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TRANSIENT RESPONSE PROCESS IN HYDRAULIC SYSTEMS WITH DIRECT OPERATED PRESSURE RELIEF VALVES

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Abstract: The transient response characteristic of the inlet pressure in hydraulic systems with direct operated pressure relief valves has been experimentally determined in this paper. The mathematical model of the system, theoretically determined the pressure peak and oscillation frequency of the system, as well, has been shown.

1. INTRODUCTION

Many authors [1], [2], [3] have investigated the transient response characteristics of hydraulic systems with direct operated pressure relief valves. For determination of the pressure peak and oscillation frequency of the transient response there are not presented simplified expressions.

Activating the directional control valves in the hydraulic systems with direct acting pressure relief valves it 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 that can cause undesirable consequences.

Experimentally and theoretically determined transient response of hydraulic systems is presented in this paper and mathematical expressions for quality determination of the transient response are shown, as well. This will allow the designers to assess in advance the dynamics of the created system.

2. A FUNCTIONAL DIAGRAM FOR TRANSIENT RESPONSE CHARACTERISTICS DETERMINATION

On *fig.1* a functional diagram of the test stand with direct acting pressure relief valve type Bosch, 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.

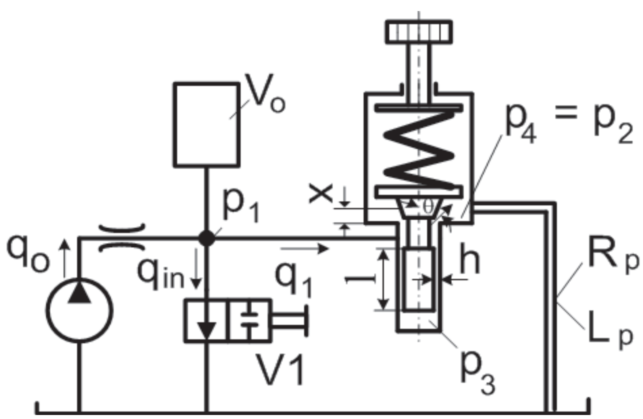

Fig.1. Functional diagram of the test stand

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 OF THE TRANSIENT RESPONSE

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

- *Equation of continuity in front of the valve*

$$q_0 = q_{in} + q_v + q_1 \tag{1}$$

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

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

$$q_1 = q_2 = q_3 + A_k \frac{dx}{dt} \tag{2}$$

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*

$$m\frac{d^2x}{dt^2} + c(h_0 + x) + r_h x p_{1,2} = A_k(p_3 - p_4) - F_T \tag{3}$$

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 $x=0$; $r_h = \mu\pi d. sin2\theta$ –coefficient of the hydrodynamic force; 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 between the piston of the closing element and the body of the valve:

$$p_3 = p_1 - R_{a,l}A_k\frac{dx}{dt} - R_{a,m}(A_k\frac{dx}{dt})^2 - L_a A_k \frac{d^2x}{dt^2} \tag{4}$$

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

$$p_2 = R_{p,l}q_2 + R_{p,m}q_2^2 + L_p\frac{dq_2}{dt} \tag{5}$$

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.

4. EXPERIMENTAL AND THEORETICAL CHARACTERISTICS

The experimental characteristics of the valve are determined with pressure transducer. The data are stored in the computer through 14 bit analog-digital convertor with sample time of 100 μs. The steady state flow is 25 l/min.

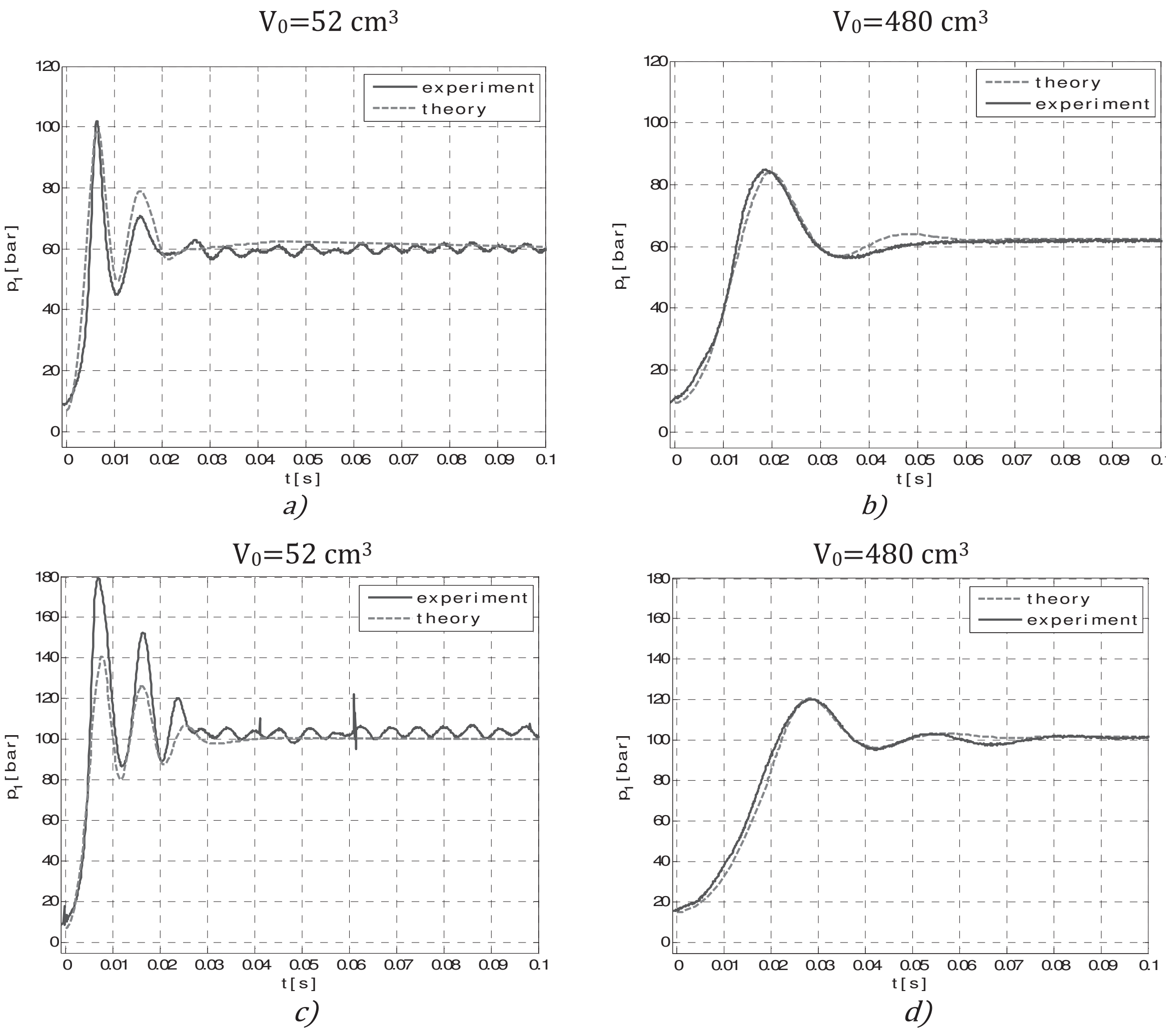

Fig.2 Experimental and theoretical dynamic characteristics for the valve type Bosch for different values of the pressure and volume at the inlet port

As can be seen at *fig.2*, at higher inlet volume there is good agreement between experimental and theoretical results. For inlet volume of oil $V_0 = 52\ \mathrm{cm}^3$ and the pressure value of $p_0 = 100\ \mathrm{bar}$ the theoretical transient response shows lower oscillation.

5. LINEAR APROXIMATION OF THE TRANSIEN RESPONSE

Presented transient responses at *fig.2* have the form of the transient response of a linear second-order system. The parameters of this system can be approximately determined if it is analyzed the linear model of the system around the steady state values of the parameters p_0 and q_0. On *fig. 3* is shown a structural diagram of the investigated system. The transient response of closing of the directional control valve *V1* is neglected. $P_{1,2}, Q_0, Q_1, P_2$ и Q_v are Laplace dimensionless parameters to the relative coordinates – pressure drop in the valve $\frac{\Delta p_{1,2}}{(p_{1,2})}$, 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:

$$W_k = \frac{P_{1,2}}{Q_1} = k_{pq}\frac{T_k^2 s^2 + 2\xi_k T_k s + 1}{T_0^2 s^2 + 2\xi_0 T_0 s + 1} \qquad (6)$$

where $k_{pq} = k_{st}\frac{q_0}{(p_{1,2})_0}$ is the slope of the dimensionless static characteristic of the valve; $T_k = \sqrt{\frac{m+L_a A_k^2}{c+r_h(p_{1,2})_0}}$ 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.

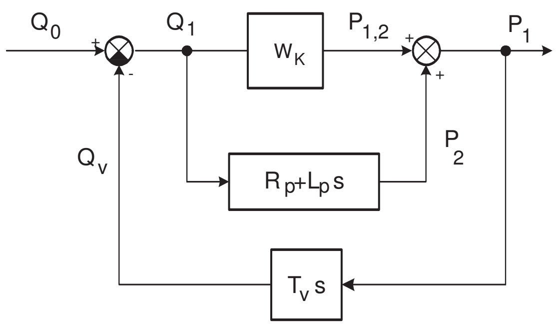

Fig.3 Block diagram of the linear model

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 $L_p = L_{p,t}\frac{q_o}{(p_{1,2})_0}$:

$$P_2 = R_p Q_1 + L_p s Q_1 \qquad (7)$$

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

$$Q_v = T_v s P_1 \qquad (8)$$

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

Solving the equations (6), (7) and (8), the transfer function of the whole system $W_o = \frac{P_1}{Q_0}$ is obtain:

$$W_0 = \frac{a_1 + a_2 s + a_3 s^2 + a_4 s^3}{b_1 + b_2 s + b_3 s^2 + b_4 s^3 + b_5 s^4} \qquad (9)$$

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 (6), (7) and (8). An analysis of the characteristic polynomial shows that the values of the roots are strongly different, so it is possible to approximate the transient response of the system with second order system. The values of the coefficients b_1, b_2, b_3 for the simplified system are:

$$b_1 = 1$$

$$b_2 = 2\xi_0 T_0 + k_{pq} T_v + R_p T_v \quad (10)$$
$$b_3 = T_0^2 + k_{pq} 2\xi_k T_k T_v + R_p 2\xi_0 T_0 T_v + L_p T_v$$

From these values it is possible to calculate the frequency of oscillation of the system $\omega_s = \frac{\sqrt{(1-\xi_s^2)}}{\sqrt{b_3}}$ and the overload $p_{1,max} = p_0(1 + e^{-\xi_s \pi/\sqrt{1-\xi_s^2}})$. Here $\xi_s = \frac{b_2}{2\sqrt{b_3}}$ is a relative damping coefficient of the approximated second order system.

In the following *Table 1* 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 (10) if the inlet flow is $25\ l/min$. The time of closing of the directional control valve *V1* is around *0.018 s*.

Table 1.

	p_0=60 bar, V_0=52 cm³	p_0=60 bar, V_0=480 cm³	p_0=100 bar, V_0=52 cm³	p_0=100 bar, V_0= 480 cm³
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	$p_{1,max}$=72 bar ω_s=689 rad/s	$p_{1,max}$=82 bar ω_s=257 rad/s	$p_{1,max}$=128 bar ω_s=679 rad/s	$p_{1,max}$=135 bar ω_s=272 rad/s

As can be seen from the above table the approximated model deviates maximum of around 20% from the experimental data. This is due to relatively slow closing of the directional control valve *V1*, which reflects the results. The data of the experimental and non-linear model good match.

6. CONCLUSION

The approximated model allows simple calculation of the parameters of the overload. As can be seen, the direct acting pressure relief valve from Bosch 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.

REFERENCES

1. Backé W., Murrenhoff H. Grundlagen der Ölhydraulik. Institut für fluidtechnische Antriebe und Steuerungen. Technische Hochschule Aachen. 1994
2. Will D., Ströhl H., Gebhardt N. Hydraulik. Springer-Verlag Berlin. 1999.
3. Brodowski W. Beitrag zur Klärung des stationären und dynamischen Verhaltens direktwirkender Druckbegrenzungsventile. Dissertation. 1974.
4. Komitowski M. Fluid Power Elements. Tehnika, Sofia, 1985.

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