Position Control of Asymmetric Hydraulic Cylinder with Two Chambers Connected*

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Abstract

This paper is concerned with position control of asymmetric hydraulic cylinder system. The hydraulic system consists of an asymmetric hydraulic cylinder with two chambers connected by an orifice, a two-position, two-way proportional valve and a load force. The system structure and the control principle were introduced in this paper. The influence of some structural parameters on the system dynamic performance was analyzed. The analysis and simulation results indicate that a two-position, two-way proportional valve can achieve position control of the asymmetric hydraulic cylinder system. A good dynamic performance can be achieved after using the control strategy proposed by this paper.

Key words: Asymmetrical Hydraulic Cylinder, Position Control, Flow Compensation, Proportional Valve, Control Strategy

Notation

- A Area of the orifice flow
- A_1 Piston area in the non-rod chamber
- A_2 Piston area in the rod chamber
- B_r Liquid viscosity damping coefficient
- C_d Flow coefficient of the orifice
- D Derivative gain of PID controller
- D_0 Diameter of the orifice
- f Frequency of the valve
- F Load
- F_f Friction force between the cylinder and the piston
- I Integral gain of PID controller
- m Mass of the piston and the rod
- p_c Pressure of the non-rod chamber
- p_n Nominal flow rate of the valve
- p_s Supply pressure
- P Proportional gain of PID controller
- q_1 Flow through the orifice from the oil supply
- q_2 Flow through the valve back to the reservoir

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- q_L Working flow
- q_n Nominal pressure of the valve
- t Time
- V Volume of the non-rod chamber
- x Displacement of the rod
- x_{in} Target position
- β_e Fluid bulk modulus
- ρ Density of the oil

1. Introduction

Hydraulic systems are widely used in many fields as aviation, navigation, weapon industry, and most applications of hydraulic systems involve position control systems, which may be linear or rotational⁽¹⁾. Asymmetric hydraulic cylinders play very important roles in linear position control systems owing to advantages such as low cost, higher carrying capacity, simple structure and less workspace⁽²⁾. There are many works focusing on these types of systems, where the fluid power sent to the asymmetric hydraulic cylinder is regulated in various ways, i.e., with one four-way, three-position servo valve, one three-way, two-position servo valve⁽³⁻⁵⁾, or with digital modulated valves⁽⁶⁾. Generally, the ports of hydraulic cylinders are connected with valves. But on some very special occasions, the two chambers of cylinders are connected by an orifice, through which oil will flow from one to the other in the course of operation. Because of the rare applications, this structural system has not been paid enough attention in the past decades.

If the two chambers of the cylinder were connected with an orifice, a single on/off valve could carry out the position control of the asymmetric hydraulic cylinder system⁽⁷⁾. However, on/off valve controlled systems have inherent ripple, poor static performance, and the frequency and amplitude cannot be adjusted independently for their interaction. Thus, it would be only suitable for the occasions with low controlling accuracy requirements⁽⁸⁾.

In this paper, single, two-position, two-way proportional valve was used to carry out the position control of the asymmetric hydraulic cylinder system. The dynamic model of the system was established, and the control strategies were provided. Both the influences of the proportional valve's response frequency and the area of orifice on system performances were analyzed.

2. System structure and working principle

Previous investigations have indicated that pressure jumps happen in the cylinder around $\dot{x} = 0$ as the asymmetric cylinder controlled by a symmetric valve, due to the asymmetry of the hydraulic cylinder^(9,10).

The analysis of the pressure characteristics inside the two chambers indicates that the structure of the hydraulic cylinder (Fig.1) can avoid pressure jumps effectively and has advantages such as low cost, and simple structure^(7,11). However, the dynamic characteristics of the asymmetric cylinder system get worse because the two chambers are connected, and the internal leakage of cylinder is increased⁽¹¹⁾. It may be one of the major causes of that the system structure has not been applied in the industry widely in the past years. Fig.1 shows the structure of the hydraulic cylinder, which consists of an asymmetric hydraulic cylinder and an orifice. The non-rod chamber is connected with the rod chamber through the orifice.

Fig.1 Schematic of the hydraulic cylinder

As shown in Fig.2, the hydraulic system consists of the asymmetrical hydraulic cylinder, an oil source, a displacement sensor, two pressure sensors, a controller and a proportional valve. The rod chamber connects with the oil supply and the non-rod chamber connects with the valve. The target position, the signals of pressure sensors and the displacement sensor can be used as inputs for the controller, and the controller computes the input signal of the proportional valve to achieve the position control of the asymmetric hydraulic cylinder system according to the control rules.

Fig.2 Schematic diagram of the system

According to flow continuity, the working flow is written as:

$$q_L = q_1 - q_2 \quad (1)$$

The flow rate at the orifice may be written as⁽¹²⁾:

$$q_1 = C_d A \sqrt{\frac{2}{\rho} (p_s - p_c)} \quad (2)$$

Substituting Eq. (2) into Eq. (1), the working flow is expressed as:

$$q_L = C_d A \sqrt{\frac{2}{\rho} (p_s - p_c)} - q_2 \quad (3)$$

According to continuity of fluid:

$$\dot{p}_c = \frac{\beta_e}{V} \left\{ C_d A \sqrt{\frac{2}{\rho} (p_s - p_c)} - q_2 - \frac{dV}{dt} \right\} \quad (4)$$

Based on Newton's second law, the kinematic equation of the piston can be written as:

$$m\ddot{x} = p_c A_1 - p_s A_2 - F - F_f \text{sign}(\dot{x}) - B_f \dot{x} \quad (5)$$

According to the kinematic equation of the piston, the control principle of system can

be expressed as follows:

As the size of valve port opening is reduced, the flow entering the non-rod chamber through the orifice is more than the flow out of the non-rod chamber through the valve, the pressure inside the non-rod chamber is increasing gradually, and the piston will move forwards under $p_c A_1 > p_s A_2 + F + F_f$. When the size of valve port opening is increased, the flow entering the non-rod chamber through the orifice is smaller than that out of the non-rod chamber through the valve; the pressure in the non-rod chamber will decrease gradually, and the piston will move backwards under $p_c A_1 + F + F_f < p_s A_2$. Obviously, the two-position, two-way proportional valve acts as a new variable orifice; when the orifice is reduced, the throttling action is increased, the pressure in the non-rod chamber increases, so it pushes the piston forwards. As the orifice increases, the throttling action is reduced, and the pressure of the non-rod chamber decreases with the piston moving backwards⁽¹¹⁾. As a result, the position control can be realized effectively by adjusting the port opening of the two-position, two-way proportional valve.

3. Control strategy and system model

From the system structure and working principle, the proportional valve acts as a throttle valve. The essence of position control of the asymmetrical cylinder system through adjusting the size of the proportional valve front opening is to control the oil volume in the non-rod chamber. The piston must track the input target position, or else, the controller computes the size of port opening of the proportional valve according to the difference between the present position and the target position. When the present position of the piston is equal to the target one, the piston is required to be still. Due to pressure difference, the oil in the rod chamber will flow through the orifice to the non-rod chamber. If the piston needs to be motionless, then the non-rod chamber should keep its oil volume unchanged. Since the oil volume in the non-rod chamber is constant, the flow entering the non-rod chamber through the orifice must be equal to that out of the chamber through the valve.

There are many investigations on the control strategy of the asymmetrical cylinder systems⁽¹³⁻¹⁵⁾. Under the condition of the system structure presented in this paper, if the traditional PID controller is adopted and the friction between the piston and the internal wall of cylinder is neglected, the control process can be expressed as follows: when the piston reaches the target position, the position error is "0", the output signal of PID is "0" and the valve closes; the oil volume of the non-rod chamber increases because $q_2 = 0$ and $q_1 > 0$, so the piston moves forwards. Then, the position error of the piston is not equal to "0", the output value of PID controller is not "0" and the valve opens, the oil volume of the non-rod chamber decreases and the piston moves backwards when $q_2 > q_1$, the position error is zero again when the piston reaches the target position; so the piston repeatedly oscillates around the target position. Namely, under PID controller, the piston cannot be stabilized at the target position but oscillates in a narrow range around the target position; and the opening size of the valve port keeps periodically oscillatory at some certain ranges, which is not good for control performance. The cause for the piston oscillating around the target position is the change of the oil volume in the no-rod chamber, the essence of which is the oil flowing to the no-rod chamber from the rod one through the orifice due to the pressure difference between the two chambers during the process. When the piston reaches the target position, the flow compensation can make an equal between the flow entering the no-rod chamber through the orifice and the flow out of the chamber through the valve, so as to keep the oil volume in the no-rod chamber and the piston position unchanged, and to fade the oscillation.

According to the special structure of the asymmetrical cylinder system, the flow compensation method and the traditional PID controller are co-operated to carry out the position control of the hydraulic system. Figure 3 illustrates the proposed control scheme.

Fig.3 Control scheme of the asymmetrical cylinder system

From the analysis above, the control strategy adopted in this hydraulic system can be expressed as follows: The flow compensation signal is employed in this controller. The value of the flow compensation signal can be calculated by the compensation flow according to the pressure and flow rate characteristics of the proportional valve. As the piston reaches the target position, the proportional valve will be in a proper opening size according to the adjustment of the compensation signal, which ensures that the flow entering the non-rod chamber through the orifice is equal to that out of the chamber through the valve.

AMESIM stands for Advanced Modeling Environment for performing Simulations of engineering systems. It is based on an intuitive graphical interface in which the system is displayed throughout the simulation process⁽¹⁶⁾. Common hydraulic components models of hydraulic systems are provided in the model libraries of AMESIM. According to the hydraulic system structure, the dynamic model of the system can be established by connecting the corresponding hydraulic components. After compiling the established model and setting the structure parameters of the hydraulic components, the simulation model of hydraulic system is obtained. The simulation model under AMESIM is shown in Fig.4.

Fig.4 Simulation model under AMESIM

Establishing an S-Function with the same name of the AMESIM model, and the co-simulation model under SIMULINK is shown in Fig.5, which is implemented by importing the AMESIM model into Simulink and setting the simulation parameters.

Fig.5 Simulation model under SIMULINK

S-Function block(AMESim:Project_) stands for the AMESIM model except for the AMESim/Simulink block in Fig.4, the rest part of simulation model under SIMULINK in Fig.5 corresponds to the AMESim/Simulink block in Fig.4. The structure of the controller block in SIMULINK is shown in Fig. 3, and the calculation procedure can be expressed as follows:

Defining e as the position error:

$$e = x - x_{in} \quad (6)$$

The output signal of the controller can be expressed as:

$$u = \frac{C_d A}{q_n} \sqrt{\frac{2}{\rho} \left(\frac{p_s}{p_c} - 1 \right) p_n} + (Pe + I \int e dt + D \frac{de}{dt}) \quad (7)$$

Where $\frac{C_d A}{q_n} \sqrt{\frac{2}{\rho} \left(\frac{p_s}{p_c} - 1 \right) p_n}$ is the value of flow compensation, $(Pe + I \int e dt + D \frac{de}{dt})$ is the value of the PID controller.

4. Simulation and analysis

Based on the system structure and control strategy, the structure parameters of hydraulic components such as the hydraulic cylinder, the proportional valve are set in the model under AMESIM: the piston diameter is 50 mm; the rod diameter is 28 mm; the max stroke is 30 mm; the total mass of the moving part in hydraulic cylinder is 1.5 Kg; the nominal flow-rate of the proportional valve is 7.6 L/min under the condition that the nominal pressure is 3.5 MPa; the response frequency of the proportional valve is 30 Hz; the orifice diameter is 2 mm. The environmental parameters of simulation and solver are set under Simulink as follows: the supply pressure is 9.5 MPa; the load is 5400 N; The controller parameters of PID controller are designed as $P=85$, $I=2$ and $D=1$; the simulation time is 6 s; and the solver is ode15s (stiff/NDF) (ode15s routine built in MATLAB) with variable step. Others are left at the default settings. The control requirements of system are that the steady-state error is less than 0.03 mm; and the maximum percent overshoot is less than 4%.

4.1 Simulation results of PID controller and PID controller with flow compensation

During the simulation process, different target positions are simulated respectively.

Under PID controller, the simulation results are shown as follows: The input step target position curve and the current position of the piston are given in Fig.6. The absolute error is shown in Fig.7, and the output of PID controller is shown in Fig.8.

Fig.6 Position track under PID controller

Fig.7 Absolute position error under PID controller

Fig.8 Control signal of the valve under PID controller

From Fig.6, Fig.7 and Fig.8, the piston oscillates around the target position. The opening size of the valve port oscillates at a certain range.

Under PID controller with the flow compensation, the input step target position curve and the current position of the piston are given in Fig.9. The absolute error is shown in Fig.10. The flow rate at the orifice is shown in Fig.11.

Fig.9 Position track under PID controller with the flow compensation

Fig.10 Absolute position error under PID controller with the flow compensation

Fig.11 Flow rate at the orifice under PID controller with the flow compensation

It can be seen from Fig.9 and Fig.10 that the piston on the track of target position can be realized effectively. The position error is about 2×10^{-5} m. As shown in Fig.11, the flow rate at the orifice changes within a rather small range. The simulation results indicate that the system structure and the control method proposed in this paper can achieve the position control of the asymmetrical hydraulic cylinder with a good control performance.

The contrast between PID controller and the controller method presented in this paper indicates that the control result of PID controller with flow compensation is superior to that of PID controller.

4.2 Influences of the orifice size on the system

In the system model under PID controller with the flow compensation, the orifice diameter is configured to different values such as 2.05 mm, 2 mm and 1.95 mm, without changing other parameters. Figure 12 shows the absolute position error and Fig.13 shows the flow rate at the orifice under different orifice diameters. The simulation results under different orifice sizes are shown in Table 1.

Fig.12 Absolute position error under PID controller with the flow compensation

Fig.13 Flow rate at the orifice under PID controller with the flow compensation

Table 1 Simulation results under different orifice sizes

Target position (mm)	Absolute error(m)			Relative error		
	$D_0=1.95$ (mm)	$D_0=2$ (mm)	$D_0=2.05$ (mm)	$D_0=1.95$ (mm)	$D_0=2$ (mm)	$D_0=2.05$ (mm)
1	-2.0×10^{-5}	-7.48×10^{-6}	5.83×10^{-6}	2.07%	0.75%	0.58%
8	5.3×10^{-6}	1.68×10^{-5}	2.81×10^{-5}	0.06%	0.21%	0.35%
1	-1.2×10^{-5}	-2.02×10^{-6}	8.66×10^{-6}	1.2%	0.2%	0.87%

From Fig.12 and Table 1, the steady-state accuracy of the piston position changes when the diameter of orifice is changed. The absolute position error of the piston varies around 2×10^{-5} mm. The steady-state accuracy relative to the current target position increases when the diameter of the orifice is enlarged, while it declines evidently when the diameter of the orifice is reduced. Figure 13 shows that the flow rate at the orifice fluctuates within a small range under the condition that the diameter of orifice is 1.95 mm as the piston moves

backwards. In summary, the flow rate increases with the diameter of the orifice increasing. It varies around 2.81 L/min.

4.3 Influences of the proportional valve's response frequency

In the established system model under PID controller with the flow compensation, taking the signal shown in Fig.6 as the input target position, the response frequency of the proportional valve is configured to 5, 15, 30 and 45 Hz, and other parameters of this model are kept unchanged. Figure 14 shows the absolute error of the piston position and Fig.15 shows the flow rate of the valve.

Fig.14 Absolute position error under PID controller with the flow compensation

Fig.15 Flow rate of the valve under PID controller with the flow compensation

As shown in Fig.14, the curves are corresponding to the absolute error of the piston position at response frequency 5, 15, 30 and 45 Hz, respectively. The steady state error changed in a wide range and the stabilization time of the system increases under the condition that the response frequency of the proportional valve is 5, 15 Hz, and the absolute error has few differences between the response frequency 30 Hz and 45 Hz.

As shown in Fig.15, the curves stand for the flow rate through the proportional valve at response frequency 5, 15, 30 and 45 Hz, respectively. The flow rate fluctuate in wide ranges as the response frequency of the valve is 5 Hz and 10 Hz, respectively, and have rather small fluctuation for response frequency 30 Hz and 45 Hz.

From the analysis above, the steady state error and the stabilization time of the system increase when the response frequency of the proportional valve drops. But the static accuracy cannot be improved more if the frequency is greater than 30 Hz.

5. Conclusions

The simulation results and analysis lead to the following conclusions:

(i) The system structure presented in this paper can achieve the position control of asymmetric hydraulic cylinder system and get a good control performance. The traditional PID controller cannot get satisfied control results under the structure. In contrast, the steady state error and control effect are improved by the control method presented in this paper.

(ii) The system has less steady state error when the machining error in the orifice varies within some ranges under the flow compensation control strategy. This indicates that the control method has good robustness.

(iii) The simulation results demonstrate that the steady-state accuracy decreases dramatically as the response frequency of the proportional valve is too low. But the static accuracy cannot be improved effectively if the frequency is greater than 30 Hz.

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