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OPTIMIZATION OF THE DESIGN PARAMETERS OF MECHATRONIC HYDRAULIC DRIVE WITH A VARIABLE-DISPLACEMENT PUMP

Introduction. Mobile working machines with hydraulic drives, based on variable-displacement pumps, are widely used in construction and municipal engineering, transport and agriculture. This enables wide-range regulation of the working members motion parameters, providing high efficiency values [1, 2].

Leading manufacturers of variable-displacement pumps equip them with modern electrohydraulically-controlled regulators. Wide use of controllers in hydraulic drives of mobile working machines with variable-displacement pumps makes it possible to provide significant improvement of the hydraulic drive characteristics, to create conditions for the machine working cycle automation as well as better working conditions for the machine operator [1, 3].

Aim. In order to ensure high-quality operations, performed by mobile working machines, hydraulic drives should have certain static, dynamic and power characteristics, which essentially depend on their structure and design parameters.

As a rule, changes of certain design parameters lead to improvement of definite characteristics and, at the same time, to worsening of others. Therefore, selection of the hydraulic drive design parameters is a rather complicated procedure that should be performed with the application of optimization approaches.

This work aims at the selection of optimal values of the hydraulic drive design parameters on the basis of optimization criterion that includes several operation quality indicators.

Research. In Vinnytsia National Technical University a mechatronic hydraulic drive circuit, based on a variable-displacement pump and a controller with analogs inputs and outputs, has been developed. The circuit is presented in Figure 1.

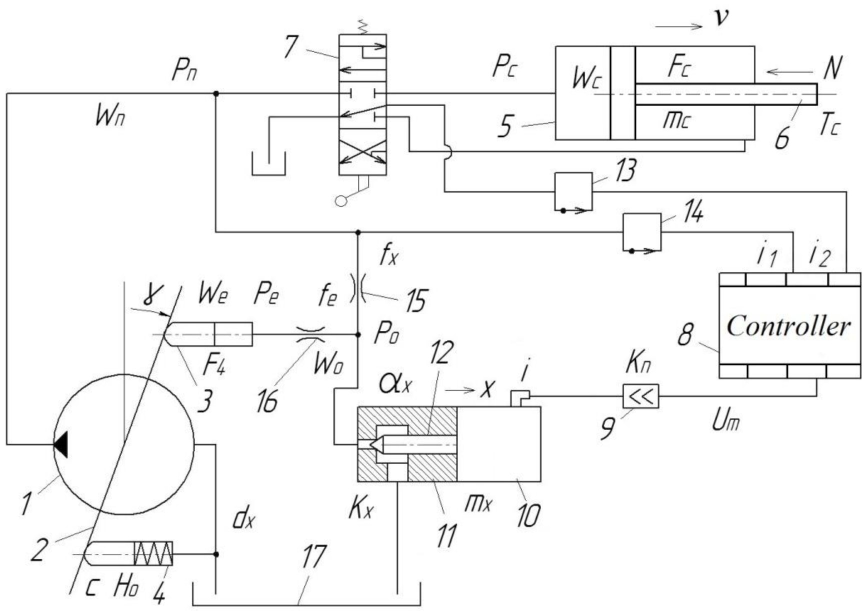

Figure 1 – Circuit of the mechatronic hydraulic drive

The circuit includes pump 1 with faceplate 2, servocylinder 3 and spring 4. Pump 1 actuates piston 6 of hydraulic cylinder 5, which is under the action of load N. Start-up and slowdown of the hydraulic cylinder is provided by directional control valve 7. Controller 8 receives signals i_1 and i_2 from pressure sensors 14, 13 and, according to a special algorithm, generates control signal U_m supplied to electromagnet 10 and regulator 11 through amplifier 9. In accordance with control signal, servovalve 12 generates the value of pressure p_o, which will keep power P_n, supplied by variable pump 1, at a constant level irrespective of speed v of hydraulic cylinder piston 6 or load N acting on hydraulic cylinder 5. Maintaining constant value of power P_n, supplied by pump1 to the hydraulic cylinder, makes it possible to use full capacity of the internal combustion engine of the machine, which provides mechatronic hydraulic drive operation. Application of the controller in the developed circuit enables pump control algorithms formation, taking into account characteristics of the mechatronic hydraulic drive.

While designing mechatronic hydraulic drives for mobile working machines, definite static, dynamic and power characteristics must be provided. At the design stage characteristics could be estimated on the basis of investigation of working processes, occurring in hydraulic drives, with the application of mathematical models.

Mathematical modelling results. Mathematical model of the mechatronic hydraulic drive includes equations of the moments acting on faceplate 2 of pump 1 (Equation 1), equation of the forces acting on piston 6 of hydraulic cylinder 5 (Equation 2) and on servovalve 12 (Equation 3), equation of the voltage drop in the circuit of electromagnet 10 (Equation 4) as well as flow continuity equation for the hydraulic lines between pump 1, directional control valve 7 and throttle 15 (Equation 5), between throttle 15, damper 16 and servovalve 11 (Equation 6), between directional control valve 7 and hydraulic cylinder 5 (Equation 7), between damper 16 and servocylinder 3 (Equation 8).

The mathematical model is developed under the following assumptions and simplifications. The hydraulic drive lump parameters are considered; temperature of the working fluid remains unchanged during transient process; wave processes in the pipelines are not taken into account; flow rates through the throttle and valve members are constant; the hydraulic drive operation mode is cavitation-free; volumes of the hydraulic cylinder and servocylinder chambers remain unchanged during transient process; pressure losses in the hydraulic lines are not taken into account with the exception of pressure hydraulic line that includes the directional control valve; dry friction force in the hydraulic cylinder does not depend on the piston motion speed; the amplifier operation is simulated by the proportional element.

$$I\frac{d^2\gamma}{dt^2} = c(H_o + l\cdot\sin\gamma)\cdot l - p_e F_4 l - \left(\frac{\pi\cdot\rho\cdot\nu\cdot d_4\cdot l_4}{\varepsilon_o} + \frac{\pi\cdot\rho\cdot\nu\cdot d_5\cdot l_5}{\varepsilon_o}\right)\cdot l^2\cos\gamma\cdot\frac{d\gamma}{dt}; \quad (1)$$

$$m_c\frac{d\nu}{dt} = p_c F_c - N - \pi D_c(q_o + k_q p_c)\cdot sign(\nu); \quad (2)$$

$$m_x\frac{d^2x}{dt^2} = p_o\frac{\pi d_x^{\,2}}{4} - k_e\cdot i - \left(\frac{\pi\cdot\rho\cdot\nu\cdot d_x\cdot l_x}{\varepsilon_x}\right)\frac{dx}{dt}; \quad (3)$$

$$\left(A_u + B_u\cdot p_n + \frac{C_u}{p_c}\right)\cdot k_u = R_e\cdot i + L_e\frac{di}{dt}; \quad (4)$$

$$d_7\cdot d_8\cdot k_1\cdot n_n\cdot tg\gamma - k_n p_n = a + bh + c(p_n - p_c) + dh^2 + e(p_n - p_c)^2 +$$

$$+ f(p_n - p_c)\cdot h + \mu f_x\sqrt{\frac{2|p_n - p_o|}{\rho}} + \beta W_n\frac{dp_n}{dt}; \quad (5)$$

$$\mu f_x \sqrt{\frac{2|p_n - p_o|}{\rho}} = \mu \cdot \pi \cdot d_x \cdot x \cdot \sin\frac{\alpha_x}{2} \cdot \sqrt{\frac{2p_x}{\rho}} + \beta W_o \frac{dp_o}{dt}; \tag{6}$$

$$a + bh + c(p_n - p_c) + dh^2 + e(p_n - p_c)^2 + f(p_n - p_c)h = F_c \cdot v + \beta W_c \frac{dp_c}{dt}; \tag{7}$$

$$\mu f_e \sqrt{\frac{2|p_o - p_e|}{\rho}} \cdot sign(p_o - p_e) = \beta W_e \frac{dp_e}{dt} - F_4 \frac{d\gamma}{dt} l - A_e p_e + B_e t^o + C_e \varepsilon_o{}^2 \tag{8}$$

In the mathematical model the following notation is used: p_n, p_c, p_o, p_e – pressures at the output of pump 1, at the input of hydraulic cylinder 5, at the input of servovalve 12, at the input of servocylinder 3; x – servovalve 12 position coordinate; γ – rotation angle of faceplate 2; i – value of the current in electromagnet 10; v – motion speed of piston 6; I – inertia moment of faceplate 2; c – stiffness of spring 4; H_o – pre-compression of spring 4; l, l_4, l_5 – arms of action of cylinders 3 and 4; l_x – length of contact of servocylinder 3 with the body; F_4, F_5, F_c – areas of pistons of servocylinders 3 and 4, area of the piston of hydraulic cylinder 5; ρ – density of the working fluid; ν – kinematic viscosity coefficient; β – working fluid compression ratio; μ – flow coefficient of the throttling members; d_4, d_x, D_c – diameters of servocylinder 3, seats of servovalve 12 and of the hydraulic cylinder piston; m_c, m_x – the working mechanism mass reduced to piston 6, mass of servovalve 12; N – the working mechanism force reduced to piston 6; T_c – friction force in hydraulic cylinder 5; ε_o, ε_x – diametric clearances between the servocylinder, servovalve and their bodies; A, B, C – coefficients in the formula of the controller output signal dependence on the input signals; a, b, c, d, e, f – coefficients in the formula, expressing dependence of the flow rate of directional valve 7 on opening h and pressure drop p_n - p_c; A_e, B_e, C_e – coefficients in the formula, expressing dependences of leakage from the chamber of servocylinder 3; k_n, k_u, k_e, k_g, k_q – coefficients of the leakage in pump, 1 of amplifier 9, electromagnet 10, sensors 13, 14, friction force in the hydraulic cylinder; R_e, L_e – active resistance and induction of the coil of electromagnet 10; Q_n – flow rate of pump 1; f_x, f_e – areas of throttle 15 and damper 16; W_n, W_o, W_c, W_e – volumes of hydraulic lines between pump1 and directional control valve 7, between throttle 15 and servovalve 12, between directional control valve 7 and hydraulic cylinder 5, between damper 16 and servocylinder 3; α_x – inclination angle of the edge of servovalve 12; i_1, i_2 – values of the currents at the outputs of pressure sensors 14 and 13 ; d_7 – diameter of the circle contact of the pump pistons with the faceplate; d_8 – diameter of the variable pump pistons; k_1 – number of the variable pump pistons; n_n – rotation speed of the pump driving shaft; q_0 – specific friction force in the hydraulic cylinder piston seal.

Controller 8 generates control algorithm for regulator 11 so that the pump operates in a constant-power mode, $P_n=Q_n \cdot p_n=Const$. This enables full use of the machine drive engine capabilities. To provide the hydraulic drive operation in the P_n=const mode, the controller must generate dependence $U_m=A_u+B_u \cdot p_n \cdot k_g+C_u/(p_c \cdot k_g)$, where A_u= 2,66·10^{-2}, B_u= -5,77·10^{-2}, C_u= -2,907·10^{-5}. Dependence of U_m on the values of pressures p_n and p_c is generated taking into account transfer ratio of pressure sensors 13, 14 as well as data of the basic dependence $Q=f(N)$.

Flowchart of operation algorithm of controller 8, realized in MATLAB Simulink environment, is presented in Figure 2. In the flowchart the following modules are used: Quantizer modules, which provide quantization of the input signals $i_1=k_g \cdot p_n$ and $i_2=k_g \cdot p_c$ with the interval of 0.01 s, Uniform Encoder modules for encoding signals in the controller with resolution of 8 bit and Uniform Decoder modules for decoding the controller output signals with resolution of 8 bit.

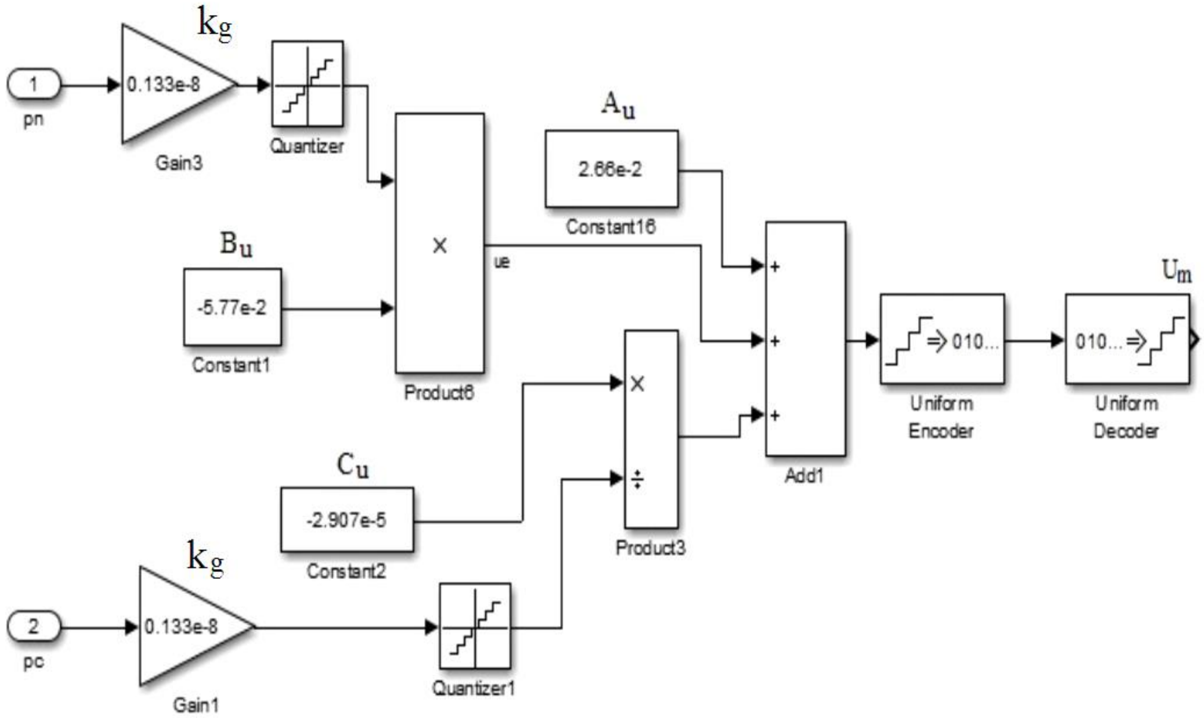

Figure 2 – Flowchart of the controller mathematical model

Mathematical model of the mechatronic hydraulic drive was generated in MATLAB Simulink environment and processed, using Rosenbrock numerical method, with absolute accuracy of $\varepsilon_a=10^{-6}$ and relative accuracy of $\varepsilon_b=10^{-3}$. The mathematical model parameters were as follows: m_c=1000 kg, $W_n=1\cdot10^{-3}$ m^3, $W_c=4\cdot10^{-3}$ m^3, $\beta=0.6\cdot10^{-9}$ m^2/H, μ=0.67, ρ=900 kg / m^3, $F_n=4.2\cdot10^{-4}$ m^2, $F_c=25\cdot10^{-4}$ m^2, $l=64\cdot10^{-3}$ m, I=0.02 kg·m^2, $d_x=1.6\cdot10^{-3}$ $^{\text{m}}$, $W_e=0.02\cdot10^{-3}$ m^3, $W_o=0.02\cdot10^{-3}$ m^3.

Simulation research of the working processes in the hydraulic drive was conducted for stepwise change of load N at hydraulic cylinder 5. Time variation of pressure values p_c, p_n, and p_o in the hydraulic drive were calculated for load, changing from 0 to N=30000 N, as well as digital signal U_m (Figure 3). Transient process simulation confirms operability of the developed circuit and of the control algorithm as well as makes it possible to determine the hydraulic drive characteristics.

The calculated transient processes, occurring in the hydraulic drive, make it possible to determine dynamic characteristics of the hydraulic drive: regulation time t_p, overshoot value σ and their dependence on the change of design parameters.

Transient processes, occurring in the hydraulic drive under dynamic modes, are of oscillatory nature in all operation modes. The highest values of regulation time t_p and overshoot value σ are observed under maximal values of load N.

The mechatronic hydraulic drive should satisfy the following requiements:

- regulation time t_p<1,2 s;
- overshoot σ <70 %;
- power losses in the variable-displacement pump control system P_y<1,5 KW.

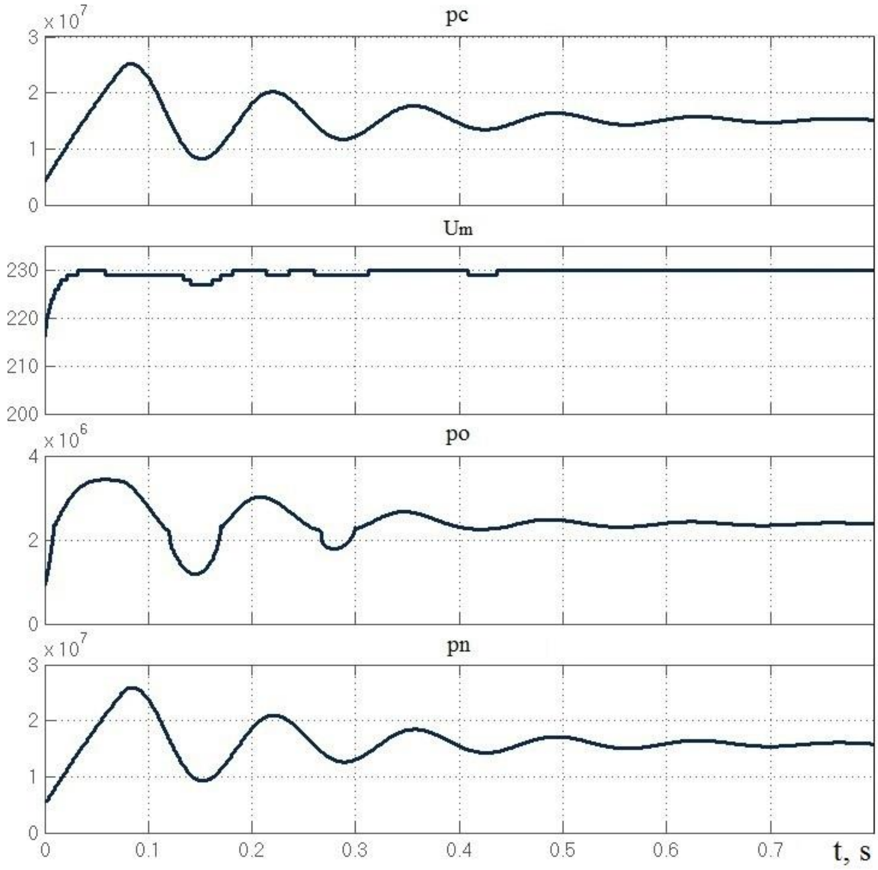

Figure 3 – Transient process in the mechatronic hydraulic drive under stepwise change of the load

Static characteristic of the mechatronic hydraulic drive, obtained as a result of research, is presented in Figure 4.

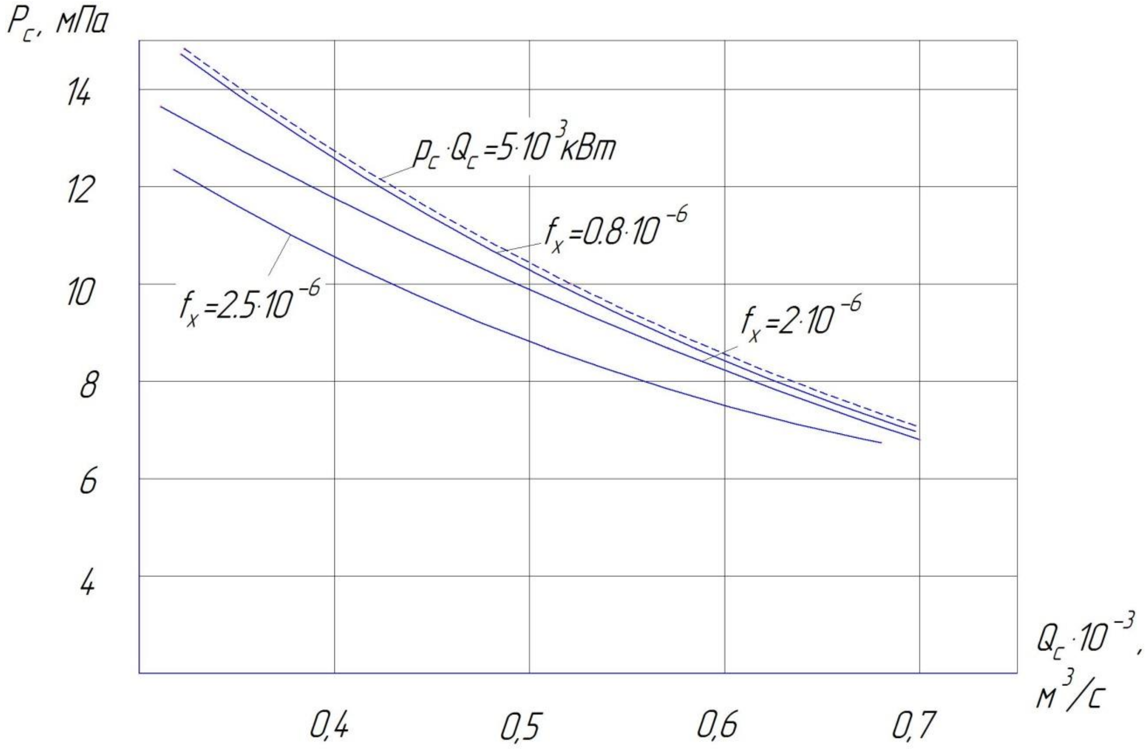

Figure 4 – Static characteristic of the mechatronic hydraulic drive

The characteristic is of the form of hyperbole with dependence $p_c \cdot Q_c = P_c$ being provided.

Using the developed mathematical model, the dependence of the pump regulator design parameters on the accuracy of maintaining hyperbolic dependence $p_c \cdot Q_c = P_c$ was investigated. The accuracy of maintaining hyperbolic dependence $p_c \cdot Q_c = P_c$ was estimated by the deviation of dependence $A_p = \frac{P_{max} - P_{min}}{P_{min}} \cdot 100\%$ while design parameters were changed in certain ranges.

Influence of the following parameters on the value of A_p was investigated:

- K_x – gain coefficient of the working port of servovalve 12;
- f_x – area of throttle 15;
- f_e – area of damper 16.

Design parameters were changed within the following ranges:

$$K_x = (1 \ldots 6{,}0) \cdot 10^{-3} \text{ m};$$
$$f_x = (0{,}8 \ldots 2{,}5) \cdot 10^{-6} \text{ m}^2;$$
$$f_e = (1 \ldots 4{,}0) \cdot 10^{-6} \text{ m}^2.$$

The nature of the effect of throttle area f_x is shown in Figure 4. Maximal value of deviation A_p, when f_x is changed from $0{,}8 \cdot 10^{-6}$ to $2{,}5 \cdot 10^{-6}$, is A_p=16%.

When the value of K_x is changed in the above-mentioned range, the value of deviation A_p does not exceed 3,5%, while for the change of f_e the deviation is A_p<1,6%.

Conclusion. If at the design stage it is necessary to meet strict requirements to the accuracy of static characteristic, the design parameters should be chosen taking into consideration their influence on the value of deviation A_p. The value of f_x should be selected from smaller values of the range. The values of f_e and K_x do not have any significant influence on the value of A_p.

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