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Simulation and experiment on characteristics of cartridge proportional lowering valve for electro-hydraulic lifter[1]

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Abstract. To improve the response characteristics and control precision of heavy-tractor electro-hydraulic hitch system, a cartridge proportional lowering valve of heavy tractor electro-hydraulic hitch was taken as the research object, and its internal structure characteristics and working principle were analyzed in full consideration of the valid displacement range of each hydraulic part, and a nonlinear mathematical model of the proportional lowering valve based on boundary conditions was developed using the state-space method. Then based on Matlab/Simulink, the static and dynamic performances of the lowering valve were discussed by using the fourth-order Runge-Kutta algorithm, and meanwhile, the experimental study on the proportional lowering valve was carried out on a closed-center load-sensing hydraulic test rig. By comparing with the dynamic simulation results, the average value of the steady flow error was less than 1.6 l/min, thus the correctness of the nonlinear mathematical model of the proportional lowering valve was proved. The results showed that the hysteresis error of steady flow rate of the lowering valve kept stable at around 3 % with differential pressure 4 MPa between the inlet and the outlet, thereof, the maximum flow could reach about 66 l/min. Moreover, the lowering proportional valve possessed good flow response characteristics when it was subjected to a step change in the load and the valve opening. Therefore, it can meet the performance requirements when used to control tillage depths of the mounted implements.

Key words. Proportional lowering valve, nonlinear modeling, boundary condition, dynamic performance, electro-hydraulic lifter.

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1. Introduction

As the core component of heavy-duty tractor electro-hydraulic lifts, the proportional drop valve controls the lowering movement of mounted implements during the plowing operation, to ensure the field work equality of tractor working unit. At present, the cartridge electro-hydraulic proportional control valve has been widely used in the field of agricultural machinery and engineering machinery, since it can effectively reduce hydraulic shock and system heating phenomenon (Dobrinska et al., 1981 [1], Lee et al., 2011 [2], Renn et al., 2005 [3]). Also, it has the advantages of strong anti-pollution ability, safety stability when power off, and easy to install and maintain. A wide range of studies have been carried out about the influence of the structural parameters of proportional control valve on the operating characteristics. Vaughan et al. (1996) [4], Canuto et al. (2014) [5] and Radpukdee et al. (2009) [6] established the mathematical model and simulation model of the proportional control valve based on the flow characteristics of different throttles, and obtained the influence of valve structure and performance parameters on the dynamic characteristics of flow in the opening and closing process. Liu et al. (2002) [7] established an EASY 5 model for a two position solenoid operated cartridge valve. And the model included the solenoid force, spring force, damping force, flow force and nonlinear mass flow rate, also it could be used to analyze cartridge valve as well as simulate system or controller performance. Opdenbosch et al. (2007) [8], Opdenbosch et al. (2013) [9] established a mathematical model used for analyzing the dynamic characteristics of the components of the proportional solenoid valve for the automatic transmission, and the coupling simulation was carried out, which could lay the foundation for optimizing the design of the solenoid valve. Zavarehi et al. (1999) [10] provided a step-by-step methodology for nonlinear modeling, parameter determination, model validation, and performance evaluation of solenoid-controlled pilot-operated spool valves. And the model considered such nonlinearities as pilot spool dead band, spool friction, flow coefficient variability, and leakage. Quan et al. (2010) [11] and Fu et al. (2008) [12] applied the displacement-flow feedback principle to the 3-position 4-way proportional directional valve, and established the mathematical model of the whole valve based on the structure and working principle.

2. Mathematical modeling

The internal structure of the valve is shown in Fig. 1.

2.1. Valve port pressure-flow equation

The flow of the orifice of the pilot spool inlet oil passage q_{LPV0} is given by the formula

$$q_{\mathrm{LPV0}} = \frac{C_{\mathrm{dPV0}} \cdot \pi d_{\mathrm{PV0}}^2}{4} \sqrt{\frac{2}{\rho} \left(p_{\mathrm{L}} - p_{\mathrm{LPV}} \right)}, \tag{1}$$

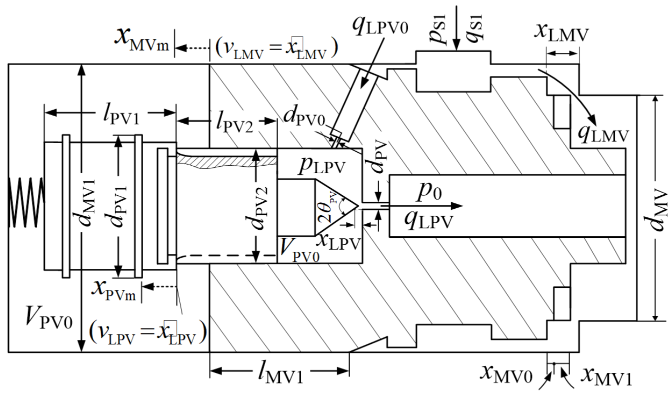

Fig. 1. Diagram of the internal structure and modeling analysis of proportional lowering valve

where C_{dPV0} is the flow coefficient of the orifice of the pilot spool inlet oil passage, ρ is the oil density, d_{PV0} is the diameter of the orifice of the pilot spool inlet oil passage, p_{L} is the inlet pressure of the main valve, and p_{LPV} is the inlet pressure of the pilot valve.

The flow of the pilot valve orifice q_{LPV} is given as

$$q_{\mathrm{LPV}} = C_{\mathrm{dPV}} A_{\mathrm{LPV}} \sqrt{\frac{2}{\rho}\left(p_{\mathrm{LPV}} - p_0\right)}, \quad x_{\mathrm{LPV}} - x_{\mathrm{LMV}} \geq 0\,, \tag{2}$$

where C_{dPV} is the flow coefficient of the pilot valve orifice, x_{LPV} is the displacement of the pilot valve spool, x_{LMV} is the displacement of the main valve spool, and A_{LPV} is the cross-section area of the pilot valve orifice that is given by the formula

$$A_{\mathrm{LPV}} \approx \pi d_{\mathrm{PV}} \left(x_{\mathrm{LPV}} - x_{\mathrm{LMV}}\right) \sin\theta_{\mathrm{PV}}\,. \tag{3}$$

Here, d_{PV} is the diameter of the orifice of the pilot valve seat and θ_{PV} is the half cone angle of the pilot valve spool.

The flow of the main valve orifice q_{LMV} is expressed by the formula

$$q_{\mathrm{LMV}} = C_{\mathrm{dMV}} A_{\mathrm{LMV}} \left(x_{\mathrm{LMV}}\right) \sqrt{\frac{2}{\rho}\left(p_{\mathrm{L}} - p_0\right)}\,, \tag{4}$$

where C_{dMV} is the flow coefficient of the main valve orifice, and A_{LMV} is the cross-section area of the main valve orifice given as

$$A_{\mathrm{LMV}} = \begin{cases} 0, & 0 \leq x_{\mathrm{LMV}} \leq x_{\mathrm{MV0}}, \\ w_{\mathrm{MV}}(x_{\mathrm{LMV}} - x_{\mathrm{MV0}}), & 0 \leq x_{\mathrm{LMV}} - x_{\mathrm{MV0}} \leq x_{\mathrm{MV1}}, \\ w_{\mathrm{MV}} x_{\mathrm{MV1}} + \pi d_{\mathrm{MV}}(x_{\mathrm{LMV}} - x_{\mathrm{MV0}} - x_{\mathrm{MV1}}), & x_{\mathrm{LMV}} > x_{\mathrm{MV0}} + x_{\mathrm{MV1}}. \end{cases} \tag{5}$$

Here, d_{MV} is the diameter of the orifice of the main valve, x_{MV0} is the overlap of the main valve opening, x_{MV1} is the maximum opening of the main valve small rectangular opening, and w_{MV} is the area gradient of the valve port when the main valve is of small rectangular opening. Its value is

$$w_{\mathrm{MV}} = nS = nd_{\mathrm{MV}} \arcsin(L/d_{\mathrm{MV}}) \,, \tag{6}$$

where n is the number of small rectangular openings in the main valve, L is the width of the small rectangular opening in the main valve, and S is the corresponding arc length of the width of the small rectangular opening in the main valve.

2.2. Flow continuity equation

The basic equation reads

$$q_{\mathrm{LPV0}} - q_{\mathrm{LPV}} = \left[V_{\mathrm{PV0}} - \frac{\pi d_{\mathrm{PV}}^2}{4} x_{\mathrm{LPV}} - \frac{\pi(d_{\mathrm{MV1}}^2 - d_{\mathrm{PV}}^2)}{4} x_{\mathrm{LMV}}\right] \dot{p}_{\mathrm{LPV}}/\beta_{\mathrm{e}} - \\ - \frac{\pi d_{\mathrm{PV}}^2}{4} \dot{x}_{\mathrm{LPV}} - \frac{\pi(d_{\mathrm{MV1}}^2 - d_{\mathrm{PV}}^2)}{4} \dot{x}_{\mathrm{LMV}} \,, \tag{7}$$

where β_{e} is the volume elastic modulus of oil, d_{MV1} is the diameter of main valve spool guide section, and V_{PV0} is the oil volume in the front chamber of pilot valve before the pilot valve closes.

The total oil inflow of the oil inlet of proportional drop valve q_{LL} is given by the formula

$$q_{\mathrm{LL}} = q_{\mathrm{LPV0}} + q_{\mathrm{LMV}} + \pi(d_{\mathrm{MV1}}^2 - d_{\mathrm{MV}}^2)\dot{x}_{\mathrm{LMV}}/4 \,. \tag{8}$$

2.3. Force balance equation of valve core

The balance of forces in the system is described by the equation

$$F_{\mathrm{LPVD}} - \pi d_{\mathrm{PV}}^2(p_{\mathrm{LPV}} - p_0)/4 = m_{\mathrm{PV}}\ddot{x}_{\mathrm{LPV}} + B_{\mathrm{PV}}\dot{x}_{\mathrm{LPV}} + B_{\mathrm{MPV}}(\dot{x}_{\mathrm{LPV}} - \dot{x}_{\mathrm{LMV}}) + \\ + k_{\mathrm{PV}}(x_{\mathrm{LPV}} + x_{\mathrm{PV0}}) + F_{\mathrm{LPVs}}, \quad x_{\mathrm{LPV}} - x_{\mathrm{LMV}} > 0, \quad x_{\mathrm{LPV}} < x_{\mathrm{PVm}} \,, \tag{9}$$

where the electromagnetic force F_{LPVD} acting on the pilot valve spool caused by proportional electromagnet is given as

$$F_{\mathrm{LPVD}} = (u - k_{\mathrm{v}}\dot{x}_{\mathrm{LPV}})k_{\mathrm{f}} \,. \tag{10}$$

Here, u is the driving voltage of proportional electromagnet, k_{f} is the driving coefficient of the electromagnet, k_{v} is the back electromotive force coefficient of velocity, m_{PV} is the total mass of pilot valve spool and its connectors, B_{PV} is the viscous damping coefficient of the movement of electromagnetic core, and B_{MPV} is the viscous damping coefficient of the movement of pilot valve spool relative to the main spool. These two damping coefficients are given by formulas

$$B_{\mathrm{PV}} = \pi\mu d_{\mathrm{PV1}} l_{\mathrm{PV1}}/\delta_{\mathrm{PV1}}\,, \tag{11}$$

where μ is the dynamic viscosity of hydraulic fluid, d_{PV1} is the diameter of pilot valve electromagnetic core, $_{\mathrm{PV1}}$ is the lead length of pilot valve electromagnetic core, and δ_{PV1} is the radial mating clearance between the pilot valve electromagnetic core and guide hole, and

$$B_{\mathrm{MPV}} = \frac{\pi\mu d_{\mathrm{PV2}} l_{\mathrm{PV2}}}{\delta_{\mathrm{PV2}}}(1 - \frac{x_{\mathrm{LPV}} - x_{\mathrm{LMV}}}{l_{\mathrm{PV2}}}) = B_{\mathrm{MPV0}}(1 - \frac{x_{\mathrm{LPV}} - x_{\mathrm{LMV}}}{l_{\mathrm{PV2}}})\,. \tag{12}$$

Further, in (9)–(12) δ_{PV2} is the radial mating clearance between the pilot valve spool and guide hole, k_{PV} is the spring stiffness of pilot valve, x_{PV0} is the pre-compressing of pilot valve spring, x_{PVm} is the maximum displacement of pilot valve spool and F_{LPVs} is the steady-state fluid force acted on the pilot spool that is given as

$$F_{\mathrm{LPVs}} = C_{\mathrm{dPV}} C_{\mathrm{vPV}} \pi d_{\mathrm{PV}} \left(x_{\mathrm{LPV}} - x_{\mathrm{LMV}}\right) \sin\left(2\theta_{\mathrm{PV}}\right) \left(p_{\mathrm{LPV}} - p_0\right)\,. \tag{13}$$

where C_{vPV} is the flow velocity coefficient of the orifice of pilot valve.

Another force equation is

$$\begin{cases} \frac{\pi(d_{\mathrm{MV1}}^2 - d_{\mathrm{MV}}^2)}{4} p_{\mathrm{L}} + \frac{\pi(d_{\mathrm{MV}}^2 - d_{\mathrm{PV}}^2)}{4} p_0 - \frac{\pi(d_{\mathrm{MV1}}^2 - d_{\mathrm{PV}}^2)}{4} p_{\mathrm{LPV}} = \\ = m_{\mathrm{MV}} \ddot{x}_{\mathrm{LMV}} + B_{\mathrm{MV}} \dot{x}_{\mathrm{LMV}} + B_{\mathrm{MPV}} \left(\dot{x}_{\mathrm{LMV}} - \dot{x}_{\mathrm{LPV}}\right) + F_{\mathrm{LMVs}}, \\ 0 \le x_{\mathrm{LMV}} \le x_{\mathrm{MVm}} \quad \text{and} \quad x_{\mathrm{LPV}} > x_{\mathrm{LMV}} \\ \\ \text{or} \\ \\ \frac{\pi(d_{\mathrm{MV1}}^2 - d_{\mathrm{MV}}^2)}{4} p_{\mathrm{L}} + \frac{\pi d_{\mathrm{MV}}^2}{4} p_0 - \frac{\pi d_{\mathrm{MV1}}^2}{4} p_{\mathrm{LPV}} + F_{\mathrm{LPVD}} = \\ \left(m_{\mathrm{PV}} + m_{\mathrm{MV}}\right) \ddot{x}_{\mathrm{LMV}} + \left(B_{\mathrm{PV}} + B_{\mathrm{MV}}\right) \dot{x}_{\mathrm{LMV}} + k_{\mathrm{PV}} \left(x_{\mathrm{LMV}} + x_{\mathrm{PV0}}\right) + F_{\mathrm{LMVs}}, \\ 0 \le x_{\mathrm{LMV}} \le x_{\mathrm{MVm}} \quad \text{and} \quad x_{\mathrm{LPV}} = x_{\mathrm{LMV}}. \end{cases} \tag{14}$$

Here, x_{MVm} is the maximum displacement of main valve spool and B_{MV} is the viscous damping coefficient of the movement of main valve spool given by the formula

$$B_{\mathrm{MV}} = \pi\mu d_{\mathrm{MV1}} l_{\mathrm{MV1}}/\delta_{\mathrm{MV1}}, \tag{15}$$

where, l_{MV1} is the seal length of main valve spool guide section, δ_{MV1} is the radial mating clearance between main spool and guide hole, and F_{LMVs} is the steady-state fluid force acted on the main spool. Its value is

$$F_{\mathrm{LMVs}} = \begin{cases} 0, \quad \text{for} \quad 0 \le x_{\mathrm{LMV}} \le x_{\mathrm{MV0}}, \\ 2C_{\mathrm{dMV}} C_{\mathrm{vMV}} w_{\mathrm{MV}} (x_{\mathrm{LMV}} - x_{\mathrm{MV0}})(p_{\mathrm{L}} - p_0) \cos\theta_{\mathrm{MV}} \\ \qquad \text{for} \quad 0 < x_{\mathrm{LMV}} - x_{\mathrm{MV0}} \le x_{\mathrm{MV1}}, \\ 2C_{\mathrm{dMV}} C_{\mathrm{vMV}} [w_{\mathrm{MV}} x_{\mathrm{MV1}} + \pi d_{\mathrm{MV}} (x_{\mathrm{LMV}} - x_{\mathrm{MV0}} - x_{\mathrm{MV1}})] \cdot \\ \cdot (p_{\mathrm{L}} - p_0) \cos\theta_{\mathrm{MV}} \\ \qquad \text{for} \quad x_{\mathrm{MV0}} + x_{\mathrm{MV1}} < x_{\mathrm{LMV}} \le x_{\mathrm{MVm}}. \end{cases} \tag{16}$$

Here, C_{vMV} is the flow velocity coefficient of the orifice of main valve and θ_{MV} is the jet angle of the orifice of main valve.

2.4. State equation of proportional lowering valve

According to the above established mathematical model, the chosen state variables are $x_1 = p_{\mathrm{LPV}}$, $x_2 = x_{\mathrm{LPV}}$, $x_3 = x_{\mathrm{LMV}}$, $x_4 = v_{\mathrm{LPV}}$, $x_5 = v_{\mathrm{LMV}}$, and the state equations are obtained:

$$\begin{cases} \dot{x}_1 = \frac{4\beta_{\mathrm{e}}}{4V_{\mathrm{PV0}} - \pi d_{\mathrm{PV}}^2 x_2 - \pi\left(d_{\mathrm{MV1}}^2 - d_{\mathrm{PV}}^2\right)x_3} \cdot \\ \quad \cdot [C_{\mathrm{dPV0}} \frac{\pi d_{\mathrm{PV0}}^2}{4} \sqrt{\frac{2}{\rho}\left(p_{\mathrm{L}} - x_1\right)} - C_{\mathrm{dPV}} \pi d_{\mathrm{PV}} \left(x_2 - x_3\right) \sin\theta_{\mathrm{PV}} \sqrt{\frac{2}{\rho}\left(x_1 - p_0\right)} + \\ \quad \frac{\pi d_{\mathrm{PV}}^2}{4} x_4 + \frac{\pi\left(d_{\mathrm{MV1}}^2 - d_{\mathrm{PV}}^2\right)}{4} x_5], \\ \dot{x}_2 = x_4, \\ \dot{x}_3 = x_5, \\ \dot{x}_4 = \begin{cases} 0 & x_2 = x_{\mathrm{PVm}} \text{ and } f_{v\mathrm{LPV}} > 0, \\ f_{v,\mathrm{LMV}} & x_2 = x_3 \text{ and } f_{v\mathrm{LPV}} < f_{v\mathrm{LMV}}, \\ f_{v\mathrm{LPV}} & \text{else,} \end{cases} \\ \dot{x}_5 = \begin{cases} 0 & x_3 = 0 \text{ and } f_{v\mathrm{LMV}} < 0, \text{ or } x_3 = x_{\mathrm{MVm}} \text{ and } f_{v\mathrm{LMV}} > 0, \\ f_{v\mathrm{LMV}} & \text{else.} \end{cases} \end{cases} \tag{17}$$

In the above equation

$$\begin{cases} f_{v\mathrm{LPV}} = \frac{1}{m_{\mathrm{PV}}}[F_{\mathrm{LPVD}} - \frac{\pi d_{\mathrm{PV}}^2}{4}\left(x_1 - p_0\right) - B_{\mathrm{MPV0}}\left(1 - \frac{x_2 - x_3}{l_{\mathrm{PV2}}}\right)\left(x_4 - x_5\right) - \\ \quad - B_{\mathrm{PV}} x_4 - k_{\mathrm{PV}}\left(x_2 + x_{\mathrm{PV0}}\right) - C_{\mathrm{dPV}} C_{\mathrm{vPV}} \pi d_{\mathrm{PV}} \left(x_2 - x_3\right) \sin\left(2\theta_{\mathrm{PV}}\right)\left(x_1 - p_0\right)], \\ f_{v\mathrm{LMV}} = \begin{cases} \frac{1}{m_{\mathrm{MV}}}[\frac{\pi(d_{\mathrm{MV1}}^2 - d_{\mathrm{MV}}^2)}{4} p_{\mathrm{L}} + \frac{\pi(d_{\mathrm{MV}}^2 - d_{\mathrm{PV}}^2)}{4} p_0 - \frac{\pi(d_{\mathrm{MV1}}^2 - d_{\mathrm{PV}}^2)}{4} x_1 - \\ \quad - B_{\mathrm{MV}} x_5 - B_{\mathrm{MPV0}}\left(1 - \frac{x_2 - x_3}{l_{\mathrm{PV2}}}\right)\left(x_5 - x_4\right) - F_{\mathrm{LMVs}}] \text{ for} \\ \quad 0 \le x_3 \le x_{\mathrm{MVm}} \text{ and } x_2 > x_3, \\ \frac{1}{m_{\mathrm{PV}} + m_{\mathrm{MV}}}[\frac{\pi\left(d_{\mathrm{MV1}}^2 - d_{\mathrm{MV}}^2\right)}{4} p_{\mathrm{L}} + \frac{\pi d_{\mathrm{MV}}^2}{4} p_0 - \frac{\pi d_{\mathrm{MV1}}^2}{4} x_1 + F_{\mathrm{LPVD}} - \\ \quad - \left(B_{\mathrm{PV}} + B_{\mathrm{MV}}\right) x_5 - k_{\mathrm{PV}}\left(x_3 + x_{\mathrm{PV0}}\right) - F_{\mathrm{LMVs}}] \text{ for} \\ \quad 0 \le x_3 \le x_{\mathrm{MVm}} \text{ and } x_2 = x_3. \end{cases} \end{cases} \tag{18}$$

The corresponding boundary conditions are: if $x_3 < 0$, then $x_3 = 0$. If $x_3 > x_{\mathrm{MVm}}$, then $x_3 = x_{\mathrm{MVm}}$. If $x_2 < x_3$, then $x_2 = x_3$. If $x_2 > x_{\mathrm{PVm}}$, then $x_2 = x_{\mathrm{PVm}}$. If $x_3 = 0$ and $x_5 < 0$ or $x_3 = x_{\mathrm{MVm}}$ and $x_5 > 0$, then $x_5 = 0$. If $x_2 = x_3$ and $x_4 < x_5$, then $x_4 = x_5$. Finally, if $x_2 = x_{\mathrm{PVm}}$ and $x_4 > 0$, then $x_4 = 0$.

2.5. Steady-state characteristic equation of proportional lowering valve

When the pilot valve orifice of the proportional lowering valve is closed, the orifice of pilot valve tends to open due to the driving force generated by the oil pressure at both ends of main spool and the steady-state fluid force it receives. Therefore, the close state of the orifice of pilot valve can only be instantaneous existence and cannot keep steady. When the proportional lowering valve is working in the steady state, the orifice of the pilot valve is opened, $\dot{p}_{\rm LPV}=0$, $\dot{x}_{\rm LPV}=\dot{x}_{\rm LMV}=0$, $\ddot{x}_{\rm LPV}=\ddot{x}_{\rm LMV}=0$.

From equation (7), it can be obtained that $q_{\rm LPV0}=q_{\rm LPV}$, thus

$$p_{\rm LPV}=\begin{cases} p_{\rm L}, & x_{\rm LPV}=0\\ \dfrac{\left(\frac{C_{\rm dPV0}d^2_{\rm PV0}}{4}\right)^2 p_{\rm L}+[C_{\rm dPV}d_{\rm PV}(x_{\rm LPV}-x_{\rm LMV})\sin\theta_{\rm PV}]^2p_0}{\left(\frac{C_{\rm dPV0}d^2_{\rm PV0}}{4}\right)^2+[C_{\rm dPV}d_{\rm PV}(x_{\rm LPV}-x_{\rm LMV})\sin\theta_{\rm PV}]^2}, & x_{\rm LPV}>0. \end{cases} \tag{19}$$

From the formulas (14) and (16), it can be calculated that

$$\frac{\pi\left(d^2_{\rm MV1}-d^2_{\rm MV}\right)}{4}p_{\rm L}+\frac{\pi\left(d^2_{\rm MV}-d^2_{\rm PV}\right)}{4}p_0-\frac{\pi\left(d^2_{\rm MV1}-d^2_{\rm PV}\right)}{4}p_{\rm LPV}=F_{\rm LMVs}\,. \tag{20}$$

1) When

$$0<x_{\rm LMV}\le x_{\rm MV0}\quad\text{or}\quad 0\le x_{\rm LPV}\le x_{\rm MV0}+\frac{C_{\rm dPV0}d^2_{\rm PV0}}{4C_{\rm dPV}d_{\rm PV}\sin\theta_{\rm PV}}\sqrt{\frac{d^2_{\rm MV}-d^2_{\rm PV}}{d^2_{\rm MV1}-d^2_{\rm MV}}}$$

then

$$x_{\rm LMV}=\begin{cases} 0\quad\text{for}\quad x_{\rm LPV}\le\frac{C_{\rm dPV0}d^2_{\rm PV0}}{4C_{\rm dPV}d_{\rm PV}\sin\theta_{\rm PV}}\sqrt{\frac{d^2_{\rm MV}-d^2_{\rm PV}}{d^2_{\rm MV1}-d^2_{\rm MV}}},\\[2ex] x_{\rm LPV}-\frac{C_{\rm dPV0}d^2_{\rm PV0}}{4C_{\rm dPV}d_{\rm PV}\sin\theta_{\rm PV}}\sqrt{\frac{d^2_{\rm MV}-d^2_{\rm PV}}{d^2_{\rm MV1}-d^2_{\rm MV}}}\quad\text{for}\\ <x_{\rm LPV}-\frac{C_{\rm dPV0}d^2_{\rm PV0}}{4C_{\rm dPV}d_{\rm PV}\sin\theta_{\rm PV}}\sqrt{\frac{d^2_{\rm MV}-d^2_{\rm PV}}{d^2_{\rm MV1}-d^2_{\rm MV}}}\le x_{\rm MV0}. \end{cases} \tag{21}$$

2) When $0<x_{\rm LMV}-x_{\rm MV0}\le x_{\rm MV1}$, then

$$x_{\rm LPV}-x_{\rm LMV}=\frac{C_{\rm dPV0}d^2_{\rm PV0}}{4C_{\rm dPV}d_{\rm PV}\sin\theta_{\rm PV}}\cdot$$

$$\cdot\sqrt{\frac{\frac{\pi(d^2_{\rm MV}-d^2_{\rm PV})}{4}+2C_{\rm dMV}C_{\rm vMV}w_{\rm MV}\left(x_{\rm LMV}-x_{\rm MV0}\right)\cos\theta_{\rm MV}}{\frac{\pi(d^2_{\rm MV1}-d^2_{\rm MV})}{4}-2C_{\rm dMV}C_{\rm vMV}w_{\rm MV}\left(x_{\rm LMV}-x_{\rm MV0}\right)\cos\theta_{\rm MV}}}\,. \tag{22}$$

3) When $x_{\mathrm{MV0}} + x_{\mathrm{MV1}} < x_{\mathrm{LMV}} \leq x_{\mathrm{MVm}}$, then

$$x_{\mathrm{LPV}} - x_{\mathrm{LMV}} = \frac{C_{\mathrm{dPV0}} d_{\mathrm{PV0}}^2}{4 C_{\mathrm{dPV}} d_{\mathrm{PV}} \sin\theta_{\mathrm{PV}}} \sqrt{\frac{\frac{\pi(d_{\mathrm{MV}}^2 - d_{\mathrm{PV}}^2)}{4} + W}{\frac{\pi(d_{\mathrm{MV1}}^2 - d_{\mathrm{MV}}^2)}{4} - W}}\,, \tag{23}$$

where

$$W = 2 C_{\mathrm{dMV}} C_{\mathrm{vMV}} \left[w_{\mathrm{MV}} x_{\mathrm{MV1}} + \pi d_{\mathrm{MV}} (x_{\mathrm{LMV}} - x_{\mathrm{MV0}} - x_{\mathrm{MV1}}) \right] \cos\theta_{\mathrm{MV}}\,.$$

By formula (8), the oil flow flowing out from hydraulic cylinder when the proportional lowering valve works in a steady state can be obtained in the form

$$q_{\mathrm{LL}} = q_{\mathrm{LPV0}} + q_{\mathrm{LMV}} = \frac{C_{\mathrm{dPV0}} \pi d_{\mathrm{PV0}}^2}{4\sqrt{1 + \left(\frac{C_{\mathrm{dPV0}} d_{\mathrm{PV0}}^2}{4 C_{\mathrm{dPV}} d_{\mathrm{PV}} (x_{\mathrm{LPV}} - x_{\mathrm{LMV}}) \sin\theta_{\mathrm{PV}}} \right)^2}} \sqrt{\frac{2}{\rho}(p_{\mathrm{L}} - p_0)} +$$

$$+ \begin{cases} 0 \quad \text{for} \quad 0 \leq x_{\mathrm{LMV}} \leq x_{\mathrm{MV0}}, \\ C_{\mathrm{dMV}} w_{\mathrm{MV}} (x_{\mathrm{LMV}} - x_{\mathrm{MV0}}) \sqrt{\frac{2}{\rho}(p_{\mathrm{L}} - p_0)} \quad \text{for} \quad 0 \leq x_{\mathrm{LMV}} - x_{\mathrm{MV0}} \leq x_{\mathrm{MV1}}, \\ C_{\mathrm{dMV}} \left[w_{\mathrm{MV}} x_{\mathrm{MV1}} + \pi d_{\mathrm{MV}} (x_{\mathrm{LMV}} - x_{\mathrm{MV0}} - x_{\mathrm{MV1}}) \right] \sqrt{\frac{2}{\rho}(p_{\mathrm{L}} - p_0)} \\ \text{for} \quad x_{\mathrm{LMV}} > x_{\mathrm{MV0}} + x_{\mathrm{MV1}}. \end{cases} \tag{24}$$

3. Simulation analysis

The custom function module of Function in MATLAB/Simulink is used to establish the simulation model of the proportional lowering valve.

Given the driving voltage of 3.8 V, load pressure of 0 MPa, and the simulation time of 20 s, the dynamic response characteristic curve of the flow rate of proportional lowering valve, when the load pressure steps from 5 to 10 MPa, is shown in Fig. 2, upper part. As is shown in Fig. 2, upper part, when the load pressure is 5 MPa, the flow of proportional lowering valve stabilizes at about 21.5 l/min after rapid adjustment. With a step change of the load pressure to 10 MPa, the system flow increases rapidly and reaches 28.5 l/min after stabilization. The adjustment time is about 0.2 s, and overshoots about 7 %, which can meet the requirements of the steady-state flow output characteristic of proportional lowering valve when the electro-hydraulic lifter of heavy-duty tractor is in the field operating environment. Given the load pressure of 10 MPa, oil return pressure of MPa, and simulation time of 20 s, the dynamic response characteristic curve of the flow rate of proportional lowering valve, when the driving voltage steps from 3.4 to 3.8 V, is shown in Fig. 2, bottom part. As is shown in Fig. 2, bottom part, when the driving voltage is 3.4 V, the flow of proportional lowering valve stabilizes at about 5 l/min after rapid adjustment. As the driving voltage increases, the main spool opening increases. With a step change of the driving voltage to 3.8 V, the system flow increases rapidly and reaches 22.5 l/min after stabilization. The adjustment time is about 0.15 s, and overshoots about 6.7 %, which can meet the requirements of the dynamic speed regulation characteristic of the proportional lowering valve.

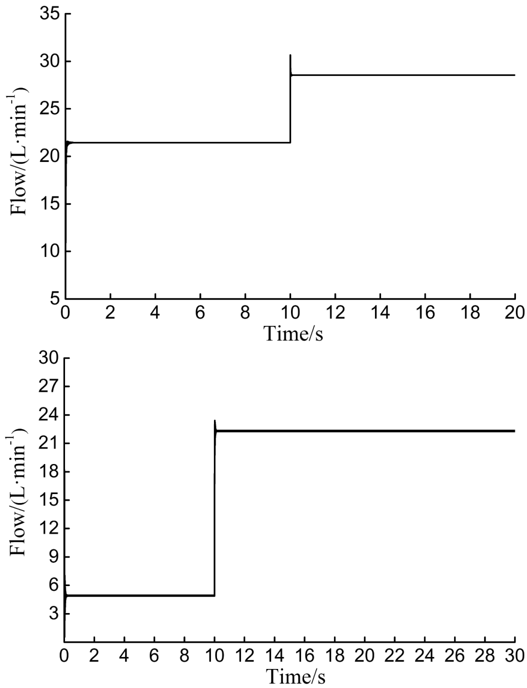

Fig. 2. Flow rate of proportional lowering valve under a step change in drive voltage: up–step change in load, bottom–step change in driving voltage

4. Test of the proportional lowering valve

The indoor test platform of proportional lowering valve load-sensing hydraulic system is constructed.

The throttle valve is connected to the outlet of load-sensing variable pump and used to regulate the flow rate of the hydraulic system. A proportional relief valve is set in parallel to the throttle valve to provide the required load pressure for the proportional lowering control valve. Meanwhile, through the throttle valve outlet the load pressure can be fed back to the flow control valve of load-sensing variable pump for pressure compensation. The safety relief valve is used for overload protection of the hydraulic system and the opening pressure of the valve is 20 MPa. Moreover, the opening pressure of the proportional relief valve and the valve core opening of the proportional lowering valve can be timely controlled by electrohydraulic proportional controller and proportional control amplifier, respectively.

By setting the input voltage of proportional lowering valve to 3.8 V, and the opening pressure of proportional relief valve controlled by electrohydraulic proportional controller stepping from 5 to 10 MPa at the time of 10 s, the dynamic response characteristic curves of the return oil pressure of proportional lowering valve, load pressure and outlet pressure of load-sensing pump are shown in Fig. 3, upper part, while the system flow characteristic curve is shown in Fig. 3, bottom part.

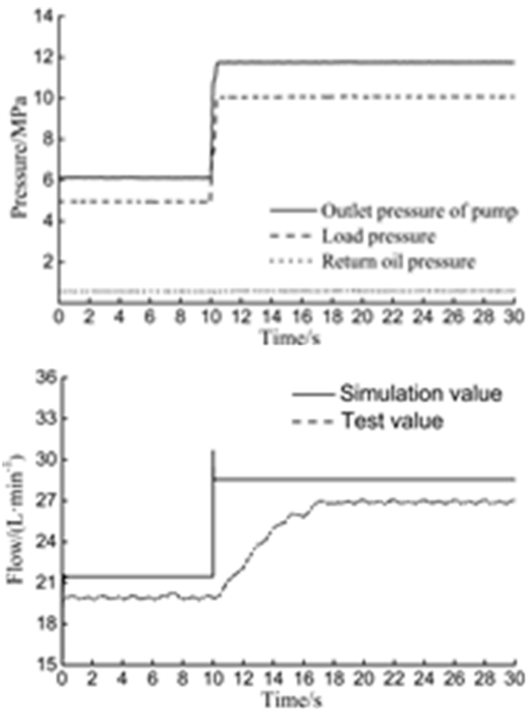

Fig. 3. Dynamic response curves under a step change in load of proportional lowering valve: up–pressure, bottom–flow

As can be seen from Fig. 3, upper part, the return oil pressure of proportional lowering valve nearly maintains at 0.5 MPa. When the load pressure steps from 5 to 10 MPa, the pressure built and pressure relief time of the system is about 0.5 s, meanwhile, the load pressure and the outlet pressure of pump change almost simultaneously during the process. Besides, the compensation pressure of load-sensing variable pump increases with the increasing of system flow and line pressure loss, averaging approximately 1.5 MPa. It can be seen from Fig. 3, bottom part, that the system flow increases from 20.3 to 27 l/min in a parabolic manner when the load makes a step change, and the adjusting time is about 6.5 s. Comparing with the simulation curves, it can be seen that with the increase of system flow, the internal leakage of pipeline and variable pump increases, and the steady flow error increases from 1.2 to 1.5 l/min, which satisfies the requirements of the proportional lowering valve output flow with load changing when the electrohydraulic lifter operates in the field.

By setting the opening pressure of proportional lowering valve to 10 MPa, and the input voltage of the valve stepping from 3.4 to 3.8 V at the time of 10 s, the dynamic response characteristic curves of the return oil pressure of proportional lowering valve, load pressure and outlet pressure of load-sensing pump are shown in Fig. 4, upper part, the system flow characteristic curve is shown in Fig. 4, bottom part.

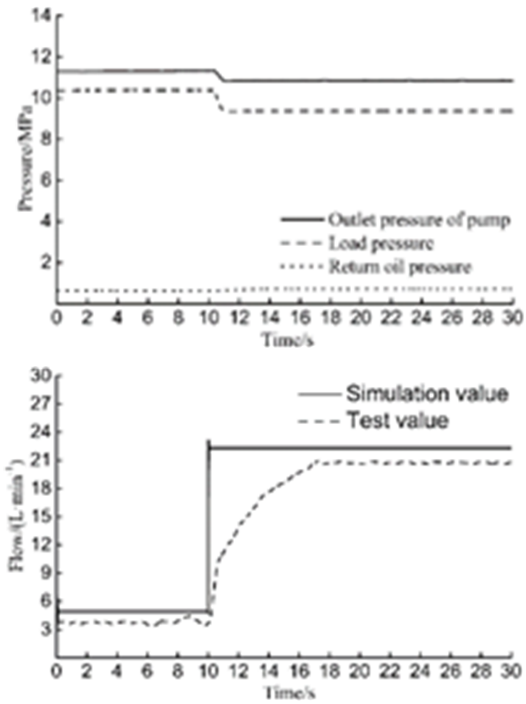

Fig. 4. Dynamic response curves under step change in valve opening of proportional lowering valve: up–pressure, bottom–flow

As can be seen from Fig. 4, upper part, the return oil pressure of proportional lowering valve nearly maintains at 0.7 MPa when the valve spool opening makes a step change. The load pressure, affected by the fluctuation of the system flow, makes a step change from 9.5 to 10.3 MPa. Meanwhile, the pressure built and pressure relief time of the system is about 1 s, the load pressure overshoot is less than 1.9 %, and the average compensation pressure is about 1.2 MPa. From Fig. 4, bottom part, it can be seen that the system flow, affected by the spool opening, makes a step change from 4 to 21 l/min and the adjustment time is about 7 s. Comparing with the simulation curves, it can be seen that with the increase of system flow, the internal leakage of pipeline and variable pump increases, and the steady flow error increases from 1 to 1.5 l/min or so, which satisfies the requirements of the dynamic speed regulation characteristic of proportional lowering valve when the electrohydraulic lifter operates in the field.

5. Conclusion

After full consideration of the effective travel ranges of the internal hydraulic components in proportional lowering valve of the electro-hydraulic lifter, a nonlinear mathematical model of the valve is established based on the boundary conditions, which makes a much more detailed description of the working characteristics of the valve.

The simulation results of the proportional lowering valve show that, with the increasing of pilot spool lift amount, the throttle opening of the main spool becomes larger and the spool moves smoothly, thereby the hydraulic impact is small. Besides, the displacement difference between the two spools is mainly determined by factors such as the diameter of the pilot valve inlet orifice, the diameter of the pilot valve seat orifice, the diameter of the main valve orifice and the diameter of the guide part of main valve spool. And the value of the displacement difference is almost the same, the main spool follows the movement of the pilot spool and the system flow can be indirectly controlled by the pilot valve, so the system has a good load speed regulation characteristic.

By comparing the test and simulation results of the dynamic characteristics of proportional lowering valve, the average error of the steady-state value of the system flow is less than 1.6 l/min, which verifies the correctness of the nonlinear mathematical model of the proportional lowering valve. Due to the factors such as the inertia of the load-sensing variable pump mechanism, the internal leakage and the oil compression of the hydraulic system pipeline, the overall inertia of the hydraulic system increases, resulting in slow system flow response. When the input voltage keeps constant, the proportional lowering valve can adjust the steady-state output flow in real-time as the load pressure changes. Besides, when the pressure difference between the two ends of the valve port remains unchanged, the dynamic speed regulation can be achieved by changing the input voltage. And this can meet the control requirements of the proportional lowering valve during the process of the electro-hydraulic lifter operating in the field.

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