Modeling and Simulation of Conversion Booster Cylinder Control System

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Abstract. This paper is based on the accelerated fatigue test of aviation hydraulic actuators. For the case where aviation hydraulic actuators transmission medium is different from the testing rig transmission medium, a new type of conversion booster cylinder is designed. This conversion booster cylinder set pressurization, medium conversion, pumping aviation oil as one, and laid the foundation for the pressure pulse fatigue test of aviation hydraulic actuator cylinders. Combined with the principle of valve-controlled asymmetric cylinder, a mathematical model of the control system of the pressurized cylinder was established using the method of parameter weighted averaging to unify the models in both directions. Simulink software is used to simulate the system, and the required time domain and frequency domain performance indexes are obtained. The working condition of confining pressure is simulated and the effectiveness of the design of the conversion booster cylinder is proved.

Key words: Hydraulic; Booster cylinder; Valve-controlled asymmetric cylinder; Simulink.

1. Introduction
In order to carry out the accelerated life test of the aviation hydraulic actuator, it is necessary to perform pressure pulse fatigue loading on cylinders. The transmission medium of pump station of the testing rig is No. 1 standard hydraulic oil, and the transmission medium of the aviation hydraulic actuator is No. 10 aviation hydraulic oil. Since the transmission medium is different, and the maximum pressure of the pressure pulse fatigue loading is greater than the output pressure of the testing rig pump station, the loading needs to be completed by using a conversion booster cylinder. The conversion booster cylinder combines the functions of pressurization and medium conversion to convert the medium-pressure standard oil output from the pump station into high-pressure aviation oil required for pressure pulse fatigue loading.

2. Mechanical Structure of the Conversion Booster Cylinder
As shown in Figure 1, the conversion booster cylinder has 4 ports, port A and port B are connected to port A and port B of the servo valve, and the transmission medium is No. 1 standard hydraulic oil. Port
C is connected to the aviation hydraulic oil tank through the check valve. Port D is connected to the tested cylinder through the hydraulic hose. The transmission medium of port C and port D is No. 10 aviation hydraulic oil.

When the testing rig begins to work, the pump station starts. The controller generates a square wave signal of -10V to +10V, driving the servo valve to circulate between the left and right positions, thereby driving the piston of the conversion booster cylinder to reciprocate left and right. With the help of two check valves the conversion booster cylinder continuously pumps the aviation oil from the aviation hydraulic oil tank through the port C and port D into the tested cylinder. The air in the tested cylinder is discharged to the aviation hydraulic oil tank through the oil return passage until it filled with oil. When the aviation oil fills the entire cylinder, the booster cylinder piston is located at the leftmost point. At this time, close the oil return passage and close the tested cylinder. The controller stops outputting the square wave signal and turns to closed-loop control. Through the left and right movement of the piston of booster cylinder, the aviation oil pressure in the test cylinder changes according to the predetermined load spectrum.

![Fig. 1 Structure of conversion booster cylinder](image)

1-front end cap, 2-cylinder sleeve, 3-piston, 4-combined sealing, 5-cylinder body, 6-rear end cap

3. Mathematical Model of the Booster Cylinder Control System

The essence of the pressure pulse fatigue loading system is the valve-controlled asymmetric cylinder system. Unlike the axial tension-compression loading system, the output of the pressure pulse fatigue loading system is the load pressure, while the output of the axial tension-compression loading system is the force and displacement. The structure of the pressure pulse fatigue loading system is shown in Figure 2. The volume of the tested cylinder is $V_c$, the pressure is $P_3$, the pressure bearing area is $A$, and the small annular area of the rod cavity is $A_1$. As shown in Figure 2, the other parts are the same as the axial loading structure. Since the tested cylinder is a closed volume and the volume of the medium is hardly changed, the load types are mass load and elastic load. Similar to axial loading, because the effective area of the two chambers of the piston is different, the flow continuity equation is related to the direction of the piston speed $y$. It is defined as the forward model when the pressure of the tested cylinder is increased, and the negative model when the pressure of the tested cylinder is decreased.
Fig. 2 Schematic diagram of valve controlled pressure cylinder

(a) Servo valve flow equation
When $x_v > 0$, we have:

$$q_L = K_{v1}x_v - K_{v1}p_L$$

(1)

similarly, when $x_v < 0$, we have

$$q_L = K_{v2}x_v - K_{v2}p_L$$

(2)

(b) Flow continuity equation
When the booster cylinder piston moves forward, that is, when the pressure in the tested cylinder increases, the flow continuity equation is:

$$q_L = A_V \frac{dy}{dt} + C_v p_L + \frac{hV_3}{2\beta_v} \frac{dp_L}{dt}$$

(3)

When the booster cylinder piston moves backward, that is, when the pressure in the tested cylinder decreases, the flow continuity equation is:

$$q_L = A_V \frac{dy}{dt} + C_v p_L + \frac{hV_3}{2\beta_v} \frac{dp_L}{dt}$$

(4)

(c) Force balance equation
When the booster cylinder is working, that is, when the pressure pulse is loading, the booster cylinder piston is taken as the research object. Since the pressure pulse fatigue loading medium in the tested cylinder is aviation oil, and the volume change can be neglected, only the inertia force and hydraulic spring force is considered.

For the tested cylinder filled with aviation oil:

$$\beta_{es} = \frac{\Delta p_3}{\Delta V_3 / V_3} = \frac{\Delta p_3}{(y \cdot A_v) / V_3}$$

(5)

where,

$\beta_{es}$ — Effective bulk modulus of aviation oil, Pa;

$\Delta p_3$ — The amount of pressure change of the tested cylinder, Pa;

$\Delta V_3$ — The effective volume change of the tested cylinder, $m^3$;

$V_3$ — The initial volume of the tested cylinder, $m^3$;
$y$ — The piston displacement, m;
Therefore the hydraulic spring force is:
\[
F_s = \Delta p_s A_s = \frac{\beta_s A_s^2}{V_s} y
\] (6)

Therefore, we can get when the booster cylinder is operating, that is, when the pressure pulse is
loading, the force balance equation of the booster cylinder is:
\[
A_{bl} P_L = m \frac{d^2 y}{dt^2} + \frac{\beta_{bl} A_{bl}^2}{V_s} y
\] (7)

After the Laplace transform, the equations for the forward movement of the booster cylinder piston

Eliminate the intermediate variable, and simplify the transfer function of the system when the booster
cylinder piston moves forward:
\[
\Delta P \over X_s = \frac{\beta_s A_s K_{v1}}{s(\frac{\omega_{hi}^2}{\omega_{hi}^2} + \frac{2\zeta_{hi}}{\omega_{hi}} s + 1)}
\] (9)

where,
\[
\omega_{hi} — \text{Hydraulic natural frequency when the booster cylinder piston moves forward}
\]
\[
\omega_{hi} = \frac{2\beta_s A_{bl} A_s}{h V_p m}
\] (10)

$\zeta_{hi} — \text{Hydraulic damping ratio when the booster cylinder piston moves forward}$
\[
\zeta_{hi} = K_{v1} \frac{\beta_s m}{2 h V_p A_{bl} A_s} + \frac{B_p}{2 \beta_s m A_{bl} A_s}
\] (11)

where,
\[K_{v1} — \text{Total pressure flow coefficient when the booster cylinder piston moves forward}\]
\[K_{v1} = K_{v1} + C_p
\] (12)

In the same way, after Laplace transform, the equations for the backward movement of the booster
cylinder piston can be obtained.
\[
Q_v(s) = K_{v2}X_v(s) - K_{v2}P_2(s)
\]
\[
Q_v(s) = A_2Ys + C_P P_1 + \frac{h_2V_0}{2\beta_x}P_2s
\]
\[
A_{v2}P_1 = m_2Ys^2 + \frac{\beta_2A_2^2}{V_3}Y
\]

Eliminate the intermediate variable, and simplify the transfer function of the system when the booster cylinder piston moves backward:

\[
\frac{\Delta P_1}{X_v} = \frac{\beta_2A_2K_{v2}}{s\left(\frac{\omega_{h2}^2}{\omega_{h2}^2} + 2\zeta_{h2}\frac{\omega_{h2}}{\omega_{h2}} + s + 1\right)}
\]

where,

\(\omega_{h2}\) — Hydraulic natural frequency when the booster cylinder piston moves backward:

\[
\omega_{h2} = \sqrt{\frac{2\beta_2A_0A_4}{h_2V_0\beta_x}}
\]

\(\zeta_{h2}\) — Hydraulic damping ratio when the booster cylinder piston moves backward:

\[
\zeta_{h2} = K_{v2}\sqrt{\frac{\beta_2m_1}{2h_2V_0A_0A_4}} + \frac{B_x}{2\sqrt{2\beta_2m_1A_0A_4}}
\]

\(K_{v2}\) — Total pressure flow coefficient when the booster cylinder piston moves backward:

\[
K_{v2} = K_{v2} + C_P
\]

When the booster cylinder piston moves forward and backward, the transfer function of the system can be integrated as one:

\[
\frac{\Delta P_1}{X_v} = \frac{\beta_2A_2K_{v2}}{s\left(\frac{\omega_{h}^2}{\omega_{h}^2} + 2\zeta_{h}\frac{\omega_{h}}{\omega_{h}} + s + 1\right)}
\]

where,

\(K_v\) — zero position flow gain of zero-opening slide valve

\[
K_v = \frac{\alpha}{1 + \alpha}K_{v01} + \frac{1}{1 + \alpha}K_{v02}
\]

\(\omega_h\) — Hydraulic natural frequency of the booster cylinder system:

\[
\omega_h = \frac{\alpha}{1 + \alpha}\omega_{h0} + \frac{1}{1 + \alpha}\omega_{h2}
\]

\(\zeta_h\) — Hydraulic damping ratio of the booster cylinder system:

\[
\zeta_h = \frac{\alpha}{1 + \alpha}\zeta_{h0} + \frac{1}{1 + \alpha}\zeta_{h2}
\]

\(\alpha\) — Flow ratio:
\[ \alpha = \frac{q_2}{q_1} = \frac{A_k - A_i}{A_1} \]  

(22)

For valve-controlled symmetrical cylinder systems:

\[ \alpha = 1, \quad K_{\phi 1} = K_{\phi 2} \]  

(23)

Similar to the axial tension-compression loading, in this subject, the servo valve system can be considered as a second-order oscillation link, so its transfer function is:

\[ G_{SV}(s) = \frac{q_{SV}(s)}{u(s)} = \frac{K_{SV}}{s^2 + \frac{2\zeta_{SV}}{\omega_{SV}}s + 1} \]  

(24)

- \( K_{SV} \) — Servo valve flow gain
- \( \omega_{SV} \) — Natural frequency of servo valve
- \( \zeta_{SV} \) — Damping ratio of servo valve

4. Control System Simulation

The transfer function of the MOOG D633-303B servo valve uses the following data.

\[ \frac{Q}{\Delta U} = \frac{6.8 \times 10^{-6}}{s^2 + \frac{2 \times 0.7}{80} s + 1} \]  

(25)

According to the value of relevant parameters, the transfer function of the booster cylinder is

\[ \frac{\Delta P}{Q_L} = \frac{\beta_{SV} A_1}{V_1 A_k} = \frac{6 \times 10^4 \times 1.96 \times 10^{-3} / 7.46 \times 10^{-4} \times 2.14 \times 10^{-3}}{s^2 + \frac{2 \times 0.0124}{5323} s + 1} \]  

(26)

According to the above data, build a simulation model in Simulink, as shown in Figure 3.

![Simulink simulation model of pressure pulse fatigue loading control system](image)

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![Simulink simulation model of pressure pulse fatigue loading control system](image)

The system input is a step signal. When the PID controller is set to \( K_p = 0.05, K_i = 0, K_d = 0 \), the simulation result is shown in Figure 4:
The open-loop Bode diagram of the pressure pulse fatigue loading control system is drawn by Simulink's Linear Analysis tool, as shown in Figure 5:

Analyze the simulation results: for a given step signal of pressure pulse fatigue loading control system, the adjustment time is less than 0.4s. The overshoot is about 5%. The steady state error is less than 0.05%. The amplitude margin of the system is about 5dB. The phase margin is about 70 degrees. Therefore, the pressure pulse fatigue loading control system is stable, and the conversion booster cylinder is designed successfully.

As with the axial tension-compression control system, the closed-loop Bode diagram of the pressure pulse fatigue loading control system is shown in Figure 6:
Fig. 6 The closed-loop Bode diagram of the pressure pulse fatigue loading control

According to the closed-loop Bode diagram, it can be known that the bandwidth of the system is 49 rad/s, which is 7.8 Hz.

5. Conclusion and Analysis
In this paper, the mathematical model of the designed conversion booster cylinder system is carried out. For the case that pressure pulse fatigue loading of the booster cylinder is asymmetric cylinder, a method of weighted average equivalent pressure area and hydraulic spring stiffness theory are proposed. A linear model of zero-opening valve-controlled asymmetric cylinder system is established, and the system weighting average is used to unify the different system models in two directions.

According to the simulation results, the booster cylinder system is stable. The bandwidth is above 7 Hz, which fully meets the needs of pressure pulse fatigue loading. The time series pressure required by the load spectrum can be applied to the tested cylinder. The system can meet the working requirements of the cylinder fatigue test. Therefore, the conversion booster cylinder and its control system meet the requirements of the aviation hydraulic actuator fatigue test.

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Junbin Yang, male, born in 1991, engineer of AVIC China Aero-Polytechnology Establishment, major in life testing and life prediction of mechanical product.

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