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ANALYSIS OF SWITCHING TIME FOR PILOT OPERATED DIRECTIONAL CONTROL VALVE

ANALIZA CZASU PRZESTEROWANIA ROZDZIELACZA STEROWANEGO POŚREDNIO

Abstract

The paper presents an analysis of switching time for pilot operated directional control valve [1–3] in a typical hydraulic system, in which the actuator is a hydraulic cylinder. A mathematical model has been built which contains equations of motion for the valve and actuator components, the balance of fluid flow, flow through the gap, and the principle of conservation of momentum and the flow curves of pressure relief valves [6, 8]. The system of equations is solved by fourth-order Runge-Kutta method. A survey to find minimal switching time has been conducted [4]. Positive overlap value, the coefficient of its movement resistance and control pressure value have significant impact on the switching time in the tested valve.

Keywords: *modeling of hydraulic systems*, *spool type directional control valve*

Streszczenie

W artykule przedstawiono analizę czasu przesterowania suwakowego rozdzielacza pośredniego działania [1–3] w typowym układzie hydraulicznym, w którym elementem wykonawczym jest siłownik hydrauliczny. Zbudowano model matematyczny zawierający równania ruchu elementów zaworu i siłownika, bilansu przepływu cieczy, przepływu przez szczelinę i zasady zachowania pędu oraz charakterystyki zaworów przelewowych [6, 8]. Układ równań rozwiązano metodą zmiennokrokową Runge-Kutty 4 rzędu. Przeprowadzono badania polegające na poszukiwaniu najkrótszego czasu przesterowania [4]. Istotny wpływ na czas przesterowania w badanym rozdzielaczu ma wartość dodatniego przekrycia suwaka, współczynnik oporów jego ruchu oraz wartość ciśnienia sterownia.

Słowa kluczowe: *modelowanie układów hydraulicznych*, *rozdzielacz hydrauliczny suwakowy*

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Designations

m_s, m_t – computational mass of directional control valve spool (or cylinder rod and piston) respectively, plus the mass of retainer, spring and volume mass of attached liquid
V_i – the volume of the *i*-th line of the hydraulic system
B_i – equivalent volumetric elastic modulus for *i*-th line of the hydraulic system
Q_i – volumetric flow rate at the input for *i*-th line of the hydraulic system
Q_p – pump volumetric efficiency
p_i – pressure at *i*-th line of the hydraulic system
p_{z1} – cracking pressure for pressure relief valve at the system input
dp_{z1} – excess of cracking pressure for relief valve at the system input
p_{z2} – cracking pressure for pressure relief valve at the cylinder output
dp_{z2} – excess of cracking pressure for pressure relief valve at the cylinder output
α_s, α_t – kinematic coefficient of friction for the valve spool (or cylinder rod and piston)
c – centering spring stiffness coefficient for directional control valve
F_{ss0} – centering spring preload force for directional control valve
F_{ss} – centering spring force for directional control valve
F_{ts}, F_{tt} – kinematic friction force of directional control valve (or cylinder rod and piston)
x_1 – coordinate of the spool position (from the neutral position)
F_{hs} – control pressure force acting on the surface of the valve spool
A_s – active area of the valve spool affected by pressure p_1
F_{ht} – resultant pressure force acting on the surface of the piston/piston rod
F_{ds} – resultant hydrodynamic force acting on the spool of directional control valve,
x_2 – coordinate of position of the cylinder piston/piston rod (from the maximum extended position)
v_2 – speed of movement of the piston/piston rod
S_1 – sectional area of the piston rod
S_k – active cross-sectional area on the side of the cylinder piston rod
S_s – width of the flow gap at the valve spool
S_k – minimum cross section of stream at the valve channel
x_{p1} – overlap value for the valve spool
μ_1 – liquid outflow coefficient
ρ – fluid density
α_i – fluid inlet/outlet angle for *i*-th control edge
v_{Qi} – fluid flow speed at the *i*-th control edge

1. Introduction

Control of working movements at the machinery or equipment with hydraulic drive is usually carried out by means of directional control valves, usually spool type [1, 2]. Start or change of direction of the working movement is caused by the spool shift. Therefore, switching of the directional control valve is forcing the system, so switching time is important in the valve operation. By changing the switching time, we can change time constants for the whole hydraulic system [11]. In industrial applications, there are many solutions to allow forcing of hydraulic directional control valves, for example in the form of slots cut in the spool, proportional spool position control by a given course, etc. [5, 7, 10, 12]. All of these solutions increase the time constant of the directional control valve, which usually has a positive effect on the elimination of hydraulic hammer and pressure excess at start of the system. However, in some applications of directional control valves, such as safety devices, in order to obtain a rapid response of the machinery or equipment, the switching time should be as short as possible [9]. In this paper, the task undertaken is to determine parameters associated with the spool which affect the speed of the opening of flow path at the valve. The study has been conducted using the hydraulic pilot operated directional control valve type WEH22E [13].

2. Object of the study

General view of the directional control valve is shown in Fig. 1. It is built of the main body 1, the spool 2, the retainers 3 and the centering spools 4 , the covers 5 forming the pressure chambers 6 and 7 and the pilot valve 8. Switching the valve is caused by the fluid under pressure provided into the chamber 6 (or 7) and the simultaneous relief of the opposite chamber 7 (or 6). As a result of the force of the fluid pressure, the spool overcoming the force of the spring , motion resistance and hydrodynamic force, moves toward the extreme position forming different configuration of connections.

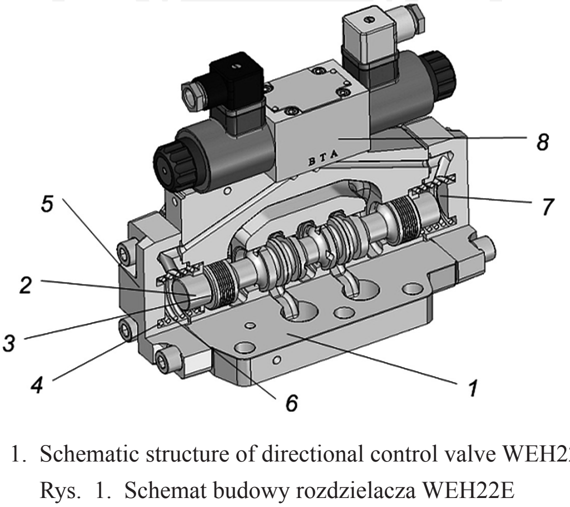

Fig. 1. Schematic structure of directional control valve WEH22E

Rys. 1. Schemat budowy rozdzielacza WEH22E

3. Mathematical model

Operation of directional control valve WEH22E has been examined in the hydraulic system, the schematic diagram of which is shown in Figure 2. Hydraulic cylinder 4 is assumed to be working element. The system is supplied by a fixed displacement pump of capacity Q_p. The pump is connected with a hydraulic directional control valve by means of the conduit with volume V_1 and pressure p_1. This line is characterized by constant volumetric elastic modulus B_1. At the output of the directional control valve and the inlet of the hydraulic line having a volume V_2 and equivalent volumetric elastic modulus B_2, flow is Q_1. Pressure in this line is p_2. The liquid of flow rate Q_2 flows into the hydraulic cylinder and flows out of flow rate Q_3 to the hydraulic line V_3 with an equivalent volumetric elastic module B_3. In this line, there is pressure p_3. The cylinder load is simulated by means of pressure relief valve 6. On energizing the solenoid 3, pilot valve 1 gives pressure into control chamber of the spool initiating the process of moving the spool and allowing the flow of fluid in the system. It is assumed that the system is in thermal equilibrium, and the mass of the liquid is concentrated in specific points of the system, that is, the valve spool, the individual sections of the line and the rod (and the piston) of the cylinder. The pressure in the drain line is omitted. Under these assumptions, the mathematical model will constitute the differential equations for the valve spool movement and cylinder piston rod movement, equations of flow continuity and the principle of conservation of momentum as well as equations of flow through the slot and performance curves of pressure relief valves.

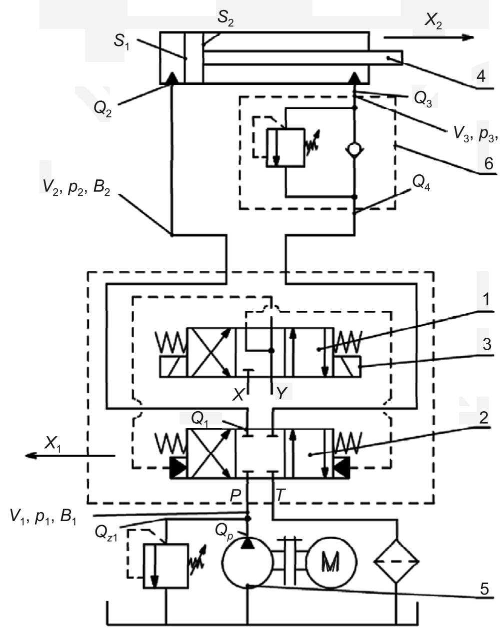

Fig. 2. Hydraulic diagram with the marked system parameters

Rys. 2. Schemat hydrauliczny wraz z oznaczeniami parametrów układu

Equations of the forces acting on the spool of directional control valve:

$$m_s \frac{d^2 x_1}{dt^2} + F_{ts} + F_{ss} + F_{ds} = F_{hs} \tag{1}$$

Equation of the cylinder piston rod motion:

$$m_t \frac{d^2 x_2}{dt^2} + F_{tt} = F_{ht} \tag{2}$$

Flow balance equation for volume V_1:

$$Q_p - Q_1 - Q_{z1} = \frac{V_1}{B_1}\frac{dp_1}{dt} \tag{3}$$

Flow balance equation for volume V_2:

$$Q_1 - Q_2 = \frac{(V_2 + S_1 \cdot x_2)}{B_2}\frac{dp_2}{dt} \tag{4}$$

Flow balance equation for volume V_3:

$$Q_3 - Q_4 = \frac{(V_3 - S_2 \cdot x_2)}{B_3}\frac{dp_3}{dt} \tag{5}$$

Volumetric flow rate Q_2 and Q_3 can be determined from the relation:

$$Q_3 = S_2 \cdot v_2, \quad Q_2 = S_1 \cdot v_2 \tag{6}$$

Volumetric flow rate through the slot Q_1 can be determined:

$$Q_1 = \mu_1 \cdot S_s(x_1) \cdot \sqrt{\frac{2 \cdot (p_1 - p_2)}{\rho}} \tag{7}$$

Function $S_s(x_1)$ is determined on the base of measurements on 3D model and approximated by the function:

$$\begin{aligned} &S_s(x_1) = a_1 \cdot (x_1 - x_{p1}) + a_2 \cdot (x_1 - x_{p1})^2 \quad \text{for} \quad S_s < S_k \\ &S_s = S_k \quad \text{for} \quad S_s \geq S_k \\ &S_s = 0 \quad \text{for} \quad x_1 < x_{p1} \end{aligned} \tag{8}$$

Volumetric flow rate Q_4 can be determined from the valve performance curve and describe by the relation:

$$\begin{gathered} Q_4 = k_4 \cdot (p_3 + dp_{z2}) \quad \text{for} \quad p_3 > p_{z2} \\ Q_4 = 0 \quad \text{for} \quad p_3 \leq p_{z2} \end{gathered} \tag{9}$$

Volumetric flow rate Q_{z1} can be determined from the valve performance curve and describe by the relation:

$$\begin{gathered} Q_{z1} = k_{z1} \cdot (p_1 + dp_{z1}) \quad \text{for} \quad p_1 > p_{z1} \\ Q_{z1} = 0 \quad \textit{for} \quad p_1 \leq p_{z1} \end{gathered} \tag{10}$$

Spring force F_{ss} can be determined from the relation:

$$F_{ss} = F_{ss0} + c \cdot x_1 \tag{11}$$

Viscous friction force of the spool F_{ts} and piston rod F_{tt} can be determined from the relation:

$$F_{tt} = \alpha_t \cdot x_2, \quad F_{ts} = \alpha_s \cdot x_1 \tag{12}$$

Forces from control pressure acting on the spool F_{hs} and cylinder piston rod F_{ht} can be determined from the relation:

$$F_{ht} = A_1 \cdot p_2 - A_2 \cdot p_3, \quad F_{hs} = A_a \cdot p_1 \tag{13}$$

The resultant value of hydrodynamic forces acting on the valve spool can be calculated from the relation (Figure 3):

$$F_{hd1} = \rho \cdot [Q_1(v_{Q1} \cos\alpha_1 - v_{Q2} \cos\alpha_2) - Q_4(v_{Q4} \cos\alpha_4 - v_{Q5} \cos\alpha_5)] \tag{14}$$

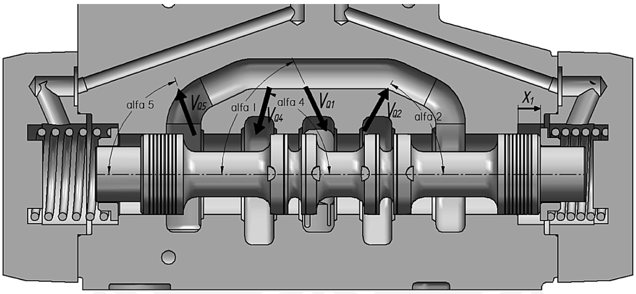

Fig. 3. Inlet and outlet angle of fluid stream on control edges

Rys. 3. Kąt wlotu i wylotu strugi cieczy na krawędziach sterujących

4. Test results

In order to determine the time for full opening of the directional control valve, system of equations (1) to (14) is resolved, using the data from technical information of the directional control valve WEH22E. It was assumed that the control pressure (X) is obtained from the main supply line of the system (P). Moreover, the figures are taken for the cylinder: diameter D = 100 mm, d_t = 56 mm and stroke 1000 mm. Forcing at the input is pump with constant flow Q_p = 480 dm^3/min. Example response of the spool to step forcing for standard value of positive spool overlap (x_{p1} = 5 mm) and reduced overlap (x_{p1} = 0.5 mm) are shown in Fig. 4. Fig. 5 and 6 show respectively pressure course in the volume V_1 and piston rod velocities v_2. As follows from the courses, the lower value of overlap parameter (x_{p1} = 0.5 mm), the valve spool moves a little more slowly to the end position. This is due to a lower pressure during the start-up of the system (Fig. 5). Reducing the spool overlap value, however, results in earlier opening of the flow ways and the piston rod faster begins to move, as is shown in Fig. 6.

Reduction of the directional control valve overlap allows to reduce the response time of the valve in the initial phase of its operation. In the final stage, speed of the spool movement decreases due to the pressure drop at pressure line, from where control pressure is taken. Excessive reduction of the spool overlap value will result in increased internal leakage. The study for various combination of parameters have shown that in addition to reducing the spool overlap value, reduction of spool movement resistance and increase of control pressure have a significant impact on minimizing the switching time.

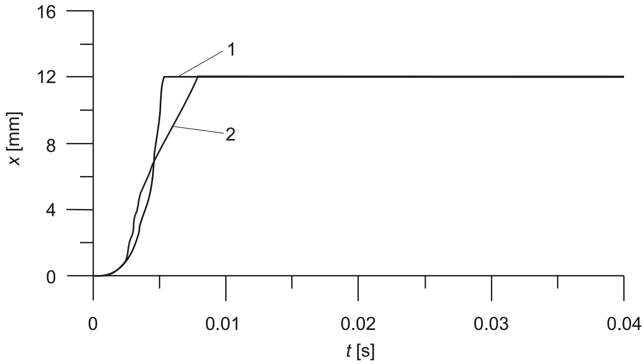

Fig. 4. Spool valve movement as a response to step forcing

Rys. 4. Przemieszczenie suwaka rozdzielacza jako odpowiedź na wymuszenie skokowe

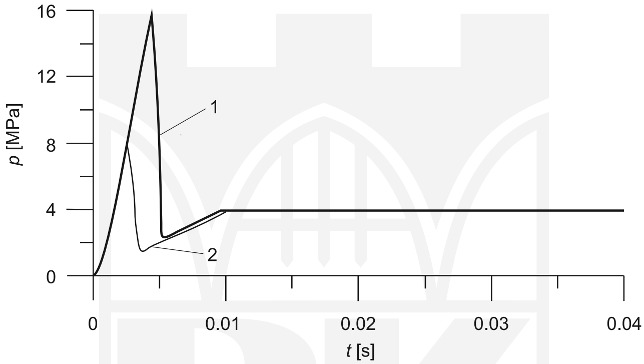

Fig. 5. Pressure course at the system inlet

Rys. 5. Przebieg ciśnienia na wejściu do układu

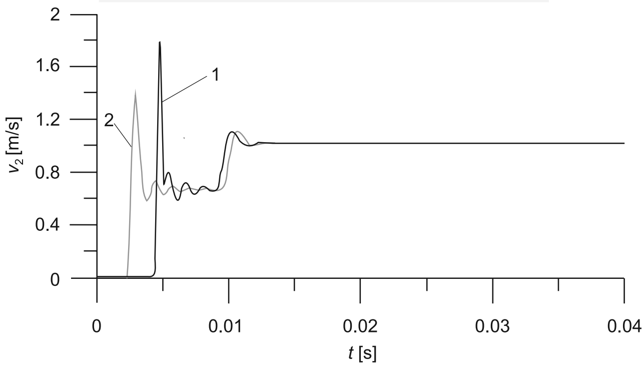

Fig. 6. Velocity course for the cylinder piston rod

Rys. 6. Przebieg prędkości tłoczyska siłownika

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