DESIGN OPTIMIZATION OF THE FLOW RATE REGULATOR FOR AXIAL-PLUNGER LS PUMP USING LP SEARCH METHOD

Y. Burennikov

Vinnytsia National Technical University, e-mail: yu.burennikov@bk.ru.

Abstract: *This paper considers the procedure and results of optimization of the pump regulator for a powerful hydraulic system with a hydraulic cylinder. Such hydraulic systems have found wide application in concrete mixers which are increasingly used at present. The regulator design optimization is performed according to the non-linear mathematical model in MATLAB-Simulink software environment using the method of LP search in the space of four parameters of the pump control system. The optimal combination of the design parameters of pump regulators satisfies the requirements on ensuring operation stability of the pump with a regulator, regulation time not exceeding 0.22 s, overshoot that does not exceed 21% and stabilization error not higher than 4.4%.*

Keywords: *optimization, pump regulator, mathematical model, LP search.*

1. Introduction

During last decades development of the hydraulic drives of working machines for construction industry, agriculture and other branches is characterized by a strong tendency towards wide application of variable axial-piston pumps (APP) with flow regulators. This is explained by the desire of developers and consumers of hydraulic equipment to expand functional capabilities, to implement energy-saving modes and optimal utilization of the engine power during operation processes.

Leading companies of the USA, Western Europe and Japan are producing a wide range of variable APP that are characterized by a high technical level and could be equipped with various control systems. In Ukraine there is a production and technological potential as well as experience of manufacturing variable positive-displacement machines. However, the control system of variable APP, and especially its flow regulator, require improvement in terms of providing appropriate combination of static, dynamic and energy-saving characteristics that should correspond to world standards.

Research work on the development of variable APP is also carried out in Ukraine. In Vinnytsia National Technical University APP control system with a profiled port of the spool has been designed. It provides pump operation both in load-sensing and constant-power modes.

2. Mathematical model

Fig. 1 presents the developed circuit that includes variable APP 1, hydraulic cylinder 16, proportional directional control valve 7 with a device for tracing load pressure of the consumers (LS signal), flow regulator 8 with spool 9 and spring 10 that is connected to hydraulic lines 3, 15 and provides APP flow stabilization mode by controlling the flow from hydraulic line 3 to control cylinder 4. Faceplate 2 of the pump is under the influence of action of cylinders 4, 5 and spring 6. At the discharge of the control cylinder 4 throttle 14 is located. Power regulator 11 with spool 12 and spring 13 changes the pump displacement according to the pressure so that APP constant power mode is provided.

The control system operates in the following way. When consumers are disconnected, control line 15 of the regulator is communicated with the discharge, pressure in it is practically absent. Flow, created by the pump, shifts spool 9 to the right compressing spring 10 and flows to control cylinder 4. Pressure p_C, created in the cylinder, moves the faceplate so that the pump flow will be minimal and compensate only for leakage in the control system and hydraulic drive.

In the steady-state mode pressures p_H and p_1 as well as the force of spring 10, acting on spool 9, will determine such position of it that pressure p_C in cylinder 4 will balance the torque at faceplate 2, created by pressure p_H in cylinder 5 together with spring 6. Spring 10 is selected so that it could keep constant pressure differential $\Delta p_{LS}=p_H-p_1$ at the port of directional control valve 7. Any change in Δp_{LS} will cause corresponding change of the inclination angle γ directed towards the error reduction. E.g., when p_{LS} increases, spool 9 of flow regulator 8 will be shifted to the right, the flow to cylinder 4 will increase and angle γ will be reduced until Δp_{LS} reaches the predetermined value. Changing the area of working port f_{gp} by means of directional valve 7, it is possible to change flow Q_H to hydraulic cylinder 16.

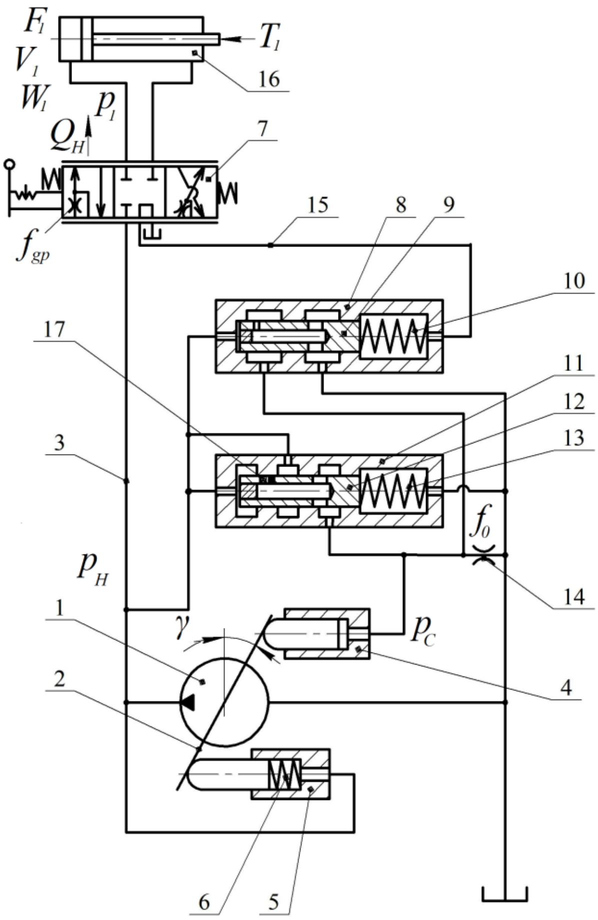

Figure 1: – *The control system circuit of variable APP with the combined flow regulator.*

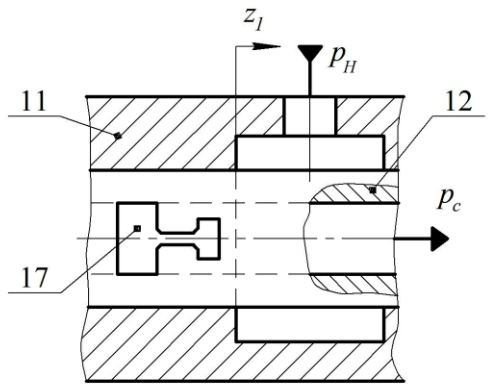

Figure 2: – *The profiled port of the combined flow regulator spool*.

Figure 3 presents a static characteristic of the variable APP in the flow rate regulation mode. Along with the possibility of proportional control of the flow rate Q_H, stabilization of the flow rate Q_H is provided when the change of pressure value p_1 occurs at the cylinder input depending on the value of technological value.

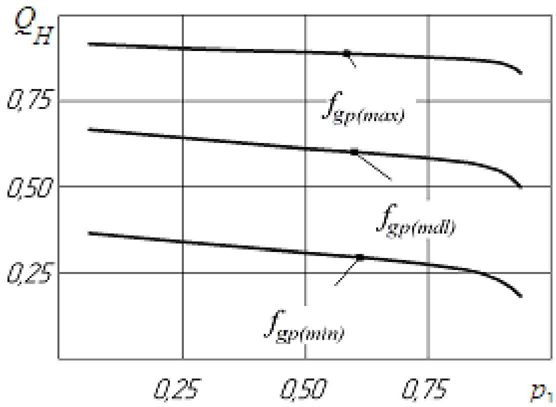

Figure 3: – *Static characteristic of the variable APP in the flow rate regulation mode.*

If the value of pressure p_H increases considerably, power regulator 11 will operate. The increased pressure p_H, that acts on the left end of spool 12, shifts it to the right, compresses spring 13 and opens the profiled working port of the spool (Fig. 2) that determines flow Q_C to control cylinder 4 depending on coordinate z_1 of the spool displacement. In control cylinder 4 pressure p_C is created, which determines inclination angle γ of faceplate 2 of the pump and changes the pump displacement, depending on the pressure, so that the constant product of the pressure and pump flow is provided, which, in its turn, ensures the driving power stability.

Static characteristic of APP in the constant power mode is presented in Fig. 4.

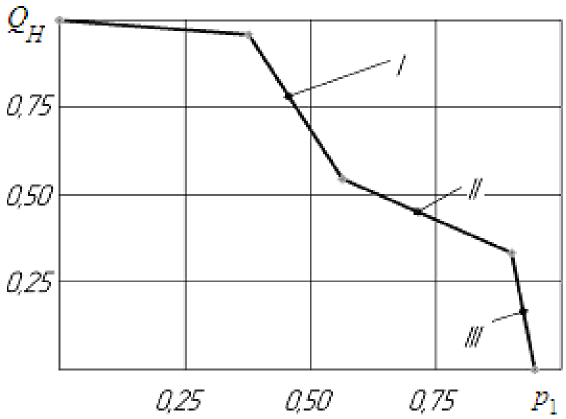

Figure 4: – *Static characteristic of the variable APP in the constant power mode.*

Presence of the profiled port in spool 12 of the power regulator 11 (Fig. 2) and corresponding variable gain coefficient k_{Z1} of the spool port depending on the displacement of spool z_1 provides two portions on the static characteristic which approximate the total static characteristic to a hyperbolic curve characterizing pump operation in a constant-power mode. Portion *I* is provided by the opening of the working port in the power regulator spool with the gain coefficient k_{Z11} for displacement of spool $z_{11} < z_1 \le z_{12}$. Portion II for z_1 displacement value bigger than that of z_{12} is provided by a simultaneous opening of the working port with the value $f_1(z_1) = (z_{12} - z_{11}) \cdot k_{Z11}$ and opening of the working port with the gain coefficient k_{Z12}. Limitation of the maximal pressure is realized through k_{Z12} with the displacement value of z_1 bigger than that of z_{13}.

Depending on the spool displacement coordinate z_1, the area of the profiled port of spool 12 is determined by the relationship

$$f_1(z_1) = \begin{cases} 0{,}1 \cdot 10^{-6}\, m^2 \ , \ 0 \le z_1 \le z_{11}; \\ k_{Z11}(z_1 - z_{11}), \ z_{11} < z_1 \le z_{12}; \\ k_{Z11}(z_{12} - z_{11}) + k_{Z12}(z_1 - z_{12}), \\ \ z_{12} < z_1 \le z_{13}; \\ k_{Z11}(z_{12} - z_{11}) + k_{Z12}(z_{13} - z_{12}) + \\ + k_{Z13}(z_1 - z_{13}), \ z_1 > z_{13}; \end{cases} \quad (1)$$

where the opening of the value of $0{,}1 \cdot 10^{-6}$ (for positive overlap with the profiled port) takes into account the working fluid overflow into the gap between the spool and the regulator body.

Mathematical model of the variable APP control system was elaborated using the following assumptions: parameters of the control system components are lumped; the volume of the control system hydraulic lines does not change during transient processes; the lengths of the control system hydraulic lines are comparatively small and so wave processes are not taken into account; the coefficients of the flow rate through the throttling and spool elements are constant; the mode of the working fluid flow in the control system is cavitation-free; pressure losses in the control system hydraulic lines are not taken into account as they are not considerable as compared with those at local resistances; hydrodynamic forces at the flow regulator spools are not taken into account because flows through the working ports of the spools are inconsiderable; compliance coefficient of the working fluid and the hydraulic lines is taken into account as the pressure-depending value (Equation 8).

Mathematical model of the variable APP control system includes flow continuity equations (2), (6), equations of the torques acting on the faceplate of the variable APP (3), equation of forces acting on the spools of the complex flow regulator (4-5), equation (7) of the dependence of the working port in spool 9 and equation (1) of the dependence of the area of profiled working port in spool 12 as well as the refined dependences of the resistance moment of the pump faceplate (9) [3] and flow rate through the profiled working port of spool (10).

$$\begin{aligned} &\frac{\pi d_7^2}{4} \cdot d_8 \cdot k_1 \cdot n \cdot tg\gamma = \\ &= \mu \cdot f_{gp} \cdot \sqrt{\frac{2|p_H - p_1|}{\rho}} \cdot sign(p_H - p_1) + \\ &+ \mu \cdot f(z) \cdot \sqrt{\frac{2|p_H - p_C|}{\rho}} \cdot sign(p_H - p_C) + \\ &+ Q_C + \beta(p) \cdot W_H \cdot \frac{dp_H}{dt} \end{aligned} \quad (2)$$

$$\begin{aligned} I \frac{d^2\gamma}{dt^2} &= p_H \cdot F_5 \cdot \ell - p_C \cdot F_4 \cdot \ell + \\ &+ M_C - b_\gamma \frac{d\gamma}{dt} \end{aligned} \quad (3)$$

$$\begin{aligned} m_p \frac{d^2 z}{dt^2} &= p_H \frac{\pi d_p^2}{4} - p_1 \frac{\pi d_p^2}{4} - c_p(z_p + z) - \\ &- b_p \frac{dz}{dt} - T_p \cdot sign \frac{dz}{dt} \end{aligned} \quad (4)$$

$$m_{p1}\frac{d^2z_1}{dt^2}=p_H\frac{\pi d_{p1}^2}{4}-c_{p1}\left(z_{p1}+z_1\right)-b_{p1}\frac{dz_1}{dt}-\\-T_{p1}\cdot sign\frac{dz_1}{dt}\qquad(5)$$

$$\mu\cdot f(z)\cdot\sqrt{\frac{2\left|p_H-p_C\right|}{\rho}}\cdot sign\left(p_H-p_C\right)+Q_C=\\=\mu\cdot f_0\cdot\sqrt{\frac{2p_C}{\rho}}+\beta(p)\cdot W_C\cdot\frac{dp_C}{dt}\qquad(6)$$

$$f(z)=\begin{cases}0{,}1\cdot10^{-6}\,m^2, & 0\le z\le(1{,}0\cdot10^{-3})\,m,\\ k_Z\cdot z, & (1{,}0\cdot10^{-3})<z\le(10\cdot10^{-3})\,m;\end{cases}\qquad(7)$$

$$\beta(p)=\begin{cases}2\cdot10^{-9}\left(1-0.03\frac{p_n}{10^5}\right) & p_n\le23\cdot10^5\\ 0{,}6\cdot10^{-9} & 23\cdot10^5<p_n\le300\cdot10^5.\end{cases}\qquad(8)$$

$$M_C=28{,}39-7{,}42\frac{p_H-7\cdot10^6}{3\cdot10^6}-\\-7{,}64\frac{Q_H-0{,}417\cdot10^{-3}}{0{,}25\cdot10^{-3}}-2{,}99\frac{t^0-40}{20}-\\-6{,}64\left(\frac{p_H-7\cdot10^6}{3\cdot10^6}\right)\left(\frac{Q_H-0{,}417\cdot10^{-3}}{0{,}25\cdot10^{-3}}\right);\qquad(9)$$

$$Q_C=0{,}61\cdot10^{-3}+0{,}23\cdot10^{-3}\left(\frac{p_H-11\cdot10^6}{5\cdot10^6}\right)-\qquad(10)\\-0{,}10\cdot10^{-3}\left(\frac{p_C-3\cdot10^6}{2\cdot10^6}\right)+\\+0{,}56\cdot10^{-3}\left(\frac{z_1-1{,}6\cdot10^{-3}}{0{,}6\cdot10^{-3}}\right)+\\+0{,}21\cdot10^{-3}\left(\frac{p_H-11\cdot10^6}{5\cdot10^6}\right)\left(\frac{z_1-1{,}6\cdot10^{-3}}{0{,}6\cdot10^{-3}}\right)+\\+0{,}10\cdot10^{-3}\left(\frac{p_C-3\cdot10^6}{2\cdot10^6}\right)\left(\frac{z_1-1{,}6\cdot10^{-3}}{0{,}6\cdot10^{-3}}\right);$$

$$\mu f_{gp}\sqrt{\frac{2\left|p_H-p_1\right|}{\rho}}=\beta(p)W_1\frac{dp_1}{dt}+v_1F_1;\qquad(11)$$

$$m_1\frac{dv_1}{dt}=p_1F_1-T_1-b_1v_1;\qquad(12)$$

In equations (1–12) the following designations are used: p_H – pump output pressure; p_C – pressure in cylinder 4 for controlling position of the pump faceplate; p_1 – load pressure; Q_H – flow to the actuator; Q_C – flow rate through the profiled working port of the spool 2; γ- rotation angle of the pump faceplate; z, z_1 – displacement coordinates of the spools; areas of the working ports: f_{gp} – of the proportional directional valve 7; f_0 – of the throttle 14 in the discharge hydraulic line; $f(z)$ – of the spool 9; $f(z_1)$ – of the spool 12 with a profiled port; k_Z – gain coefficient of the port of spool 9; k_{Z11}, k_{Z12}, k_{Z13} – gain coefficients of the profiled port of spool 12; F_1, F_4, F_5 – areas of the hydro cylinder and control cylinders; I – inertia moment of the pump faceplate; d_7 – diameter of the pump pistons; d_8 – diameter of location of the pistons in the pump rotor; k_1 – the number of pump pistons; v_1 – speed of the hydro cylinder; n – speed of the pump shaft rotation; ℓ – the arm of control cylinders action on the pump faceplate; M_C – resistance moment of the pump faceplate; m_p, m_{p1}, m_1 – masses of the spools and hydro cylinder; d_p, d_{p1} –diameters of the spools; c_p, c_{p1} – stiffness of the spool springs; z_p, z_{p1} – initial compression of the spool springs; T_p, T_{p1} – dry friction forces of the spools; b_p, b_{p1}, b_y, b_1 – damping coefficients of the spools the pump faceplate and hydro cylinder; $\beta(p)$ – coefficient that takes into account total deformation of the working fluid and hydraulic lines; W_H, W_C – volumes of hydraulic lines in the corresponding parts of the control system; ρ – working fluid density; μ – flow rate coefficient; t^0 – temperature of the working fluid; T_1 – force of the hydro cylinder.

3. Optimization

Equations of the mathematical model were solved using MatLab Simulink software package.

While studying the mathematical model, transient processes of the variables describing the control system state have been obtained: pressures p_H and p_C, the pump faceplate rotation angle γ and displacement coordinates z, z_1 of the spools. According to the form of transient processes, obtained by means of mathematical modeling, stability was investigated and quality indicators of the transient process of the system state variables were determined: regulation time t_p and overshoot σ according to pressure p_H. The mathematical model also makes it possible to determine stabilization error A of the value of flow rate supplied to the hydraulic cylinder from the variable pump.

Optimization criterion of the hydraulic drive with a variable pump has been elaborated. It includes such indicators as stability, flow rate stabilization error A $\leq$ 5%, regulation time t_p<0.31, overshoot $\sigma \leq 30\%$.

As optimization parameters the following design parameters of the control system are considered: d_p – diameter of the regulator spool, c_p, c_{p1} – stiffness of the regulator spool, f_0 –pump regulator throttle area.

The optimization is conducted in both hydraulic drive operation modes using LP search method [3] for the following ranges of optimization parameters variations:

$d_p=(5…8)\cdot10^{-3}$m, $c_{p1}=(1,0…3,5)\cdot10^4$ N/m,

$c_p=(1,0…2,5)\cdot10^4$ N/m, $f_0=(0,8…3,0)\cdot10^{-6}$ m^2.

The optimization results are presented in the table.

Table: Optimization results

Optimization parameters			Optimization criterion indicators				№ of the test
Flow rate regulation mode							
d_p	c_p	f_0	Stabi-lity	δ, %	t_p, c	σ, %	10
$8\cdot10^{-3}$	$1,0\cdot10^4$	$1,2\cdot10^{-6}$	+	4,4	0,22	21	
Constant power mode							
d_p	c_{p1}	f_0	Stabi-lity	δ, %	t_p, c	σ, %	25
$8\cdot10^{-3}$	$5,3\cdot10^4$	$1,2\cdot10^{-6}$	+	–	0,20	18	

4. Conclusion

For the developed variable pump control system such combination of the design parameters has been determined, which provides APP characteristics corresponding to the elaborated optimization criterion. For the spool of flow rate regulator the following values of parameters are recommended:

$d_p=8\cdot10^{-3}$ m, $c_p=10\cdot10^4$ N/m, $f_0=1,2\cdot10^{-6}$ m^2.

For the constant power spool $d_p=8\cdot10^{-3}$ m, $c_p=5/3\cdot10^4$ N/m, $f_0=1,2\cdot10^{-6}$ m^2.

With such combination of the design parameters of pump regulators the requirements on ensuring operation stability of the pump with a regulator are satisfied with regulation time not exceeding 0.22 s, overshoot that does not exceed 21% and stabilization error not higher than 4.4%.

References

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