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Simulation and Experimental Analysis of Hydraulic Directional Control for Displacement Controlled System

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ABSTRACT The characteristics of a hydraulic directional control method for the displacement controlled system are studied. The directional control is realized by the pilot-operated check valve. The reversing operation of the pilot-operated check valve system is investigated with simulation and experiment. The relationship between the valve spool motion and the circuit pressure has been analyzed during the directional control. The pressure reversing points during the directional control can be used to determine the operation state of the pilot-operated check valve system. The results indicate that the valve spool motion can be confirmed by the pressure reversing points. The response becomes faster with a smaller diameter control piston. A smaller hydraulic chamber volume is also an effective method to improve the dynamic response. Higher circuit pressures decrease the dynamic response. The results can be applied for the optimized design of the hydraulic directional control method with the pilot-operated check valve.

INDEX TERMS Directional control, dynamic modeling, hydraulic circuit, pressure reversing, pump controlled system.

I. INTRODUCTION

The hydraulic fluid power system is widely employed in industry applications. With different hydraulic components, the hydraulic fluid power system can be divided into different types, such as the valve controlled system [1] and the displacement controlled system [2]. The valve controlled system has fast response [3], [4]. The displacement controlled system achieves higher energy efficiency. For instance, the displacement controlled actuation can realise 15% fuel savings on a wheel loader [5], 20% fuel savings on a skid steer loader [6], and 40% fuel savings on an excavator [7]. However, the hydraulic fluid power system still lacks energy efficiency compared with the electric system. It is remarkably inefficient with efficiencies varying from 6% to 40% [8].

There are different types of hydraulic circuits used in the displacement controlled systems. The two types are described as the open circuit and the closed circuit [9]. In the open circuit system, the control valve plays an important role in the directional control [10]. Unavoidable throttle loss is caused by the control valve [11]–[13]. The valve also makes the control of the displacement controlled system become complicated. Different from the open circuit system, both chambers of the actuator are connected to the pump ports directly without control valves in the closed circuit system [14], [15]. The closed circuit system eliminates the valve throttle loss. Further, the energy recovery methods for the kinetic and the potential energy are probable [16].

With different actuators, the closed circuit system has different performance. A symmetric actuator, such as double rods cylinder, is limited by the small output force and installation space [9]. For a differential cylinder, the oil flows go through the two chambers are different. The asymmetric flows result in poor control performance [17], [18]. To simplify the directional control and compensate the asymmetrical flow, a pilot-operated check valve (PO check valve) system was proposed for the displacement controlled system [19]–[22]. The PO check valve system has been used for the power steering systems [23], [24]. An adaptive velocity controller was designed [25]. The new system promises improved energy efficiency [26]. The PO check valve dynamics are usually neglected in the hydraulic fluid power system modelling due to their fast nature [27]. Only the Flow characteristic of the valve is considered.

Compared with the valve controlled system, the displacement controlled system with the PO check valve system need improve the dynamic response. The dynamic response of the system is affected by the PO check valve performance. Further, the response of the system is related to the safety when the system is used for the hydraulic braking. To satisfy the further requirements on the dynamics and safety of the displacement controlled system with the PO check valve system, the valve spool motion needs a clear understanding. A reliable and simple method for the valve spool response analysis is needed, since the measurement of the spool displacement is difficult and the displacement sensor for high pressure application is also not cheap.

In this paper, the dynamic performance of the PO check valve system for the displacement controlled system are studied. A PO check valve model considering the interaction of the valve spools is built up. The directional control by the PO check valve system is investigated with simulation and test in detail. The pressure break points during the directional control are used to analyse the valve spool motion of the PO check valve system. The fundamental experimental results of the displacement controlled motor system with the PO check valve system are investigated. It is aimed to identify the main influence factors in the directional control of the displacement controlled system with the PO check valve system.

II. SYSTEM DESCRIPTION

Figure 1 provides two kinds of the displacement controlled system with the PO check valve system. Figure 1 (a) is a pump controlled cylinder system and Fig. 1 (b) is a pump controlled motor system. The actuator is controlled and adjusted by the variable pump with different speeds or displacements. For the pump controlled cylinder system, the pump inlet and outlet are connected to the piston and rod sides of the actuator. The different pressures between the actuator's uneven sides keep the low-pressure side of the actuator connected to a low-pressure source. The connection is realised by the PO check valves. The pressure levels of both sides are adjusted by pressure relief valves to protect from over pressurization.

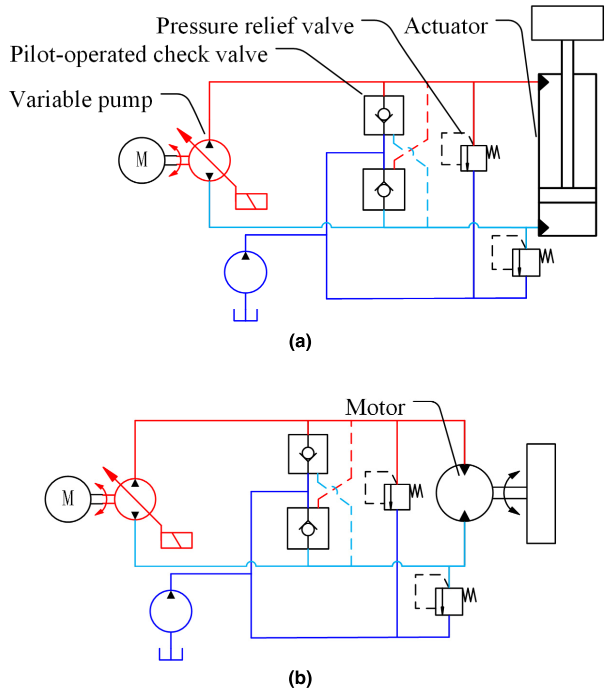


FIGURE 1. Hydraulic schematic of a pump controlled system. (a) A pump controlled cylinder system. (b) A pump controlled motor system.

III. MATHEMATICAL MODELS

The PO check valve dynamic performance affects the directional control dynamics. To obtain the dynamic performance of the PO check valve system, a test bench has been built up, as shown in Fig. 2 [28]. The test bench consists of electric motor, bidirectional variable displacement pump, PO check valve and pressure relief valve. The parameters of the test bench are listed in Table 1. The bidirectional variable displacement pump is used to simulate the pressure variation during the directional control. By adjusting the displacement of the pump, the flow rate and flow direction can be changed. The operational pressures also change with the pump displacement. To identify the PO check valve dynamic performance, the operational pressures are measured. The measured pressures have exclusive characteristics due to the motion of the valve spool.

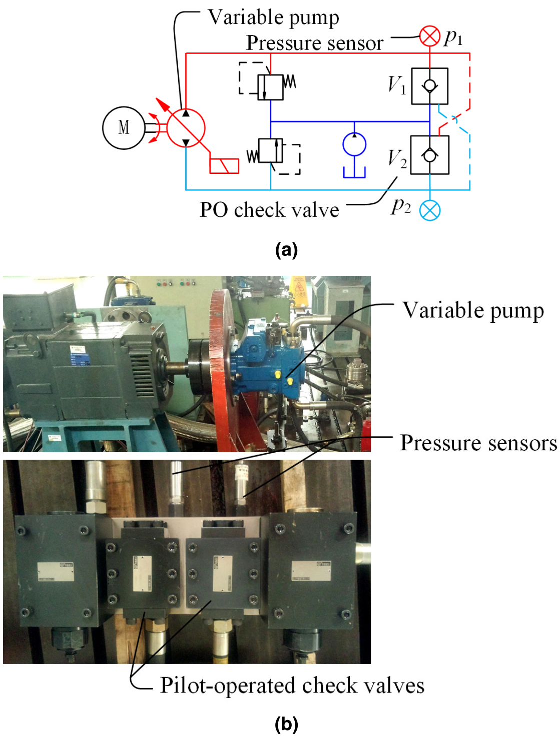


FIGURE 2. Test method of the system. (a) Schematic of test bench. (b) Apparatus of test bench.

TABLE 1. **Units for magnetic properties.**

Component	Parameter	Value
Pilot-operated check valve	Rated pressure	31.5 MPa
	Rated flow	400 l/min
Pressure sensor	Output	4~20 mA
	Range	0~40 MPa
Displacement sensor	Output	4~20 mA
	Range	0~500 mm
Data acquisition system	Channel	16
	Sampling frequency	1000 Hz

To investigate the directional control dynamics, mathematical models of the test bench components have been built up. The output flow and input torque of the bidirectional variable displacement pump are given by:

$$Q_p = V_{max} n_p r \eta_v \tag{1}$$

$$T_p = \Delta p V_{max} r \eta_m \tag{2}$$

where V_{max} is pump displacement, n_p is pump speed, Δp is differential pressure between the inlet and the outlet, r is pump displacement ratio, η_v and η_m are volumetric efficiency and mechanical efficiency, respectively.

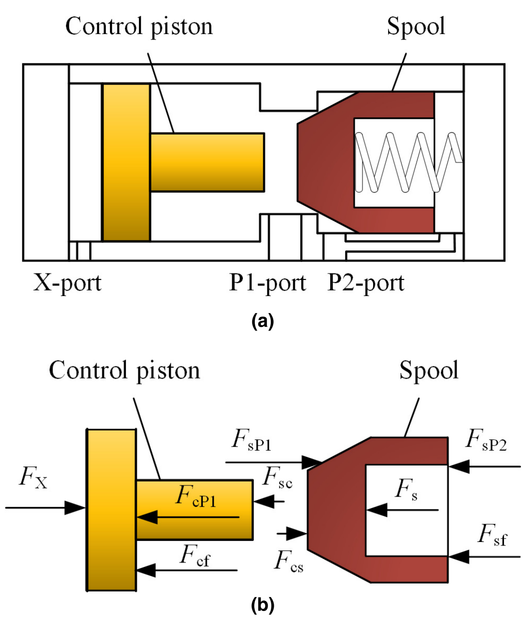


FIGURE 3. **The pilot-operated check valve. (a) Configuration of the valve. (b) Dynamic model of the valve.**

Figure 3 presents the configuration of the PO check valve. The positive direction is set from the control piston to the spool. And the spool velocity is greater than zero when the moving direction is consistent with the positive direction. It can be seen that the valve allows the oil to flow in two directions with the X-port pressure. When the oil flow from the P1-port to the P2-port, the spool motion is described by:

$$m_s \ddot{x} = F_{sP1} - F_{sP2} - F_s + F_{sf} sign(\dot{x}) \tag{3}$$

where x is spool displacement, m_s is spool mass, F_s is spring force, F_{sf} is friction force, F_{sP1} is spool force applied by the P1-port oil, F_{sP2} is spool force applied by the P2-port oil.

The spring force is calculated by

$$F_s = k_s(x - x_{s0}) \tag{4}$$

where k_s is spring stiffness, x_{s0} is spring pre-compression amount.

The friction force is given by

$$F_{sf} = c_s \dot{x} + f \tag{5}$$

where c_s is viscous damping coefficient, f_s is dynamic frictional resistance.

The spool forces applied by the P1-port oil and the P2-port oil are given by

$$F_{sP1} = \frac{p_{P1} \pi D^2}{4} \tag{6}$$

$$F_{sP2} = \frac{p_{P2} \pi D^2}{4} \tag{7}$$

where D is valve port diameter, p_{P1} and p_{P2} are oil pressures of the P1-port and the P2-port, respectively.

A small gap exists between the spool and the control piston. When the X-port pressure becomes larger and the gap is still larger than zero, the control piston motion is described by:

$$m_c \ddot{s} = F_X - F_{cP1} + F_{cf} sign(\dot{s}) \tag{8}$$

where s is piston displacement, m_c is piston mass, F_{cf} is friction force, F_X is piston force applied by the X-port oil, F_{cP1} is piston force applied by the P1-port oil.

The friction force is given by

$$F_{cf} = c_c \dot{s} + f_c \tag{9}$$

where c_c is viscous damping coefficient, f_c is dynamic frictional resistance.

The piston force applied by the X-port oil is

$$F_X = \frac{p_X \pi d^2}{4} \tag{10}$$

where p_X is X-port pressure, d is control piston diameter.

The piston force applied by the P1-port oil is

$$F_{cP1} = \frac{p_{P1} \pi d^2}{4} \tag{11}$$

When the spool and the control piston contact with each other, the motions of the control piston and spool are described:

$$\begin{cases} m_s \ddot{x} = F_{cs} + F_{sP1} - F_{sP2} - F_s - F_{sf} sign(\dot{x}) \\ m_c \ddot{s} = F_X - F_{sc} - F_{cP1} - F_{cf} sign(\dot{s}) \end{cases} \tag{12}$$

where F_{cs} and F_{sc} are interactive forces between the spool and the control piston, respectively. Further, the forces applied by the P1-port oil are calculated by the following equations:

$$F_{cP1} = \frac{p_{P1}\pi(d^2 - d_{rod}^2)}{4} \tag{13}$$

$$F_{sP1} = \frac{p_{P1}\pi(D^2 - d_{rod}^2)}{4} \tag{14}$$

where d_{rod} is control piston rod diameter.

The interactive force is calculated considering the damping effect of a thin oil layer.

$$F_{cs} = F_{sc} = \frac{3\pi\mu R^4 \dot{x}_{gap}}{2x_{gap}^3} \tag{15}$$

where μ is oil dynamic viscosity, R is equivalent contact radius, and x_{gap} is gap between the spool and the control piston. The gap is expressed as

$$x_{gap} = x_{gap0} - (x - s) \tag{16}$$

where x_{gap0} is initial gap when the valve closes.

The flow rate through the PO check valve is given by:

$$Q_v = CA_v\sqrt{\frac{2\,|p_{P1} - p_{P2}|}{\rho}} \tag{17}$$

where A_v is valve flow area, C is flow coefficient, ρ is oil density.

The valve flow area is calculated by

$$A_v = \pi x \sin\alpha(D - x\sin\alpha\cos\alpha) \tag{18}$$

where α is half poppet angle of the valve.

The pressure response of the hydraulic chamber can be defined by:

$$\frac{dp}{dt} = \frac{B_{oil}\,(Q - dV/dt)}{V} \tag{19}$$

where p is pressure, B_{oil} is oil bulk modulus, Q is instantaneous flow, V is instantaneous volume.

The dynamics of the pressure relief valve are ignored in the simulation.

IV. RESULTS AND DISCUSSION

A. VALVE SPOOL DYNAMICS

The measured hydraulic circuit pressure dynamics of the test bench are shown in Fig. 4. The pump displacement changes from the maximum positive value to the maximum negative value in the tests. The pump displacement signal is the control signal of the servo varying mechanism. It is not the real-time pump displacement. Thus, there is a phase difference between the measure pressure and the control signal. Further, there is a clear pressure reversing process with the change of the displacement. The test results also prove a good repeatability.

The comparison between the simulated and measured results is shown in Fig. 5. The simulation model parameters are presented in Table 2. It seems that the simulated results have the same trend with the measured results. The mathematical model can describe the dynamics of the PO check valve. It seems that the spool motion can be judged through the pressure dynamics although the measurement of the displacements is difficult, as shown in Fig. 5 (a). A clear pressure reversing between 168.8 s and 168.9s is being measured. Further, it seems that the pressure p_2 increases only after

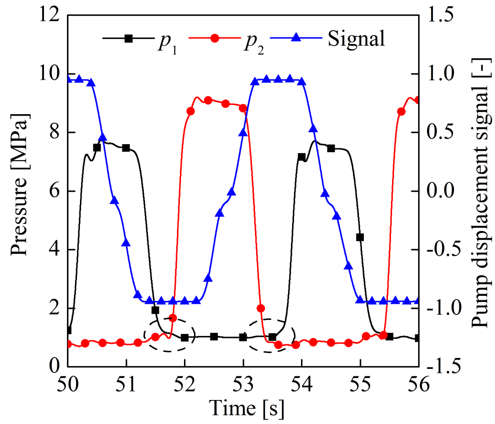
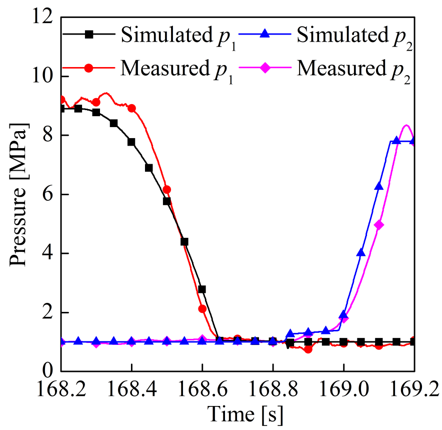
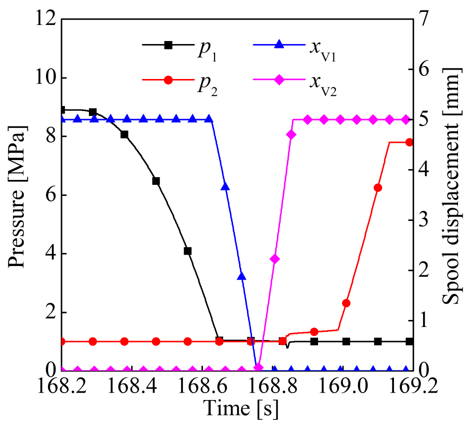


FIGURE 4. Measured pressures and pump displacement signal during the directional control.



(a)



(b)

FIGURE 5. Simulated and measured results during the directional control. (a) Simulated and measured pressures. (b) Simulated pressures and spool displacements.

TABLE 2. Main parameters used in the simulation.

Parameter	Value	Parameter	Value
V_{max}	125 ml	d_{rod}	10 mm
η_v	0.98	x_{gap0}	2 mm
η_m	0.9	α	45°
m_s	0.050 kg	k_s	10 N/mm
m_c	0.050 kg	D	29 mm
B_{oil}	1000 MPa	d	50 mm

the V2 is completely closed. The decreasing of the pressure p_1 and increasing of the pressure p_2 lead to the opening of the V1, as shown in Fig. 5 (b). The dynamic responses are affected by the hydraulic circuit parameters such as the dead volume and the oil bulk modulus. The valve parameters also affect the dynamic responses.

B. PARAMETER INFLUENCES

It is important to improve the dynamic response of the hydraulic fluid power system. Thus, the influencing parameter effects on the dynamic response are analysed. The effect of the hydraulic chamber volume is shown in Fig. 6. The volume influence on the pressure mainly appears in the starting and ending process. Different volumes affect the pressure slope. The influences of the different circuit chamber volumes are decoupled. The interactive effects are very small. A smaller hydraulic chamber volume is an effective method to improve the dynamic response.

The simulated results with different diameters of the control piston are presented, as shown in Fig. 7. It can be seen that the response becomes faster with a smaller diameter of the control piston. It is because that a smaller diameter of the control piston can reduce the resistance force F_X applied by the X-port pressure. Therefore, the spool closing speed increased significantly. The spool opening speed is not improved obviously. The diameter of the spool is mainly determined by the rated flow of the PO check valve. A smaller diameter is beneficial for increasing the dynamic response since the spool becomes lighter. However, it also causes a larger throttle loss.

The effects of the circuit pressure are shown in Fig. 8. The time-consuming of the directional control increases with higher pressures. It can be seen that the lower pressures can make the pressure exchange response faster. However, the time of the whole process becomes longer. The pressure slope is mainly determined by the oil bulk modulus and the hydraulic chamber volume. With a higher oil bulk modulus, a faster response can be achieved, as shown in Fig. 9. When the circuit pressures increase, it is necessary to ensure an adequate response speed by reducing

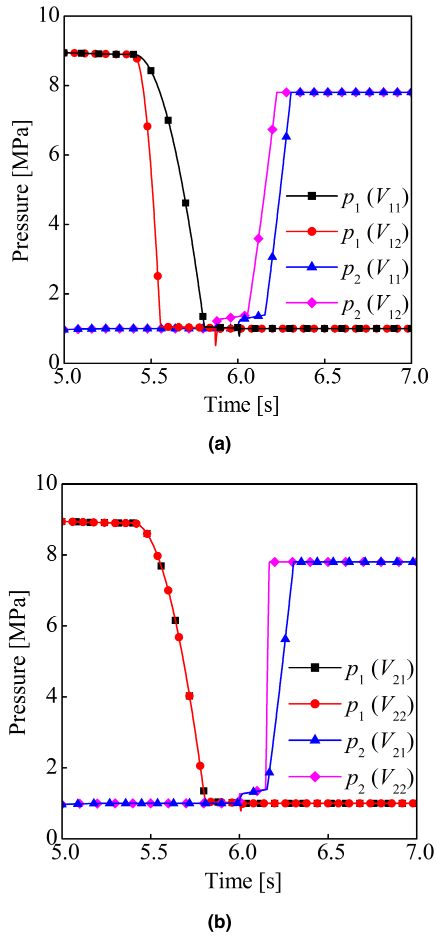
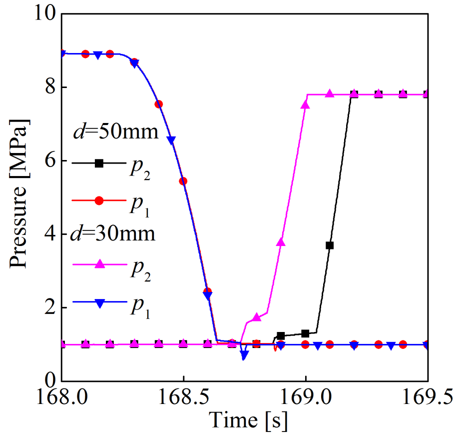


FIGURE 6. Effects of different hydraulic chamber volumes. (a) Different chamber volumes in p_1 side. The hydraulic chamber volume V_{11} is larger than the V_{12}. (b) Different chamber volumes in p_2 side. The hydraulic chamber volume V_{21} is larger than the V_{22}.

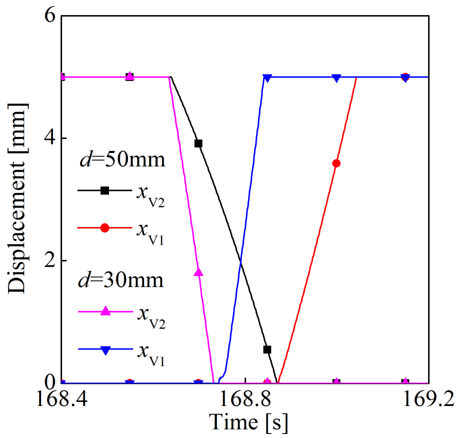
the hydraulic chamber volume and increasing the oil bulk modulus.

C. APPLICATION IN DISPLACEMENT CONTROLLED MOTOR SYSTEM

The test bench of the displacement controlled pump system using the hydraulic directional control method has been built up. The experimental results of the directional control are shown in Fig. 10. There are the two pressure reversing operations in the test. When the motor pressure change, the pressure reversing operation starts. After time t_1, the motor changes the rotating direction. The high and low pressures of the motor ports exchanged. The same operation appears at the time t_2. The pressure reversing is realised in the same way. Fig. 10 (b) is the reversing process enlarged. After the reversing of the pressure of the motor ports, the regenerative reversing process starts. The motor speed decreases. The system kinetic energy makes the pressure difference between the motor ports increase. The motor operates in the pump mode. At the time t_2, the motor speed is braked to zero with a total time of about 3 s. And then the motor changes



(a)



(b)

FIGURE 7. Effects of different control piston diameters. (a) Simulated pressures. (b) Simulated displacements.

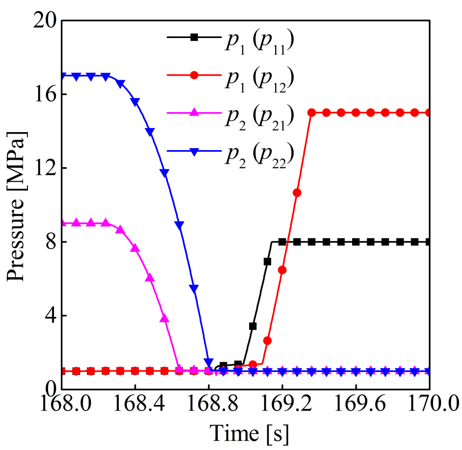


FIGURE 8. Effects of different pressures. p12 and p22 are higher than p11 and p21.

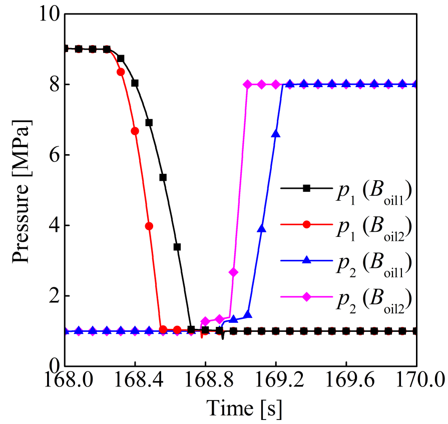
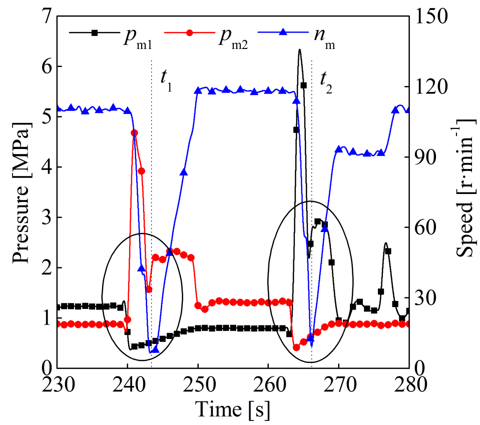
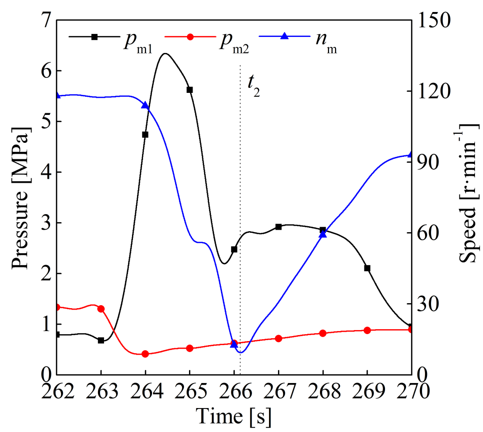


FIGURE 9. Effects of different oil bulk modulus. The oil bulk modulus B_{oil1} is less than the B_{oil2}.



(a)



(b)

FIGURE 10. Measured results of the motor operation characteristics during the directional control. (a) Variation of the motor pressures and speed. (b) Measured parameters enlarged.

the rotating direction and complete the reversing process. In practice, the hydraulic directional control method should be optimized according to the system response demand. It is aimed to meet the requirements of the system response and ensure the operational security. Further, the cost should be limited.

V. CONCLUSIONS AND FUTURE WORK

The dynamic performance of the hydraulic directional control method for the displacement controlled system is investigated. The valve port pressure responses during the directional control are measured and discussed. The results suggest the following.

(1) The PO check valve spool motions can be determined by the measured pressure reversing points. A clear pressure exchange can be found when the spool contacts or leaves the valve seat. The PO check valve model can be used for the system dynamic analysis.

(2) The response becomes faster with a smaller diameter of the control piston. A smaller hydraulic chamber volume is also an effective method to improve the dynamic response. Higher circuit pressures decrease the directional control response.

(3) The hydraulic directional control method can be used for the displacement controlled motor system. The regenerative pressure reversing can be realised and the regenerative energy can be stored with the method.

APPENDIX

A_v	flow area of the pilot-operated check valve [mm^2]
$B_{oil}, B_{oil1}, B_{oil2}$	oil bulk modulus [MPa]
c_c	viscous damping coefficient
c_s	viscous damping coefficient
C_c	flow coefficient
d	control piston diameter [mm]
d_{rod}	control piston rod diameter [mm]
D	valve port diameter [mm]
f_c	dynamic frictional resistance[N]
f_s	dynamic frictional resistance [N]
F_{cP1}	piston force applied by the P1-port oil [N]
F_{cf}	friction force [N]
F_{cs}	contact force between the spool and the control piston [N]
F_s	spring force [N]
F_{sc}	contact force between the spool and the control piston [N]
F_{sf}	friction force [N]
F_{sP1}	spool force applied by the P1-port oil [N]
F_{sP2}	spool force applied by the P2-port oil [N]
F_X	piston force applied by the X-port oil [N]
k_s	spring stiffness [N/mm]
m_c	piston mass [kg]
m_s	spool mass [kg]
p	Pressure [MPa]
α	half poppet angle of the valve [$\circ$]
Δp	differential pressure between inlet and outlet [MPa]
η_v	volumetric efficiency
η_m	mechanical efficiency
μ	oil dynamic viscosity [N·s/m^2]
ρ	oil density [kg/m^3]

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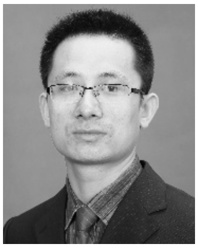

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