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Modelling of operational cycle of hydraulic drive of lifting mechanism on the basis of axial piston hydraulic machines with discrete control

Ilia Nikolenko, Andrey Ryzhakov

National Academy of Environmental Protection and Resort Development,
181, Kievskaya str. , 95493 Simferopol, Autonomous Republic of Crimea, Ukraine
E-mail: napks@napks.edu.ua

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Summary. The mathematical model of hydraulic drive of lifting mechanism on the basis of discretely controlled bent axial piston hydraulic machines is developed. The model allows investigating of features of transients in hydraulic drive and control unit associated with the change of its operation mode, including physical and mechanical properties of power fluid.
Key words: axial piston hydraulic machines, hydraulic drive, mathematical model, discrete control, transients.

INTRODUCTION

The use of axial piston adjustable pumps and hydraulic motors in hydraulic drives of lifting mechanisms provides an ability of machines operating elements' speed control, with virtually no loss of energy by changing of hydraulic machines displacement. For lifting mechanisms it is enough to ensure the functioning of their operating elements in two modes: speed range operation (for operating element movement without load or with a cargo at the mean phase of lifting) and slow mode (for phases of cargo breakaway and completion of cargo movement). To ensure these two operation modes, one adjustable hydraulic machine with two fixed values of displacement is enough. Application of the second adjustable hydraulic machine in hydraulic drive allows increasing the number of different speeds of lifting mechanism up to four. Thus there is no need for manual or proportional control of lifting mechanism hydraulic drive, that requires rather complicated scheme of automatic control and increases the initial cost of hydraulic equipment and its maintenance and repair. Availability of mechanism's several speeds provides increasing of performance and workmanship of lifting machine, as well as reliability of the hydraulic drive.

Changing time parameters of discrete control of hydraulic drive, one can affect the nature of transients in the hydraulic system [1] arising from a change of APH and reducing the negative impact of pressure fluctuations in the hydraulic system.

Design solution and features of individual devices of APH displacement discrete control system affect the time value change of hydraulic machine displacement and nature of pressure fluctuations in the discharge line and in the line of displacement control system.

STATEMENT OF TASK

The objective of this paper was the creation of a mathematical model of lifting device hydraulic drive which allows carrying out numerical experiments for studying of progress of the transients and stable processes in hydraulic drive depending on its design features, physical and mechanical properties of power fluid and forces acting on the operating element of a lifting machine.

Results of numerical experiments can be used for optimization of discrete control procedure and technical parameters of hydraulic drive elements.

MATHEMATICAL MODEL

Hydraulic drive of lifting device comprises two bent axial piston hydraulic machines (P – Pump and M – hydraulic motor) with discrete two-level control of displacement (Fig. 1.). Pressure value in the discharge line of hydraulic drive is hydraulically controlled by overflow valve with direct action (V), with zero overlap, comprised of throttle (Th1).

Discrete change of displacement of the hydraulic machines P and M [2] is fulfilled by differential connection of hydraulic cylinders Hc, located in the control line (dashed line in Fig. 1.). Connection of hydraulic cylinders Hc is carried out by hydraulic distributors Hd.

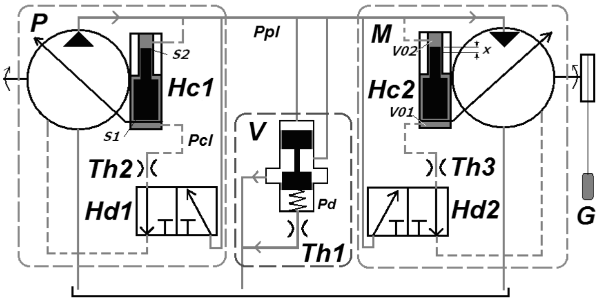

Fig. 1. Basic diagram of hydraulic drive of lifting device on the basis of discrete control axial-piston hydraulic machines

When developing a mathematical model of the hydraulic drive operation [3] it was assumed that:

– wave actions in pipelines are neglected [4] because of the short length of pipelines and minimum pressure loss at power fluid passing,
– no wave actions accompanying the transition of hydraulic cylinders of pumping unit from zone of suction to discharge zone and vice versa,
– little effect of friction forces in the piston supports on dynamics of pumping unit movement.

The following systems of equations are the basis of the mathematical model. Changes of pressure p_{pl} in discharge line of hydraulic drive and hydraulic motor speed w_m are found by simultaneous solving of equation of continuity (1), equation of moments (2), kinematic equations for hydraulic machines (3) and the equations that simulate the operation of overflow valve (4, 5):

$$Q_p - C_{pn}p_{pl} - \frac{V^p + V^{tr}}{E_a\left(p_{pl}, m_0\right)}\frac{dp_{pl}}{dt} - Q_V - Q_{Th1} -$$
$$-Q_{op} = Q_m + C_{mn}p_{pl} + \frac{V^M}{E_a\left(p_{pl}, m_0\right)}\cdot\frac{dp_{pl}}{dt}, \quad (1)$$

$$C_{p(m)n} = 1{,}23\cdot 10^{-8}\left(W^{p(m)}\right)^{0{,}7} / \mu(p_{pl}),$$

$$\beta = 4{,}2\cdot 10^4\cdot\mu(p_{pl})\cdot\left(W^m\right)^{1{,}6},$$

$$W_0^m p_{pl}\eta_m\eta_g - M_c - \beta\omega_m = J\frac{d\omega_m}{dt}. \quad (2)$$

The meanings of (1) and (2): $V^{p(m)}$ – P (M) hydraulic machine displacement; $C_{p(m)n}$ – leakage factor in P (M); V^{tr} – pressure pipeline volume; β – fluid friction coefficient M; J – integral inertia, reduced to the shaft. M_c – moment of resistance on the shaft M; $\eta_{m,g}$ – mechanical and hydromechanical efficiency coefficient. The model took into account that pump delivery flow Q_p and flow rate of the motor M are time variant in accordance with the kinematic dependencies for axial piston hydraulic machines [6]:

$$Q_{p(m)} = \sum_{i=1}^{k} Q_{ip(m)} = \frac{\pi d_{p(m)}^2}{4}\sum_{i=1}^{k}\frac{dx_{ip(m)}}{dt},$$

$$k = \frac{z\pm 1}{2}; \quad R' = \frac{\beta_{R'}\left(\gamma_{\max}, z\right)\cdot d_{p(m)}}{2}, \quad (3)$$

$$R = k_D\cdot R'; \quad d_{p(m)} = 2\cdot\beta_R\left(\gamma_{\max}, z\right)\cdot\sqrt[3]{V^{p(m)}}.$$

In equations (3): x_i – hydraulic machine i- piston stroke [5]; z – number of pistons of internal cylinder block ; d – diameter of the piston; and coefficient values $\beta_R\left(\gamma_{\max}, z\right)$ and $\beta_{R'}\left(\gamma_{\max}, z\right)$ are given in reference [7, 8].

The model of overflow valve V is given by equations (4):

$$m\frac{d^2z}{dt^2} = \left(p_{pl} - p_d\right)S_V - \left(z + z_0\right)\cdot c - F_{gtr} - F_g, \quad (4)$$

$$F_g = \rho Q_V^2\left(\frac{1}{S_V} - \frac{\cos(\theta/2)}{S_p}\right); S_p = \pi d_p\sin\frac{\theta}{2}\cdot z,$$

$$F_{gtr} = \frac{\left(p_{pl} - p_d\right)}{2}b\delta - \frac{\mu b\left(L + x\right)}{\delta}\cdot\frac{dx}{dt}.$$

In equations (4): p_d – pressure in damping cavity of overflow valve V; z – offset of shut-off-and-regulating element of the valve V; F_{gtr} and F_g – liquid friction forces and hydrodynamic force acting on shut-off-and-regulating element ; c – spring stiffness V; θ – cone angle of shut-off element V. Power fluid flow rates Q_V through the valve V and throttle Th1 are given by the equation (5):

$$Q_V = \pi\mu_{Th1}d_p\sin\frac{\theta}{2}\sqrt{\frac{2p_{pl}}{\rho}}z, \quad (5)$$

$$S_V\frac{dz}{dt} = Q_{Th1} + \frac{V_{Th1}}{E_a}\frac{dp_d}{dt}; \quad Q_{Th1} = \mu_{Th1}S_{Th1}\sqrt{\frac{2p_d}{\rho}}.$$

Hydrodynamic processes of power fluid motion in the control line of hydraulic drive between the discharge line and cavities of hydraulic cylinders Hc, at displacement decreasing, are described by differential equation (6), and at displacement increasing – by the equation (7):

$$Q_{Th} = \mu_{Th}\cdot S_{Th}\cdot\sqrt{\frac{2}{\rho}\left(p_{pl} - p_{cl}\right)} = S_{1Hc}\frac{dy}{dt} +$$
$$+\frac{V_{01Hc} + S_{1Hc}\cdot y}{E_a}\cdot\frac{dp_{cl}}{dt}, \quad (6)$$

$$Q_{Th} = \mu_{Th}\cdot S_{Th}\cdot\sqrt{\frac{2}{\rho}\left(p_{cl} - p_0\right)} + \frac{dp_{cl}}{dt}\cdot$$
$$\cdot\frac{V_{01Hc} + S_{1Hc}\cdot y}{E_a} = -S_1\frac{dy}{dt}. \quad (7)$$

Relationship of the mathematical model of the stroker (6, 7) with the equation of continuity of the initial model (1) is carried out by means of equations (8 and 9 – in the cases of displacement increasing and decreasing, respectively) describing the flow rate of power fluid Q_{op} to ensure the operation of the stroker:

$$Q_{op} = Q_{Th} - S_{2Hc} \cdot \frac{dy}{dt}, \quad (8)$$

$$Q_{op} = \frac{V_{02Hc} + S_{2Hc} \cdot (y_{\max} - y)}{E_a} \cdot \frac{dp_{pl}}{dt} - S_{2Hc} \cdot \frac{dy}{dt}. \quad (9)$$

The meanings of (6 – 9): p_{cl} – power fluid pressure in the control line (Fig. 1); y – offset of hydraulic cylinder piston Hc; $S_{1,2Hc}$ - area of larger and smaller sections of hydraulic cylinder piston Hc, respectively; $V_{01,2Hc}$ – appropriate minimum cavity volumes of hydraulic cylinder.

Dynamics of hydraulic cylinder piston Hc of hydraulic machine stroker is described [9, 10] by equations (10):

$$m\frac{d^2y}{dt^2} = -S_{2Hc} \cdot p_{pl} + S_{1Hc} \cdot p_{cl} - F_{gtr} - -sign\left(\frac{dx}{dt}\right) \cdot F_{tr} - F_G - F_{in}, \quad (10)$$

$$F_{gtr} = \mu \frac{S_{tr}}{\Delta}\frac{dy}{dt};\ F_G = Mg \cdot \frac{l}{L} \cdot \cos\gamma,$$

$$F_{tr} = \nu_{tr} \cdot \left(z \cdot S_{bc} - k \cdot S_s\right) \cdot p_{pl},$$

$$F_{in} = \left(m_p + \frac{1}{2}m_{pit}\right)\omega_{P,M}^2 \frac{r}{L}\sum_{j=1}^{k}\left[z_j(t) \cdot \cdot \sin\left(\omega_{P,M} \cdot t - \delta_j\right)\right],$$

$$\gamma[\deg ree] = -4.5[\frac{\deg ree}{cm}]y + \gamma_{\max}.$$

The meanings of (10): F_{gtr} – viscous friction force of the hydraulic cylinder piston; F_G – force applied to hydraulic cylinder piston Hc by weight of hydraulic machine pumping unit on the control block; F_{tr} – sliding frictional force of the hydraulic distributor; F_{in} – oscillating inertial force of the pistons of pumping unit; γ – bent angle of cylinder block. S_{bc} – sectional area of the piston of pumping unit; S_s – the effective surface area of the distributor, on which squeezing pressure of discharge line acts; δ – phase shift between the pistons of pumping unit; m_p and m_{pit} – masses of the piston and piston rod of pumping unit, respectively.

For lifting device, external forces of torque value M_c acting on the axis of hydraulic motor M, that is a member of the moment equation (2) is found by solving [11] of equations (11). For generation of the equations (11), as a part of the engineering approach, let us present the hoisting rope as a bar, endowed with longitudinal stiffness k of the rope. Tension in the rope is assumed to be uniformly distributed over its cross-section:

$$\frac{d^2h}{dt^2} = g - \frac{k}{M}h - 2 \cdot \delta \cdot \frac{dh}{dt} + a_i, \quad (11)$$

$$a_i = \frac{d\upsilon}{dt} = \frac{D_b}{2} \cdot \frac{d\omega_M}{dt},$$

$$\omega_0 = \sqrt{k / M},\ 2 \cdot \delta = \tau_{tr} \cdot \omega_0^2,\ k = E_d \cdot S / L,$$

$$E_d = \frac{\rho_L}{S} \cdot a^2,\ M_c = k \cdot y \cdot n \cdot D_b / 2.$$

In the equations (11): h – shifting of cargo with weight $G = Mg$; δ – damping factor; a_i – inertia of acceleration; D_b – diameter of the lifting device unit; τ_{tr} – time constant of internal friction; E_d – dynamic modulus of elasticity; a – wave velocity in spiral ropes. Dependences of main parameters that characterize physical and mechanical properties of power fluid – adiabatic bulk modulus E_a, density – ρ and dynamic viscosity – μ from pressure of power fluid p and percentage of non-dissolved air [12] are given by equations (12):

$$\mu(p) = \mu_0 \cdot \exp\left(\frac{p - p_0}{k_\mu}\right), \quad (12)$$

$$E_a(p) = \frac{kE_a^p\left[\left(E_a^0 / E_a^p\right)^{1/A} + \bar{m}\left(p_p^0\right)^{1/k}\right]}{k\left(E_a^0 / E_a^p\right)^{1/A} + \bar{m}\frac{E_a^p}{(p_i + 1)}\left(p_p^0\right)^{1/k}},$$

$$\rho(p) = \rho_0\left(\left(\sqrt[A_a]{\frac{E_a^0}{E_a^p}} + \bar{m}\sqrt[k]{p_p^0}\right)(1 - m_0)\right)^{-1},$$

$$E_a^p = A_a p + B_a,\ E_a^0 = A_a p_0 + B_a,$$

$$p_p^0 = (p_0 + 1)/(p + 1),\ \bar{m} = \frac{m_0}{1 - m_0},$$

$$k = c_p / c_V.$$

In equations (12): μ_0, E_a^0, ρ_0 – the values of dynamic viscosity, bulk modulus and density of power fluid under pressure p_0. The parameters A_a, B_a и k_μ are determined from experimental data for a particular power fluid.

RESULTS AND DISCUSSION

The system of equations (1 – 12) contains the nonlinear differential equations, which eliminates the possibility of its analytic solving. The system was integrated numerically by Bogacki – Shampine using the program Matlab. The results of numerical experiments allow conducting work analysis of both individual hydraulic drives and the whole hydraulic system. The results of numerical modelling of complete cycle of hydraulic drive operation were obtained that provide two possible speeds of lifting mechanism. Complete cycle of operation of lifting mechanism includes: cargo breakaway at the lowest speed, cargo movement at the maximum speed of movement and the completion of movement at the lowest speed. To analyze the effect of different parameters on the hydraulic drive operation various factors were changed: time change τ_0 of hydraulic drive operation mode and physical and mechanical conditions of power fluid – viscosity μ, density ρ, adiabatic bulk modulus E_a and percentage of undissolved air in power fluid – m_0. Also different models describing the dependence of values μ, ρ and E_a from pressure were compared. The values of the parameters that were used for graphing in Figures 2 – 4 are shown in Table 1.

Table 1. Model varied parameters values

№	τ_{01}, s	τ_{02}, s	μ, MPa s	ρ, MPa s²/cm²	E_a, MPa	m_0, %
1	1	3.5	$\mu(p)$	$\rho(p)$	$E_a(p)$	0
2	1	3.5	1.78 ^{10}-8	8.9 ^{10}-8	$E_a(p)$	0
3	1	3.5	1.78 10^{-8}	8.9 10^{-8}	1880	0
4	0.25	3.5	$\mu(p)$	$\rho(p)$	$E_a(p)$	0
5	0.25	3.5	$\mu(p)$	$\rho(p)$	$E_a(p)$	5
6	-	-	$\mu(p)$	$\rho(p)$	$E_a(p)$	0

The parameters' values $\mu(p), \rho(p)$ and $E(p)$ are considered as functions of pressure in this part of hydraulic drive and were found by equations (12). Time τ_{01} defines the beginning of hydraulic motor M displacement decrease with values $V_M = 112cm^3$ up to $V_M = 56cm^3$, and time τ_{02} – the beginning of its increase up to the initial value.

In Fig. 1. dynamics of power fluid pressure in discharge line of the hydraulic drive under different operation modes are shown.

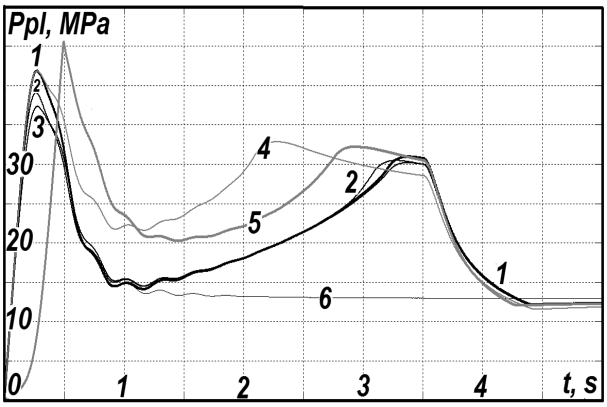

Fig. 2. Pressure in discharge line of lifting device hydraulic drive under different operating conditions

The graph 6 (Fig. 2.) shows the change of pressure in the discharge line of hydraulic drive during the transition process associated with actuation of hydraulic drive. Change of motor displacement for case 6 was not carried out. For transients, shown in Figures 1 – 3 (Fig. 2.), decrease of motor displacement (from value – $V_M = 112cm^3$ up to $V_M = 56cm^3$) began upon expiration of time $\tau_{01} = 1s$ after hydraulic drive starting, the reverse change V_M began in time period $\tau_{02} = 3.5s$ from the moment of process starting. For processes in graphs 4 and 5 displacement decrease V_M began at $\tau_{01} = 0.25s$. Duration of stable process, between changes of displacement V_M, was given as a minimum value in order to reduce computation time.

A given increase of hydraulic motor displacement causes pressure boost in the discharge line of hydraulic drive up to value – $p_{pl} = 3/4 \cdot p_{pl}^{\max}$, where $p_{pl}^{\max}$ – maximum pressure surge at actuation of hydraulic drive without changing of hydraulic motor displacement. In graphs 2 and 3 the results of calculations for the transient process similar to process 1 are shown, but density and viscosity parameters were considered as depended constant values (Table 1.). From these calculations it follows that such a simplification of the model affects only the analysis of non-stable processes and leads to a relative error in the amplitude of pressure pulse, which does not exceed 12%.

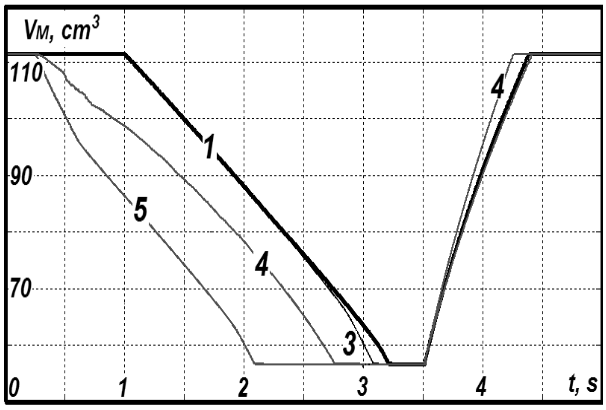

Fig. 3. Changes of hydraulic motor displacement at varying of cargo lifting speed

For processes 1 – 3, two times decrease of displacement motor M takes up some time -about 2.25s. The reverse process is faster – for a period of time about 1s (Fig. 3). The difference in response time of the control system is observed, despite the fact that the differential circuit connection of hydraulic cylinder Hc2, piston sectional area ratio is equal to $S_{2Hc}/S_{1Hc} = 2$. It can be explained if we consider that in the control line P and M the throttle Th is located which slows down pressure increase in the control line at displacement decrease of the hydraulic machine. Dynamics of pressure change in the control line for processes 1 and 5 is shown in Fig. 4. Comparison of graphs 5 in Figures 3 and 4 leads to the conclusion that the high rate of motor displacement decrease, at the initial phase of the process is directly connected with pressure surge of the control line.

Dependencies for the transient process 5, in contrast to previous relationships, were considered the influence of undissolved air bubbles on physical and mechanical properties of power fluid.

This feature leads to increase compressibility of power fluid, time delay in reaching of peak pressure in the transient process (Fig. 2.) and increase the value of pressure starting level in the control line (Fig. 4).

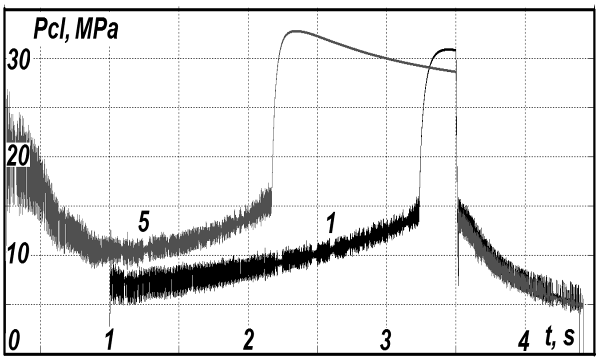

Fig. 4. Pressure change in the control line at hydraulic motor displacement change

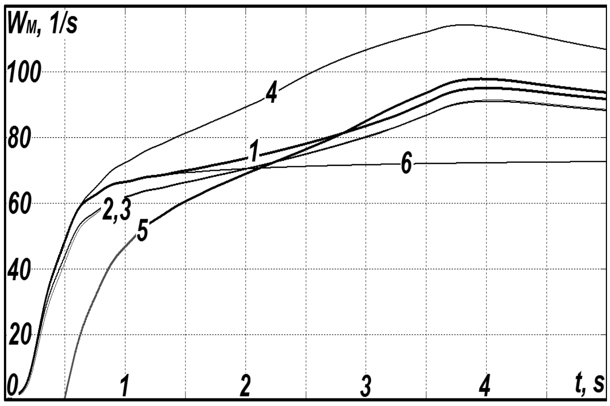

Fig. 5. Dynamics of hydraulic motor shaft acceleration under different operation modes of hydraulic drive of lifting device. Figure 5 shows the time dependence of angular speed of hydraulic motor shaft under different operation conditions of hydraulic drive. By comparison of curves 1 and 6 it can be seen that the expected increase of hydraulic motor shaft speed, at changing its displacement by two, will be resulted in 30%. More significant speed increase at the beginning of cargo lifting can be achieved by use of control 4

CONCLUSIONS

The numerical experiments show the possibility of use of the developed model to simulate the transient and stable state processes in hydraulic drive of lifting mechanism on the basis of discretely controlled bent axial piston hydraulic machines.

The numerical calculations within the framework of the developed model can be used for optimization of the control procedure of lifting mechanism for road building machine for the purpose of decrease of negative impact of pressure fluctuations on the hydraulic system, increase of its performance and workmanship of lifting machine, as well as reliability of the hydraulic drive.

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МОДЕЛИРОВАНИЕ РАБОЧЕГО ЦИКЛА ГИДРОПРИВОДА МЕХАНИЗМА ПОДЪЕМА НА ОСНОВЕ АКСИАЛЬНО-ПОРШНЕВЫХ ГИДРОМАШИН С ДИСКРЕТНЫМ УПРАВЛЕНИЕМ

Аннотация. Разработана математическая модель гидропривода механизма подъема на основе аксиально-поршневых дискретно регулируемых гидромашин с наклонным блоком цилиндров. Модель позволяет исследовать особенности переходных процессов в гидроприводе и блоке управления, связанных с изменением режима их работы, с учетом физико-механических свойств рабочей жидкости.

Ключевые слова: аксиально-поршневые гидромашины, гидропривод, математическая модель, дискретное регулирование, переходные процессы.