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Modeling and dynamic characteristics analysis on a three-stage fast-response and large-flow directional valve

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ABSTRACT

The large transient power hydraulic systems, characterized by high pressure, large transient flow and high output power, have widespread industrial applications in converting powerful hydraulic energy to kinetic energy in a transient period. A conventional large flow rate directional valve is unable to be used in these applications due to the slow response. A directional control valve with fast response and high flow capacity simultaneously is presented for the large transient power hydraulic system in this paper. The valve utilizes a three-stage structure with two high-speed on/off solenoid valves as the pilot stage and two cartridge poppet valves as the secondary stage to overcome the fundamental trade off between valve response and flow capacity. A precise mathematical model of this valve considering both turbulent flow and laminar flow is developed. A test apparatus which has the ability to provide and measure transient large flow is built. The flow rate is estimated based on the pressure dynamics. The property parameters in the simulation model are optimized against measured data. According to the dynamic characteristics analysis, the valve response is split into the starting delay and opening time. The step response is rapid enough to provide a large transient flow, while the high flow capacity is not reduced due to the fast response. The main control pressure is characterized by its change time and critical open pressure and these two parameters determine the main-stage response. Some key structural factors concerning with these two parameters are discussed in detail and optimize to further reduce the response time.

1. Introduction

The large transient power hydraulic systems have widespread industrial applications such as hydraulic operating mechanism for high voltage circuit breaker [1] and high-speed hydraulic catapult system [2]. They need to convert powerful hydraulic energy to kinetic energy in a transient period. The control for these systems is difficult due to the high hydraulic pressure, large transient flow and high output power [1]. The control valve is one of the most important components in these hydraulic systems. In order to meet the demand of large flow for hydraulic system, the control valve needs to have high flow capacity. In existing research, the two-stage or three-stage valve structure gives an attractive solution to attain a high flow capacity for the control valve. There were literatures on multi-stage structure used in proportional/servo valve [3–7], directional on/off valve [8] and high-speed switching valve [9]. In addition to high flow capacity, the large transient power hydraulic systems have tougher requirements of response than general hydraulic systems. However, the design of this kind of control valve is difficult because there is a conflict between large flow and short response time due to the factors like high flow forces [10], large spool mass and stroke, etc.

Most current methods to reduce the response time are by means of fast response electromechanical devices. High-speed on/off solenoids are usually used in high-speed switching valves as electromechanical devices. Their step response time is less than 3.5 ms in the present literature [9,11–14]. Moreover, it can obtain a larger magnetic force and a lower power by adjusting the structural parameters of the solenoid, proposed by Tao et al. [15]. Compared with the high-speed solenoid, the piezoelectric system is an alternative way to reduce the response time. Piezoelectric actuators have advantages of a fast response and high output force [16,17]. However, their drawback is small stroke even at a large applied voltage, which restricts orifice gap and in turn limits maximum achievable flow rate [18]. Additionally, Magnetorheological (MR) fluids are a kind of potentially simpler and more reliable material as electromechanical devices. The particular advantage of MR valves is no moving parts and can offer fast switching speed [19]. The reported response times of MR fluid and MR devices cover a broad range of 0.1–100 ms, depending on the method applied [20]. However, a potential disadvantage of this actuator is insufficient block force [21]. It means the MR fluid is not suitable for the large power hydraulic systems. Therefore, comparing these

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Nomenclature

A_a	chamber A area (m^2)
A_b	chamber B area (m^2)
A_{c2}	secondary valve control chamber area (m^2)
A_{c3}	main valve control chamber area (m^2)
A_{cp}	cylinder piston side area (m^2)
A_{cr}	cylinder rod side area (m^2)
A_{in}	pilot valve inlet chamber area (m^2)
A_K	pilot valve chamber K area (m^2)
A_{out}	pilot valve outlet chamber area (m^2)
A_P	main valve chamber P area (m^2)
A_T	main valve chamber T area (m^2)
A_Z	main valve chamber Z area (m^2)
A_{Vi}	valve orifice area (i = 1, 2, 3) (m^2)
B	magnetic flux density (V s/m^2)
b_m	viscous friction coefficient
C_d	discharge coefficient of orifice 4
C_i	discharge coefficient of orifice 5
C_{qi}	valve discharge coefficient for turbulent flow (i = 1, 2, 3), C_{q1} = 0.707, C_{q2} = 0.72, C_{q1} = 0.804
C_{qli}	valve discharge coefficient for laminar flow (i = 1, 2, 3), 0.04
C_{ri}	radial clearance (i = 1, 2, 3) (m)
C_v	velocity coefficient
D_{cp}	cylinder piston diameter (m)
D_{cr}	cylinder rod diameter (m)
d_c	clearance between the spool and the sleeve (m)
d_{Hi}	spool hydraulic diameter (i = 1, 2, 3) (m)
d_{si}	spool diameter (i = 1, 2, 3) (m)
d_o	orifice 4 diameter (m)
d_{so}	orifice 5 diameter (m)
F_e	magnetic force (N)
F_{fi}	flow force (N) (i = 1, 2, 3)
F_m	solenoid output force (N)
F_{s1}	pilot valve spring force (N)
F_{s2}	secondary valve spring force (N)
F_{sm}	solenoid spring force (N)
F_{vi}	spool viscous damping force (i = 1, 2, 3) (N)
i_m	solenoid current (A)
k_{s1}	pilot valve spring stiffness (N/m)
k_{s2}	secondary valve spring stiffness (N/m)
k_{sm}	solenoid spring stiffness (N/m)
L_m	inductance (H)
L_{si}	total spool length in contact with sleeve for damping (i = 1, 2, 3) (m)
m_i	spool mass (i = 1, 2, 3) (kg)
N	coil turns number
p_A	chamber A pressure (Pa)
p_B	chamber B pressure (Pa)
p_{c2}	secondary valve control chamber pressure (Pa)
p_{c3}	main valve control chamber pressure (Pa)
p_{c3cr}	critical open value of main valve control chamber pressure (Pa)
p_{cp}	cylinder piston side pressure (Pa)
p_{cr}	cylinder rod side pressure (Pa)
p_{in}	pilot valve inlet chamber pressure (Pa)
p_K	pilot valve chamber K pressure (Pa)
p_{out}	pilot valve outlet chamber pressure (Pa)
p_P	main valve chamber P pressure (Pa)
p_T	main valve chamber T pressure (Pa)
p_Z	main valve chamber Z pressure (Pa)
Δp_i	valve pressure difference (i = 1, 2, 3) (Pa)
q_i	valve flow rate(i = 1, 2, 3) (m^3/s)
q_o	orifice 4 flow rate (m^3/s)
q_{so}	orifice 5 flow rate (m^3/s)
r_c	main spool port fillet (m)
Re_i	Reynolds number (i = 1, 2, 3)
Re_{cri}	critical Reynolds number (i = 1, 2, 3), Re_{cr1} = 100; Re_{cr2} = 100; Re_{cr3} = 254
R_m	electrical resistance (Ω)
R_{eddy}	eddy currents resistance (Ω)
s_i	spool stroke (i = 1, 2, 3) (m)
Δt_1	starting delay (s)
Δt_2	opening time (s)
U_L	inductance voltage (v)
U_{in}	input voltage (v)
v_{cy}	cylinder velocity (m^3)
V_A	chamber A volume (m^3)
V_B	chamber B volume (m^3)
V_{c2}	secondary valve control chamber volume (m^3)
V_{c3}	main valve control chamber volume (m^3)
V_{cy}	cylinder piston side volume
V_{in}	pilot valve inlet chamber volume (m^3)
V_K	pilot valve chamber K volume (m^3)
V_{out}	pilot valve outlet chamber volume (m^3)
V_P	main valve chamber P volume (m^3)
V_T	main valve chamber T volume (m^3)
V_Z	main valve chamber Z volume (m^3)
x_{cy}	cylinder displacement (m)
x_{gap}	gap between the solenoid and pilot spool (m)
x_i	spool displacement (i = 1, 2, 3) (m)
x_{lap}	spool underlap (m)
x_m	solenoid displacement (m)
x_{s1}	pilot valve spring precompression length (m)
x_{s2}	secondary valve spring precompression length (m)
x_{sm}	solenoid spring precompression length (m)
x_{test}	measured data
x_{sim}	simulation data
α	poppet half angle (deg)
β	oil bulk modulus (Pa)
δ	air gap in solenoid (mm)
μ	dynamic viscosity of oil (Pa s)
ν	kinematic viscosity (m^2/s)
θ	jet angle of the fluid (deg)
ρ	density of oil (kg/m^3)
φ	flux (Wb)

electromechanical devices, high-speed on/off solenoid is an appropriate choice for electromechanical devices in high flow capacity valve.

Another method to shorten the response time is optimizing structural parameters of the control valve. Liu et al. [22] researched the effect of return spring stiffness and pre-tightening on the dynamic characteristics of three-way proportional reducing valve. Hu et al. [9] pointed out that the time between the step of the control signal and the response of the pilot valve was important for reducing the valve response lag in the three-stage high-speed switching valve. They proposed that a smaller gap was helpful in improving the valve response. Fu et al. [6] studied the parameters influence on main stage dynamic characteristics in the large flow cartridge servo proportional valve. The optimal parameters were selected to lower the response time, reduce the vibration frequency and overshoot of the valve step response.

Previous research on the high flow capacity and fast response valve focused on the proportional/servo or high-speed switching valve. However, the valve only needs to control the flow direction in the large transient power hydraulic systems. It should have both