Study on Simulation of System Dynamic Characteristics of Hydraulic Scissor Lift Based on Load-Sensing Control Technology

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Abstract. In view of the large energy consumption of crane hydraulic system with quantitative pump, load-sensitive control technology can be used to solve this problem. Firstly, the transfer function of the system is deduced by the method of principle analysis. Then a comparative study is carried out based on digital simulation technology. The dynamic characteristics of load-sensitive hydraulic system are analyzed comprehensively.

1. Introduction
The scissor lift is a commonly used lifting equipment in engineering. It is designed by connecting rod mechanism and has compact structure, as shown in Figure 1. Because the hydraulic transmission system has the characteristics of balanced transmission, large power-weight ratio and easy to operate, the scissor lift mostly uses hydraulic pressure as the power component.

At present, the hydraulic transmission system of scissor lift mostly adopts the quantitative pump oil supply system. Because the flow rate of the pump is constant, it causes a large overflow loss, so the energy consumption and efficiency of the system are relatively high. In order to solve this problem, load-sensitive pump can be used instead of quantitative pump, so that the pump can change its displacement according to the change of external load, and realize dynamic matching of power. This paper is based on digital simulation technology to analyze and study the application of load sensing technology to scissor lift.

2. Load sensing control principle
The structure of the load sensing pump is shown in Figure 2. Among them, 1 is a variable pump, 2 is a variable hydraulic cylinder, 3 is a pressure cut-off valve, 4 is a load-sensitive control valve, P_L control valve is a load feedback pressure, P_S control valve is the output pressure of the pump.
Figure 1. Mechanism sketch of scissor lift  
Figure 2. Structural of Load-sensing Pump

The spring of the pressure cut-off valve sets the maximum pressure of the system. The output pressure of the pump is connected to the left side of the load-sensitive control valve, and the feedback pressure of the load is connected to the right side. At the same time, the spring on the right side sets a fixed value to maintain a fixed pressure difference between the output pressure and the load pressure of the pump. Through the dynamic balance of the three, the opening of the load-sensitive valve is controlled, the piston displacement of the variable hydraulic cylinder is further controlled, and the variable mechanism of the pump is finally controlled to realize the dynamic adjustment of displacement.

3. Mathematical Model of Load-sensing Pump  
The structure and characteristics of the system can be qualitatively understood by constructing the system transfer function of the load-sensing pump. The mathematical modeling process is as follows.

[1]

Force balance equation of load-sensing valve core is

\[(p_S - p_L)A_v - K_vx_0 = M_s \frac{d^2x}{dt^2} + B \frac{dx}{dt} + K_vx_v\]  

Where,  \(p_S\) is pump output pressure, MPa; \(p_L\) is load feedback pressure, MPa; \(A_v\) is control area of load-sensing valve, \(\text{cm}^2\); \(K_v\) is Spring stiffness, N/cm; \(x_0\) is initial displacement of spring, cm; \(M_s\) is spool quality, kg; \(B\) is equivalent viscous damping coefficient, Ns/cm; \(x_v\) is Spool displacement, cm.

Transfer function of load-sensitive valve spool obtained by Laplace transformation of equation (1).

\[W_v(s) = \frac{x_v(s)}{E(s)} = \frac{1/K_v}{s^2 / \omega_n^2 + 1}\]  

Where, \(E(s)\) is pressure deviation signal, \(E(s) = [p_S(s) - p_L(s)]A_v\); \(\omega_n\) is natural frequency of load-Sensing valve, \(\omega_n = \sqrt{K_v / M_s}\).

Flow equation of load-sensing valve is

\[Q = c_d \omega x_v \sqrt{\frac{2\Delta p}{\rho}}\]  

Where, \(c_d\) is flow coefficient, \(\text{cm}^3/(\text{s} \cdot \text{MPa})\); \(\omega\) is area gradient of load-sensing valve port; \(\Delta p\) is pressure difference, MPa; \(\rho\) is hydraulic oil density, kg/m\(^3\).
Flow gain coefficient of load-sensing valve is

\[
K_q = \begin{cases} 
  \frac{\partial Q}{\partial x_v} = c_d \omega x_v \sqrt{\frac{2(p_2 - p_1)}{\rho}} & x_v \geq 0 \\
  c_d \omega x_v & x_v \leq 0 
\end{cases}
\] (4)

Pressure gain coefficient of load-sensing valve is

\[
K_p = \begin{cases} 
  -\frac{\partial Q}{\partial \Delta p} = \frac{c_d \omega x_v}{\sqrt{2(p_3 - p_1) \rho}} & x_v \leq 0 \\
  \frac{c_d \omega x_v}{\sqrt{2(p_2) \rho}} & x_v \geq 0 
\end{cases}
\] (5)

Where, \( p_1 \) is maximum cavity pressure of variable cylinder piston in left shift; \( p_2 \) is maximum cavity pressure of variable cylinder piston in right shift.

When the demand of load flow decreases, the flow linearization equation of load-sensing valve is

\[
q_{v1} = K_q x_v - K_p p_1 \quad \text{(When the demand of load flow decreases)}
\]

\[
q_{v1} = K_q x_v - K_p p_2 \quad \text{(When the demand of load flow increases)}
\] (6)

The dynamic equation of the swashplate is

\[
(p_3 A_1 - p_2 A_2) r_0 = \frac{J}{r_0} \frac{d^2 x_p}{dt^2} \quad \text{(When the swing angle of the pump increases)}
\]

\[
(p_1 A_1 - p_3 A_2) r_0 = \frac{J}{r_0} \frac{d^2 x_p}{dt^2} \quad \text{(When the swing angle of the pump decreases)}
\] (7)

Where, \( A_1 \) is large cavity section area of variable cylinder, cm²; \( A_2 \) is large cavity section area, cm²; \( r_0 \) is distance from variable cylinder axis to swashplate axis, cm; \( J \) is moment of inertia, kg·cm²; \( x_p \) is piston displacement of variable cylinder, cm.

When the swing angle of the swashplate becomes smaller, the flow continuity equation of the load sensing valve is

\[
q_{v2} = A_1 \frac{dx_p}{dt} + \frac{V}{\beta} \frac{dp_1}{dt} + c_0 p_1 \quad \text{(When the swing angle of the swashplate becomes smaller)}
\]

\[
q_{v2} = A_1 \frac{dx_p}{dt} + \frac{V}{\beta} \frac{dp_2}{dt} + c_0 p_2 \quad \text{(When the swing angle of the swashplate becomes larger)}
\] (8)
Where, \( V \) is variable cylinder cavity volume, \( \text{cm}^3 \); \( \beta \) is effective volumetric elastic modulus of oil, MPa; \( c_0 \) is leakage coefficient of variable cylinder, \( \text{cm}^3/(s\cdot\text{MPa}) \).

From equation (3) ~ (8), we can get

\[
2K_q x_v = 2A_1 \frac{dx_p}{dt} + \frac{V}{\beta A_1 r_0^2} \frac{2J}{\beta} \frac{d^2x_p}{dt^2} + (K_p + c_0) \frac{2J}{A_1 r_0^2} \frac{d^2x_p}{dt^2}
\]

The transfer function of the swashplate is

\[
W_2(s) = \frac{x_v(s)}{x_p(s)} = \frac{K_q / A_1}{s^2 + \omega_n^2 + 2\delta_n / \omega_n + 1}
\]

Where, \( \omega_n \) is natural frequency of swashplate, \( \omega_n = \sqrt{A_1 \beta r_0^2 / (VJ)} \); \( \delta_n \) is damping coefficient, \( \delta_n = \omega_n (K_q + c_0) J / (2A_1 r_0^2) \);

Flow gain equation of pump is

\[
Q_p = -K_q nx_p
\]

Where, \( K_q \) is displacement gradient of pump, \( \text{cm}^3/r \); \( n \) is pump speed, \( r/s \).

Transfer function of pump flow and displacement of variable cylinder is

\[
W_3(s) = \frac{-Q_p(s)}{x_p(s)} = K_q n
\]

Variable displacement pump output flow and pressure characteristics is

\[
-Q_p(s) + Q_L - c_i = \frac{V_i \ddp}{\beta} \frac{dt}{dt}
\]

The transfer function of pump output pressure is

\[
W_4(s) = \frac{P(s)}{I(s)} = \frac{1}{s + \omega_r} \frac{c_i}{1 + s / \omega_r}
\]

Where, \( I(s) \) is flow deviation signal; \( V_i \) is volume of output capacitor of pump; \( \omega_r \) is turning frequency of inertial element; \( c_i \) is leakage coefficient of spring cavity of variable cylinder.

In summary, the block diagram of the system can be obtained, as shown in Figure 3. The open-loop transfer function of load-sensitive pump is

\[
W(s) = \frac{1}{s + \frac{1}{\omega_r} \left( \frac{s^2}{\omega_r^2} + 1 \right) \left( \frac{s^2}{\omega_r^2} + 2\delta_n / \omega_n + 1 \right)}
\]
4. Simulation Modeling of Load-sensing System

The AMESIM platform is chosen for simulation analysis. The software is easy to realize the simulation analysis of multi-disciplinary collaboration. Based on the mechanical component library, hydraulic component library and signal library provided by the software, the simulation model can be built as shown in Figure 4. It should be noted that in order to simplify the problem, some components are omitted in the modeling.

In order to facilitate the comparison, two sets of models of Load-sensing hydraulic system and quantitative pump hydraulic system are established, and the two systems are analyzed by synchronous signal excitation. The load-sensitive hydraulic system is on the left side of Figure 4 and the quantitative pump hydraulic system is on the right side of Fig. 4.

In the simulation model, the main component parameters are set as follows. The maximum displacement of the pump is set to 0.1L. The motor speed is set to 1000r/min. The differential pressure of the Load-sensing valve is set to 2 MPa. Pressure globe valve is set at 16 MPa. The relief valve is set to 16 MPa. The piston diameter of hydraulic cylinder is 200 mm, the piston rod diameter is 120 mm, and the stroke is 2 m. The maximum lifting height of the scissor lift is 10m, and the weight of the lifting platform is 50Kg.

5. Simulation and Analysis of Load-sensing System

5.1. No-load Operation Simulation of Equipment

Two kinds of hydraulic systems use the same PID control and power amplifier, and drive the electro-hydraulic servo valve by the same driving signal. The longitudinal displacement of the platform driven by two systems is shown in Figure 4. Through comparison, it can be found that the corresponding speed
of the quantitative pump hydraulic system (Line 1) is slightly faster than that of the Load-sensing hydraulic system (Line 2). However, by comparing the longitudinal velocity, as shown in Fig. 5, the impact of the former is significantly greater than that of the latter, so from the perspective of stationarity, the latter is better than the former.

By comparing the pressure of the rodless chamber of the driving cylinder, as shown in Figure 6, the pressure of the quantitative pump hydraulic system (Line 1) should be obviously higher than that of the Load-sensing hydraulic system (Line 2). By comparing the outlet pressure of the pump, as shown in Figure 7, the former has reached the maximum set pressure of the system in both working sections. In conclusion, the former has a relatively large overflow energy consumption, while the latter does not, reflecting the advantages of energy saving.

By observing the difference between pump output pressure and load pressure in load-sensitive hydraulic system, as shown in Fig. 8. The pressure difference is a constant value for most of the time, except for the fluctuation of the equipment during the rising and falling stages. This is because the pump can dynamically adjust the displacement through the feedback load pressure, as shown in Fig. 9. At the beginning of lifting, the pressure difference decreases and the displacement of the pump is large. As the distance from the target position gets closer and closer, the pressure difference will increase, so the displacement of the pump will be reduced by the load-sensing valve. So the flow rate of the pump decreases gradually, as shown in Fig. 10.
5.2. Equipment Operation Simulation with External Load Disturbance

In order to compare the anti-jamming performance of the two systems, a downward force of 10000N is added to the lifting platform, the action time is 5-25s, and the action force is a step form. The ideal displacement (Line 1) and external load signals (Line 2) are shown in Fig. 11.

By comparing the longitudinal displacement of the platform driven by two systems, as shown in Fig. 12, it can be seen that the external load has little effect on the response speed and positioning accuracy of the equipment. By comparing longitudinal velocity, as shown in Fig. 13, it can be found that the moment of loading and unloading still cause’s vibration. The dotted line frame area in Figure 12 is partially enlarged and displayed as shown in Figure 13. It can be found that the amplitude of the quantitative pump hydraulic system (Line 1) is larger than that of the load-sensitive hydraulic system (Line 2), and the oscillation of the former lasts for about 2.3s, while that of the latter lasts only 0.3s, and then decays rapidly. By comparing the output pressure of the pump, as shown in Figures 14 and 15, it can also be found that the load-sensitive hydraulic system (Line 2) has better stability and anti-interference when the load changes.
5.3. System Performance of Pressure Difference of Different Load-sensing Valves

In order to analyze the influence of different pressure differences set by load sensitive valves on the dynamic characteristics of the system, batch analysis was carried out with 3MPa (Line 1), 2 MPa (Line 2) and 1 MPa (Line 3) respectively. The vertical displacement of the elevator platform is shown in Fig. 17. It can be seen that the greater the pressure difference set by the load sensitive valve, the faster the response speed of the system. At the same time, it can be found that when the pressure difference is set to 2 MPa and 3 MPa, the difference is not obvious. When the differential pressure is set to 1 MPa, the response speed is obviously reduced.

By tracking and observing the difference between the output pressure and load feedback pressure of the three systems, as shown in Figure 18, it can be found that the greater the pressure difference of the load sensitive valve, the greater the fluctuation of the difference in the transient state.

By tracking and observing the output pressure of the pump, as shown in figs. 19 and 20, it can be found that the system with larger pressure difference setting has better rigidity under loading. When the system is set to 3MPa, the pressure fluctuation is only 0.3s, and when the system is set to 1MPa, the pressure fluctuation is more than 5s. But when the load is unloaded, the higher the pressure, the greater the pressure rebound.
6. Conclusion
This paper studies the characteristics of shear crane system based on load sensing technology. Firstly, the transfer function of the system is deduced based on the physical laws and theorems of the joints of the system. Then a comparative study is carried out based on digital simulation technology.

By comparing the indexes of quantitative pump hydraulic system and load sensitive hydraulic system, it can be found that the latter is superior to the former in energy saving, stability and anti-interference. By comparing the dynamic characteristics of the system under different setting pressure difference, it can be found that when the pressure difference of load sensitive valve is set at about 2 MPa, the comprehensive performance of the system is the best one.

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