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Dynamic Processes in Hydraulic Systems during simultaneous Operation of a Pressure Relief and Pressure Reducing Valve

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Abstract

This paper deals with modelling and simulation of dynamic processes in hydraulic system with simultaneous operation of a directly operated pressure relief valve and a directly operated three-way pressure reducing valve. A comprehensive nonlinear mathematical model is deduced in order to predict the performance of the studied system. The proposed model takes into consideration most nonlinearities of the studied valves. A computer simulation, based on the proposed model, is performed to predict the transient performance of the system. The validity of the proposed model is assessed experimentally in the transient mode of operation. The results show significant agreement between simulation and experiments.

Keywords: Pressure relief valve, Pressure reducing valve, dynamic process, hydraulic system

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Introduction

The pressure control valves can perform different functions in the hydraulic systems, such as establishing a maximum pressure, reducing pressure in some circuit lines, and establishing sequence movements, among other functions. The main function of these valves consists of providing a balance between the pressure difference and the force load on a spring. Most of these valves can be positioned in many different levels, between totally open and totally closed, depending on the flow and the differential pressure. The pressure control valves are usually named according to their primary functions, and their basic function is to limit or to determine the pressure of the hydraulic system for the attainment of a certain function of the equipment in motion. In order to protect a hydraulic circuit against overloads and limit the work pressure, pressure relief valves are used. The main function of theses valves is to limit the maximum working pressure in the hydraulic system. They are normally positioned after the hydraulic pump. However, many times, there are hydraulic circuits where diverse lines are fed by one same source, but it must work at different levels of pressure. For this reason, pressure reducing valves are used. The main function of these valves is to maintain a preset downstream reducing pressure regardless of changing upstream pressures. They normally work with hydraulic actuators (cylinders or motors). These actuators will respond to change of pressure or flow and will open or close the pressure reducing valve.

Many authors have independently researched dynamic characteristics of directly operated pressure relief valves and pressure reducing valves. (Brodowski, 1974) 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 (Lichtarovitz at al., 1965), (Wobben, 1978), (Stone, 1960). During the unsteady process the flow presumably passes in and out of laminar and turbulent regions. So, a model which describes both regimes simultaneously is needed (Borutzky at al., 2002). Another, empirical model for the discharge coefficient has been presented by (Wu at al., 2002). The hydrodynamic reaction force of the flow has a high impact on the dynamic characteristics (Zehner, 1987), (Watton, 1988), (Osama, 2016). In the dynamic regimes it can even cause unstable work of the valve.

The dynamics of a directly operated pressure relief valve with directional damping can be studied through a bond graph simulation technique (Dasgupta & Karmakar, 2002). 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. They have compared their theoretical research with the experimental dynamic characteristic presented by (Watton, 1988). Few publications are found to deal with the performance of pressure reducing valves. (Osama, 2016) developed a mathematical model and assessed experimentally in the steady-state and transient modes of the three-way reducing valve by changes in the pressure of the primary circuit in the hydraulic system. (Will & Gebhardt, 2011) developed a mathematical model of the two-way reducing valve by changes of the flow through the valve, without experimental confirmation. (Storr, 1967) and (Zimmerman, 1984) in their dissertations, developed a dynamic model for pilot operated pressure reducing valves.

This paper deals with modeling and simulation of dynamic processes in a hydraulic system with simultaneous operation of a directly operated pressure relief valve and a directly operated three-way pressure reducing valve. So, it is important to analyze the dynamic characteristics of a hydraulic system with the mutual influence of the two valves with each other.

The purpose of the present study can be explained on the basis of the hydraulic system for regulating the clamping force with a three-way reducing valve (see Figure 1). After activating the solenoid 'a' of the directional control valve 5, the working fluid flows through the reducing valve 2 to the hydraulic cylinder 3, which moves its piston towards the clamped part 4 with a speed determined by the throttle 6. At the end of the stroke z, the cylinder piston reaches the clamped part 4 and its speed becomes zero. This sudden change in speed leads to an increase in the pressure in the volume V and, accordingly, the clamping force. The change in the flow rate q_2 and, respectively, q_1 leads to a change in the flow rate q_r through the pressure relief valve 1 and the pressure p_1 also changes in the studied transient process.

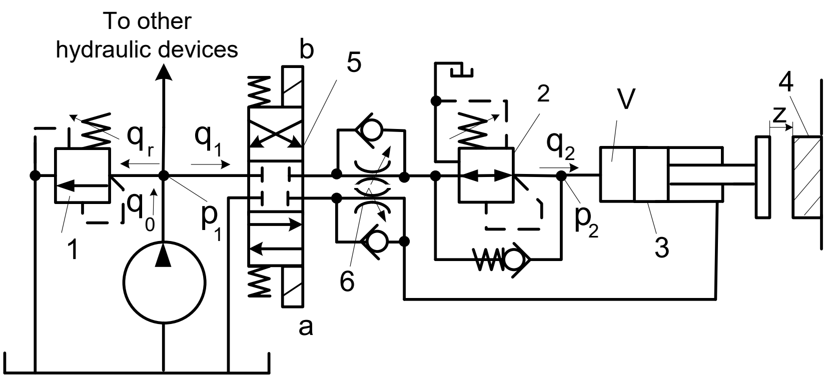


Figure 1. Hydraulic circuit for controlling the clamping force with a three-way pressure reducing valve

In this work, the transient processes of the reduced pressure p_2 and the relief pressure p_1, depending of the flow rate q_2 flowing through the pressure reducing valve and the flow rate q_r flowing through the pressure relief valve, have been experimentally and theoretically researched.

Valve Component Description and Schematics Diagram of the Test Rig

Pressure relief valve

The role of the pressure relief valve is to limit a system pressure downstream the valve. The subject of research was Rexroth's directly operated pressure relief valve type *DBD*. Figure 2 shows the basic components of the studied valve. This valve basically consists of body 1, adjusting spring 2, poppet 3 with damping piston 6, 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 adjustment on spring 2, poppet 3 moves against spring 2 and the valve opens. Hydraulic fluid can now flow from port *P* towards port *T*.

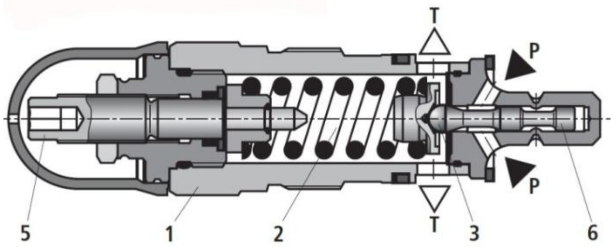
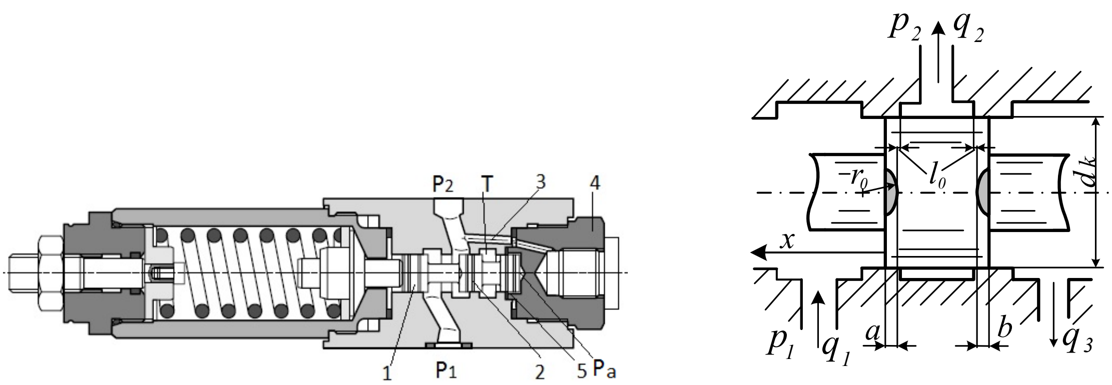


Figure 2. Schematic diagram of the valve

Pressure reducing valve

The design of the studied Bosch's three-way pressure reducing valve type 0811150233 has been presented on Figure 3. The middle spool land 2 of the control spool 1 (see Figure 3b) has a positive overlap of the two control edges l_0 ($l_0 \approx 0.1\ mm$), one between the inlet pressure p_1 and the reduced pressure p_2, the second between the reduced pressure p_2 and the tank *T*. There are control notches at the edges of the control spool with circle segments with a radius of $r_0 = 3\ mm$ and width of $a = 0.75\ mm$ and $b = 0.35\ mm$ (see Figure 3b). The movement of the control spool 1 damps by the hydraulic resistance of the canal 3 and the gap 5 between the closure nut 4 and the cylindrical surface of the spool 1 (see Figure 3a).



a) b)

Figure 3. Three-way pressure reducing valve (BoschRexroth)

a) design; b) control spool land

A schematic diagram of the experimental test rig for determining the transient processes of the system on Figure 1 is shown on Figure 4. The pump transmits flow of $q_0 = 40\ l/min$ to the pressure relief valve 1 and the pressure reducing valve 2. The pressure relief valve 1 is set to $p_{10} = 80\ bar$. The volume of fluid in the pipelines between the two valves is $V_1 = 95\ cm^3$. The pressure reducing valve 2 is set to the pressure of $p_{20} = 20\ bar$, with a flowrate of $q_{20} = 10\ l/min$ setted at the flow control valve 4. The pressure relief valve is opened allowing the rest of the flow $q_{r0} = 30\ l/min$ to pass to the tank. The flow q_2 passes through the pipeline with the linear resistance r_p and the inertial resistance L_p to the volume V and to the directional control valve 3. The outlet of the directional control valve 3 is connected to the tank T by a two-way flow control valve 4. The volume V corresponds to the chamber of the piston side of the hydraulic cylinder in which the reduced pressure acts during the clamping (see Figure 1). The researches were carried out at three levels of the fluid volume $V = 105, 250$ and $700\ cm^3$.

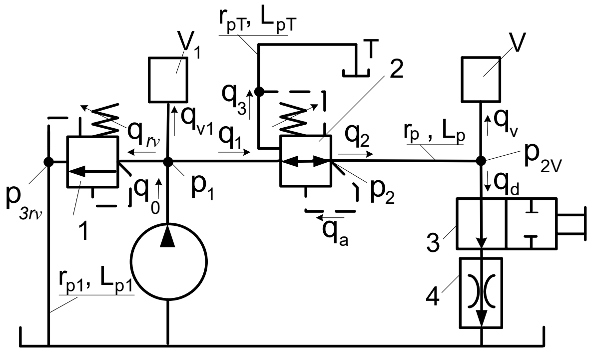


Figure 4. Schematic diagram of the experimental test rig

A transient response process in the system on Figure 4 is created with the closing of the directional control valve with mechanical control 3, by means of a knock. In this way, a sudden reduction to zero of the piston velocity of the clamping hydraulic cylinder at the end of its stroke is simulated. The closing time of the directional control valve 3 in this study is about $t_1 = 3\ ms$.

With sudden closing of the directional control valve, the flow q_d is reduced to zero and the transient process begins in the system. The flow q_2 from the pressure reducing valve keeps entering through the pipeline in the volume V, and increases the pressure p_2 to its maximum peak $p_{2v,max}$. Increasing the pressure p_2, the control plunger 1 of the pressure reducing valve (see Figure 3) shifts to the closing position thus the flows q_1 and q_2 are reduced. This leads to a change of the flow q_r through the pressure relief valve, which increases to the flow of the pump q_0. After the transient process is completed, the inlet pressure is stabilized to the steady-state value of $p_{1e} = 88\ bar$.

The control plunger 1 of the three-way pressure reducing valve exceeds the position $x = 0$, so $q_1 = 0$ and the control edge (see Figure 3a) opens the passage to the tank T. The pressure reducing valve now works as a pressure relief valve, through it beginning to pass a flow rate of q_3 from the volume V and the pressure p_2 quickly decreases. After the transient response process passes, the steady-state values of the reduced pressure $p_{2e} = 30\ bar$, of the displacement of the plunger $x = 0$ and of the flow $q_2 = 0$ have been established.

Mathematical Modelling of the Dynamic Processes in the System

To model the studied hydraulic system, a few assumptions are made in developing the nonlinear model. It is assumed that the fluid is nonviscous with high bulk modulus. This assumption is close to reality under most realities. The hydraulic pump delivers a constant supply of pressure. The tank pressure is constant and equal to the atmospheric pressure. During the transient mode of operation, the flow rates passing through the valve throttling areas are assumed of high Reynolds number. The discharge coefficient in this throttling areas changes in a complicated manner depending on the Reynolds number of the streaming flow in the throttling areas. This coefficient here is assumed to be constant, independent of the flow rate and the opening throttling areas (Osama, 2016).

Mathematical Modelling of the Transient Process of the Pressure Relief Valve

Mathematical model of the pressure relief valve is described by the following equations:

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

$$q_0 = q_r + q_{v1} + q_1 \tag{1}$$

where q_r, q_{v1}, and q_1 are the flow rate through throttling area in the pressure relief valve, the flow rate which enters in the volume V_1 and the flow rate entering in the pressure reducing valve, respectively.

The flow which enters in the volume V_1 can be expressed by the equation of the compressibility effect in the volume V_1:

$$q_{v1} = \frac{V_1}{E} \cdot \frac{dp_1}{dt} \tag{2}$$

where E is the bulk modulus of the oil.

Equation of continuity in the valve in front of the control orifice and after it is

$$q_r = q_{rv} + A_{rv} \cdot \frac{dx_{rv}}{dt} \tag{3}$$

where: A_{rv} – the area of the valve poppet; q_{rv} – the flow through the control orifice in the relief valve.

The equation of motion of the valve poppet is

$$m_{rv} \cdot \frac{d^2x_{rv}}{dt^2} + c_{rv} \cdot (h_0 + x_{rv}) + r_{h,rv} \cdot x_{rv} \cdot p_{1,3rv} = A_{rv} \cdot (p_3 - p_{3rv}) - F_T \tag{4}$$

where: $m_{rv} = m_{p,rv} + \frac{1}{3} m_{s,rv}$ – the equivalent mass of the valve poppet $m_{p,rv}$ and the spring $m_{s,rv}$ c_{rv} – the stiffness of the spring; h_0 – the deformation of the spring when $x_{rv} = 0$; $r_{h,rv}$– the coefficient of the hydrodynamic force of the pressure relief valve; F_T – friction force between the valve poppet and the body of the valve.

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

$$p_3 = p_1 - R_{a,l} \cdot A_{rv} \cdot \frac{dx_{rv}}{dt} - R_{a,m} \cdot \left(A_k \cdot \frac{dx_{rv}}{dt}\right)^2 - L_a \cdot A_{rv} \cdot \frac{d^2 x_{rv}}{dt^2} \tag{5}$$

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.

Mathematical Modelling of the Transient Process of the Pressure Reducing Valve

The equations describing the dynamic processes of the three-way pressure reducing valves vary depending on the displacement x and the corresponding changes in the cross section of the throttle edges. When the valve operates as a pressure reducing valve, the displacement of the control spool is negative and changes from a starting value x_0 to the positive overlap area $-l_0 \le x < l_0$. When the valve operates as a relief valve, the displacement has a positive value of $x \ge l_0$.

In the reducing mode of operation when $x < -l_0$, the flow rate through the valve throttling area will be

$$q_2 = C_{d1} . A_1 . \sqrt{\frac{2}{\rho} p_{1,2}} - A_k . \frac{dx}{dt} \tag{6}$$

In the intermediate position of the spool when $-l_0 \le x < l_0$, all ports are closed and the outlet flow of the valve depends on the movement of the spool in the overlapped position $q_2 = A_k . \frac{dx}{dt}$. If the pressure reducing valve works in the relieving mode when $x \ge l_0$, the flow rate at the outlet port of the valve will be

$$q_2 = -C_{d2} . A_2 . \sqrt{\frac{2}{\rho} p_{2,3}} - A_k . \frac{dx}{dt} \tag{7}$$

In the above equations $A_1(x), A_2(x)$ - the throttling areas; C_{d1}, C_{d2} – the discharge coefficients; $A_k = \pi d_k^2/4$ – the area of the control spool with the diameter $d_k = 10\ mm$.

The equation of motion of the control spool of the reducing valve, depending in which mode of operation the valve is, will be describe as:

$$m\frac{d^2x}{dt^2} + c(s_0 + x) - F_{J1} = A_k p_a \pm F_f, \; if \; x < -l_0$$
$$m\frac{d^2x}{dt^2} + c(s_0 + x) = A_k p_a \pm F_f, \quad if \; l_0 \le x < l_0 \tag{8}$$
$$m\frac{d^2x}{dt^2} + c(s_0 + x) + F_{J2} = A_k p_a \pm F_f \quad if \; x \ge l_0$$

where: $m = m_s + \frac{1}{3} m_{sp}$ – the mass of the spool m_s and of the spring m_{sp}; $c = 83{,}65\ N/mm$ - the spring stiffness; $s_0 = \frac{A_k p_{2e}}{c}$ – the deformation of the spring when $x = 0$, $q_2 = 0$ and the value of the reduced pressure of $p_2 = p_{2e}$; $p_a = p_2 - r_d.A_k.\frac{dx}{dt} - L_d.A_k.\frac{d^2x}{dt^2}$ – the pressure in the control cavity of the control spool 1, determined by the pressure p_2 and the pressure losses from the linear and the inertial resistance in the damping channels 3 and 5 (see Figure 3a), F_f - the friction force (that due to its low value) can be neglected.

The hydrodynamic reaction force in the above equations, depending on the mode of operation of the valve, can be expressed as

$$F_{J1} = 2C_{d1}A_1 cos\theta_1.p_{1,2} = A_{h1}.p_{1,2}, \quad if \; x < -l_0$$
$$F_{J1} = 0; \; F_{J2} = 0, \quad if \; l_0 \le x < l_0 \tag{9}$$
$$F_{J2} = 2C_{d2}A_2 cos\theta_2.p_{2,3} = A_{h2}.p_{2,3}, \quad if \; x \ge l_0$$

F_{J1} , F_{J2} - the hydrodynamic reaction forces in reducing and relieving mode, expressed by the respective areas $A_{h1}(x)$, $A_{h2}(x)$ and the pressure drops $p_{1,2}$ and $p_{2,3}$; θ - the flow streamline angle;

The values of the discharge coefficient C_d and the flow streamline angle θ in (6 - 9) depend on the design of the pressure reducing valve. They can be determined by different methodologies (Borutzky at al., 2002), (Lichtarovitz at al., 1965), (Wobben, 1978). For the simplicity of calculations here, the discharge coefficient C_d and the flow streamline angle θ are assumed to be constant, independent of the flow rate and the geometry of the opening area.

The pressure drop in the pipeline connecting the valve and the volume V can be express as

$$p_{2v} = p_2 - r_{p,l}q_2 - r_{p,m}q_2^2 - L_p\frac{dq_2}{dt} \tag{10}$$

where: $r_{p,l}$ – the coefficient of linear laminar friction loss; $r_{p,m}$ – the coefficient of local loss; L_p – the inertia resistance of the *8 mm* diameter pipeline and *0.85 m* long.

The continuity equation at the outlet of the pressure reducing valve will be

$$\frac{V}{E}\frac{dp_{2v}}{dt} = q_1 - A_k\frac{dx}{dt} - q_d \tag{11}$$

where: $q_d = q_{20}(1 - \frac{t}{t1})$ for $t \leq t_1$ and $q_d = 0$ for $t > t_1$ - the approximate variation of the flow through the directional control valve 3 (see Figure 4) in the transition process.

Experimental and Theoretical Transient Response

The experimental research of the dynamic characteristics of the system presented on Figure 1 is carried out according to the diagram of the test rig shown on Figure 4.

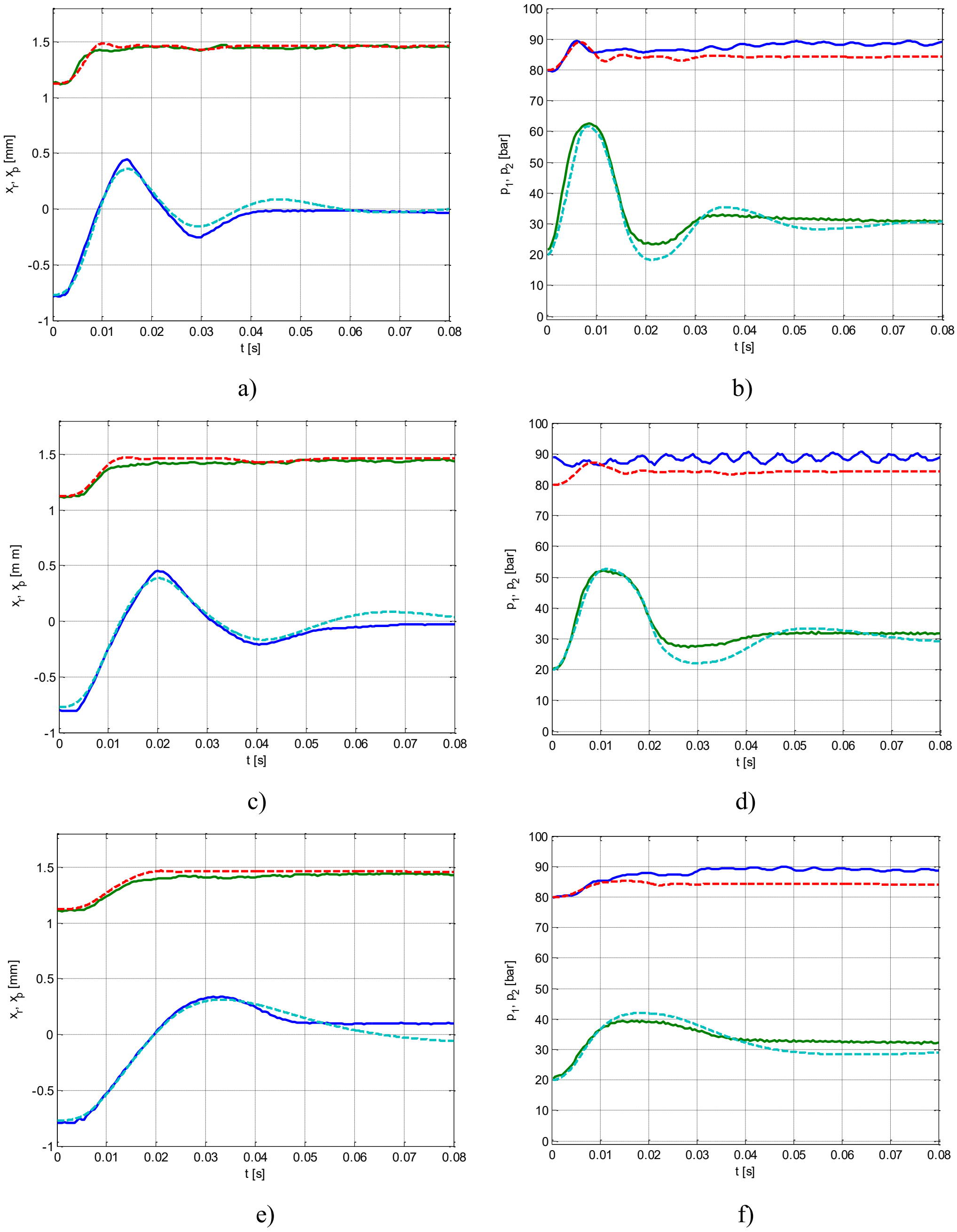


Figure 5. The dynamic characteristics of the pressure reducing valve and the pressure relief valve

The pressures p_1, p_{2V} and the displacements of the control spools of the pressure reducing valve x and the pressure relief valve x_r were measured with pressure transducers and displacement transducers, products of Bosch. The flows through the pressure reducing valve and pressure relief valve have been measured in static mode of operation by means of a rolling vane hydraulic motor type Indramat SM110.1. The experimental data was recorded on a computer by means of the National Instrument Data Acquisition Card NI USB-6009 and are plotted on Figure 5.

The research of the dynamic characteristics of the analyzed hydraulic system with simultaneous operation of a pressure relief valve and a pressure reducing valve have been done for different volumes of oil V at outlet port of the reducing valve, simulating different sizes of the hydraulic cylinder in the real system. On Figure 5a the transient processes of the displacement of the relief valve x_r and the reducing valve x for the volume $V = 105\ cm^3$ have been presented. On Figure 5b the transient process of the pressures p_1 and p_2 for the volume $V = 105\ cm^3$ have bin presented. On Figure 5c and Figure 5d the same transient processes respectively have been presented for the volume $V = 250\ cm^3$, and on Figure 5e and Figure 5f the same transient processes respectively have been presented for the volume $V = 700\ cm^3$. It can also be noticed that there is a good match between the theoretically simulated transient processes with the experimental ones.

In the transient process the pressure p_2 with the volume $V = 105\ cm^3$ is increasing from the starting value of $p_{20} = 20\ bar$ to the maximum value of $p_{2V,max} = 60\ bar$ and then it is established at the stationary value of $p_{2e} = 30\ bar$ (see Figure 5b). This double rise of the pressure in the transition process would also correspond to a greater clamping force of the clamped part. An increase in the value of the pressure p_{2V} also occurs at the other volume values V. Increasing the volume V, the maximum pressure value $p_{2V,max}$ decreases.

From the presented diagrams, it can be concluded that the variation of the inlet pressure p_1 does not depend on the oil volume V, i.e. on the size of the hydraulic cylinder, driven in the system (see Figure 1). It is evident that during the transient process the pressure p_1 slighty increases from the initial value. The new steady-state value of the pressure p_1 depends on the slope of the static characteristics, as a result of the increased oil flow through the the relief valve. It can be seen from Figure 5 that the maximum pressure values $p_{2V,max}$ are obtained when the values of the displacement of the pressure reducing valve is $x = 0$. At this point the connection to the inlet pressure p_1 is interrupted. The maximum values of the displacement in the relief mode of operation x_{max} are approximately the same, close to the segment height of the throttling area $b = 0.35\ mm$ (see Figure 3b).

As can be seen from Figure 5b, the displacement of the control spool of the pressure reducing valve x at a volume $V = 105\ cm^3$ is performed rapidly, from $x = -0.75\ mm$ to the value of $x = 0$ for only $0.01\ s$. Beginning from this point up to $0.0225\ s$ the pressure reducing valve works as a pressure relief valve, whereby the pressure connection with the inlet p_1 is interrupted. The time of the transient process for different volumes of V of the studied three-way pressure reducing valve is approximately the same and it is around $t_0 = 0.035\ s$

(see Figure 5) when the set value of $p_{2e} = 30\ bar$ is achieved. Only the pressure peak of the pressure p_2 decreases when the volume of V increases.

Conclusion

The transient processes with change of the flow rate through the three-way direct-acting pressure reducing valves show a relatively high momentum increase of the reduced pressure and respectively of the clamping force in the transition processes. Reduction of the maximum value of the reduced pressure in the transient processes can be achieved by reducing the flow rate, respectively the velocity speed of the clamping piston. Input pressure variation in the transition process has little effect on the maximum value of the reduced outlet pressure.

A decrease in the variation of the reduced pressure in the considered transient processes can also be achieved introducing pilot operated pressure reducing valves. However, this leads to an increase in energy losses in the system due to the continuous flow of pilot oil through the valve towards the tank.

This article examines the unique effects of line-of-sight (LOS) and multipath propagation on single-input single-output (SISO), multiple-input single-output (MISO), and multiple-input multiple-output (MIMO) systems, outlining the challenges and opportunities presented by each propagation scenario. By understanding these interactions, we can gain insights into how modern wireless systems optimize signal quality and spectral efficiency in diverse environments.

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