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Research paper

STATIC AND DYNAMIC CHARACTERISTICS OF DIRECT OPERATED PRESSURE RELIEF VALVES

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Abstract: *The static and dynamic characteristics of direct operated pressure relief valve, manufactured by BoschRexroth are presented in this study. Linear and nonlinear mathematic model of the valve in hydraulic system with volume of oil at its inlet and pipeline at its outlet are given. The linear model allows determining, with sufficient approximation, the maximum pressure values and the frequency of oscillations of the transient response.*

Key words: *static and dynamic characteristics, pressure relief valve, transient response, linear model,*

1. INTRODUCTION

Many authors [1], [2], [3], [5] have investigated the static and dynamic characteristics of direct operated pressure relief valves. For determination of the pressure peak and oscillation frequency of the transient response there are not presented simplified expressions for fast engineering calculations.

Activating the directional control valves in the hydraulic systems with direct acting pressure relief valves occurs a transition process in which it is possible the pressure to reach values several times higher than the steady state value. This occurs overload of the system that can cause undesirable consequences.

Experimental and theoretical investigations of the transient response in hydraulic systems with this type of valves are presented. The coefficient of the hydrodynamic force acting on the poppet to close the valve is obtained from the experimental static characteristics.

2. A FUNCTIONAL DIAGRAM FOR DETERMINATION OF THE TRANSIENT RESPONSE OF THE VALVE

On *fig.1* a functional diagram of the test stand with direct acting pressure relief valve type BoschRexroth, 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 pressure pulsation after the pump, it is included a throttle with high inertial resistance.

The design of the poppet of this type of valve is characterized with turning the streaming flow of the oil with the ring 1, Fig.2.

This turning of the streaming flow is leading to decreasing of the component of the hydrodynamic force F_h which, together with the spring force F_s, influence to close the valve. With this design modification the error in the static characteristic is decreased. The value of the hydrodynamic force depends on the diameter and the shape of the ring 1 of the poppet and it is difficult to define it. Thus, it is necessary to use the experimental static characteristics with the displacement of the poppet of the valve measured, to determine the coefficient of the hydrodynamic force.

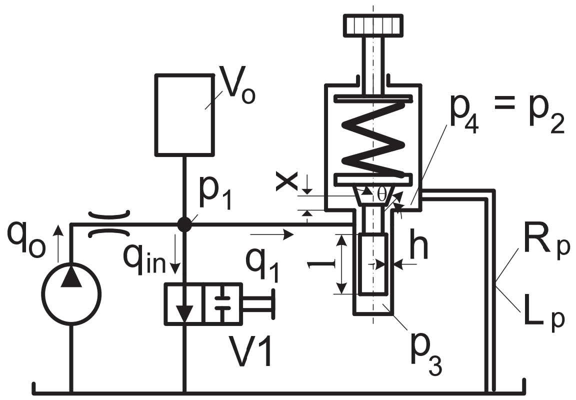

Fig.1. Functional diagram of the test stand

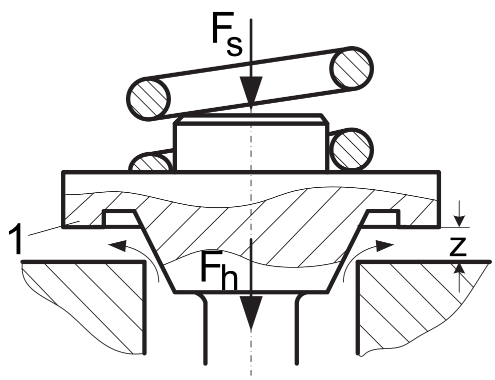

Fig.2. Hydrodynamic force modification

Experimental static characteristics of the specified direct operated pressure relief valve are presented on fig.3.

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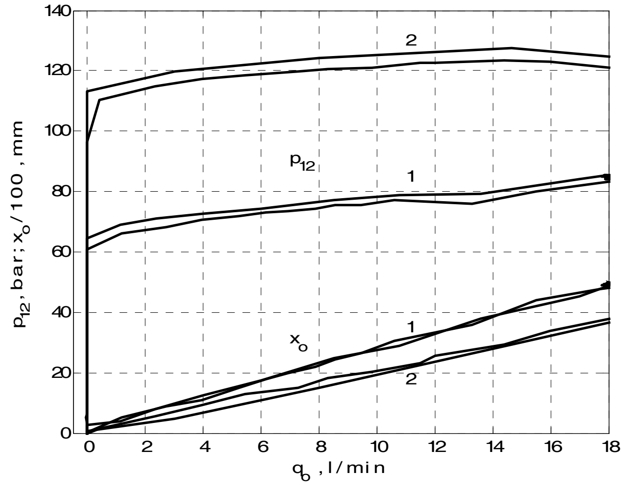

Fig.2. Experimental static characteristic of the specified valve

For a given flow q_0, a pressure drop $p_{1,2}$, a poppet area A_κ and a valve spring constant c and measured pressure – flow constant k_{st} of the static characteristic of the valve and displacement of the poppet x_0 of the valve (fig.3), the coefficient of the hydrodynamic force r_h is obtain by the expression [4]:

$$r_h = \frac{\frac{k_{st} \cdot q_0 \cdot A_k}{x_0} - c}{p_{1,2}} \qquad (1)$$

A rapid closure of the opening x_v of the directional control valve *V1* with plunger diameter d_v, creates a transient response. Pump flow q_0 enters in the inlet volume V_0 and the inlet pressure p_1 is increasing. The valve opens when the set pressure p_0 is reached and through the outlet pipeline the oil q_1 flows back in the tank.

3. MATHEMATICAL MODEL

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

- *Equation of continuity in front of the valve*

$$q_0 = q_{in} + q_v + q_1 \qquad (2)$$

where: $q_{in} = (1 - \frac{t}{t_1}) \cdot \mu_v \cdot \pi \cdot d_v \cdot x_v \sqrt{\frac{2}{\rho} \cdot p_1}$ –the flow through the directional control valve *V1*, which closes for time t_1; t – the time; μ_v – the flow coefficient through the directional control valve; d_v-the diameter of the directional control valve; $q_v = \frac{V_0}{K} \cdot \frac{dp_1}{dt}$ – the flow which enters in the volume V_0; $q_1 = \mu \cdot \pi \cdot d \cdot x \cdot sin\vartheta \sqrt{\frac{2}{\rho} \cdot p_{1,2}}$ – the flow through the valve with diameter d, the angle of flowing θ, the opening x, the pressure drop $p_{1,2}$ and the flow coefficient μ.

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

$$q_1 = q_2 = q_3 + A_k \frac{dx}{dt} \qquad (3)$$

where: A_k-area of the closing element of the valve; q_3 – flow through the control orifice in the valve.

- *Equation of motion of the closing element of the valve*

$$m \frac{d^2x}{dt^2} + c(h_0 + x) + r_h x p_{1,2} = A_k(p_3 - p_4) - F_T \qquad (4)$$

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

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

$$p_3 = p_1 - R_{a,l} A_k \frac{dx}{dt} - R_{a,m} (A_k \frac{dx}{dt})^2 - L_a A_k \frac{d^2x}{dt^2} \qquad (5)$$

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.

The pressure in the upper region of the closing element is obtain analogically when for this type of the valve is $p_4=p_2$.

- *Equation of flowing in the outlet pipeline*

$$p_2 = R_{p,l} q_2 + R_{p,m} q_2^2 + L_p \frac{dq_2}{dt} \qquad (6)$$

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

4. EXPERIMENTAL AND THEORETICAL CHARACTERISTICS

The measurement instruments were previously calibrated. A pressure transducer manufactured by BoschRexroth was used for pressure measurement. For displacement of the valve a position sensor manufactured by BoschRexroth was used. The data are stored in the computer through 14 bit data acquisition card manufactured by National Instruments. The steady state flow is 25 l/min.

As can be seen at *fig.4*, there is good agreement between experimental and theoretical results. For inlet volume of oil $V_0=52\ cm^3$ and the pressure value of $p_0=100$ bar the theoretical and the experimental transient response are little different because of the error in the simulation process at the beginning when the valve is not still open.

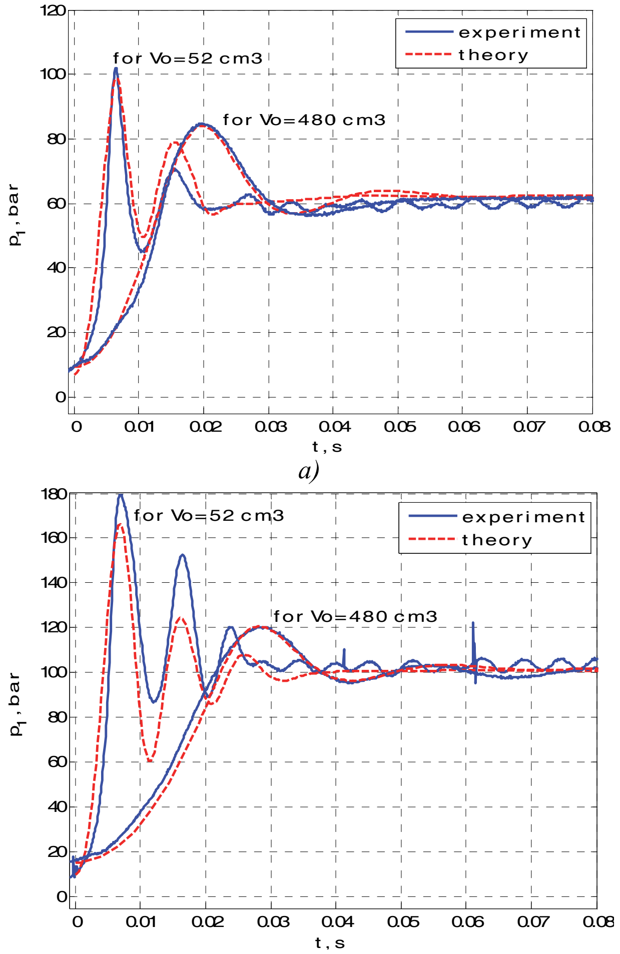

Fig.4. Experimental and theoretical dynamic characteristics of the specified valve for different pressures and volumes at inlet port

5. LINEAR APROXIMATION OF THE TRANSIEN RESPONSE

The linear approximation of the specified hydraulic system is shown on *fig.5*, where the transient process of the directional control valve *V1* is neglected. $P_{1,2}$, Q_0, Q_1, P_2 and Q_v are Laplace dimensionless parameters to the relative coordinates – pressure drop in the valve $\frac{\Delta p_{1,2}}{(p_{1,2})_0}$, inlet flow $\frac{\Delta q_0}{q_0}$, flow through the valve $\frac{\Delta q_1}{q_0}$, outlet pressure $\frac{\Delta p_2}{(p_{1,2})_0}$ and flow into the compressible volume of oil at inlet port $\frac{\Delta q_v}{q_0}$.

The transfer function W_k of the valve, presented in [4], is:

$$W_k = \frac{P_{1,2}}{Q_1} = k_{pq}\frac{T_k^2 s^2 + 2\xi_k T_k s + 1}{T_0^2 s^2 + 2\xi_0 T_0 s + 1} \quad (7)$$

where $k_{pq} = 1$ is the slope of the dimensionless static characteristic of the valve around the point of the regime after the transient process or with flow q_0 and $(p_{1,2})_0$; $T_k = \sqrt{\frac{m+L_a A_k^2}{c+r_h(p_{1,2})_0}}$ and $\xi_k = \frac{R_a A_k^2}{2T_k[c+r_h(p_{1,2})_0]}$ - time constant and relative damping coefficient of the closing element of the valve; $T_0 = \frac{T_k}{\sqrt{1+2k_{x,p}}}$ and $\xi_0 = \frac{\xi_k T_k + k_{x,p} T_A}{T_k\sqrt{1+2k_{x,p}}}$ - time constant and relative damping coefficient of the control system of the valve; $k_{x,p} = \frac{A_k(p_{1,2})_0}{[c+r_h(p_{1,2})_0]x_0}$ and $T_A = \frac{A_k x_0}{q_0}$ are respectively opening coefficient of the valve and time constant; s- Laplace operator.

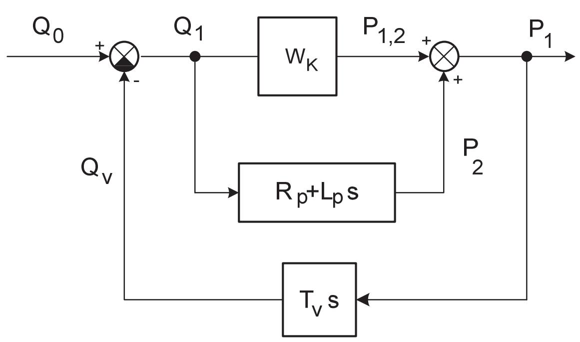

Fig.5. Block diagram of the linear model

The outlet flow from the valve is flowing through the pipeline with linear resistance in dimensionless coordinates $R_p = R_{p,l}\frac{q_0}{(p_{1,2})_0}$ and inertial resistance $L_p = L_{p,t}\frac{q_0}{(p_{1,2})_0}$:

$$P_2 = R_p Q_1 + L_p s Q_1 \quad (8)$$

The inlet pressure P_1 changes the flow Q_v, which enters in the volume V_0 according to the expression:

$$Q_v = T_v s P_1 \quad (9)$$

where $T_v = \frac{V_0}{K}\cdot\frac{(p_{1,2})_0}{q_0}$ is a time constant of the compressible volume of oil.

Solving the equations (7), (8) and (9), the transfer function of the whole system $W_0 = \frac{P_1}{Q_0}$ is obtain:

$$W_0 = \frac{a_1 + a_2 s + a_3 s^2 + a_4 s^3}{b_1 + b_2 s + b_3 s^2 + b_4 s^3 + b_5 s^4} \quad (10)$$

where the coefficients a_1, a_2, a_3, a_4 and b_1, b_2, b_3, b_4 and b_5 are directly obtain from the equations (7), (8) and (9) and there are presented in the following table:

Table 1:The coefficients of the linear model

$a_1 = 1 + R_p$	$b_1 = 1$
$a_2 = k_{pq}2\xi_k T_k + R_p.2\xi_0 T_0 + L_p$	$b_2 = 2\xi_0 T_0 + k_{pq}T_v + R_p T_v$
$a_3 = k_{pq}T_k^2 + R_p T_0^2 + 2.\xi_0 T_0.L_p$	$b_3 = T_0^2 + k_{pq}2\xi_k T_k T_v + R_p 2\xi_0 T_0 T_v + L_p T_v$
$a_4 = L_p T_0^2$	$b_4 = k_{pq}.T_k^2.T_v + R_p.T_0^2.T_v + 2.\xi_0 T_0.L_p.T_v$
	$b_5 = L_p T_0^2 T_v$

The analyses of the zeros and the poles of the transfer function W_0 shown that it can be approximated if the values of the coefficients a_3, a_4 and b_4, b_5 are neglected and the dynamics of the whole system can be approximately expressed with the following transfer function:

$$W_0 = \frac{a_1 + a_2 s}{b_1 + b_2 s + b_3 s^2} \tag{11}$$

The transient response of the pressure $p_1(t)$ with step input of the flow from zero to q_0, obtained from (11), is:

$$\frac{p_1(t)}{p_0} = a_1\left(1 - sign\varphi_1 \frac{A}{\sqrt{1-\xi^2}} e^{-\xi\omega_n t}.\sin\left(\omega_n\sqrt{1-\xi^2}.t + \varphi_1 - \varphi_2\right)\right) \tag{12}$$

where $\omega_n = \frac{1}{\sqrt{b_3}}$; $\xi = \frac{b_2}{2\sqrt{b_3}}$; $\varphi_1 = arctg\frac{a_2\omega_n\sqrt{1-\xi^2}}{a_1 - a_2\xi\omega_n}$; $\varphi_2 = arctg\frac{\sqrt{1-\xi^2}}{-\xi}$; $A = \sqrt{a_1^2 - 2a_1a_2\xi\omega_n + a_2^2\omega_n^2}$.

The maximal value of the pressure can be calculated from (12), as follows:

$$\frac{p_{1,max}}{p_0} = a_1\left(1 - sign\varphi_1 \frac{A.\sin(n\pi - \varphi_2)}{\sqrt{1-\xi^2}} \cdot e^{-\frac{\xi(n\pi - \varphi_1}{\sqrt{1-\xi^2}}}\right) \tag{13}$$

where the coefficient n depends of the sign of the expression $a_1 - a_2\xi\omega_n$: $n=0$ if $\varphi_1<0$ and $n=1$ if $\varphi_1>0$. For lower values of the time constant T_v of the inlet oil volume V_0 and high resistance R_p, $a_2\xi\omega_n > a_1$ and respectively $sign\varphi_1 = -1$ and $n=0$.

In the following table the values of the overload and the frequency of the oscillation are shown from the experiments, the theoretical nonlinear model and the approximated linear model (12).

Table 2. Comparison of the results

	p_0=60 bar, V_0=52 cm^3	p_0=60 bar, V_0=480 cm^3	p_0=100 bar, V_0=52 cm^3	p_0=100 bar, V_0= 480 cm^3
Experiment	$p_{1,max}$=100 bar ω_s=742 rad/s	$p_{1,max}$=85 bar ω_s=206 rad/s	$p_{1,max}$=180 bar ω_s=709 rad/s	$p_{1,max}$=120 bar ω_s=225 rad/s
Non-linear model	$p_{1,max}$=98 bar ω_s=778 rad/s	$p_{1,max}$=84 bar ω_s=227 rad/s	$p_{1,max}$=140 bar ω_s=650 rad/s	$p_{1,max}$=120 bar ω_s=217 rad/s
Approximated model	$p_{1,max}$=109 bar ω_s=671 rad/s	$p_{1,max}$=90 bar ω_s=250 rad/s	$p_{1,max}$=173 bar ω_s=775 rad/s	$p_{1,max}$=154 bar ω_s=264 rad/s

6. CONCLUSION

The approximated model (12) allows relatively fast calculation of the maximal pressure of the transient process. As can be seen, the direct acting pressure relief valve from BoschRexroth shows a relatively large amount of dynamic overload of the hydraulic systems. In cases where this overload cannot be accepted it is advisable to use pilot operated pressure relief valves.

To assess the dynamics of transient response process with pressure relief valves it is necessary to know the displacement x of the poppet of the valve in the static characteristics, as well.

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