoot of the shift valve, bar; opop is the output oil pressure of the shift valve, bar; op is the main oil pressure, bar; jlk1 is the diameter of orifice 1, mm; jlk3 is the diameter of orifice 3, mm; zgl is the coverage amount of the shift valve core, mm; scazd is the spring compression at zero displacement, mm; sr is the shift valve spring stiffness, N/mm.

3) Optimize algorithm configuration

In multi-objective optimization problems, each objective usually conflicts with the others, resulting in no unique optimization solution, but a set containing multiple solutions, which is called the Pareto optimal solution set. The image formed by these solutions in the objective function space is called the Pareto front. The final solution of the multi-objective optimization problem needs to choose a compromise solution that can take into account each objective in the Pareto optimal solution set[32-34]. The concept of the Pareto solution is shown in Figure 15. The solution $x* \in X$ is called a Pareto optimal solution when there is no other solution $x \in X$ such that $f(x) < f(x*)$, meaning it is impossible to further improve the value of any objective function without deteriorating the values of other objective functions.

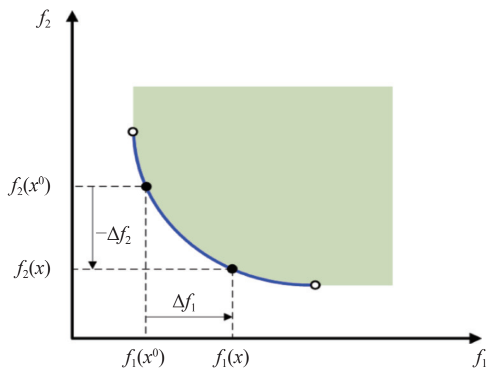


Figure 15 The concept of the Pareto solution

NSGA-II[35-37] is a genetic algorithm used to solve multi-objective optimization problems. It simulates natural selection and genetic principles, and balances the diversity and uniformity of solutions through non-dominated sorting and crowded distance selection mechanisms, to find an effective compromise between multiple objectives and consider multiple optimization objectives at the same time. Therefore, this paper selects the NSGA-II algorithm to optimize the parameters that affect the response characteristics of the proportional electromagnetic valve. The optimization process of the NSGA-II algorithm is shown in Figure 16.

Based on Isight and Amesim, a proportional solenoid valve parameter optimization design platform was created. The specific parameters of the NSGA-II algorithm in Isight software are set as follows: Population Size is 28; Number of Generations is 50; Crossover Probability is 0.9; Crossover distribution index is 20; Mutation distribution index is 20. The iterative optimization process of oil pressure output step response time t_s and oil pressure response overshoot σ_p of the shift valve is shown in Figure 17.

Based on the NSGA-II optimization algorithm, the optimal solution set of the key parameters of the proportional solenoid valve structure is obtained after 1402 runs of calculation. The results of some iterative data in the optimization process are listed in Table 3.

Figure 18 shows the Pareto optimal front of the proportional solenoid oil pressure output step response time t_s and oil pressure overshoot σ_p. The scatter points in the figure represent a series of solutions obtained in the multi-objective optimization process, which provide different trade-offs between the two objectives of reducing response time and controlling overshoot. Most of the

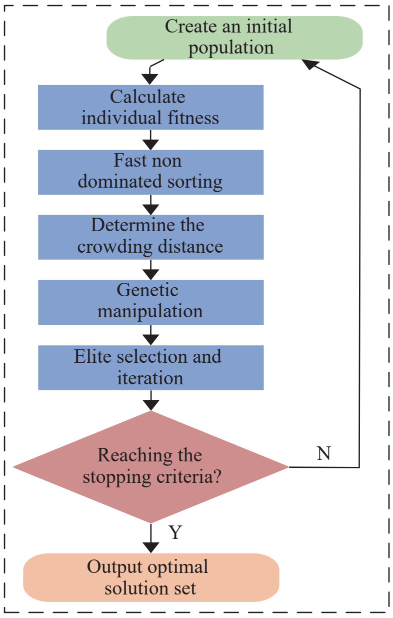


Figure 16 NSGA-II algorithm optimization process

Table 3 Partial running data sample information

Number	jlk1	jlk3	op	scazd	sr	zgl	t_s	σ_p
1	0.97	0.90	27.17	2.64	40.03	–1.87	0.24	4.94
2	1.89	1.00	30.45	3.86	9.39	3.23	0.22	0.73
3	0.71	0.57	32.49	5.94	17.22	–2.23	0.32	6.13
4	0.82	0.47	29.50	5.11	16.87	–1.79	0.27	7.40
5	1.96	0.62	32.38	10.93	22.07	1.79	0.11	1.31
6	0.82	0.34	26.05	4.89	25.86	1.77	0.23	1.12
…	…	…	…	…	…	…	…	…
1000	1.26	0.57	25.65	7.34	5.89	5.15	0.15	0.73
1001	1.32	1.22	30.74	8.93	14.41	1.69	0.21	0.32
…	…	…	…	…	…	…	…	…
1401	0.85	0.64	28.38	5.30	44.40	–3.40	0.18	1.89

solutions are concentrated in the lower region of σ_p, indicating that the system overshoot is small in these solutions, while the variation range of t_s is large, indicating that the optimization space of response time is wide. After a thorough examination of every Pareto optimal solution set, the best compromise solution is chosen as the ultimate multi-objective optimization solution for the proportional solenoid valve's structural parameters.

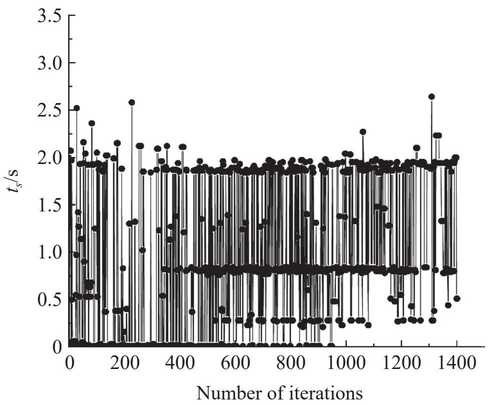
In order to intuitively display the optimization results, the data before and after the optimization of the structural parameter design variables of the proportional solenoid valve are listed in Table 4. After optimization, the main oil pressure op is 29.4 bar, the diameter of jlk1 is 0.85 mm, the diameter of jlk3 is 0.5 mm, the covering amount zgl of the shift valve core is –1.5 mm, the spring compression at zero displacement scazd is 3.1 mm, and the shift valve spring stiffness sr is 12 N/mm.

Table 4 Data comparison before and after proportional solenoid valve structural parameter design variable optimization

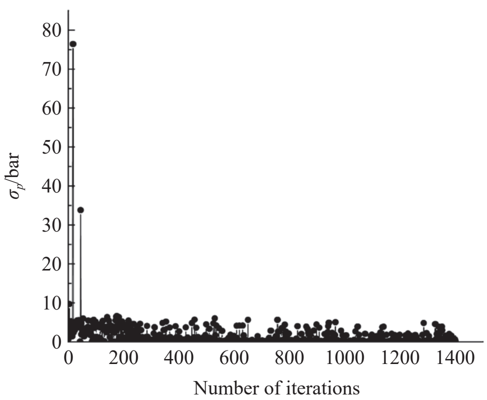
Design variable name	Initial value	Lower limit value	Upper limit value	Optimized value
op/bar	30	22	33	29.4
jlk1/mm	0.9	0.5	2	0.85
jlk3/mm	0.4	0.3	1.5	0.5
zgl/mm	–2.44	–5	6	–1.5
scazd/mm	3.5	2	20	3.1
sr/(N/mm)	10	5	50	12

4.3 Optimization verification of structural parameters of proportional solenoid valve

According to the optimization results of structural parameters of the proportional solenoid valve in Table 4, the values of each parameter obtained after multi-objective optimization are assigned to the Amesim simulation model of the electro-hydraulic shifting system of the proportional solenoid valve in Section 3 to verify the feasibility of the NSGA-II algorithm optimization. Figure 19 shows the dynamic response curve of the oil pressure output of the shift valve before and after optimization. It can be observed from Figure 19a that the dynamic response curve of the oil pressure output of the shift valve in the oil filling stage after optimization is consistent with the ideal oil filling pressure change curve in the shifting process, mainly including the following key stages: rapid oil filling stage, buffer boost stage, step boost stage, and oil pressure stability stage.



a. Oil pressure output step response time t_s



b. Oil pressure response overshoot σ_p

Figure 17 Iterative optimization process

In the rapid oil filling stage, the oil pressure rises rapidly to quickly fill the clutch cavity and eliminate the friction plate clearance. In the buffer boost stage, the oil pressure rise rate slows down, which is to prevent hydraulic impact and overshoot caused by excessive oil pressure rise and achieve smooth combination of friction plates. In the step boost stage, the oil pressure rises sharply to the system oil pressure in a short time, completing the rapid coupling of the clutch. During the oil pressure stabilization phase, the oil pressure reaches and maintains the target value, ensuring that the clutch maintains stable operation.

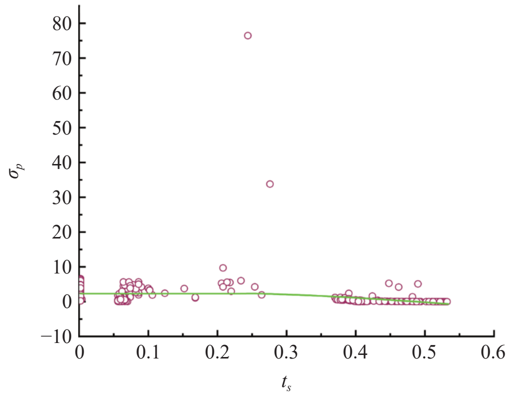
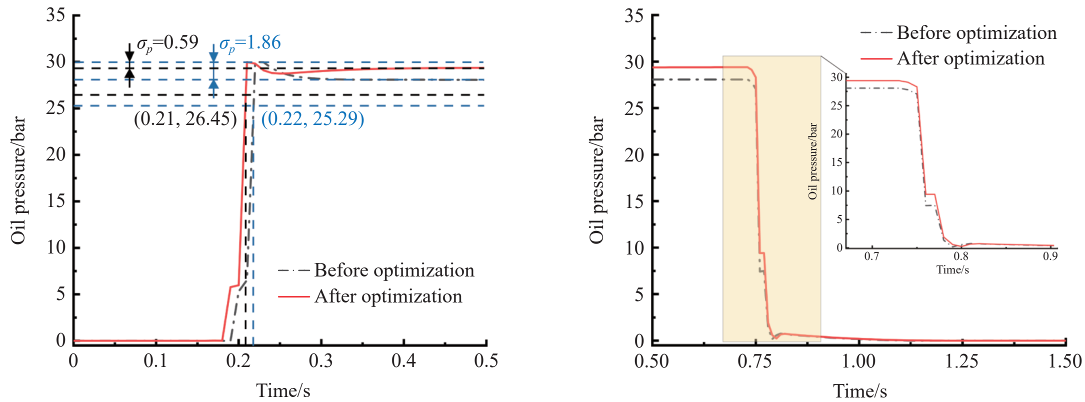


Figure 18 Oil pressure output step response time t_s and oil pressure overshoot σ_p Pareto optimal frontier solution set

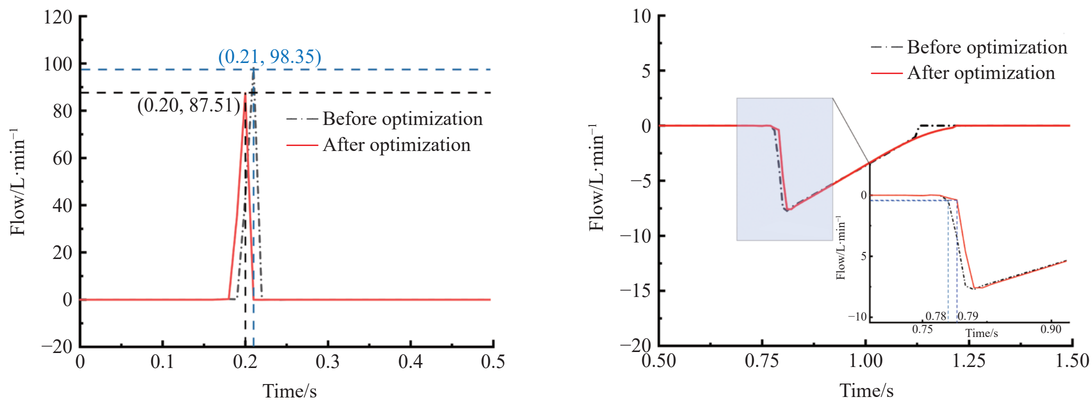
Figure 19b shows the oil pressure change curve of the shift valve before and after optimization in the oil drainage stage. The oil drainage time of the shift valve before and after optimization is basically the same, and the oil pressure drops rapidly from a higher pressure value in a short time, showing a certain decreasing trend.

After the optimization of the NSGA-II algorithm, the oil pressure step response time of the shift valve is shortened to 0.21 s, the oil pressure response overshoot is reduced to 0.59 bar, the output oil pressure step response time of the shift valve is shortened by 4.55%, and the oil pressure response overshoot is reduced by 68.28% compared with that before optimization. Figure 20 shows the dynamic response curve of the flow output of the shift valve before and after optimization. It can be seen that after the optimization of Figure 20a, the flow rate rapidly increases to a high enough value at the beginning of clutch coupling in the oil filling process, with a high transient flow rate to push the piston to move quickly and realize the rapid coupling of the clutch. When the clutch piston is completely coupled, the gap between the clutch friction plate and the steel plate disappears, and the clutch piston cavity flow rate is at a low level. After optimization, the flow response time during the oil filling process is 0.2 s, which is 4.76% shorter than that before optimization. After optimization in Figure 20b, at the beginning of the oil drainage process, the flow rate has a delay of 0.01 s, and the overall flow transition of the optimized system is smoother.



a. Oil-filled state b. Oil-drained state

Figure 19 Optimize the dynamic response curve of the oil pressure output before and after shift valves



a. Oil-filled state b. Oil-drained state

Figure 20 Dynamic response curve of flow output of the shift valve before and after optimization

5 Conclusions

1) Through the identification of key structural parameters of the proportional solenoid valve, the key parameters affecting its response characteristics have been studied and determined. This provides a basis for parameter sensitivity analysis in the multi-objective optimization of the proportional solenoid valve. Subsequently, an optimization model targeting response speed and control stability can be constructed to achieve optimal solutions for parameter combinations.

2) After multi-objective optimization, the step response time of the shift valve's output oil pressure has been shortened by 4.55% compared to before optimization, and the overshoot of oil pressure response has been reduced by 68.28%. The optimization results have improved the system's response speed and stability, positively impacting the reliability of the hydraulic system.

3) After optimization, the flow response time during the oil filling process has been shortened by 4.76% compared to before optimization, which enables the flow rate during the oil filling stage to meet the working requirements more quickly, thereby effectively reducing the time required to establish the conditions for shifting oil pressure.

There are some limitations in this study. The simulation model does not fully consider nonlinear factors such as oil temperature changes and component wear under actual working conditions. When establishing the kinematic equations for the shift valve, the study assumes an ideal oil source and ignores the leakage problem of the oil passage. In the complex working conditions of a real cotton picker, these factors may interfere with the actual dynamic characteristics of the system. Further research will be conducted in these two areas in the future.

Acknowledgements

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