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MODELING AND SIMULATION TESTS ON LIFT VALVES OPERATING IN THE HYDRAULIC SYSTEM OF A PRESSURE CASTING MACHINE

MODELOWANIE ORAZ BADANIA SYMULACYJNE ZAWORÓW WZNIOSOWYCH PRACUJĄCYCH W UKŁADZIE HYDRAULICZNYM ODLEWNICZEJ MASZYNY CIŚNIENIOWEJ

This article presents dynamic analysis of a selected hydraulic system provided with a lift valve, whose schematic diagram corresponds to the control of positive-displacement pumps in the feeding system of the pressure casting machine. A mathematical model of the lift valve which illustrates the effect upon the working dynamics of the hydraulic system exerted by control nozzle diameter values was presented.

Finally the simulation results, for different values of nozzle diameters were graphically compared together.

Keywords: lift valves, model testing, hydraulic systems, pressure casting machines

Niniejszy artykuł przedstawia analizę dynamiczną wybranego zespołu hydraulicznego z zaworem wzniosowym, którego schemat ideowy odpowiada sterowaniu zespołem pomp wyporowych w układzie zasilacza odlewniczej maszyny ciśnieniowej. W artykule zaprezentowano model matematyczny zaworu wzniosowego, który uwzględnia wpływ jaki na dynamikę pracy układu hydraulicznego wywiera odpowiedni dobór średnic dysz sterujących. Wyniki badań symulacyjnych uzyskane dla różnych średnic dysz sterujących zostały porównanie w sposób graficzny.

1. Introduction

Hydraulic systems with lift valves have been widely applied in various machines and devices, especially in those which operate on high flow rates of working medium. The dynamics in systems provided with such valves is highly influenced by correct selection of control nozzles. This article presents dynamic analysis of a selected hydraulic system provided with a lift valve, whose schematic diagram corresponds to the control of positive-displacement pumps in the feeding system of the pressure casting machine. A mathematical model of the lift valve as presented in this article illustrates the effect upon the working dynamics of the hydraulic system exerted by control nozzle diameter values.

2. Construction of the lift valve

Cartridge two -way lift valves are also known and named in Poland as 'logical elements'. Lift valves are two-way valves with two working positions: 'open' and 'closed'. They are designed for systems with branching parts which, however, cannot be too complex. Lift valves, if properly controlled, may influence the value and direction of the flow or pressure of the working medium.

An exemplary standard lift valve for controlling the flow direction is shown in Fig. 1. The valve consists of: body 1, mushroom 2, spring 3 and lid 4. Those parts are usually fixed in the control block 5.

As can be seen in Fig. 1, control block 5 contains ports A and B for making connections to other main circuit parts. Block 5 is also provided with duct X which drains the fluid to the valve controller.

Either connection or disconnection of ports A and B depends on the areas A_1, A_2 and A_3, on the pressure exerted on them and on the spring force. Those three areas are very important for the valve operation, and are determined as follows:

- A_1 – area in port A, in other words – base area,
- A_2 – area in port B, equal to 50% of base area in standard valves (e.g. Mannesmann Rexroth); other values also possible,
- A_3 – area from the spring side – the sum of areas A_1 and A_2.

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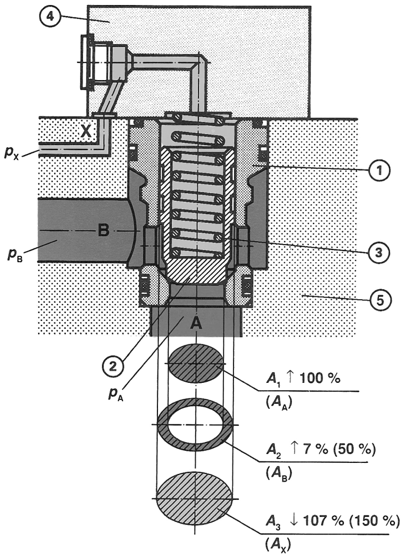

Fig. 1. Lift valve: 1 – body, 2 – mushroom, 3 – spring, 4 – lid, 5 – control block [1]

3. Mathematical model of the valve

So as to investigate the effect of flow nozzle diameters exerted upon the lift valve operation, there has been elaborated a mathematical model for a valve operating in the relief system of the positive displacement pump. Its diagram is shown in Fig. 2 together with physical parameters illustrating the system operation.

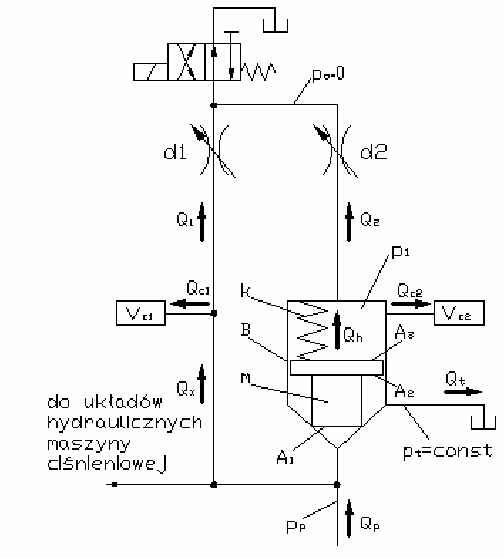

Fig. 2. Diagram of the system used for constructing a lift valve model

While analyzing the operation of the system, it can be found that if the slide distributor has not been re-set (as in Fig 2), then the flow rate Q_p from the positive displacement pump reaches directly the working liquid tank, whereas the pump runs idly. In the case of re-setting the slidedistributor to position 2, the flow rate from the outlet of the positive displacement pump reaches the hydraulic system, whereas the pump remains in the pressure operation regime. The time of switching from the pressure regime to idle running (and vice versa) is determined by the diameters of flow nozzles d_1 and d_2 in the circuit controlling the lift valve operation. If they are selected incorrectly, there may occur water hammers which affect considerably the working dynamics in the entire hydraulic system. This article describes modeling and simulation tests performed on a hydraulic system with a lift valve and the slide distributor not re-set. In such a case, the system dynamics is determined chiefly by the selection of nozzle diameter d_2. In order to present the operation of the lift valve as reliably as possible, two simplifying assumptions have been made:

- The force of gravity of the mushroom is small enough and can be neglected
- The valve operates in stationary thermal conditions.

Taking into consideration the simplifying assumptions and basing upon the diagram as accepted (Fig. 2), the operation of the lift valve has been described with a system of the following equations:

- Equation of equilibrium of the forces acting upon the valve mushroom

$$F_m + F_B + F_{hd} + F_s = F_p \tag{1}$$

- Equation of balance of flow rates from the feeding side

$$Q_x + Q_t + Q_h = Q_p \tag{2}$$

- Equation of balance of flow rates from the control side

$$Q_2 + Q_{c2} = Q_h \tag{3}$$

where: F_m – inertial force, F_B – force of viscotic friction, F_{hd} – hydrodynamic force, F_s – force of spring elasticity, F_p – resultant hydrostatic force, Q_x – flow rate in control branch, Q_t – flow rate in the gap between the mushroom and valve seat, Q_h – absorbing capacity of the valve mushroom, Q_p – flow rate on the pressure side, Q_2 – flow rate through gland d_2, Q_{c2} – flow rate resulting from fluid compressibility in the control chamber. Taking into consideration the known mathematical relations describing respective components in equations (1), (2) and (3), and after performing proper transformations, there have been obtained equations in a form convenient for modeling in Matlab/Simulink:

$$D^2y = \begin{cases} \dfrac{1}{m_b}\left[p_pA_1 + p_tA_2 - p_1A_3 - h_gQ_t\sqrt{2\rho\left(p_p - p_t\right)} - BDy - k(y + y_0)\right] & for p_p > p_{p0} \\ Dy = y = 0 & for p_p \leqslant p_{p0} \end{cases} \tag{4}$$

$$Dp_p = \frac{E_c}{V_{c1}}\left(Q_p - 0.0276d_1^2\sqrt{p_p} - \mu f K_q\sqrt{p_p - p_t} - A_3Dy\right) \tag{5}$$

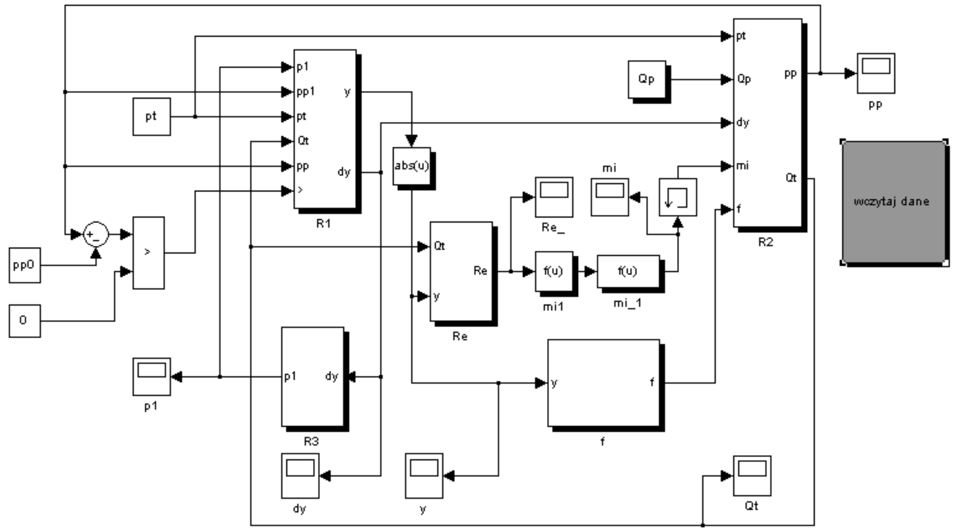

Fig. 3. General block diagram of the system

$$Dp_1 = \frac{E_c}{V_{c2}}\left(A_3Dy - 0.0276d_2^2\sqrt{p_1}\right) \tag{6}$$

where: m_b – mushroom weight + 1/3 spring weight, y – linear shift of mushroom, D – differential operator, k – spring elasticity coefficient, y_0 – initial spring deflection, B – viscotic friction coefficient for valve mushroom, h_g – factor of proportionality, ρ – oil density, p_p – pressure on pump inlet p_t – pressure on valve outlet, p_1 – pressure over mushroom, A_1 – mushroom area in port A, A_2 – mushroom area in port B, A_3 – mushroom area from the spring side, V_{c1} i V_{c2} – fluid volumes subject to compressibility, E_c – fluid compressibility modulus, d_1 – reducer diameter, d_2 – reducer diameter f – fluid flow area in valve, p_{p0} – valve opening pressure, K_g – mushroom flow coefficient.

Basing upon the References, measurements and by way of estimation, there have been deter-mined the values of mathematical model coefficients; successively, there has been constructed the general block diagram of the system with a lift valve as shown in Fig. 3.

4. Model tests

Model tests were carried on a system fed with hydraulic oil of pre-set kinematical viscosity. There was assumed a jump characteristics of the flow rate Q_p, signal applied to the valve inlet. In the simulation process, the reducer diameter d_2 was subject to changes so as to determine its effect upon the system behavior. The values of reducer diameters d_2 values for which simulation tests were performed were, respectively: 0.5; 1; 1.5; 2, 4; 10 [mm]. In compliance with the collected results, some comparative graphs for different values of reducer diameters d_2 (Figs. 4÷13) were plotted.

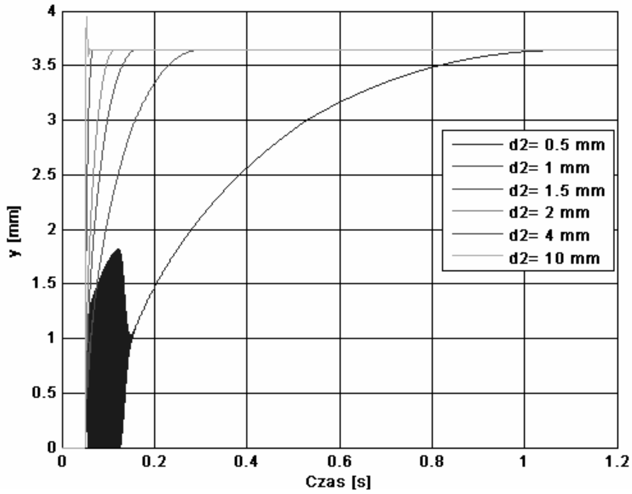

Fig. 4. Comparison of characteristic curves of mushroom shift y [mm]

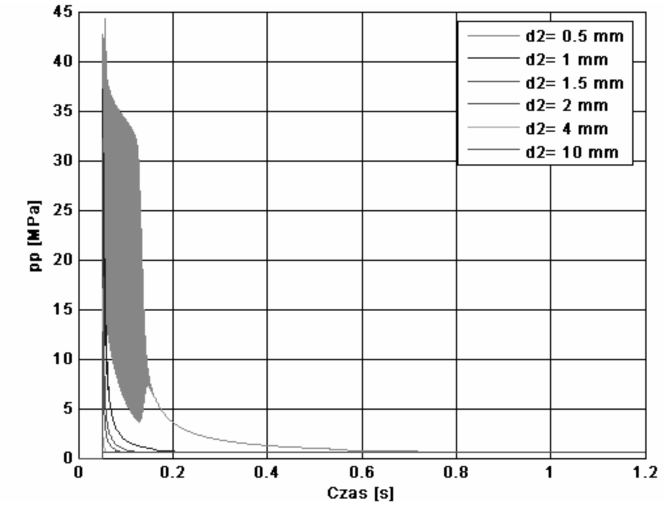

Fig. 5. Comparison of characteristic curves of feeding pressure p_p [MPa]

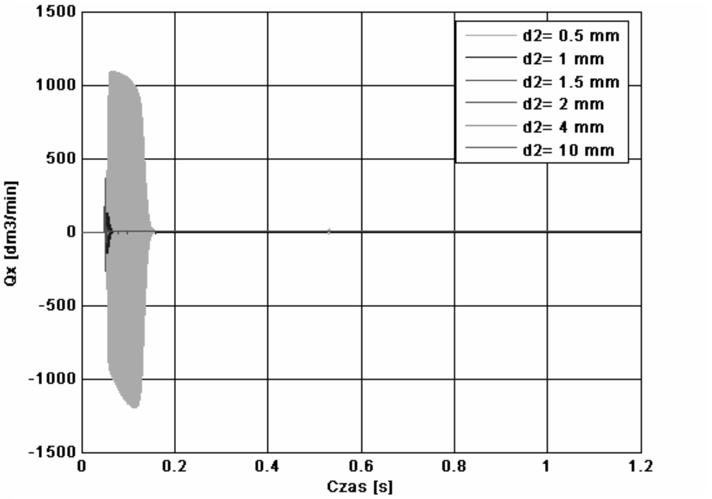

Fig. 8. Comparison of characteristic curves of flow rates Q_x [dm^3/min] at the inlet of control ducts

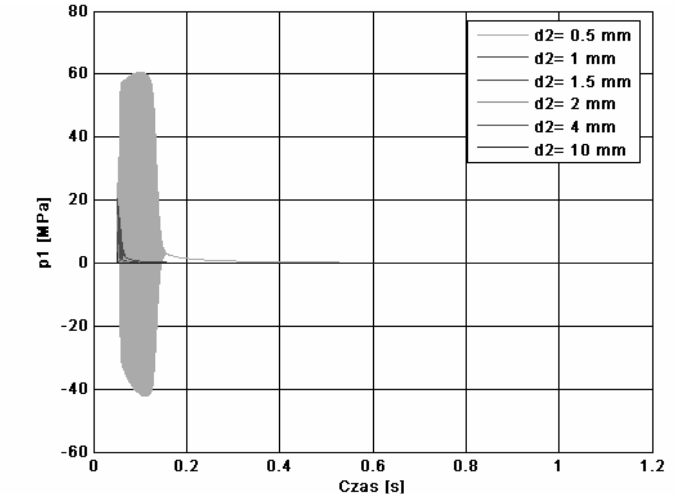

Fig. 6. Comparison of characteristic curves in mushroom chamber p_1 [MPa]

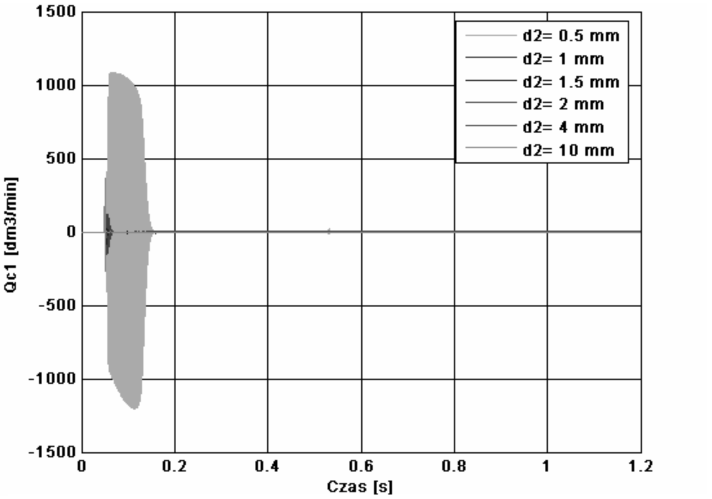

Fig. 9. Comparison of characteristic curves of flow rate Q_{c1} [dm^3/min] resulting from compressibility in control ducts

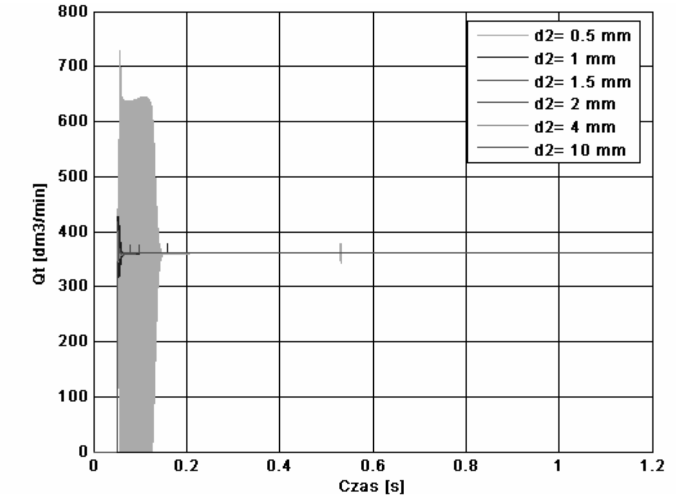

Fig. 7. Comparison of characteristic curves of flow rate Q_t [dm^3/min] through the valve slot

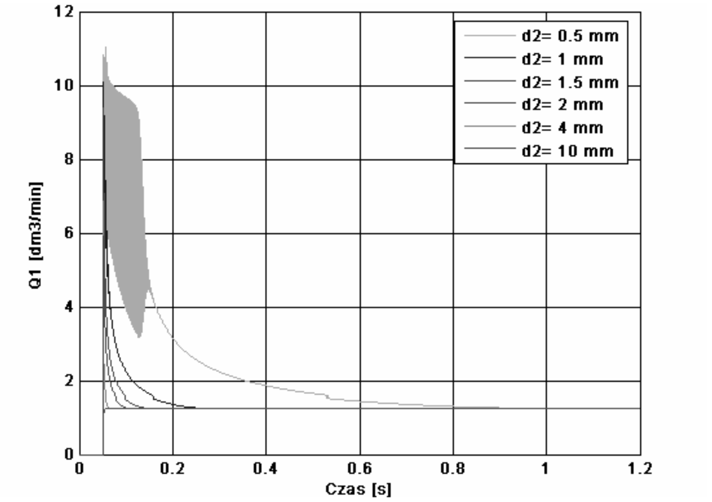

Fig. 10. Comparison of characteristic curves of flow rates Q_1 [dm^3/min] through reducer d_1

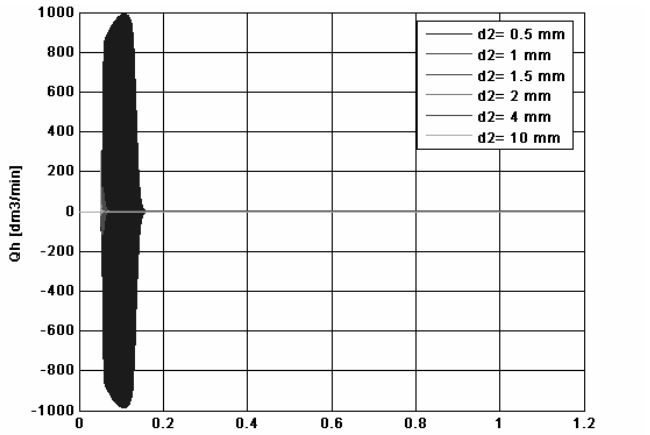

Fig. 11. Comparison of characteristic curves of absorbing capacity of mushroom Q_h [dm^3/min]

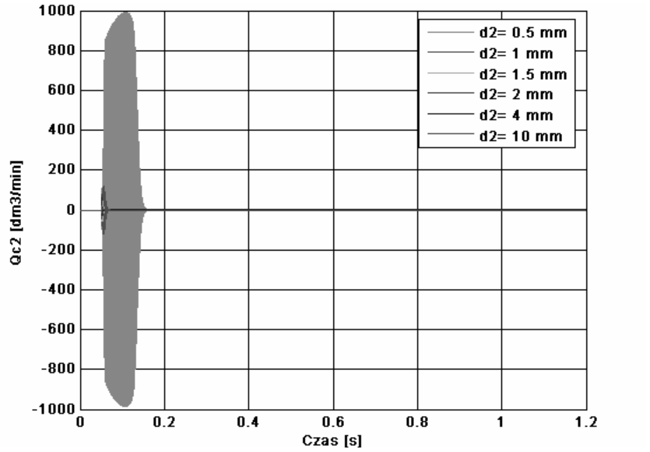

Fig. 12. Comparison of characteristic curves of flow rates Q_{c2} [dm^3/min] resulting from the compressibility in the mushroom chamber

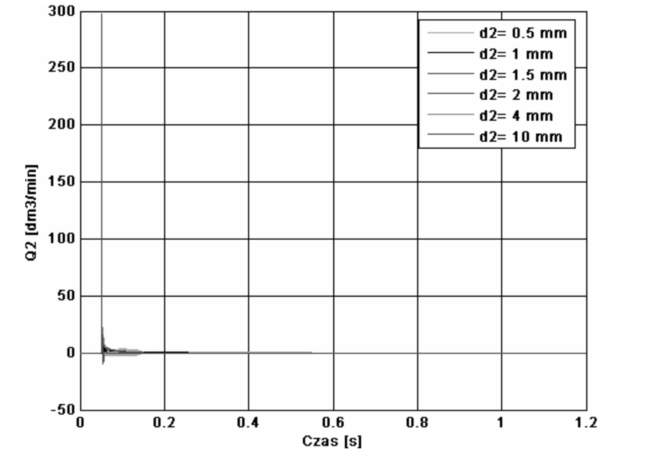

Fig. 13. Comparison of characteristic curves of flow rates Q_2 [dm^3/min] through reducer d_2

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5. Conclusions and final results

Basing upon the obtained simulation research results, be ascertained as follows:

1. There are no objections to the simulation of the valve opening process as generated by the model elaborated, and the final results may constitute the base for experimental verification.
2. While applying the jump signal of flow rate Q_p to the system, some shocks of feeding pressure p_p may appreciably exceed the permissible operation pressure value, which implies the use of d_2 reducers with larger diameters. But the reducer diameter must not be too large because it suppresses the mushroom's motion.
3. Among the reducers investigated, the best dynamic parameters were obtained for d_2 = 4 [mm]. For such a value, the pressure pulse is relatively small and no oscillations in the mushroom motion take place. There may be also applied reducers with diameters 2 [mm] and 1.5 [mm]; although the feeding pressure does not exceed the permissible range, the pressure pulse is higher.
4. Reducers of diameters over 4 [mm] cannot be applied in the system under investigation despite a lower pressure pulse because of oscillations occurring in the mushroom motion.
5. Reducers of diameters under 1 [mm] cannot be used because of very bad dynamic parameters, incl. mushroom's striking on the seat.

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