Modeling and stability of electro-hydraulic servo of hydraulic excavator

Wenhua JIA¹, Chenbo YIN², Guo LI¹, Menghui SUN¹

¹School of Mechanical Engineering, Nanjing Institute of Technology, Nanjing, China  
²School of Mechanical and Power Engineering, Nanjing University of Technology, Nanjing, China

Abstract: The condition of the hydraulic excavator is complicated and the working environment is bad. The safety and stability of the control system is influenced by the external factors. This paper selects hydraulic excavator electro-hydraulic servo system as the research object. A mathematical model and simulation model using AMESIM of servo system is established. Then the pressure and flow characteristics are analyzed. The design and optimization of electro-hydraulic servo system and its application in engineering machinery is provided.

1 Instruction

As a key part of engineering machinery, hydraulic excavators have been widely used in construction, transportation, and disaster rescue owing to their superior efficiency. However, the complicated working conditions and harsh working environment of hydraulic excavators have proposed higher requirements for their intelligent control. By combining hydraulic and electrical control technologies, the electro-hydraulic servo facilitates hydraulic excavator in remote control, self-operation, and synergetic work via electro-hydraulic integrated control. This study proposes a simulation model for electro-hydraulic servo and investigates its pressure and flow [1]. A mathematical model of electro-hydraulic servo is proposed based on force balance equation, voltage equation, and flow equation. Also, the stability of electro-hydraulic servo is studied using Bode plot and Nichols diagram.

Composition of electro-hydraulic servo control system is shown in Fig. 1: 1. valve pump, 2. One-way valve, 3. overflow valve, 4. electro-hydraulic servo valve, 5. Hydraulic cylinder, 6. displacement sensor, 7. Load, 8. servo amplifier

The input electrical signal is amplified and transduced to the electro-hydraulic servo valve, the spool shifts, the hydraulic pump initiates, resulting in displacement of the hydraulic cylinder. The output signal is then transduced to by the displacement sensor a feedback electrical signal, which is then compared to the command signal to obtain the bias voltage signal. The bias voltage signal is transmitted to the servo valve via the amplifier. The servo valve then controls the hydraulic actuator to reduce the bias between the output signal and the command signal to a designated range[2]. In this way, closed-loop control is achieved.
2 Mathematical model of electro-hydraulic servo

2.1 Servo amplifier
The bias control signal ($\Delta u$) was generated by the comparison element based on input signal and feedback signal. As the amplification element, the servo amplifier amplifies $\Delta u$, which is relatively weak, to realize the control of electro-hydraulic servo valve $^{[3,4]}$. The transfer function is as follows:

$$ I(s) = K_1 \times U(s) $$

Where $I$ is the input current to the electro-hydraulic servo valve, $U$ is the voltage signal generated by the servo amplifier, $K_1$ is the amplification coefficient.

2.2 Electro-hydraulic servo valve
A displacement feedback servo valve consists of a moving coil force motor, control slide valve, and main valve. With current flowing through, the coil in the motor generates an electromagnetic force to push the control slide valve, the top and bottom throttle shifts, and the main valve shifts. Once the displacement of the control slide valve and that of the main spool are aligned, the valve does not move further.

Moving coil force motor. The signal voltage on the control coil by the servo amplifier is the sum of the voltage drop on the resistor, self-induced back electromotive force in the control coil, and the back electromotive force induced by movement of control coil in the magnetic field. The voltage balance equation is as follows:

$$ E = (R_c + r_p)i + L_c \frac{di}{dt} + K_b \frac{dx}{dt} $$

Where $R_c$ is the resistance of the control coil, $r_p$ is the internal resistance of the amplifier, $L_c$ is the inductance of the control coil, $K_b$ is the back electromotive force constant ($K_b = B_a \pi D d N_c$).

The force balance equation for coil component is as follows:

$$ F = m \frac{d^2x}{dt^2} + B \frac{dx}{dt} + Kx $$

Where $m$ is the mass of coil, $B$ is the damping coefficient of coil, $K$ is the spring stiffness.

Laplace transform of Equation (2) and (3) results in:

$$ I_c = \frac{K_1 U - K_b s X}{(R_c + r_p)(1 + \frac{s}{\omega_0})} $$

Where $\omega_0$ — the corner frequency of the control coil ($\omega_0 = \frac{R_c + r_p}{L_c}$), $\delta_0 = \frac{1}{2} \sqrt{\frac{1}{Km}}$, $\omega_0 = \frac{K}{\sqrt{m}}$. 

![Figure 1. Composition of electro-hydraulic servo control system](image)
\[
\frac{X}{t_c} = \frac{K_t/K}{\omega_0^2 + \omega_0^2} 
\]  \hfill (5)

Control slide valve - main valve. The masses of front slide valve and main valve are \(m_1, m_2\), the dynamic damping coefficients of front slide valve and main valve are \(c_1, c_2\), the spring stiffness of front slide valve and main valve are \(K_a, K_b\), and the flow gain of front slide valve is \(K_{qp}\). The force balance equation for front slide valve and main valve is as follows:

\[
F_1 = m_1 \frac{d^2x}{dt^2} + c_1 \frac{dx}{dt} + K_a x + K_{qp} x_v \hfill (6)
\]

\[
F_2 = m_2 \frac{d^2x}{dt^2} + c_2 \frac{dx}{dt} + K_b x_v - \Delta P x A_v \hfill (7)
\]

The open capacity of front slide valve can be determined by:

\[
x_e = x - x_v \hfill (8)
\]

Excluding the effects of flow force, the transfer function can be obtained based on Equations (6), (7), and (8):

\[
x_v \frac{x_v}{x_e} = \frac{K_{qp}}{s (\omega_0^2 + \omega_{hp}^2 + 1)} \hfill (9)
\]

\[
x_v \frac{x_v}{x} = \frac{1}{s K_v + 1} \frac{1}{s (\omega_0^2 + \omega_{hp}^2 + 1)} \hfill (10)
\]

Figure 2. Diagram of transfer function of direct location feedback servo valve.

The simplified block diagram of transfer function of direct location feedback servo valve is shown in Figure2.

2.3 Hydraulic cylinder - load
Assuming that the pressure in the hydraulic cylinder is uniform, the temperature and bulk modulus of hydraulic medium are constant and the leakage of hydraulic cylinder is laminar flow. The flow equation of electro-hydraulic servo valve is as follows:

\[
Q_l = K_q x_v - K_c p_L \hfill (11)
\]

If the piston is located in the middle part, the system stability is poor, the frequency of hydraulic component is low, and the damping ratio is low. Also, \(V_1 = V_2 = V_c/2\) is valid in the hydraulic cylinder. The overall leakage coefficient \((C_{tp} = C_{ip} + \frac{C_{p}}{2})\) is used to reflect the effects of hydraulic cylinder leakage on load flow. The flow in the hydraulic cylinder can be determined by:

\[
Q_l = A_p \frac{dx_p}{dt} + C_{tp} p_L + \frac{v_c}{4\beta} \frac{dp_l}{dt} \hfill (12)
\]

Where \(A_p\) is the effective area of the piston \((m^2)\), \(x_p\) is the displacement of the piston \((m)\), \(C_p\) is the leakage coefficient of the hydraulic cylinder \((m^3 \cdot s^{-1}/Pa)\), \(\beta\) is the effective bulk modulus\((Pa)\).

The force balance equation for hydraulic cylinder is as follows:

\[
A_p p_L = m_t \frac{d^2x_p}{dt^2} + B \frac{dx_p}{dt} + Kx_p + F_L \hfill (13)
\]

Where \(m_t\) is the overall mass of piston \((kg)\), \(F_L\) is the external load on the piston \((N)\).

Equations (11), (12), and (13) are equations for displacement, pressure, flow of the hydraulic cylinder, respectively. These parameters are indicators of dynamic performance of the hydraulic cylinder. Laplace transform of these equations results in overall displacement of the piston in the hydraulic cylinder:
\[ x_p = \frac{K_q v_p e^{-K_{ce} (1 + \frac{V_t}{4\beta K_{ce}})} F_L}{m_e V e} \] (14)

Where \( K_{ce} \) is the overall flow-pressure coefficient, \( K_{ce} = K_c + C_{ip} \).

3 Modeling of electro-hydraulic servo

3.1 Establish system Modeling

In the electro-hydraulic servo [5-8], the hydraulic cylinder pushes the load, the displacement sensor at the load converts location signal to electrical signal. The bias between the actual location signal and the designated location signal is amplified and applied on the electro-hydraulic servo valve. The servo spool shifts, the flow of hydraulic system fluctuates, and the location of hydraulic cylinder is adjusted to achieve trajectory tracking. A simulation model is established shown in Fig. 3.

![Simulation model of electro-hydraulic servo system](image)

Figure 3. Simulation model of electro-hydraulic servo system.

The flow is defined as 0 when the servo valve is at the initial position. The flow is defined as +1 at an extreme position. Herein, Port P is connected to Port A and Port T is connected to Port B. The flow is defined as -1 at the other extreme position. Herein, Port A is connected to Port T and Port B is connected to Port P.

3.2 Set system parameters

The key component parameters of the simulation model are summarized in Table 1. The values of these parameters were determined based on the actual system. Fig. 4 shows the signal set-up of the signal source: the signal stays at 0 in 0~1 s, increased to 0.8 in 1 s~4 s, and stays at 0.8 in 4~6 s.

| System parameters | Value |
|-------------------|-------|
| Motor rated speed / (r.min⁻¹) | 1500 |
| Pump rated displacement / (cc/rev) | 122 |
| Pump speed / (rev/min) | 1500 |
| Rated current of servo valve / mA | 200 |
| Natural frequency / Hz | 50 |
| Damping ratio | 1 |
Servo amplifier gain 250
Diameter of hydraulic cylinder piston /mm 120
Diameter of hydraulic cylinder piston rod /mm 85
Piston block mass /kg 100
Cylinder stroke /m 1

Figure 4. Signal set-up of the signal source.

3.3 Simulation and analysis of electro-hydraulic servo
The simulation time is set to be 6 s to check the steady operation status of the system. The communication interval is set to be 0.01 s to exclude effects of system fluctuations. The pressure and flow of key components were obtained and analyzed.

Fig. 5 shows the flow curve in the hydraulic cylinder. During 0~1 s, the electrical signal is 0, the servo valve is closed, and the flow in the hydraulic cylinder is 0. After 1 s, an electrical signal is observed, the servo valve shifts, and flows in rod-containing cavity and rodless cavity vary. At the initial stage, the flow in rod-containing cavity varies smoothly, while that in rodless cavity varies drastically. After 3 s, fluctuations of flows in both cavities approach negligible. Eventually, the maximum flows in rodless cavity and rod-containing cavity are 8.66 L/min and 4.33 L/min, respectively.

Figure 5. Flow curve in the hydraulic cylinder.

Fig. 6 shows the pressure in the hydraulic cylinder. During 0~1 s, the electrical signal is 0, the servo valve is closed, and the pressures in the two cavities are 0. After 1 s, an electrical signal is observed, the servo valve shifts, and flows in rod-containing cavity and rodless cavity vary. At the initial stage, the flow in rodless cavity increases drastically and is significantly higher than the flow in rod-containing cavity. After 1.2 s, the increasing rate of the flow in rodless cavity degrades and is lower than that in rod-containing cavity. Eventually, fluctuations of flows in both cavities approach negligible. The maximum pressure in rod-containing cavity and rodless cavity are 7.28 MPa and 6.26 MPa, respectively.
Fig. 6 shows the pressure in the hydraulic cylinder. Once the electrical signal is observed, the valve opens and reached the extreme position at 1.2 s.

Fig. 7 shows the displacement of servo valve spool as a function of time. Once the electrical signal is observed, the valve opens and reached the extreme position at 1.2 s.

Fig. 7. Displacement of servo valve spool.

Fig. 8 shows the flow curve at the pump valve outlet. Once the hydraulic system is initiated, the outlet flow is maintained at 183 L/min. During 0~1 s, the electrical signal is 0 and the flow at the overflow valve increases drastically to 183 L/min. After 1 s, an electrical signal is observed, the servo valve shifts, and the flow at the overflow valve decreases gradually. As the spool of servo valve reached the extreme position, the flow at the overflow valve stays at 174.3 L/min.

Fig. 8. Flow curve at the pump valve outlet.

Fig. 8. Flow curve at the pump valve outlet.

4 Conclusions
A study of electro-hydraulic servo of hydraulic excavator using mathematical and simulation models is proposed. The simulation parameters are designed according to actual working conditions and the pressure and flow characteristics of the servo system is studied. This study provides references for the design and improvement of electro-hydraulic servo of hydraulic excavators.
Acknowledgment
This work was supported in part by The National Natural Science Fund of China, Jiangsu Natural Science Foundation, University of Jiangsu Natural Science Foundation, and SANY Co., Ltd. in Jiangsu. Lecturer, Support Fund Nos. 51505211, 51505212, BY2015005-15 and JXKJ201513.

References
[1] Cui L, Zhiyong T, Xiaobao Z, Zhongcai P. The modeling and simulation study of hydraulic intelligent power system based on AMESim [C]. Proceedings of 2014 IEEE Chinese Guidance, Navigation and Control Conference Guidance, Navigation and Control Conference (CGNCC), 2014 IEEE Chinese: 1530-1533;
[2] Ning X, Xi C, Jin Jin G. Simulation Analysis of Proportional Valve Controlled Cylinder Hydraulic System Based on AMESim [C]. Advanced Materials Research; March 2013, Vol. 668 Issue: 1: p420-425;
[3] Amirante R, Andrea Catalano L, Tamburrano P. The importance of a full 3D fluid dynamic analysis to evaluate the flow forces in a hydraulic directional proportional valve [J]. Engineering Computations, 2014, 31(5): 898-922;
[4] Ling long L, Liang qing H, Qi feng Z. Design and simulation analysis of servo amplifier [J]. Process Automation Instrumentation, 2012, 33(12): 83-85;
[5] Tivay, Ali, Zareinejad, Mohammad, Rezaei, S. Mehdi, Baghestan, Keivan. A switched energy saving position controller for variable-pressure electro-hydraulic servo systems. In Disturbance Estimation and Mitigation, ISA Transactions July 2014 53(4):1297-1306;
[6] Chiang, MH, Lee, LW, Liu, HH. Adaptive fuzzy controller with self-tuning fuzzy sliding-mode compensation for position control of an electro-hydraulic displacement-controlled system [J]. Journal of Intelligent & Fuzzy Systems, 2014, 26 2: 815-830;
[7] Milic, V, Situm, Z, Essert, M. Robust H-infinity position control synthesis of an electro-hydraulic servo system [J]. ISA TRANSACTIONS, OCT, 2010, 494:535-p542;
[8] Baghestan, Keivan, Rezaei, Seyed Mehdi, Talebi, Heidar Ali, Zareinejad, Mohammad. Research Article: An energy-saving nonlinear position control strategy for electro-hydraulic servo systems [J]. In ISA Transactions November 2015 59:268-279.