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ABOUT THE POSSIBILITY TO REDUCE HYDRAULIC LOSSES IN THE CONTROL SYSTEM OF HYDRAULIC DRIVE WITH VARIABLE–DISPLACEMENT PUMP

The paper presents the mathematical model of the automatic regulator for a variable-displacement pump. This regulator provides hybrid control means combining both pump flow rate stabilization and fixed-power mode of pump operation. There had been determined the hydraulic power losses in the suggested system. The possibilities of their reduction have been analyzed in order to eliminate high power losses in the hydraulic drive.

Keywords: *hydraulic drive, variable-displacement pump, pump regulator, flow rate regulator, pressure regulator, mathematical model, hydraulic efficiency, power, control pressure drop, stabilization error.*

Wide application of various means for automatic control of variable-displacement pumps enables to obtain the hydraulic mechanisms with different properties intended for servicing the corresponding technological processes. Designing such control systems requires to estimate power losses as well as to analyze the possibilities of their reduction in order to improve energy characteristics.

Energy characteristics of the control system depend on the power distribution in the control system in the process of operation and they can be evaluated by the hydraulic efficiency value. This work is aimed at the development of measures for reducing hydraulic losses in the control system of variable-displacement pump in order to prevent high power losses in the hydraulic drive.

Design model for control system of the hydraulic drive with a variable-displacement pump is shown in fig.1. The control system includes variable-volume pump with automatic regulator that provides pump flow rate stabilization as well as constant-power operation of the pump [1-4].

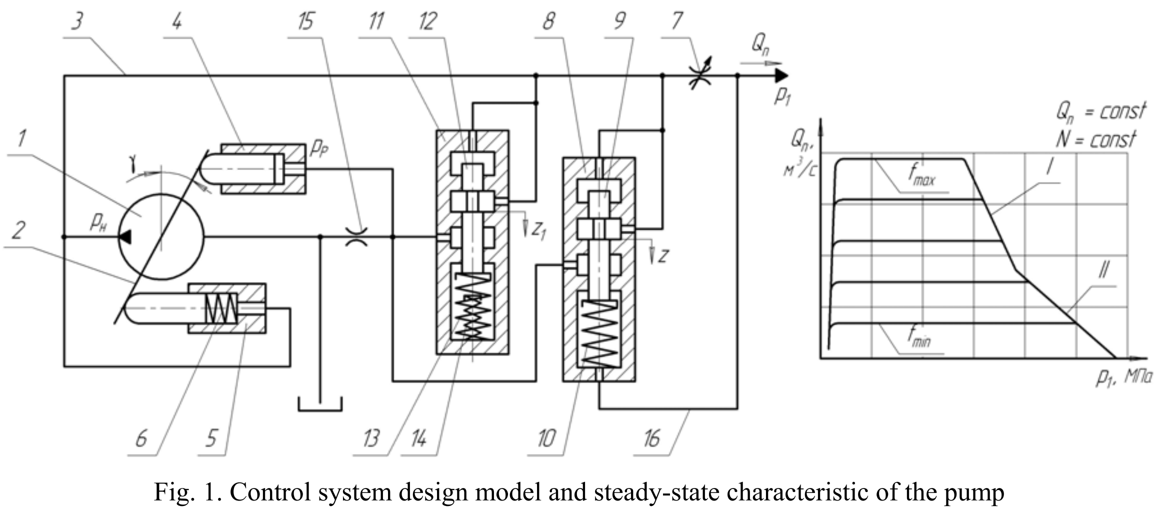

Fig. 1. Control system design model and steady-state characteristic of the pump

The circuit includes axial-piston variable-volume pump 1, variable orifice 7, flow rate regulator 8 with spool 9 and spring 10 that is connected to hydraulic lines 3 and 16 and controls the flow from hydraulic line 3 to control piston 4. Pump swashplate 2 is under the influence of pistons 4, 5 and spring 6. At the outlet of control piston 4 throttle 15 is installed. Pressure regulator 11 with spool 12 and springs 13, 14 limits maximal pressure value in the pressure line and provides constant-power mode of pump operation.

In a steady-state mode the control system operates in the following way. Pressures p_H i p_1 as well as the force of spring 10 act on spool 9 and determine its position so that the pressure p_P in piston 4 balances torque on the swashplate 2 created by pressure p_H in piston 5 together with spring 6. Spring 10 is selected so that it can maintain constant pressure drop at throttle 7. When pressure p_H grows considerably, pressure regulator 11 with spool 12 and springs 13, 14 operates so that pressure fluid flows to control piston 4 acting on the swashplate in the way that constant-power mode of pump operation is provided.

Steady-state characteristic of the pump with the proposed control system is represented by the relationship shown in fig.1, where $f_{\max}, f_{\min}$ – maximal and minimal areas of the opening of throttle 7 working port. The presence of two springs, 13 and 14, provides two sections on the steady-state characteristic, that approximate the total characteristic to a hyperbolic curve showing constant-power mode of the pump operation. Section I is provided by the operation of spring 13, and section II – by simultaneous operation of springs 13 and 14.

In accordance with the design model, the mathematical model of the variable-displacement pump control system includes equations of the forces and torques, acting on the regulating elements, and flow continuity equations for corresponding lines. Equations of the mathematical model were set up with the following assumptions: the length of hydraulic lines is small and therefore the influence of hydraulic losses and wave processes on the drive dynamics was considered to be insignificant; pressure fluid temperature is constant; compliance of the chambers and fluid compression were taken into account as averaged values for the given range of pressure change; flow coefficients of the throttle and spool elements are constant; control system operation modes are cavitation-free; hydrodynamic forces acting on the regulator spools as well as their masses were neglected. Taking into account the above-mentioned assumptions equations of the mathematical model of the pump control system are given by

$$\begin{aligned}\frac{\pi d_7^2}{4}\cdot d_8\cdot k\cdot n\cdot tg\gamma &= \mu\cdot f_{TH}\sqrt{\frac{2(p_H-p_1)}{\rho}}\cdot sign(p_H-p_1)+\\ &+\mu\cdot f(z)\sqrt{\frac{2(p_H-p_P)}{\rho}}\cdot sign(p_H-p_P)+\\ &+\mu\cdot f_1(z_1)\sqrt{\frac{2(p_H-p_P)}{\rho}}\cdot sign(p_H-p_P)+\beta_1\cdot W_H\frac{dp_H}{dt}\end{aligned} \tag{1}$$

$$I\frac{d^2\gamma}{dt^2}=p_H\cdot F_5\cdot\ell-p_P\cdot F_4\cdot\ell+M_C-b_\gamma\frac{d\gamma}{dt} \tag{2}$$

$$b_P\frac{dz}{dt}=p_H\cdot f_P-p_1\cdot f_P-c_P(z_P+z)-T_P\cdot sign\frac{dz}{dt} \tag{3}$$

$$b_{P1}\frac{dz_1}{dt}=p_H\cdot f_{P1}-c_{P1}(z_{P1}+z_1)-T_{P1}\cdot sign\frac{dz_1}{dt} \tag{4}$$

$$\begin{aligned}&\mu\cdot f(z)\sqrt{\frac{2(p_H-p_P)}{\rho}}\cdot sign(p_H-p_P)+\\ &+\mu\cdot f_1(z_1)\sqrt{\frac{2(p_H-p_P)}{\rho}}\cdot sign(p_H-p_P)=\mu\cdot f_0\sqrt{\frac{2p_P}{\rho}}+\beta_2\cdot W_P\frac{dp_P}{dt}\end{aligned} \tag{5}$$

where p_H is pump output pressure; γ – angle of the swashplate rotation; p_P – pressure in the piston for positional control of the pump swashplate; p_1 – load pressure on the actuator; F_4 – area of control piston 4; F_5 – area of the control piston 5; I – inertia moment of the pump swashplate; b_γ – kinematic viscosity coefficient of the pump swashplate; d_7 – diameter of the pump pistons; d_8 – diameter of the pistons location in the pump rotor; k – the number of pistons in the pump; n – speed of the pump shaft rotation; ℓ – arm of the control cylinders action on the pump swashplate; M_C – resistance moment of the pump swashplate that was calculated in N·m by the formula given below [2]:

$$M_C = 26{,}4 - 3{,}2\left(\frac{p_H - 9\cdot 10^6}{7\cdot 10^6}\right) - 9{,}9\left(\frac{Q_n - 0{,}67\cdot 10^{-3}}{0{,}5\cdot 10^{-3}}\right) - 2{,}2\left(\frac{p_H - 9\cdot 10^6}{7\cdot 10^6}\right)\times\left(\frac{Q_n - 0{,}67\cdot 10^{-3}}{0{,}5\cdot 10^{-3}}\right);$$

where Q_n – flow rate of the fluid supplied to the actuator; z – displacement of the spool 9 of the flow rate regulator 8; z_1 – displacement of the spool 12 of the pressure regulator 11; $f_P = \frac{\pi\cdot d_p^2}{4}$ – end surfaces area of the spool 9 of the flow rate regulator 8; d_p – end surface diameter of the spool 9 of the flow rate regulator 8; $f_{P1} = \frac{\pi\cdot d_{p1}^2}{4}$ – end surface area of the spool 12 of the pressure regulator 11; d_{p1} – end surface diameter of the spool 12 of the pressure regulator 11; c_P – strength of the spring 10 of the flow rate regulator 8; c_{P1} – total strength of springs 13 and 14 of pressure regulator 11; z_P, z_{P1} – initial compression values for the springs of the flow rate regulator 8 and pressure regulator 11 correspondingly; $f(z), f_1(z_1)$ – opening areas for the working ports of the flow rate regulator 8 and pressure regulator 11 correspondingly; b_P, b_{P1} – kinematic viscosity coefficients of the spools of the flow rate and pressure regulators correspondingly; T_P, T_{P1} – dry friction forces of the spools of the flow rate and pressure regulators correspondingly ; ρ – fluid density; μ – flow rate coefficient for the throttling elements; f_{TH} – opening area of the working port of the variable throttle 7; f_0 – opening area of the working port of the throttle 15 in the discharge line of the piston for controlling pump swashplate; W_H – fluid volume in the pressure hydraulic line section that is adjacent to the pump; W_P – pressure fluid volume in the control piston of the pump swashplate and in the hydraulic line section that is adjacent to it; β_1, β_2 – cumulative ratios of the working fluid compression by W_H і W_P volumes correspondingly.

Opening values for the working ports of the flow rate and pressure regulator spools correspondingly were approximated by the expressions

$$f(z) = 0{,}1\cdot 10^{-6}\ \text{м}^2 \text{ if } 0 \le z \le z_{\min},$$

$$f(z) = k_Z \cdot z \text{ if } z_{\min} \le z \le z_{\max};$$

$$f_1(z_1) = 0{,}1\cdot 10^{-6}\ \text{м}^2 \text{ if } 0 \le z_1 \le z_{1\min},$$

$$f_1(z_1) = k_{Z1} \cdot z_1 \text{ if } z_{1\min} \le z_1 \le z_{1\max}$$

where $z_{\min}, z_{1\min}$ – spool displacement values when throttling orifices of the flow rate and pressure regulators are opened; $z_{\max}, z_{1\max}$ – spool displacement values when throttling orifices of the flow rate and pressure regulators are closed; k_Z, k_{Z1} – opening area coefficients for the working ports of the flow rate regulator 8 and pressure regulator 11 correspondingly.

Efficiency of the power consumption in the suggested control system can be evaluated by the hydraulic efficiency value calculated by the formula:

$$\eta_G = \frac{N_U}{N_U + N_{1L} + N_{2L}},$$

where $N_U = p_1 \cdot \mu \cdot f_{TH}\sqrt{\frac{2(p_H - p_1)}{\rho}}$ – useful power determined by the pressure p_1 at the actuator input and flow $Q_n = \mu \cdot f_{TH}\sqrt{\frac{2(p_H - p_1)}{\rho}}$ through the variable throttle to the actuators; $N_{1L} = p_P \cdot \mu \cdot f_0\sqrt{\frac{2p_P}{\rho}}$ – power losses in the control system of the variable pump, that are determined by pressure p_P in the piston for positional control of the pump swashplate and of the flow $Q_Z = \mu \cdot f_0\sqrt{\frac{2p_P}{\rho}}$ through the throttle in discharge line of the control piston; $N_{2L} = p_H \cdot p_H \cdot K'_H$ – hydraulic power losses determined by the pump output pressure p_H and volumetric coefficient K'_H of the pump; $K'_H = 3{,}3 \cdot 10^{-12}$ sec·m^4/kg [2].

Equation set that describes behavior of the variable-displacement pump control system was researched using MATLAB Simulink software package.

Fig. 2 shows relationships of the efficiency of the variable pump control system for different control pressure drops $\Delta p = p_H - p_1$ across the variable throttle. In the course of research values of Q_n and Q_Z were found on the results of solving the equations of the non-linear mathematical model for initial conditions of the variables being close to steady-state conditions. As to energy characteristics, it is more efficient for the hydraulic drive to be operated under pressure p_1 below 20 MPa and high value of f_{TH}, that determines the value of flow Q_n supplied to the actuators. It should be noted that in order to prevent high power losses, it is desirable for the control pressure drop Δp across the variable throttle to be of the least possible value.

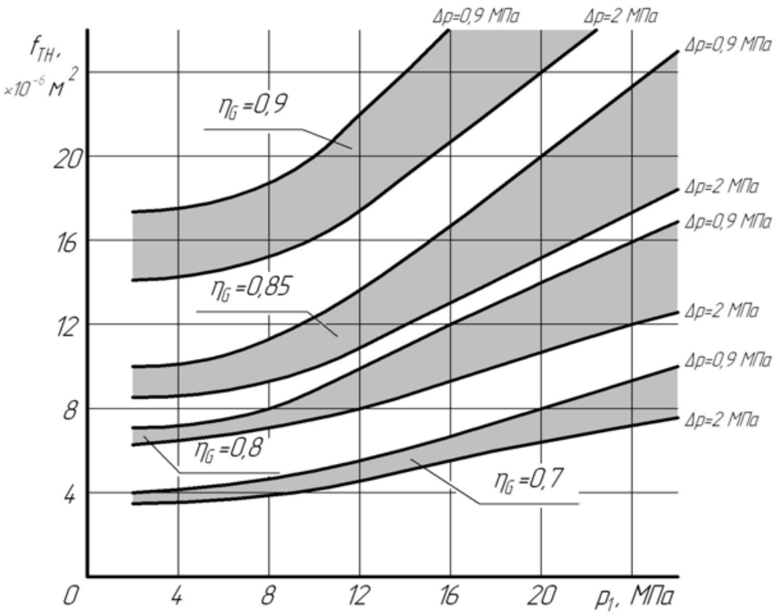

Fig. 2. The relationship between hydraulic efficiency and pressure drop across the variable throttle

Due to mathematical modeling of the operation processes in the proposed control system of the variable-displacement pump it was determined how the control pressure drop Δp influences the error δ of the flow supplied to the consumer (fig. 3). Error δ of the flow supplied to the consumer was estimated by the relationship $\delta = \dfrac{Q_{n\max} - Q_{n\min}}{Q_{n\min}} \cdot 100\%$, where $Q_{n\max}, Q_{n\min}$ are corresponding values of maximal and minimal flows supplied to the actuators during the flow rate regulator operation. Conditions under which certain technological processes are performed require that the value of δ does not exceed the pre-defined values, otherwise there can be errors in control signals, the precision of machine operation is reduced. The analysis of the obtained relationship (fig.3) shows that if the value of the control pressure drop Δp is reduced, the stabilization error δ increases. For example, when Δp decreases from 2,0 MPa to 0,5 MPa the value of δ increases from 1,8 to 6%.

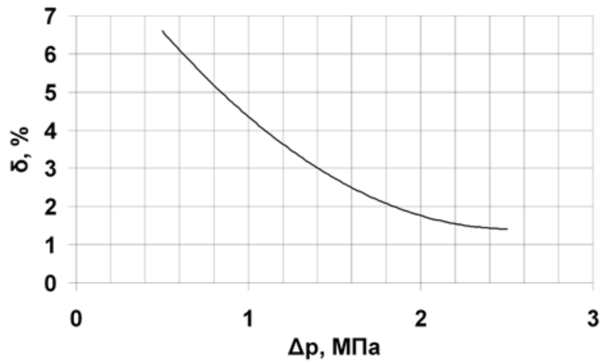

Fig. 3. The influence of the value of control pressure drop Δp across the variable throttle on the value of stabilization error δ

Therefore the reduction of the control pressure drop Δp results in the improved energy characteristics but at the same time in the considerable growth of the stabilization error δ, which can appear to be undesirable and to require the development of measures on the elimination of the above negative effect of the control pressure drop reduction.

One of the methods for stabilization error reduction is the selection of the appropriate regulator design parameters. Fig. 4 shows the influence of the spool diameter d_p, the opening area coefficient k_z of the working port, spring strength c_p of the flow rate regulator on the error stabilization value δ. Along the abscissa axis values of the non-dimensional parameters $\Pi = \Pi_a / \Pi_{\max}$ are plotted, where Π_a is actual current parameter value, $\Pi_{\max}$ – maximal value of the parameters. The maximal values of the parameters were equal to the upper limits of the following range of the parameter change: d_p = (4,0…10,0)·10^{-3} m; k_z = (0,5…4,0)·10^{-3} m; c_p = = (0,8…4,0)·10^{4} N/m. The initial values of the variables were as follows: d_p = 6,0·10^{-3} m; k_z = = 1,5·10^{-3} m; c_p = 1,5·10^{4} H/m. d_p has the most noticeable influence on the value of δ. For example, when d_p is growing from 5·10^{-3} m to 8·10^{-3} m, δ is reduced from 3,5 to 1,2%. The increase of k_z and c_p values results in the reduction of δ, but to a lesser degree than it happens when d_p is increased.

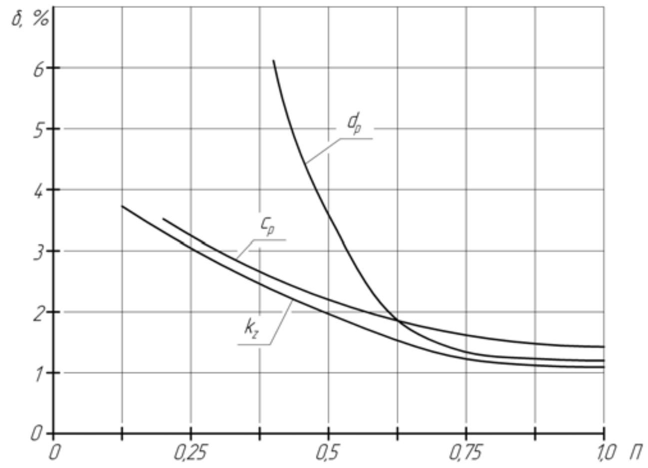

Fig. 4. Influence of the design parameters of the flow rate regulator on stabilization error value δ

The research on hydraulic losses in the control system of the hydraulic drive with variable-displacement pump makes it possible to draw the following conclusions:

1. Switching to reduced values of the control pressure drop across the variable throttle provides lower hydraulic pressure losses and therefore results in higher efficiency of the hydraulic drive. At the same time switching to a lower control pressure drop, from 2 to 0,5 MPa, is accompanied by the increased stabilization error δ in the value of flow supplied to the consumers (from 1,8 to 6%).

2. It was determined that this negative influence of the reduced control pressure drop Δp on the flow stabilization error can be compensated by appropriate selection of the design parameters for the flow rate regulator: spool diameter d_p, coefficient k_z of the working port opening area and strength c_p of the flow rate regulator spring.

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