Study on hydraulic direct-acting relief valve

V V Syrkin, P D Balakin, V A Treyer
Omsk State Technical University, 11, Mira ave., Omsk, 644050, Russia

e-mail: SyrkinVV@mail.ru

Abstract. Valves with elastic shut-off-and-regulating elements are increasingly applied in hydraulic systems, working in conditions of excessive vibrations. The objective of this study is to determine the dynamic characteristics of regulators with shut-off-and-regulating elements. The method of phase trajectories establishes the controller's parameters and modes, which provide its work stability. The authors receive dynamic characteristics providing the absence of self-oscillations at the expense of additional damping of the shut-off- and-regulating element. They make a test sample of the pressure regulator with an elastic shut-off-and-regulating element. The article defines the characteristics curve of the pressure regulator (direct operated pressure relief valve) from the initial conditions. The obtained results can be used, when designing hydraulic technological machines. Keywords – oscillation damping, an elastic shut-off-and regulating element.

1. Introduction

Modern technological machines for various purposes often have hydraulic gears and hydraulic tools, which shall comply with the highest requirements to reliability, velocity, stable vibration-free work, constancy of controlled performance, fluid flow, operating accuracy, etc. These requirements are provided by the protection and prevention structure of hydraulic systems, which ensure reliable and steady work of the hydro drive. Protection and prevention components include: safety pressure regulators and relief valves. One of the most important conditions for the work of such devices is dynamic stability. It means that continuous oscillatory processes should not appear in the valve throughout the whole flow range, especially on the main operative range. These phenomena are linked with the presence of elastic medium (working fluid), which may cause malfunction of the whole hydraulic system related to the valve. There are various constructions of devices, such as direct operated pressure relief valve and indirectly working safety valve that are subdivided into many more variant performances including relief valves with elastic shut-off- and-regulating elements. The authors consider that such valves are more prospective [7].

2. Materials

Dynamic properties of valves with direct and indirect actions in traditional performance have been studied most thoroughly. In the papers [1, 2, 3, 4] their velocity, transient and frequency characteristics, choice of design factors of structural and functional diagrams are reviewed. At the same time there is practically no information about such studies for devices with elastic shut-off-and-regulating elements. The objective of this study is to determine the dynamic characteristics of regulators with shut-off-and-regulating elements.

3. Theory

Mathematical modeling of dynamic processes of hydraulic pressure regulators typically includes the record keeping of elastic properties of pressure fluid, liquid flow forces, mass of regulator closure, viscous and dry friction during the element motion process, as well as the quadratic law of fluid discharge through the regulator working window [1, 2, 3]. Nonlinear characteristics are subjected in most cases to linearization, differential equations become linear, that allow determining the dynamic characteristics of regulators with sufficient accuracy for engineering analysis.

The regulator under study (Figure 1) contains an enclosure with the internal cylinder 1 with a circular dovetail groove, in which there is a reinforced metal spring 3, an elastic shut-off- and-regulating element 2, ensuring a desired level of hydraulic fluid pressure. There are performed coaxial input and output axial channels I and III enclosed, linked by radial channels (II) through a smaller base of a dovetail groove.

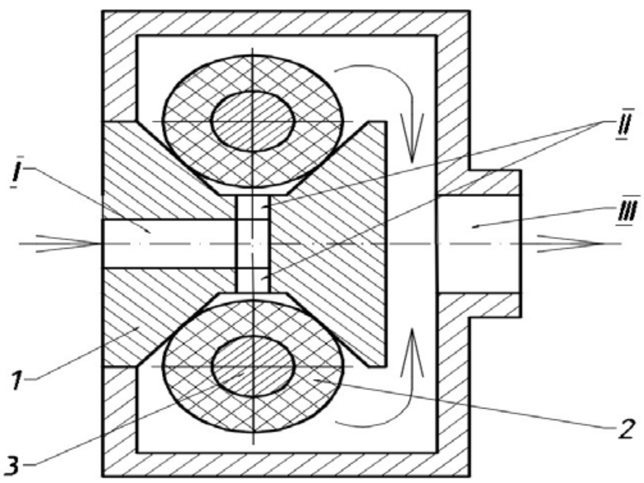


Figure 1. Hydraulic pressure regulator with shut-off and regulating the elastic element.

In the initial position the regulator (valve) is closed, the shut- off element 2 separates the channels (I) and (III). When increasing the pressure in the hydraulic system to which you are connected, the pressure regulator shut-off and regulating element 2 pressure exceeding the level regulator (pressure relief valve), deformed in the radial direction, forming an annular crack between the Conic surfaces grooves and locking element and providing a reset of the working fluid through the output channel III. By reducing the pressure shut-off element under the action of the elastic force element (springs) returns to its original position, cutting through the input and output channels III, I..

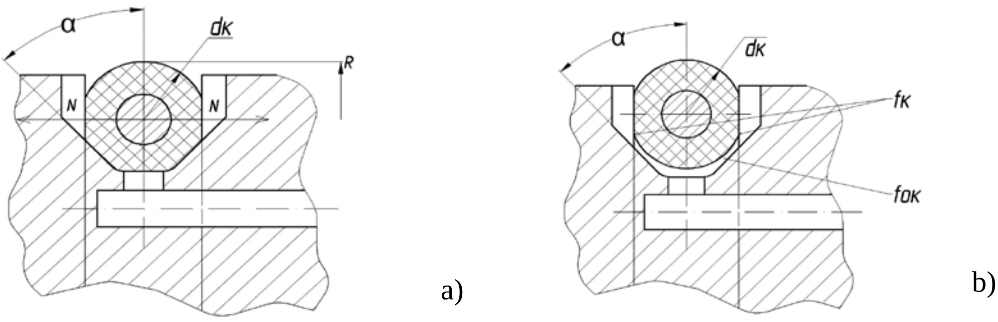


Figure 2. Position locking-regulating element a), working b).

For damping vibrations locking-regulating element in the design of the valves can be provided with vertical walls of the Groove in which the locking-regulating element (Figure 2) for sealing surfaces which creates a constant contact pressure excessive based on elastic material properties (E) and compression, when $\varepsilon<30\%$ of the pressure determined according to [1]

$$p_k = 1.25 \cdot \varepsilon \cdot E \cdot 10^{-3}\ MPa.$$

Contact area f_k locking-regulating element with vertical walls is $f_k = 2 \cdot \pi \cdot d \cdot \lambda$ defined as the grooves in which the width of the contact with the wall element is determined by the degree of deformation of this element when mounting for rubber hardness H_p = 55...75 according to Shore and it can be defined by the empirical formula $\lambda_k = 0.03 \cdot d_k \cdot \varepsilon$, mm [1].

The normal force is defined as $N_k = p_k\ f_k$ and the corresponding friction, taking into account the non-linear nature can be determined as $F_{mp} = f_{mp}\ N_k$ signsr, where r is the radial displacement locking-regulating element s-operator of differentiation, ftr-coefficient of friction between the vertical wall and locking-regulating element.

Mathematical model of the regulator with preserved fluid volume can be represented by the equations of equilibrium of the forces on the locking- regulating element and balance costs in the working area of the regulator (pressure relief valve). This assumes that the pressure at all points of pressure line is changed at the same time, the discharge coefficient of the working fluid through the working window regulator and its constant temperature, hydraulic resistance of drain line slightly, hydrodynamic forces are not counted because of small fluid flow speeds.

Applying the principle of D'alembert and the law of conservation of mass for some fixed hydraulic circuit, we get a system of differential equations that describe the dynamic processes occurring in the hydraulic circuit, which includes a pressure regulator.

$$\mathrm{M}s^2 r + \beta s r + c(r_0 + r) + F_{mp} = pf, (1)$$

where m is the mass of the regulatory element; r – move an element; β – coefficient of viscous friction; c – the stiffness of the spring; r_0 is the pretension of the spring; f – square, which operates fluid pressure; p = inlet pressure regulator; s – operator of differentiation.

Balance the cost equation for fluid pressure regulator can be represented as follows

$$Q = Q_p + Q_{сж} + Q_n,, (2)$$

where Q is the flow rate for preserved volume, $Q = f_1 su$; $Q_p = B_p f_{ok} sqrt(\Delta p)$ – liquid flow rate passing through the window; $Q_{сж} = kWsp$ – fluid consumption caused by its compression; $Q_n = f sr$ – fluid comsumption formed by the offset of the regulating element; f_1 - area of preserved volume which is bounded by the edges of the cylinder volume; B_p - constant of working window regulator; Δp - pressure difference in the regulator; k - coefficient of compressibility of fluid; W is the amount of input cavity of the regulator; u - move of preserved volume.

Now the system of differential equation takes the following form:

$$ms^2r = pf - \beta sr - c(r_0 + r) - F_{mp},;$$

$$f_1 su = B_p f_{ok} sqrt(\Delta p) + kWsp + fsr. (3)$$

The dynamic stability of the pressure regulator may be the presence or absence of autooscillations in a wide range of frequencies, from acoustic resonance phenomena to exciting fluctuations caused by fluid compressibility or slackness has been established pump drive [2]. So, the volatility of the regulator can be caused by coincidence own frequency regulator with frequency pulsation of pump pressure.

The volatility caused by Governor compressibility of fluids, can be called as follows. On the application of a pressure pulse to the regulatory body in the direction of closing pressure in pressure cavity increases. Released when compressing the volume to fill the newly received liquid. After removing the pulse pressure regulator will begin to move in the opposite direction, that is accompanied by a decrease in pressure in the pressure valve cavity. Through the disc annular gap along with the main regulator flow must pass and the consumption that is released when expansion of the liquid, resulting in additional forces, aimed in the direction of movement of the shutter (regulatory body). If it is smaller, the damping force adjuster after some time return to the established State. Otherwise, the controller will enter the mode self- excited oscillations with constant or increasing amplitude.

For linear systems, defining the boundaries of a steady- state is trivial in nature. For nonlinear systems require further analysis, particularly in terms of how to define those boundaries. It is necessary to determine the feasibility of linearization varying nonlinearity identified when solving particular tasks, and also obtained when this error.

Taking into account non-linearities this model it is advisable to apply the method of phase trajectories [5], which allows you to explore nonlinear vibrational systems without having to build a whole picture phase trajectories. When you use this method, built phase trajectory of polucikla (in the case of symmetric phase trajectories) or (in the case of asymmetrical phase trajectories) traffic system, which defines a function turned [5], on which to construct the diagram Kenigsa-Lamereâ. On the type of chart, you can judge the system limit cycles or subcarinal regimes [6]. The initial values of the system parameters and disturbing impact of maximum are chosen. In the absence of limit cycles can argue that they will be absent in any other primary values and revolting influences.

Study of dynamic stability regulator conducted Runge-Kutta numerical method. Solution algorithm of mathematical model of pressure regulator (pressure relief valve direct action) provides for the transition process and phase trajectory in polar coordinates. For the study were taken the following initial conditions:

$$p_0=2MPa;\ m = 0.1\cdot 10^{-3}\ kg;\ f = 0.5\cdot 10^{-4}\ m;$$

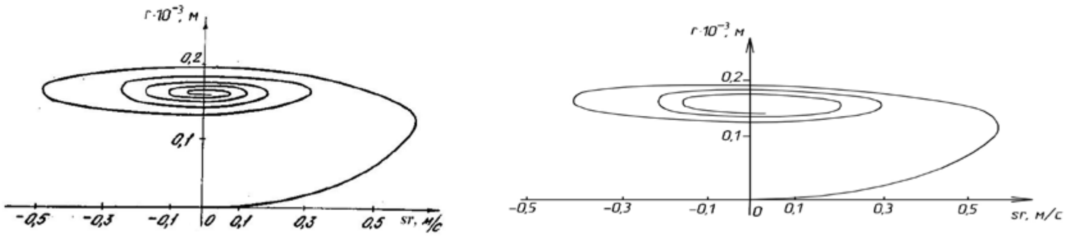
$$f_{mp}=0.075;\ d = 0.8{\cdot}10^{-2}\ m;\ b = 50N{\cdot}s/m;\ c=103N/m;$$

$$r = 3{\cdot}10^{-3}\ N/m;\ f = 0.16\cdot10^{-2}\ m^2;\ m = 0.4;$$

$$B = 0.18{\cdot}10^{-3}m^3/Ns;\ k = 0.714\cdot10^{-3}\ 1/MPa;$$

$$W = 2{\cdot}10^{-4}\ m^3.$$

Phase trajectory obtained by numerical method, compared with constructed graphically (Figure 3). The resulting difference should not exceed 5%, which can be considered as quite acceptable.

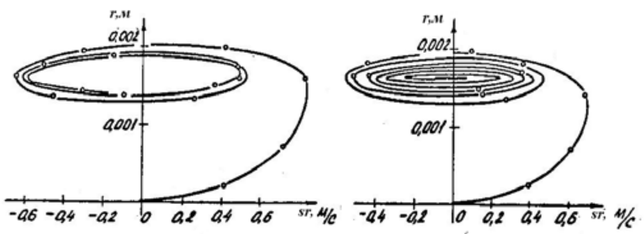


a) b)

Figure 3. Phase trajectory of locking-regulating valve element (shutter) received (a) numerical and graphical (b) methods.

The study found that the dynamic stability of safety valves direct action largely depends on: the mass closure (shutter), the stiffness of the spring, volume pressure line, dry and viscous friction.

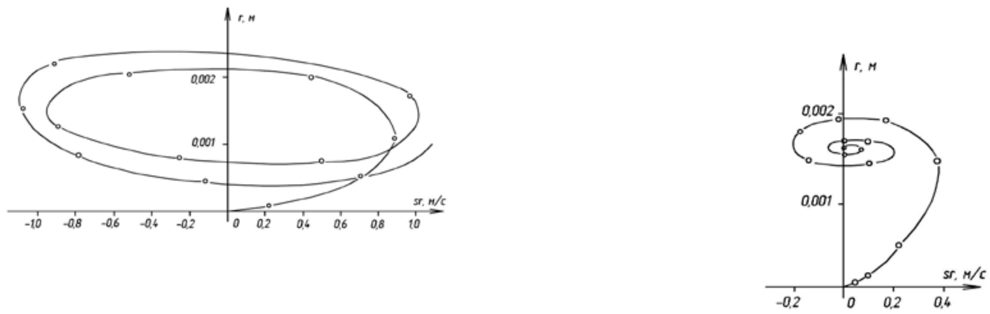
Built phase trajectory of shut-off and regulating element without (Figure 4 a) and taking into account the forces of friction (Figure 4 b) shows that in the first case there has been a limit cycle, while the second is a fading transition.



a) б)

Figure 4. Phase regulator shutter motion path excluding (a) and with (b) the force of friction bolt.

In the research process as changing initial settings were adopted: the stiffness of the spring, damping factor, consumption and the starting offset of the shutter. Relevant phase trajectories are presented in Figure 5, 6. Based on the results of the calculations are constructed diagrams Kenigsa- Lamereâ that can be used to judge the nature of transients in hydraulic circuit with pressure regulator (direct-acting relief valve with an elastic locking-regulating



element).

a) b)

Figure 5. Phase trajectories of the valve movement when changing damping coefficient of the regulator.

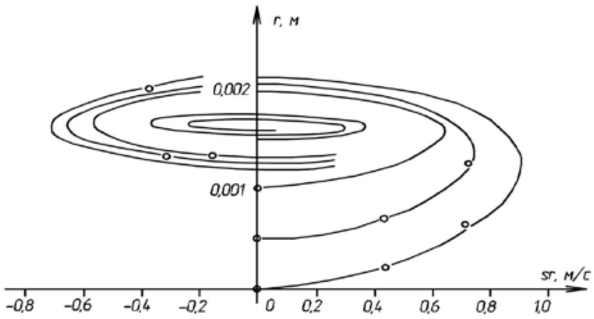


Figure 6. Phase trajectories of the regulator valve when changing the initial displacement of the r_0 valve.

4. Results

Dynamic characteristics are obtained to ensure the absence of self-oscillations due to the additional damping of the shut-off- and-regulating element. The prototype of the elastic pressure regulator with the shut-off-and-regulating element is constructed. Characteristics dependences of the pressure regulator (pressure relief valve with direct action) from the initial conditions are defined.

Figures 7, 9 present the diagrams, obtained by changing the initial values of the spring stiffness. The diagram first return function, is under the bisecting line, indicating the absence of limit cycles in the system and the presence of the damping transition process [5]

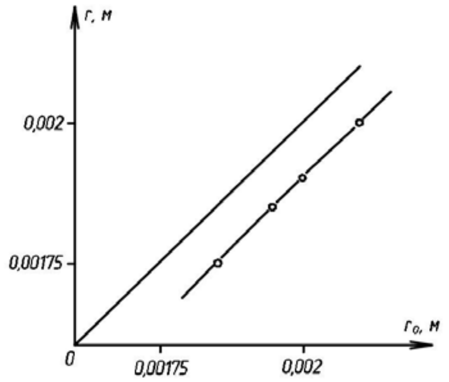


Figure 7. Kenigs-Lamereâ's diagram obtained by changing the spring stiffness.

Figures 8, 10 present the diagrams, obtained by changing the initial values of the damping coefficient and pressure. In both cases, the first return functions cross the bisector line; it shows that there is a limit cycle in the system, i.e. a damping transition process.

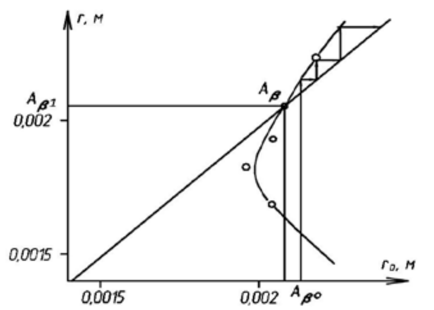


Figure 8. Kenigs-Lamereâ's diagram obtained by modification of the regulator damping coefficient β.

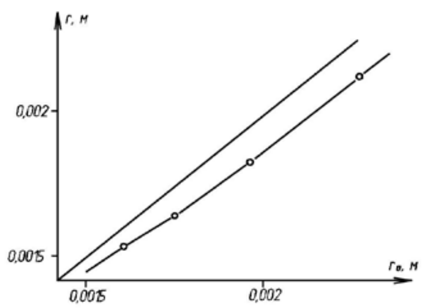


Figure 9. Kenigs-Lamereâ's diagram obtained by the modification of the input flow rate Q into the regulator.

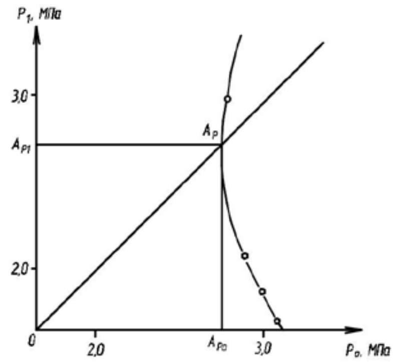


Figure 10. Kenigs-Lamereâ's diagram obtained by modification of the initial pressure p_0.

5. Discussion of results

Projecting points of intersection A_p an A_β on the relevant axes of self-excited oscillations amplitudes $A_{\beta 0}$ and $A_{\beta 1}$; Ap_0 and Ap_1 accordingly. The period of self-oscillations is determined by the phase trajectory in time of the single cycle of self-oscillations in a certain integration step.

After comparing the values of self-oscillations parameters of the researched system, which were obtained experimentally, (A_a = 1.8 MPa, f_a = 350 Hz) and in response to the numerical solution of the mathematical model (A_a = = 1.9MPa; f'_a = 330 Hz) we establish that the proposed mathematical model describes adequately the dynamic processes occurring in the hydraulic system with hydraulic regulator of the proposed design.

6. Summary and conclusions

The analysis of the results shows that the main parameters affecting the dynamic stability of the pressure regulator is the damping factor and the initial pressure. Finding the best values of these parameters was carried out through the analysis of transient characteristics of the valve when the impact on it of the standart disturbance on consumption. It has been established that valve mode is unstable in the case when the damping coefficient is < 40 Ns/m and the initial pressure is > 3 MPa. When you change other parameters limiting unstable modes were not observed. The obtained results can be used when designing hydraulics of the technological machines.

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