Optimization of the quality parameters of the transition process in hydraulic pressure regulators

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Abstract. In direct-acting hydraulic regulators, the source of harmful self-oscillations of pressure in hydraulic systems may be their own damping of these regulators. Their elimination is possible by decreasing the valve gain coefficient according to the flow rate, which is determined by the magnitude of the fluid flow increment per unit of displacement of the regulator when the working fluid is throttled in the valve itself.

The article discusses the possibility of optimizing the quality parameters of transient processes in pressure regulators with the help of their integral evaluation using the example of optimization of the gain coefficient for the flow rate and the damping coefficient. It is established that the reduction of the gain is most appropriate for its large values. A further decrease in this ratio slightly improves the quality of the transition process, and may worsen the flow characteristic of the steady-state controller due to excessive displacement of the regulator. The tests confirmed the validity of this assessment of the quality of dynamic processes in the transient modes of these regulators.

Keywords: an integral performance index of the transient process; a pressure regulator; a flow gain; a damping coefficient; a working fluid.

1. Introduction

Hydraulic pressure regulators of direct action of hydraulic drive systems and hydraulic automation with low self-damping can initiate harmful self-oscillations in the hydraulic system [1]. The self-oscillations can be repaired by reducing, for example, the pressure controller gain (the safety valve) according to the flow rate, which is determined by the increment of the hydraulic fluid consumption through the throttle regulator.

With a small flow rate, when the dynamic component of the pulsating fluid flow is insignificant in magnitude, the effect of additional dissipative losses on the change in the optimal value of viscous friction can be neglected. But with large fluctuations in fluid flow through the pressure regulator, the optimal value of the viscous friction coefficient will change significantly with a simultaneous small change in the gain.

Reducing the required optimum damping coefficient to the level of a small intrinsic damping ensures an aperiodic transient process in a direct-acting pressure regulator without additional devices [2].

2. Problem statement

The optimal flow gain of pressure regulators with a shut-off and regulating element made of elastomer [2] can be determined in a number of ways including the integral criterion of the quality of the transition process as follows [3]:

\[ I = \int_{0}^{\infty} p^{2}(t)\,dt. \]
where \( p(t) \) is a dynamic component of the fluid pressure in the pressure cavity of the pressure regulator compared with a new pressure level, which is established after all transients have died out at the step disturbance; \( t \) is the time of the process; \( I \) is the value of the integral, which expresses the total (integral) quadratic error of the dynamic pressure in the transition process.

The value of the integral expresses an assessment of the dynamic accuracy of the hydraulic circuit attenuator (the safety valve). The minimum value of this integral provides dynamic stability and rapid damping of oscillations with a relatively slight overshoot of pressure.

When calculating the integral (1), it is rational to use linearized third-order differential equations, which approximately describe dynamic processes in pressure regulators.

As a step perturbation, the pressure quantity in the pressure regulator \( p_H(t) \) at the beginning of a short pressure pipeline with a hydraulic accumulator having low dynamic resistance is chosen (Figure 1).

![Figure 1. The scheme of the pressure regulator with the perturbation pressure.](image)

1- a regulator; 2- a hydraulic accumulator; 3- a distributor.

With a step perturbation of the pressure in the pressure cavity of the regulator, a transient process occurs, and at the beginning of the short pressure pipeline, the pressure \( p_H(t) \) increases abruptly by the value of \( p_b \).

A step change of the fluid pressure in the pressure pipeline, which is made by prompt opening of the distributor 3 connecting the pipeline to the hydraulic accumulator 2, causes an increase in the flow through the throttle slot of the regulator by the value \( p_b K_0 \).

3. Theory

The differential equation in the operator form for the dynamic pressure increment \( p(t) \) in the pressure cavity of the regulator under pressure perturbation is as follows [4]:

\[
(a_1 S^3 + a_2 S^2 + a_3 S + a_4) P(S) + (a_5 S^2 + a_6 S + a_7) P_0(S) = 0,
\]

where \( a_1 = \frac{m W}{E} \); \( a_2 = m \left( \frac{K_m Z_0}{2p_0} + K_s \right) \); \( a_3 = \frac{W}{E} \); \( a_4 = F^2 + \frac{c W}{F} + K \left( \frac{K_m Z_0}{2p_0} + K_s \right) \); \( a_5 = F K_0 + c \left( \frac{K_m Z_0}{2p_0} + K_s \right) \); \( K_0 = \mu \ell \sin \frac{\alpha}{\rho} \); \( K_s = \int_{p_0}^{p_f} \left( \frac{2}{p_0} \right) \); \( a_6 = m K_0 F \); \( a_7 = K_s K_0 F \); \( a_8 = c K_0 F \); \( P(S) P_0(S) \)

– Laplace images of the dynamic equations \( p(t) \) and \( p_0(t) \); \( m \) - the mass of the moving parts of the regulator with the added-liquid mass; \( F \) - the working area of the shut-off and regulating element of the regulator; \( W \) - the volume of the pressure cavity of the regulator; \( C \) - the unit stiffness of the shut-off and regulating element in terms of the constant component of the reactive fluid flow force; \( E \) - the reduced elastic modulus of the liquid in terms of the gas component in the liquid, elasticity of the
pressure cavity of the regulator and the effect of the hydraulic accumulator; \( p_0, z_0 \) – the initial pressure and displacement of the shut-off and regulating element in steady-state to perturbation; \( K_0 \) – the coefficient of viscous friction; \( K_0 \) – the flow gain equal to the increment of the fluid flow through the throttle slot of the regulator per unit of displacement of the shut – off and regulating element; \( \mu \) – the flow coefficient in the throttle slot of the regulator; 1, \( t \) -the number and width of the grooves of the walls of the regulator, along which the shut – off and regulating element moves; \( \alpha \) – the half of the throttle angle of the regulator; \( K_0 \) - the coefficient of additional energy dissipation under sudden change of live flow; \( f_0 \) - the area of the inlet of the regulator body; \( \xi \) - the coefficient of resistance of the inlet; \( p_{\text{ref}} \) - the initial pressure in the pressure pipeline.

The transition process in a linear system with a step perturbation and zero initial conditions is equivalent to the transition process without perturbations in a linear system, for which the left part of the differential equations remains unchanged, and the right part will be zero. At the same time, the equivalent initial conditions should be chosen according to the relations [3]:

\[
p(0) = -p_z; \quad p'(0) = \frac{a_1}{a_3} Q_s = \frac{Q_s}{K_c};
\]

\[
p^*(0) = \frac{Q_s}{a_1(a_1 - \frac{a_2h_2}{a_3})} = \frac{Q_s}{K^*_z} K^*_z;
\]

where \( Q_s = K_p p_z \); \( K_c = \frac{W}{E} \); \( K^*_z = \frac{K_z}{2p_0} + K_0 \); \( p_s \) – the pressure increment in steady-state with an increase in flow rate by \( Q_s \).

The value of the integral (1) for the transient described by the equation (2) is determined as follows [3]:

\[
I = \frac{1}{a_0} \left( h_1 + \frac{a_2a_1h_1 + a_3^2h_3}{a_1a_2 - a_1a_3} \right),
\]

where \( h_1, h_2 \) and \( h_3 \) are the functions depending on initial conditions:

\[
h_1 = a_1 \frac{p'^{2}(0)}{2} + a_1 \frac{p''^{2}(0)}{2} + a_0 \frac{p^{2}(0)}{2};
\]

\[
h_2 = a_1 p^*(0) p(0) + a_1 \frac{p''^{2}(0)}{2} + a_0 \frac{p^{2}(0)}{2};
\]

\[
h_3 = a_1 \left( p^*(0) p(0) - \frac{p'^{2}(0)}{2} \right) + a_2 p'(0) p(0) + a_1 \frac{p^{2}(0)}{2};
\]

Under low unit stiffness of the shut-off and regulating element (\( c=0 \)) the characteristic \( p_0 \) is \( f(Q_0) \) and \( p_s \) is 0. In the circumstances, the functions of the initial conditions (5) are expressed as follows:

\[
h_1 = \frac{Q^2}{2K_c} \left( \frac{mK^2}{K^2} + \frac{F^2 + K_0K_s}{K_c} \right);
\]

\[
h_2 = \frac{Q^2}{2K_c} \left( K_s - \frac{mK^2}{K_c} \right); \tag{6}
\]

\[
h_3 = -\frac{Q^2m}{2K_c}.
\]

Substituting the expressions (6) and the coefficients of the equation (2) into equality (4), we obtain an integral against the parameters of the oscillatory system and the value of the step perturbation:
The coefficient of additional losses of $K_0$ in real structures can be considered constant. In this case, it is much easier to find the minimum. The optimal value of the gain $K_0$ and the coefficient of viscous friction, which are in this case variables, correspond to finding the minimum.

Representing the expression (7) in a dimensionless form, where the variable parameters $K_e$ and $K_Q$ are replaced by the corresponding dimensionless variables, we obtain:

$$I = \frac{Q^2 m}{2FK_0K_e} \left[ \frac{(mK_0 + K_e)(K_e + F^2 + \frac{K_e^2}{m})}{mK_eK_0 + (F^2 + mK_0^2)K_e + FK_e(FK_e - mK_0)} - 1 \right].$$  \hspace{1cm} (7)

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Representing the expression (7) in a dimensionless form, where the variable parameters $K_e$ and $K_Q$ are replaced by the corresponding dimensionless variables, we obtain:

$$I = \frac{I_0}{I_0} = \frac{1}{3\sqrt{3}} \begin{vmatrix} y^2 + axy + x \end{vmatrix}.$$ \hspace{1cm} (8)

where $\bar{I} = \frac{I}{I_0}$ - a dimensionless value of the integral (7); $I_0 = \frac{3Q^2}{2F^2} \sqrt{\frac{3m^3}{K_e}}$ - private value of the same integral; $x = \frac{K_0}{F^2} \sqrt{mK_e}$ - a dimensionless flow gain; $y = \frac{K_e}{F} \sqrt{\frac{m}{K_e}}$ - a dimensionless coefficient of viscous friction; $a = F \sqrt{\frac{K_e}{m}}$ - a dimensionless coefficient of additional energy dissipation.

The calculation of the expression minimum (8) in general terms is reduced to solving a system of two algebraic quintic equations for particular cases. In the calculations of the initial approximation, it is possible to determine the minimum for the limit case ($a \to 0$), wherein the hydraulic losses in local resistances ($K_0 \to 0$) and increments in the fluid flow ($\frac{K_0z_0^2}{2p_0} \to 0$), are not taken into account, which corresponds to the small initial displacement of the shut-off and regulating element ($z_0 \to 0$) and to the requirement $\frac{mK_0z_0}{2p_0} \frac{K_W}{E}$. In this case, the expression (8) is represented as

$$I = \frac{1}{3\sqrt{3}} \begin{vmatrix} y^2 + x \end{vmatrix},$$ \hspace{1cm} (9)

i.e., the calculation of the minimum is reduced to solving the simultaneous quadric equations.

Taking $\frac{\partial I}{\partial x} = 0$ and $\frac{\partial I}{\partial y} = 0$, we obtain a system of equations:

$$x^2 + 2y^2x - y^4 = 0;$$
$$3y^2 + 1)x - 2y^2 = 0,$$

which has positive roots:

$$x^* = \frac{1}{3\sqrt{3}} \quad \text{and} \quad y^* = \frac{1}{\sqrt{3}}.$$ \hspace{1cm} (11)

The obtained values of dimensionless variables $x$ and $y$ correspond to the minimum quadratic error of transients under slight displacements of the shut-off and regulating element.

The expressions (11) correspond to the dimensional values of the optimal flow gain and the viscous friction coefficient:

$$K_0^* = \frac{F}{3\sqrt{3}} \sqrt{\frac{c_\infty}{m}} \quad \text{and} \quad K_e^* = \frac{m}{\sqrt{3} \sqrt{\frac{c_\infty}{m}}}.$$ \hspace{1cm} (12)
where \( \sqrt{\frac{c_w}{m}} = \omega_c \) – own angular frequency in critical damping (at the border of stability) for the regulator with a horizontal flow characteristic and low losses; \( c_w = \frac{EF^2}{W} \) – the stiffness of the added volume of liquid.

In approximate calculations, the value of the optimal gain can be attributed to the unit of area and expressed in fractions of the natural frequency \( f_c = \frac{\omega}{2\pi} \):

\[
K_v^* = \frac{K_v}{F_c} = \frac{2\pi}{3\sqrt{3}} f_c \approx 1.2 f_c. \tag{13}
\]

It is possible to present the value of the optimal reduced damping coefficient, which determines the decrement of free oscillations of the regulatory body, by the same procedure:

\[
n^* = \frac{K_v^*}{2m} = \frac{\pi}{\sqrt{3}} f_c \approx 1.8 f_c. \tag{14}
\]

From graphs \( I = f(x) \) and \( I = f(y) \) (Figure 2), we can see that the loss of gain is appropriate for its large values. Further reduction of this coefficient slightly improves the transient process. However, it is possible to worsen the flow characteristics of the steady-state valve because of the excessive displacement of the regulating element.

![Figure 2. Dependency graphs without a dimensional integral error in the dimensionless amplification coefficient and the dimensionless ratio of "viscous" friction](image)

4. Supplementary data

In determining the effect of fluid flow on the quality of the transition process, special studies have been carried out with several versions of structures with different magnitudes of the gain coefficient for the flow. These studies have shown that a significant reduction in the gain coefficient on the flow rate ensures an adequate margin of stability in the event of an instantaneous jump in the flow rate of the fluid, and the pressure fluctuations in the pressure chamber cavity attenuate the pressure regulator intensively.

One of the possible reasons for the intense attenuation of pressure fluctuations in pressure regulators with a decrease in the gain coefficient on the flow rate is the additional energy losses in the pressure cavity and the supply channels of the regulator.

The experiments confirm the existence of an optimal flow gain under other equal conditions (preservation of the geometric dimensions of the regulator elements, its mass, friction forces, etc.).

5. Summary

Application of parametric optimization of the quality of dynamic (transient) processes in pressure regulators by their integral estimation is effective. Integral minimum (1) provides high speed of
pressure regulators (low acceleration time [3-6] and a small number of oscillations at a step perturbation).

The obtained expressions (13) and (14) for the calculation of optimum gain and damping are the result of the initial approximation, which can be refined by introducing coefficients that take into account the influence of the nonlinearity of the flow characteristics, the initial displacement of the regulating element and other factors.

6. References

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