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CONTENT

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RESEARCH ON THE INFLUENCE OF THE VOLUME OF OIL IN FRONT OF THE DIRECT OPERATED PRESSURE RELIEF VALVE ON ITS TRANSIENT PERFORMANCES

SASKO DIMITROV, DENNIS WEILER AND SIMEON PETROV

Abstract. The dynamic characteristics determine variations of the inlet pressure in front of the valve in function of the flow through the valve in time. In any hydraulic system, at valve inlet port there is some volume of compressible oil which influences the quality of the transient process. When switching the directional control valve in the hydraulic system with direct-operated pressure relief valves, a transient process occurs in which, depending on the amount of oil in front of the valve, it is possible for the pressure to reach values many times higher than the set value. This causes the system to be overloaded with undesirable consequences. This paper examines experimentally and theoretically the influence of the volume of oil in front of the valve on its transient performances. It is presented that when there is a low amount of oil in front of the valve, it reacts faster, but the system is significantly overloaded. When there is a higher amount of oil in front of the valve, it reacts more slowly and overloads the system less above the preset value of the pressure.

1. Introduction

In order to protect a hydraulic circuit against overloads and limit the work pressure, pressure relief valves are used. The main function of these valves is to limit the maximum working pressure in the hydraulic system. They are normally positioned after the hydraulic pump. In this case, the valve does not require an external power source, meaning that the fluid pressure is enough to open or close the valve. This means that the function of this class of valves is performed automatically. This makes this class of valves indispensable for the hydraulic circuit function and operation.

There are two types of pressure relief valves: direct operated and pilot operated. The direct operated pressure relief valves have higher deviation of the adjusted pressure in the transient process than pilot operated ones which leads to overloading of the hydraulic system [1]. One of the reasons for this deviation is the volume of oil in front of the valve.

Many authors have investigated the static and dynamic characteristics of direct operated pressure relief valves. Brodowski [3] has presented experimental and theoretical dynamic characteristics and shown that the magnitude of the pressure peak is far higher than the steady-state magnitude. He has also proved that the pressure peak depends on the size of the damping orifice. Many authors have worked on dependence of the discharge coefficient in the control orifice. During an unsteady process, the flow presumably passes in and out of laminar and turbulent regions.

Keywords. Schematic and functional diagram, valve components, mathematical modeling,

So, a model is needed which describes both regimes simultaneously. That kind of a model is recommended by Borutzky [8].

Another empirical model for the discharge coefficient has been presented in [7]. For the pressure relief valve, it is suitable to determine the discharge coefficient in the control orifice based on experimental static characteristics [10]. High impact to the static and dynamic characteristics has the hydrodynamic reaction force of the flow [2], [4],[9]. In the dynamic mode it can even cause unstable work of the valve.

Dasgupta and Karmakar [5] studied the dynamics of a direct operated pressure relief valve with directional damping through bond graph simulation technique. The authors concluded that some significant parameters of the valve response are identified, which can be modified to improve the dynamic characteristics of the valve. Their theoretical research they have compared with the experimental dynamic characteristic presented by Watton [6].

Although this class of valves is indispensable for the function and operation of the hydraulic systems, a review of the available research shows that their study is not well covered and there is a need for an in depth study of modeling and simulation of their performance. Therefore, a comprehensive study of the modeling and simulation of the performance of this class of valves in the transient modes of operation is carried out in this paper. A comprehensive nonlinear mathematical model, taking into account most nonlinearities of the valve, is developed. The transient performances of the studied valve are investigated theoretically and experimentally. The experimental study is also used to validate the simulation program of the studied valve in the transient modes of operation. Special attention has been paid to the influence of the volume of oil in front of the valve on the transient process.

2. Valve components description and schematic diagram

The objective of the pressure relief valve is to limit system pressure downstream of the valve. Figure 1 shows the basic components of the studied valve, while its schematic diagram is shown in Figure 2. This valve basically consists of sleeve 1, adjusting spring 2, poppet with damping piston 3, and adjustment element 5. The system pressure setting can be infinitely varied by means of adjustment element 5. Spring 2 presses poppet 3 onto its seat. Port P is connected to the system. The system pressure acts on the poppet area. When the pressure in port P rises above the value adjusted on spring 2, poppet 3 moves against spring 2 and the valve is opening. Hydraulic oil can now flow from port P towards port T. The stroke of poppet 3 is limited by embossment 6.

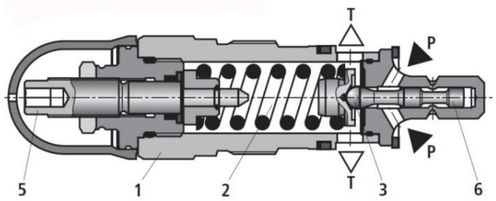


Figure 1. *Schematic diagram of the valve*

In Figure 2 the schematic diagram of the test rig with the studied pressure relief valve, volume of oil at its inlet V_0 and output pipeline with linear R_p and inertial L_p resistance is shown. To isolate the oil compressibility between the pump and the valve and for reducing the pressure pulsation of the pump, a throttle with high inertial resistance is included.

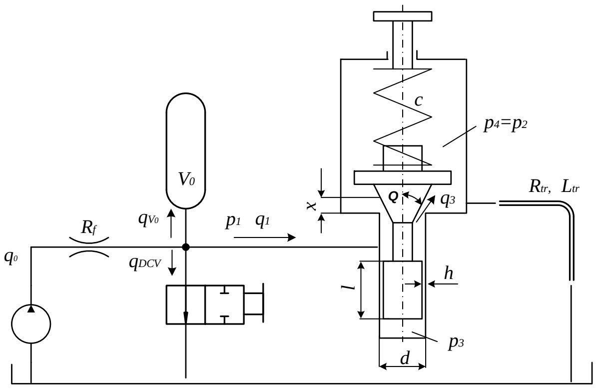


Figure 2. *Functional diagram of the test rig*

The valve is normally closed. When the pressure p_1 is lower than the value necessary to move the poppet against the spring, the main valve throttling area remains closed and the valve poppet presses onto its seat. Rapidly activating the control valve V_1, the pressure begins to rise in the volume V_0 and in front of the valve and the transient process starts. When the pressure is high enough, the valve poppet lifts from its seat and the valve opens. Thus, the valve limits additional rising of the pressure downstream of the valve.

3. Mathematical modelling of the studied valve

To model the studied valve, some assumptions are made in developing the nonlinear mathematical model. It is assumed that the tank pressure is constant at atmospheric pressure; the geometry and discharge area of the valve restriction usually change nonlinearly; the pressure losses in the short pipelines are neglected; the oil temperature and viscosity are kept constant. During the transient mode of operation, the flow rate passing through the valve throttling area is of high Reynolds number. The discharge coefficient of this throttling area changes with the Reynolds number in a complicated manner. For simplicity reasons, the discharge coefficient for the valve throttling area is taken to be constant.

Mathematical model of the system is described by the following equations:

According to Figure 2, the equation of continuity in front of the investigated pressure relief valve can be expressed as:

$$q_0 = q_{in} + q_v + q_1 \tag{3.1}$$

where q_{in}, q_v, and q_1 are the flow rate through restriction area in the directional control valve V1, the flow rate which enters in the volume V_0 and the flow rate entering in the valve, respectively. The transient variation of the restriction area of the directional control valve V1 affects the studied valve transient response. The flow rate q_{in} passing through the directional control valve is given by

$$q_{in} = (1 - \frac{t}{t_1}) \cdot \mu_v \cdot \pi \cdot d_v \cdot x_v \cdot \sqrt{\frac{2}{\rho} \cdot p_1} \tag{3.2}$$

where t_1 is the closing time of the DCV, μ_v, d_v and x_v are the discharge coefficient, the diameter of the valve spool and valve spool displacement, respectively. The flow which enters the volume V_0 can be expressed by the equation of the compressibility effect in the volume V_0:

$$q_v = \frac{V_0}{K} \cdot \frac{dp_1}{dt} \tag{3.3}$$

where K is the bulk modulus of the oil.

The equation of continuity in the valve in front of the control orifice and after is

$$q_1 = q_2 = q_3 + A_k \cdot \frac{dx}{dt} \tag{3.4}$$

where: A_k – the area of the valve poppet; $q_3 = \mu \cdot \pi \cdot d \cdot x \cdot sin\theta \cdot \sqrt{\frac{2}{\rho} \cdot p_{1,2}}$ – the flow through the control orifice in the valve.

The equation of motion of the valve poppet is

$$m \cdot \frac{d^2x}{dt^2} + c \cdot (h_0 + x) + r_h \cdot x \cdot p_{1,2} = A_k \cdot (p_3 - p_4) - F_T \tag{3.5}$$

where: $m = m_k + \frac{1}{3} m_f$ – the equivalent mass of the valve poppet m_k and the spring m_f; c – the stiffness of the spring; h_0 – the deformation of the spring when $x = 0$; r_h– the coefficient of the hydrodynamic force obtained by the expression (3.1); F_T – friction force between the valve poppet and the body of the valve.

The pressure in the lower chamber of the closing element of valve p_3 depends on the losses in orifice h between the piston of the valve poppet and the body of the valve:

$$p_3 = p_1 - R_{a,l} \cdot A_k \cdot \frac{dx}{dt} - R_{a,m} \cdot \left(A_k \cdot \frac{dx}{dt}\right)^2 - L_a \cdot A_k \cdot \frac{d^2x}{dt^2} \tag{3.6}$$

where: $R_{a,l}$, $R_{a,m}$ and $L_a = \rho \frac{l}{\pi dh}$ are linear, local, and inertial resistances in the orifice with length l.

The pressure in the upper chamber above the valve poppet is obtained analogically when for this type of the valve is $p_4 = p_2$.

The pressure drop in the outlet pipeline is

$$p_2 = R_{p,l} \cdot q_2 + R_{p,m} \cdot q_2^2 + L_p \cdot \frac{dq_2}{dt} \tag{3.7}$$

where: $R_{p,l}, R_{p,m}$ and $L_{p,t}$ respectively linear, local, and inertial resistance of the outlet pipeline with length l_p and diameter d_p.

Additional conditions were taken into account when solving the nonlinear system of the differential equations: the flow rate q_1 is zero when the valve is closed; pressure p_1

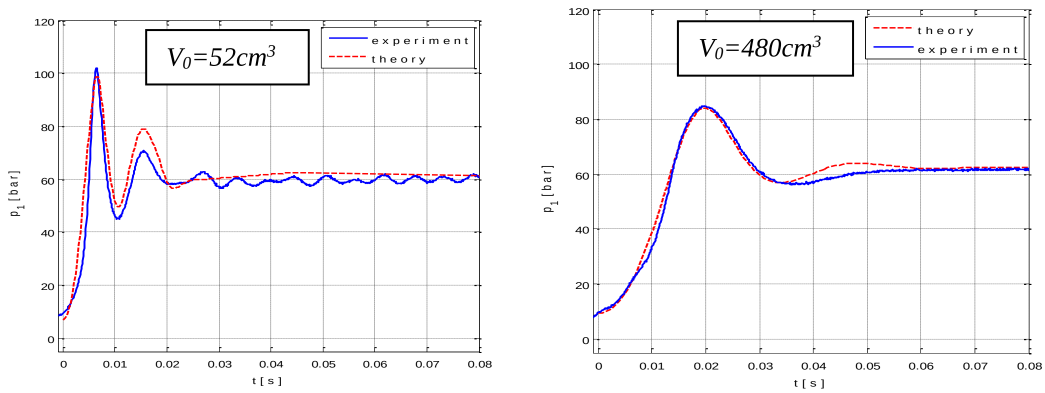
cannot be less than the absolute vacuum; displacement x of the valve poppet cannot be negative; flow rate q_{in} is zero at $t > t_1$, etc., The mathematical model (3.1) - (3.7) can be solved with computer programs for solving nonlinear differential equations. For the solution of the system of the nonlinear differential equations, the adaptive Runge-Kutta method has been used. This method based on the fourth order Runge-Kutta method estimates the truncation error at each integration step and automatically adjusts the time step size to keep the error within prescribed limits.

4. Experimental and theoretical characteristics of the researched valve

Figure 3 presents the results of experimental and theoretical studies of a BoschRexroth type valve for a pressure of $60\ bar$ and oil volumes $V_0 = 52\ cm^3$ – figure 3 - a and $480\ cm^3$ – Figure 3 - b. Closing time t_1 of the directional control valve is less than $20\ ms$ and the flow rate of the pump is $q_0 = 25\ l/min$. The outlet pipe is $12\ mm$ in diameter and $1\ m$ long. The experiment was performed with pressure and displacement transducers and was recorded on a computer.

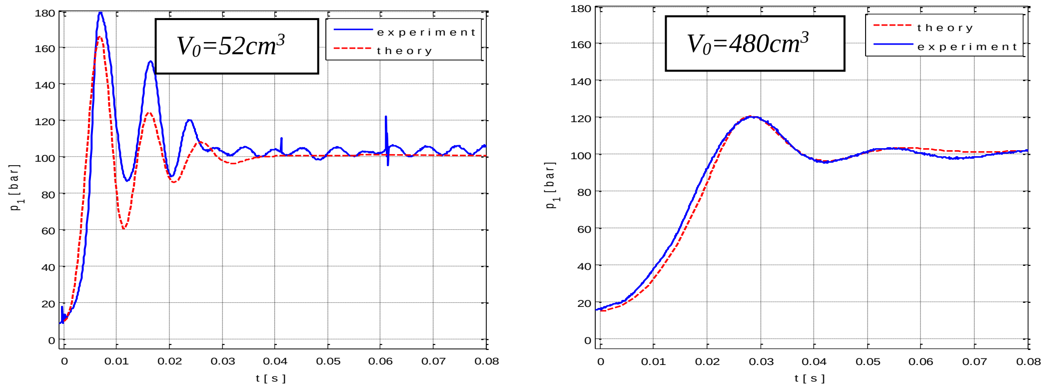
With the volume of oil at the inlet of $52\ cm^3$, a relatively large dynamic load is obtained, as the pressure reaches $100\ bar$ and the natural frequency is $742\ rad\ /\ s$. This leads to system overload, which in many cases is unacceptable. As the volume increases to $480\ cm^3$, the maximum pressure and the natural frequency of the transient process decrease to $85\ bar$ and $206\ rad\ /\ s$, respectively.

Figure 4 presents the results of experimental and theoretical studies of a BoschRexroth type valve for a pressure of $100\ bar$ and oil volumes $V_0 = 52\ cm^3$ – figure 4 - a and $480\ cm^3$ – figure 4 - b. With a volume of oil at the inlet of $52\ cm^3$, the overshoot is much higher, but with increased volume of oil, the overshoot decreases.



a b

Figure 3. *Experimental and theoretical dynamic characteristics of the specified valve for a pressure of* 60 *bar and different volumes of oil at the inlet port*



a b

Figure 3. *Experimental and theoretical dynamic characteristics of the specified valve for the pressure of* $100\ bar$ *and different volumes of oil at the inlet port*

The poppet of the valve opens when the spring-set pressure is reached. The pressure in this phase of the transient process changes at a rate determined by the flow rate of the pump, the volume of oil at the inlet and the closing time of the directional control valve t_1. The difference between the experiment and the theoretical solution is due to the change in the hydrodynamic force and the slope of the static characteristic during the transient process, which are not taken into account in the mathematical model.

5. Conclusion

The transient characteristics of a direct operated pressure relief valve are researched theoretically and experimentally. Special attention has been paid to the influence of the volume of oil in front of the valve on the transient process. A comprehensive nonlinear mathematical model of the studied valve is deduced to predict the performance of the studied valve in the transient modes of operation. The developed model, which takes into consideration most nonlinearities of the studied valve, is used to develop a computer simulation program. The transient characteristics of the studied valve are simulated using this program. The experimental work aimed at validating the proposed model and the simulation program. The results showed good agreement between simulation and experimental results in the transient modes of operation.

As shown in equation (3.3) and the diagrams in Figure 3 and Figure 4, the transient process of the direct operated relief valves directly depends on the volume of oil in front of the valve. With a lower amount of oil - $V_0 = 52\ cm^3$, the valve acts very fast. Its rise time is around $0.005\ s$, and the peak time is around $0.007\ s$, but the overshooting is very high – $67\ \% - 80\ \%$ of the set pressure. This feature of the valve overloads the entire hydraulic system. When there is higher amount of oil in front of the valve - $V_0 = 480\ cm^3$, the valve acts more slowly, but the overshoot is much more acceptable. Its rise time is around $0.013\ s - 0.022\ s$, and the peak time is around $0.02\ s - 0.028\ s$. The overshoot is around $20\ \% - 40\ \%$ of the set pressure. This is the main reason why, when the hydraulic system works with higher working pressure, it is always recommended to use a pilot operated pressure relief valve. It has better transient performances than the direct operated pressure relief valves.

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