Optimization of Design Parameters of the Counterbalance Valve for the Front-End Loader Hydraulic Drive

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1. Introduction

Most of the front-end loaders, used in Ukraine, are purchased from the manufacturers of CIS countries. These operational mobile machines are equipped with hydraulic drives, that have numerous drawbacks: dependence of hydraulic motors speed on the load, low efficiency factor, lack of proportional control [1, 2]. Such manufacturers as Caterpillar, Volvo, HydraForce, Case, Komatsu, Hidromek equip their machined with variable pumps with LS control, proportional hydraulic control valves, feed regulators and counterbalance valves [3-9]. Cost of such machinery is far higher but they are more productive and economic efficient.

To improve the front-end loaders of domestic manufacture researchers from Vinnytsia National Technical University developed hydraulic drive circuit, based on variable pump and counterbalance valve [10]. Application of the variable pump and the counterbalance valve provides load fixation in a certain position, proportionality of its motion speed control and energy saving as a result of load lowering under its own weight.

To provide the design of counterbalance valve with the required characteristics mathematical modeling is applied and optimization of design parameters.

2. Construction of calculation scheme of front-end loader boom hydraulic drive

Fig. 1 shows calculation scheme of front-end loader boom hydraulic drive. Calculation scheme is based on the construction of the boom of domestic manufacture front-end loader "БОРЕКС" model 2206. The loader is equipped with H-shape boom with two hydraulic cylinders for lifting/lowering of the boom.

Calculation scheme includes variable pump 1 with feed regulator 2 and nonadjustable throttle 3. Adjustable throttle 4 establishes the flow Q_{np} of working fluid to hydraulic cylinders 5, 6, that move boom 7 of front-end loader. Also the scheme comprises counterbalance valve 8 with nonadjustable throttle 9, main spool 10, piston 11 and spring 12. Hydraulic tank is shown under position 13.

Hydraulic drive operates in the following way. Working fluid from variable pump 1 is supplied across the working window f of adjustable throttle 4 and open main

spool 10 to piston chambers of hydraulic cylinders 5, 6. Motion of rods and lifting of boom 7 of front-end loader takes place. Fluid drain is realized from rod cambers of hydraulic cylinders 5,6 in hydraulic tank 13.
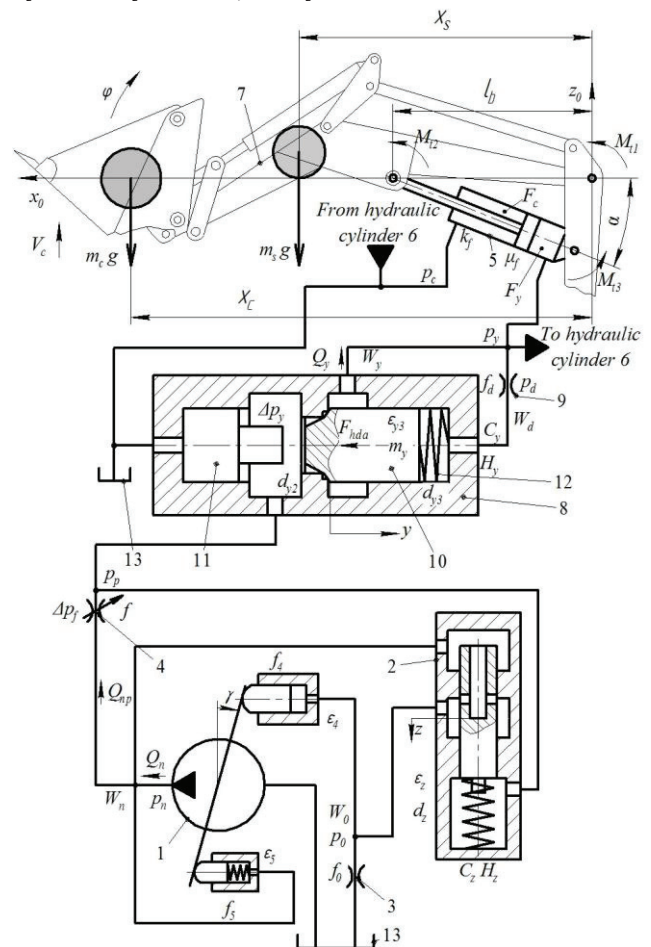

Fig. 1 Calculation scheme of front-end loader boom hydraulic drive

Pump regulator 2 determines pressure drop Δp_f on adjustable throttle 4 and maintains constant valve of the flow across it. Piston 11 of counterbalance valve 8 is in the left position. The main spool 10 of counterbalance valve 8 performs the functions of hydraulic lock and provides reliable load fixing during the shutdown.

3. Construction of mathematical model

In order to determine the mathematical model parameters, 3D models of the front-end loader boom and counterbalance valve were developed Fig. 2 shows 3D models of front-end loader boom, realized by means of the program Autodesk Inventor. Lifting/lowering of the boom is realized as a result of its rotation about a stationary axis y_0. As a result of analysis of 3D model the dependence of inertia moments $I_{xx}, I_{zz}, I_{yy}, I_{xz}, I_{yz}, I_{xy}$ of the boom on the angle of boom φ and integrated mass m of the boom with load was calculated.

In Fig. 3, the curve a shows the dependence of boom inertia tensor $I_m=f(\varphi,m)$ of 3D model of front-end loader boom on the angle of swing of a boom φ for integrated mass m of the boom with load, determined by the formula 1. In formula 1 angles $\alpha=90°$, $\gamma=90°$, $\beta=0$ characterize the positions of an axes x, z, y relatively axis y_0, and

integration mass m includes masses of load m_c and the boom of front-end loader m_s.

Fig. 2 3D model of front-end loader boom at φ angle turn

$$I_m = I_{xx}\left(\varphi,m\right)\cdot\cos^2\alpha + I_{yy}\left(\varphi,m\right)\cdot\cos^2\beta + I_{zz}\left(\varphi,m\right)\cdot\cos^2\gamma -$$

$$-2\cdot I_{xy}\left(\varphi,m\right)\cos\alpha\cdot\cos\beta - 2\cdot I_{xz}\left(\varphi,m\right)\cdot\cos\alpha\cdot\cos\gamma - 2\cdot I_{yz}\left(\varphi,m\right)\cdot\cos\beta\cdot\cos\gamma$$

(1)

Motion of front-end loader boom is accompanied by jerks due to dissipative forces, backlashes and transient processes in hydraulic drive. Boom with the load oscillates according to a random law, taking up backlashes in joints. That is why, real values of α, β, γ angles should be taken into account as stochastic values with range $\pm4°$, as it is shown in Fig. 2, b, c for different integrated mass m.
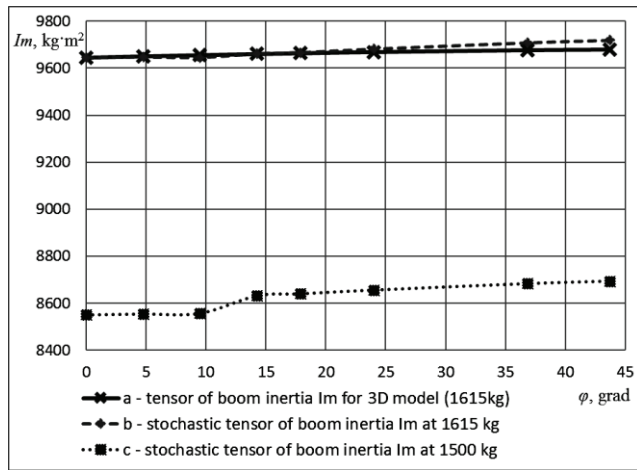

Fig. 3 Dependencies of boom inertia tensur $I_m=f(\varphi,m)$ on the angle of a swing of a boom φ and integrated mass m of the boom with load

From 3D model of front-end loader boom it is also determined the dependencies of x_c and x_s coordinates change of masses m_c and m_s center location for load and boom, correspondingly, on the angle of a swing of a boom φ.

Scheme of the counterbalance valve [11] is shown in Fig. 4, where A, B, C, D – working, T – drain hydraulic lines, correspondingly. Characteristic feature of such counterbalance valve is servo-spool with electromagnetic control. In the process of servo-spool operation spring-loader

end of the main spool is connected either with working hydraulic line D or with drain hydraulic line T while lifting (neutral position) and lowering of the load, correspondingly.
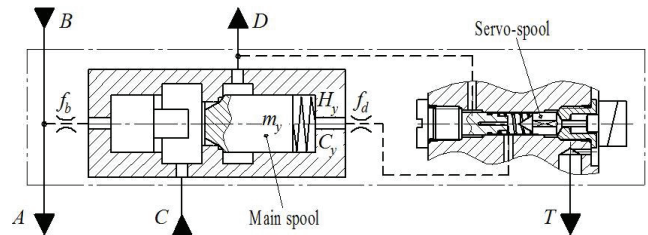

Fig. 4 Scheme of counterbalance valve in case of counter-loading
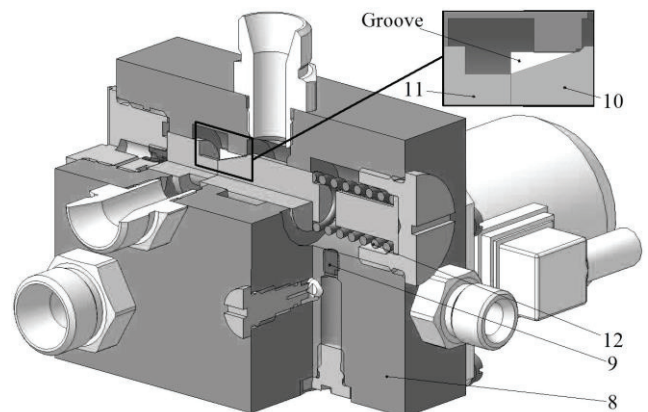

Fig. 5 3D model of counterbalance valve mock-up

Thus, servo-spool, when hydraulic distribution valve is in positions «lifting» and «neutral» gives the main spool the functions of hydraulic lock. At the same time reliable leak profess of hydraulic line D and load fixing in the required position is provided in case of the shutdown.

In «lowering» position servo-spool is switched and main spool performs the functions of the counterbalance valve. It provides the lack of cavitation processes in the working hydraulic lines, energy saving one to load lowering under its own weight, proportional speed control in case of accompanying load.

Fig. 5 shows 3D model of counterbalance valve mock-up, constructed on the base of its scheme and recommendations [11]. Working fluid flow in the canals of the counterbalance valve mock-up was studied on the base of simulation modeling. The research was performed by means of Autodesk Simulation CFD. Hydraulic forces, acting on the main spool at different initial conditions were calculated. Initial conditions include the values of input/output pressures, position of the main spool and working fluid supply across working window of counterbalance valve.

Approximation of calculation results is performed by means of DataFit program, dependence of hydrodynamic force $F_{hd}=f(p_y, Q_y)$ at the main spool on pressure p_y value at the output of counterbalance valve and Q_y supply across working window is obtained. Determination coefficient for approximated dependence in case of counter load is $R^2=0,993$. Approximation dependence is quadratic function:

$$F_{hda} = a_0 + a_1 \cdot Q_y + a_2 \cdot p_y + a_3 \cdot Q_y^2 + a_4 \cdot p_y^2 + \\ + a_5 \cdot Q_y \cdot p_y \tag{2}$$

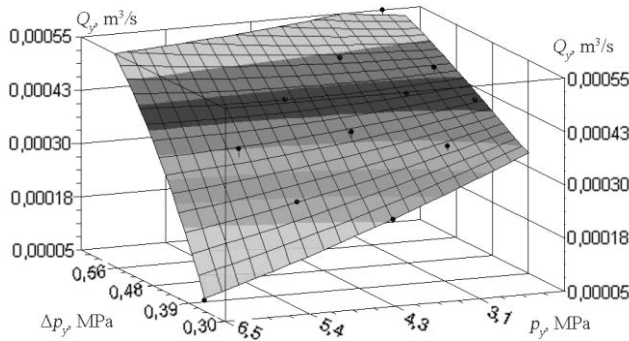where: a_0, a_1, a_2, a_3, a_4, a_5 – approximation coefficients.

Fig. 6 Dependence of feed value $Q_y=f(p_y, \Delta p_y)$ across counterbalance valve

Experimental stand was constructed in the laboratory of the university [12], where the studies of the experimental prototype of counterbalance valve were carried out. The dependence of feed value Q_y across counterbalance valve on pressure p_y value at the output and pressure drop Δp_y was determined.

Fig. 6 shows approximated results of experimental studies of feed value dependence $Q_y=f(p_y, \Delta p_y)$. Determination coefficients for the dependence $Q_y=f(p_y, \Delta p_y)$ is $R^2=0,989$. Dependence of feed value $Q_y=f(p_y, \Delta p_y)$ on the value of pressure p_y and pressure drop Δp_y at main spool of counterbalance valve has the from:

$$Q_y = c_0 + \frac{c_1}{\Delta p_y} + c_2 \cdot p_y + \frac{c_3}{\Delta p_y^2} + c_4 \cdot p_y^2 + \frac{c_5 \cdot p_y}{\Delta p_y} , \tag{3}$$

where: c_0, c_1, c_2, c_3, c_4, c_5 – approximation coefficients; $\Delta p_y = p_p - p_y$ – pressure drop on main spool 10.

On the base of calculation scheme and determined dependencies, mathematical model is composed, it comprises the system of non-linear differential equations.

Mathematical model is composed on condition of the flowering assumptions [13, 14]: lifting load, is uniformly distributed in the bucket; geometric parameters of hydraulic cylinders 5, 6 are assumed to be the same; coordinates, motion speeds and rods acceleration are assumed to be equal among themselves, correspondingly, as hydraulic cylinders 5, 6 are fixed at common frame and, rods have rigid connection; concentrated parameters of hydraulic drive are considered; temperature and viscosity of working fluid is constant; wave processes in hydraulic lines are not taken into account; compliances of cavities as well as compressibility of working fluid are taken into account as averaged values for the investigated range of pressure change; coefficient of flow across the throttles and spool elements are constant; pressure of the fluid in drain hydraulic lines is constant; operation models – without cavitation; pressure losses in hydraulic lines are not taken into account; forces of dry friction for spools are not taken into account; overflows of working fluid in pump 1, hydraulic cylinders 5, 6 and counterbalance valve 8 are not taken into account.

Mathematical model includes the equation of flows continuity equation for hydraulic lines:

$$Q_n = \mu \cdot f \cdot \sqrt{\frac{2|p_n - p_p|}{\rho}} \cdot sign\left(p_n - p_p\right) + \mu \cdot \pi \cdot d_z \cdot z \cdot \sqrt{\frac{2|p_n - p_0|}{\rho}} \cdot sign\left(p_n - p_0\right) - \frac{\pi \cdot d_z^2}{4} \cdot \frac{dz}{dt} + \beta \cdot W_n \cdot \frac{dp_n}{dt} , \tag{4}$$

where: $Q_n = 0,25 \cdot \pi \cdot (d_7)^2 \cdot d_8 \cdot k \cdot n \cdot tg(\gamma)$;

$$\mu \cdot f \cdot \sqrt{\frac{2|p_n - p_p|}{\rho}} \cdot sign\left(p_n - p_p\right) = Q_y - \frac{\pi \cdot d_z^2}{4} \cdot \frac{dz}{dt} + \beta \cdot W_p \cdot \frac{dp_p}{dt} ; \tag{5}$$

$$Q_y = 2F_y \cdot \frac{d\varphi}{dt} \cdot \frac{l_b}{\sin(\alpha)} + \mu \cdot f_d \cdot \sqrt{\frac{2|p_y - p_d|}{\rho}} \cdot sign\left(p_y - p_d\right) + \beta \cdot W_y \cdot \frac{dp_y}{dt} ; \tag{6}$$

$$\mu \cdot f_d \cdot \sqrt{\frac{2|p_y - p_d|}{\rho}} \cdot sign\left(p_y - p_d\right) = \beta \cdot W_d \cdot \frac{dp_d}{dt} - \frac{\pi \cdot d_{y3}^2}{4} \cdot \frac{dy}{dt} ; \tag{7}$$

$$\mu \cdot \pi \cdot d_z \cdot z \cdot \sqrt{\frac{2|p_n - p_0|}{\rho}} \cdot sign\left(p_n - p_0\right) = \mu \cdot f_0 \cdot \sqrt{\frac{2p_0}{\rho}} + \beta \cdot W_0 \cdot \frac{dp_0}{dt} ; \tag{8}$$

and equations of forces balance on spools of counterbalance valve 8 and feed regulator 2:

$$m_y \cdot \frac{d^2 y}{dt^2} = p_y \cdot \frac{\pi \cdot d_{y2}^2}{4} - p_d \cdot \frac{\pi \cdot d_{y3}^2}{4} - \frac{\pi \cdot \rho \cdot v \cdot d_{y3} \cdot l_{y3}}{\varepsilon_{y3}} \cdot \frac{dy}{dt} - C_y\left(H_y + y\right) - F_{hda}; \tag{9}$$

$$\frac{\pi \cdot \rho \cdot v \cdot d_z \cdot l_z}{\varepsilon_z} \cdot \frac{dz}{dt} = p_n \frac{\pi \cdot d_z^2}{4} - p_p \frac{\pi \cdot d_z^2}{4} - C_z\left(H_z + z\right). \tag{10}$$

Equations of moments balance on the faceplate of variable pump 1 and on the boom 7 front-end loader is described by the formulas:

$$J \cdot \frac{d^2 \gamma}{dt^2} = p_n \cdot f_5 \cdot l - p_0 \cdot f_4 \cdot l - \frac{\pi \cdot \rho \cdot v \cdot d_4 \cdot l_4^2}{\varepsilon_4} \cdot \frac{d\gamma}{dt} \cdot \cos(\gamma) - \frac{\pi \cdot \rho \cdot v \cdot d_5 \cdot l_5^2}{\varepsilon_5} \cdot \frac{d\gamma}{dt} \cdot \cos(\gamma) +$$
$$+ \left(m_0 + m_1 \cdot Q_n + m_2 \cdot p_n + m_3 \cdot Q_n^2 + m_4 \cdot p_n^2 + m_5 \cdot Q_n \cdot p_n\right) \tag{11}$$

$$I_m \cdot \frac{d^2 \varphi}{dt^2} = p_y \cdot 2F_y \cdot l_b \cdot \sin(\alpha) - m_s \cdot g \cdot x_s - m_c \cdot g \cdot x_c - 2\left[m_c \cdot g \cdot x_c + m_s \cdot g \cdot x_s\right] \cdot \mu_f \cdot sign\left(\frac{d\varphi}{dt}\right) \cdot l_b \cdot \sin(\alpha) -$$
$$- 2\left[\pi \cdot d_c \cdot b_c \cdot k_f + \pi \cdot d_s \cdot b_s \cdot k_f\right] \cdot sign\left(\frac{d\varphi}{dt}\right) \cdot l_b \cdot \sin(\alpha) - 2M_{t1} \cdot sign(\frac{d\varphi}{dt}) - 2M_{t2} \cdot sign(\frac{d\varphi}{dt}) - 2M_{t3} \cdot sign(\frac{d\varphi}{dt}) \tag{12}$$

In non-linear differential equations (4) – (12) of mathematical model the following designation: f – area of working window of regulating throttle 4; μ_f – friction coefficient of steel on steel; k_f – specific friction of seals in the process of operation with mineral oil; μ – flow coefficient across throttle and spool elements; ρ – density of working fluid; β – compression coefficient of working fluid and pipelines; d_{y2}, d_{y3} – diameters of left and right ends of main spool 10; l_{y3} – length of main spool 10 iз contact with the housing of the counterbalance valve 7; f_d, f_o – areas of fixed throttle; d_c, d_s, b_c, b_s – diameters and widths of hydraulic cylinders 5, 6 piston and rod seals, correspondingly; W_n, W_p, W_y, W_0, W_d – volumes of hydraulic lines; m_c, m_s, m_y – integrated masses of the load, boom 7 and main spool 10 with the spring 12, correspondingly; g – free fall acceleration; C_y, C_z – rigidities of counterbalance valve 8 and feed regulator 2 springs; H_y, H_z – preliminary compression of counterbalance valve 8 and feed regulator 2 springs; ε_4, ε_5 – slots, formed between working surfaces of plungers and housing of the pump 1; ε_z, ε_{y3} – slots, formed between working surface: spool of feed regulator 2 and its lousing, main spool 10 of counterbalance valve 8 and its housing, correspondingly; F_c, F_y – areas of rod and piston chambers of hydraulic cylinders 5, 6; J – inertia moment of pump 1 faceplate; v – viscosity coefficient of mineral oil; d_4, d_5, l_4, l_5 – diameters of plungers and lengths of plungers contact with bushings variable pump 1, correspondingly; d_z, l_z – diameter and length of spool contact with the housing of feed regulator 2; d_7, d_8 – diameters of pump 1 and contact area of pump pistons with faceplate; k – number of pistons in pump 1; n – number of pump 1 shaft rotations;

f_4, f_5 – plungers areas; z, y – coordinates of feed regulator 2 and counterbalance valve 8 spools position; p_n, p_c, p_p, p_y, p_0, p_d – pressure values in hydraulic lines; γ, φ, α –turning angles of the faceplate of pump 1, boom 7 and hydraulic cylinders 5, 6, correspondingly; m_0, m_1, m_2, m_3, m_4, m_5 – coefficient of resistance moment dependence on the value of Q_n and p_n; Q_n – value of variable pump1 feed; F_{hda} – hydrodynamic force, acting on the main spool 10; l_b – boom 7 length from rotation axis to the axis of hydraulic cylinders 5, 6 fiction; x_s, x_c – coordinates of mass center of the boom 7 and load on axis x; M_{t1}, M_{t2}, M_{t3} – friction moments, occurring in the joints of hydraulic cylinders 5, 6 and boom 7.

4. Optimization of design parameters of counterbalance valve

Theoretical studies of the impact of counterbalance valve parameters on dynamic characteristics are performed by means of mathematical modeling. Software packet MATLAB Simulink is applied for mathematical modeling. Solution of non-linear differential equations is performed by means of Rosenbrock pattern search at such initial conditions: $z(0)=0$; $y(0)=0$; $V(0)=0$; $p_n(0)=4,6 \cdot 10^6$ Pa; $p_p(0)=4,2 \cdot 10^6$ Pa; $p_y(0)=4 \cdot 10^6$ Pa; $p_0(0)=0,1 \cdot 10^6$ Pa; $p_d(0)=3,7 \cdot 10^6$ Pa; $\gamma(0) = 0,05$ rad.

Results of calculations at different operation model are shown in Fig. 7. Process of load lifting at mass $m_c = 800$ kg was modeled. Speed of load $V_c(t)$ lifting at $Q_n=20$ LPM has more oscillating transient process than at $Q_n=40$ LPM. Disturbance factor of transient process for

hydraulic drive was the transition from rest state to motion of the boom 7 with load.

Research of optimization of counterbalance valve parameters was carried out for operation mode with working fluid supply Q_n=20 LPM. As optimization criterion the following dependence is used:

$$W_{opt} = 0,6\frac{\sigma_i}{\sigma_{max}} + 0,4\frac{t_{pi}}{t_{pmax}} , \qquad (13)$$

where: i – number of the experiment; σ_i, σ_{max} – value of pressure p_n overregulation in transient processes and maximum pressure p_n overregulation in transient processes; t_{pi}, t_{pmax} – time of transient process of pressure p_n and maximum time of transient process in a series of experiment.

In the process of optimization the following parameters are used: area f_d of nonadjustable throttle 9, slot ε_{y3} between main spool 10 and housing of counterbalance valve 8, preliminary compression of the spring 11 H_y, that changed in such range: f_d=(0,8…3,14)·10^{-6} m^2, ε_{y3}=(6…41)·10^{-6} m, H_y=(2…6,5)·10^{-3} m.

Optimization was realized, applying the method of LP-search. In the process of optimization transient process for 27 experiments were calculated. Calculation is carried out for variants combinations of parameters f_d, ε_{y3},

H_y. The values of overregulation σ_i, time of transient processes t_{pi} is determined; optimization criterion was calculated by the formula (13). The results of calculation are shown in the Table.
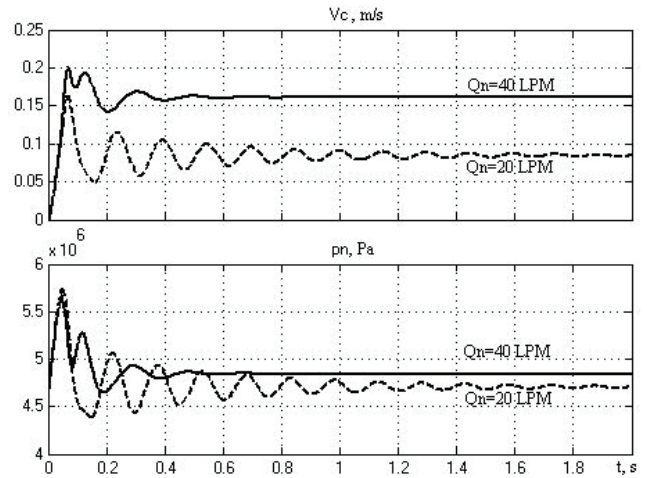

Fig. 7 Comparison of transient processes at the value of supply Q_n = 20 and 40 LPM prior the optimization

Table

Results of optimization criterion calculation

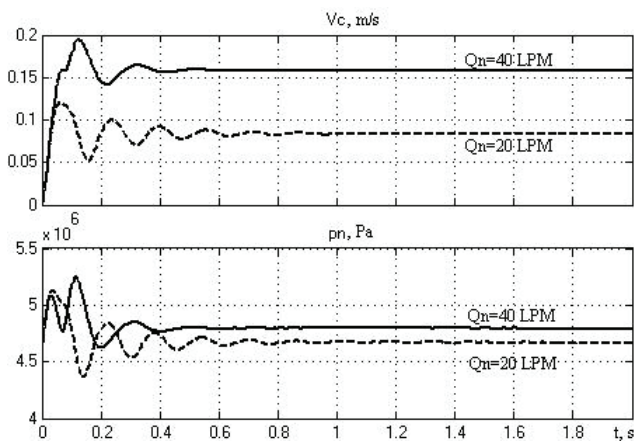| Number of research	f_d, m^2	ε_{y3}, m	H_y, m	σ_i, %	t_{pi}, s	W_{opt}
1	0,8·10^{-6}	6·10^{-6}	2·10^{-3}	19,5	0,8	0,66
2			4·10^{-3}	21,2	0,85	0,71
3			6,5·10^{-3}	23	0,9	0,76
…	…	…	…	…	…	…
10	1,76·10^{-6}	6·10^{-6}	2·10^{-3}	11,1	1,10	0,63
11			4·10^{-3}	12,7	1,15	0,68
12			6,5·10^{-3}	14,4	1,20	0,73
13		26·10^{-6}	2·10^{-3}	10,1	0,90	0,54
14			4·10^{-3}	11,6	1,00	0,6
15			6,5·10^{-3}	13,7	1,15	0,7
16		41·10^{-6}	2·10^{-3}	9,9	0,95	0,55
17			4·10^{-3}	11,4	1,10	0,64
18			6,5·10^{-3}	13,5	1,30	0,76

Fig. 8 Comparison of transient processes at the value of supply Q_n = 20 and 40 LPM after optimization

Combination of design parameters of counterbalance valve is thirteenth experiment is optimal. Transient process at variants values of the flow and optimal combinations of design parameters of counterbalance valve is shown in Fig 8.

5. Conclusions

1. Mathematical model of hydraulic drive of front-end loader with counterbalance valve is constructed. Mathematical model is constructed with account of quadratic dependence of hydrodynamic force $F_{hd}=f(p_y, Q_y)$ on the main spool of counterbalance valve, stochastic inertia tensor $I_m=f(\varphi, m)$ of front-end loader boom and experimentally determined dependence of working fluid supply $Q_y=f(p_y, \Delta p_y)$ across working window of counterbalance valve.

2. Optimization counterbalance valve value design parameters was carried out, it was calculated that optimal combination of parameters f_d=1.76·10^{-6}m^2, ε_{y3}=26·10^{-6}m, H_y=2·10^{-3}m provides: overregulation value σ_{13}=10,1%, transient process time t_{p13}=0.9s.

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OPTIMIZATION OF DESIGN PARAMETERS OF THE COUNTERBALANCE VALVE FOR THE FRONT-END LOADER HYDRAULIC DRIVE

S u m m a r y

A circuit of the front-end loader hydraulic drive has been developed and it is based on a counterbalance valve, which provides load fixation in a certain position, proportional speed control and energy saving due to the load lowering under its own weight. Mathematical models of the loader boom hydraulic drive has been elaborated and the solutions have been obtained in MATLAB Simulink environment. The mathematical model takes into account pressure losses of the counterbalance valve, the results of computer simulation of the loader boom inertia moment and hydrodynamic force at the main spool of the counterbalance valve, calculated using Autodesk software product. Optimization criterion has been developed and optimal design parameters are found.

Keywords: counterbalance valve, loader boom hydraulic drive.