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Simulation of the hydraulic system of a device with self-adaptation for power and kinematic parameters on the working body

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Introduction. Currently, Russia has adopted a course towards the creation of intelligent machines and equipment. The same holds for mobile technological machines for road construction and public utilities. Therefore, the design and creation of this type of actuators with a self-adaptation function is a critical task.
Materials and Methods. A device equipped with a hydraulic drive with self-adaptation to load and coordination of kinematic and power parameters of the principal motion and the feed movement of the working body of the rock-drilling rig, is presented. To study and design the device based on the mathematical modeling methods of a hydraulic drive and adaptive systems, a mathematical model is proposed. It is developed using the foundations of the theory of volumetric stiffness of hydraulic systems. This enables to accurately describe the impact of the dynamic properties of the hydraulic system (compressibility of the working fluid, elastic properties of pipelines, high-pressure hoses, hydraulic apparatuses) on the dynamic properties of the system as a whole.
Results. The mathematical model for a device with self-adaptation includes submodels of adaptive communication, interrelations of power, kinematic and process parameters of rock drilling, as well as mathematical description of the movement of system elements. The solution to the developed mathematical model was performed in the software environment for dynamic modeling of technical systems SimInTech. As a result, general dependences of the adaptive system on the design parameters of the system and the operating conditions are obtained.
Discussion and Conclusion. The mathematical model of the presented device shows the fundamental possibility of implementing the principle of self-adaptation in terms of load under external and internal disturbing actions during operation. The results obtained can be used under designing adaptive systems of other technological equipment, for example, for the implementation of deep drilling in workpieces with variable properties in its depth.

Keywords: hydraulic device with self-adaptation, hydraulic drive function, generalized mathematical model, adaptive communication, coordinated movements, working body, load, stabilization.

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Introduction. The course adopted in Russia on building intelligent machines is the basis for the design and creation of actuators with a self-adapting function. The solution to this problem is timely and relevant.

Drives with differential couplings of internal structure elements possess self-adapting properties [1]. This class of technical systems includes a variable rock drilling device (RF Patent No. 2582691). The self-adapting property is implemented by a device with negative feedback and positive feedback [2].

The quality of the self-adapting process is affected by external and internal actions — load variability, medium resistance, dry and viscous friction, volumetric stiffness of the fluid and pipelines, adaptive links.

Materials and Methods. At the current level of development of computer technology, the complexity and high cost of the designed equipment, methods of mathematical modeling are widely used at the development stage [3-

13]. The development of a special model for calculations and computational experiments to determine the parameters at the design stage provides a reasonable choice of the standard size of the device with self-adaptation.

The authors have developed a mathematical model for solving the problem of device design in the environment of dynamic simulation of SimInTech technical systems (Simulation In Technic) [14, 15]. This software product enables to simulate technological processes occurring in various industries with simultaneous modeling of control systems.

Fig. 1 shows a device for exercising the function of self-adaptation under the conditions of force resistance on the working body which contains fixed displacement pump (Н), security valve (КП), filter (Ф), adjustable throttles (Др1) and (Др2), flow regulator (РР), feed hydraulic cylinder (ГЦ) and main drive hydraulic motor (ГМ), hydraulic valves (Р1), (Р2) and (Р3), pressure gauges (Мн1), (Мн2) and (Мн3),

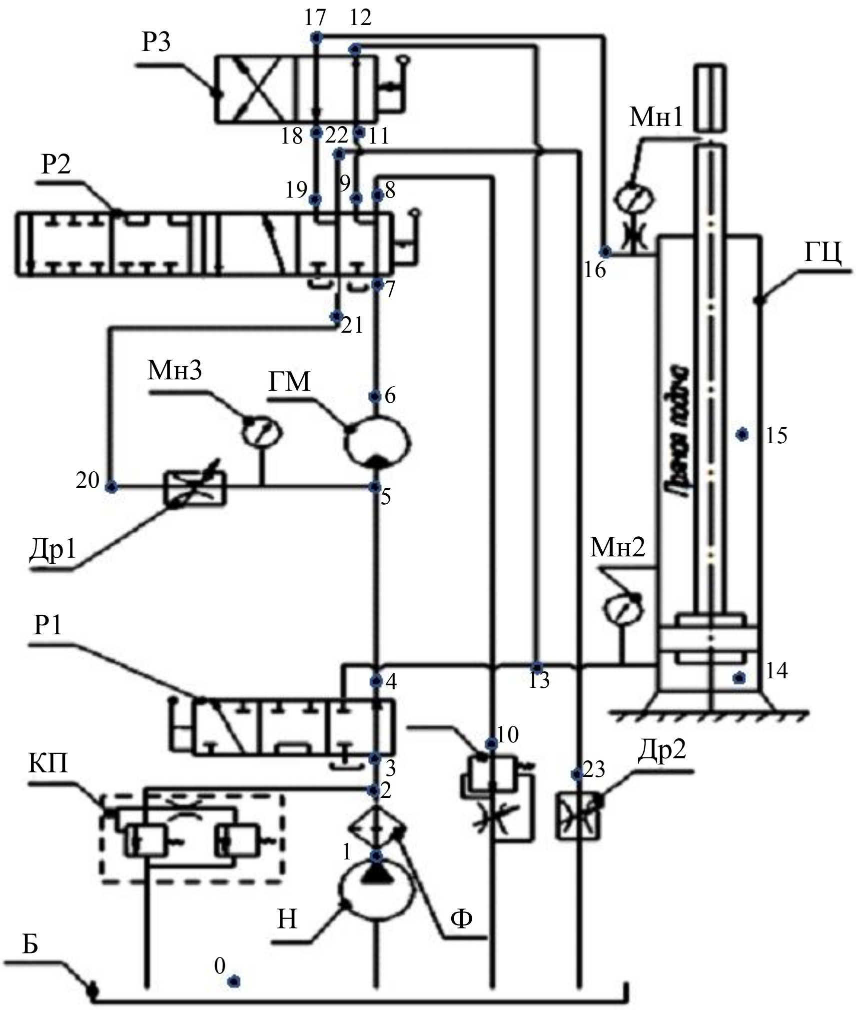


Fig. 1. Schematic diagram of the device for exercising the function of self-adaptation under the conditions of force resistance on the working body

The device is designed to exercise the function of self-adaptive load (to stabilize it) and matching output movements working on production machines and equipment, as well as functions of the actuator itself. Therefore, the device provides multi-position distributors for the formation of flows and directions of the working fluid. Such additional device functions include heating of the hydraulic oil in its start operation; “weighting” of moving parts (idle load) of the feeder when adjusting to technological mode; accelerated lifting (retraction) of the tool with rotation, but without controlling its speeds; tool feed “to the object of action” or “from the object of action” with rotation and control of tool speeds.

The mathematical dependence for the internal negative coupling of the device is established through joint solving the equations arising from the pressure balance equations in the system and the working fluid flow balance equation in the feedback system.

From the pressure balance in the system, it follows

$$\Delta p_{\text{др}5} = \frac{1}{w_{\text{гм}}} M_{\text{гм}} + \frac{1}{f_{\text{п}}} F_{\text{п}} - \left(1 - \frac{f_{\text{п.шт}}}{f_{\text{п}}}\right) \Delta p_{\text{др}10}, \quad (1)$$

where $\Delta p_{\text{др}5}$ and $\Delta p_{\text{др}10}$ — differential pressure across throttles 5 and 10; $w_{\text{гм}}$ and $M_{\text{гм}}$ — reduced volume of the hydraulic motor ГМ and the torque generated by it; $f_{\text{п}}$ and $f_{\text{п.шт}}$ — areas of the hydraulic cylinder piston in the piston and rod cavities; $F_{\text{п}}$ — resistance force from the ground, overcome when moving the rod of the hydraulic cylinder ГЦ.

The balance equation for the costs in the feedback loop has the form

$$Q_{\text{др}10} = Q_{\text{др}5} + v_{\text{п}} \cdot f_{\text{п.шт}}, \quad (2)$$

where $Q_{\text{др}5}$ and $Q_{\text{др}10}$ — the flow rate of the working fluid through the throttles Др1 and Др2; $v_{\text{п}}$ — the hydraulic cylinder ГЦ piston travel speed.

Considering that the hydraulic pump Н, which feeds the system, has a constant capacity, the flow of the working fluid through the flow regulator РР under the operation of the drive system remains constant. It becomes obvious that the flow rate of the working fluid through the throttle Др2 will also be constant. In this case, the dependence of the acceleration of the hydraulic cylinder piston ГЦ on the total load on the hydraulic motors of the main movement and the feed movement will take the form

$$\frac{dv_{\text{п}}}{dt} = -\mathrm{A}\left(\frac{1}{w_{\text{гм}}} \frac{dM_{\text{гм}}}{dt} + \frac{1}{f_{\text{п}}} \frac{dF}{dt}_{\text{п}}\right), \quad (3)$$

where $v_{\text{п}}$ — hydraulic cylinder ГЦ piston travel speed; t — time; A — feedback coefficient depending on the parameters of the nominal operating mode of the system, the design parameters of hydraulic machines, throttles and its settings.

It can be seen from the equation (3) that the positive increments of the torque on the hydraulic motor shaft and the movement of the hydraulic cylinder feed piston correspond to the negative increment of the tool feed speed. In other words, as the ground resistance increases to the rotation of the cutting tool or the movement of its feed, the tool feed rate decreases.

The direct relationship between the speeds and accelerations of the hydraulic motor shaft of the main movement and the hydraulic cylinder rod of the feed movement is as follows

$$v_{\text{п}} = \frac{1}{f_{\text{п}}} \left(w_{\text{гм}} \omega_{\text{гм}} - Q_{\text{рр}}\right),$$

or

$$\frac{dv_{\text{п}}}{dt} = w_{\text{гм}} \frac{d\omega_{\text{гм}}}{dt}. \quad (4)$$

where $\omega_{\text{гм}}$ — the angular speed of rotation of the hydraulic motor shaft; $Q_{\text{рр}}$ — the flow rate of the working fluid through the flow regulator.

From the equation (4), it can be seen that with an increase in the rotation speed of the hydraulic motor of the main movement (occurs with a decrease in resistance from the side of the treated surface), the speed of movement of the feed cylinder piston increases.

Mathematical modeling of the movements of the working elements of the device. The equations of motion of the rotor of the hydraulic motor of the main movement and the piston of the hydraulic cylinder of the tool feed have the form:

$$\mathrm{J}_{\text{гм}} \frac{d\omega_{\text{гм}}}{dt} = \Delta p_{\text{гм}} w_{\text{гм}} - M_{\text{сопр}}, \quad (5)$$

$$m_{п} \frac{dv_{п}}{dt} = f_{п}\, p_{п} - f_{п.шт}\, p_{п.шт} - F_{сопр}\,, \quad (6)$$

where $J_{гм}$ — the total moment of inertia of all rotating elements of the system reduced to the shaft of the hydraulic motor; $m_{п}$ — the mass of all moving parts of the system reduced to the hydraulic cylinder piston; $M_{сопр}$ — the total moment of resistance to the rotation of the working body reduced to the shaft of the hydraulic motor; $F_{сопр}$ — total force of resistance to the tool movement from the side of the treated surface reduced to the piston of the hydraulic cylinder; $\Delta p_{гм}$ — pressure drop across the hydraulic motor; $p_{п}$ and $p_{п.шт}$ — pressure in the piston and in the rod cavities of the hydraulic cylinder, respectively.

Modeling the properties of a hydraulic system

The mathematical model of the hydraulic system of the device for performing the function of self-adaptation under the conditions of force resistance on the working body is developed using volumetric rigidity, which provides modeling as close as possible to the real characteristics [16–20]. When modeling, special attention is paid to determining the reduced coefficient of volumetric stiffness of high-pressure hoses [16]. The resulting mathematical model includes the following equations:

— the equation of the pressure increment at various points (in Fig. 1, marked by points 1 to 23) of the hydraulic system has the form:

$$dp = C_{прi}\left(\sum Q_{вхi} - \sum Q_{исхi}\right)dt\,, \quad (7)$$

where $\sum Q_{вхi}$ and $\sum Q_{исхi}$ — the sums of all the flow rates of the working fluid entering and outgoing from the considered (*i*-th) volume of the system during the time *dt*; $C_{прi}$ — the reduced coefficient of volumetric rigidity of the selected section of the hydraulic system;

— the equation for determining the flow rate of the working fluid through various elements of the hydraulic system has the form:

$$Q_i = \mu f \sqrt{\frac{2}{\rho}\left|p_i - p_{i+1}\right|} \cdot \mathrm{sign}(p_i - p_{i+1}), \quad (8)$$

where p_i and p_{i+1} — pressure at the inlet and outlet of the hydraulic resistances; *f* — free cross-sectional area of resistance; ρ — working fluid density;

— the formula for calculating the reduced flow rate of linear resistances is as follows

$$\mu = \mu_l = \frac{1}{\sqrt{\lambda_l \frac{l_l}{d_l}}}, \quad (9)$$

where d_l and l_l — diameters and lengths of the linear section of the pipeline; λ_l — the coefficient of hydraulic friction of the pipeline;

— the formula for determining the reduced coefficient of volumetric stiffness of the metal sections of the pipeline has the form:

$$C_l = \frac{4}{\pi d^2 l}\, \frac{E_{fl}}{1 + \frac{d E_{fl}}{\delta\, E_l}}, \quad (10)$$

where *d* and *l* — the internal diameter and length of the pipeline section; δ — its wall thickness; E_{fl} and E_l — moduli of elasticity of fluid and wall material of the hydraulic line.

The reduced coefficients of the volume stiffness РВД are determined experimentally.

The performance of the hydraulic pump is determined with account for the volume losses from the formula:

$$Q_{нд} = \frac{q_{рн}\omega_{н}}{2\pi}\eta_{0,н}, \quad (11)$$

where $q_{рн}$ — the working volume of the hydraulic pump; $\omega_{н}$ — the hydraulic pump shaft speed; $\eta_{0,н}$ — the current value of the volumetric efficiency of the hydraulic pump.

The flow rate of the working fluid through the hydraulic motor is determined from the formula:

$$Q_{\text{мот}} = \frac{q_{\text{рм}}\omega_{\text{м}}}{2\pi\eta_{0,\text{м}}}, \tag{12}$$

where $q_{\text{рм}}$ — the hydraulic pump working volume; $\omega_{\text{м}}$ — the hydraulic pump shaft speed; $\eta_{0\text{м}}$ — current volumetric efficiency of the hydraulic motor.

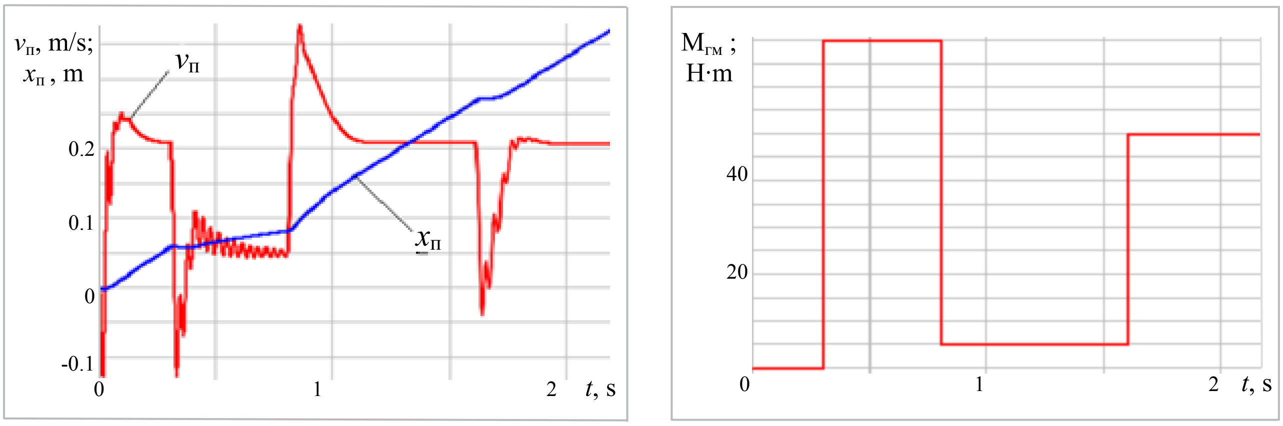
The current value of the volumetric efficiency coefficients of the hydraulic pump and the hydraulic motor are determined from the formula:

$$\eta_0 = 1 - (1 - \eta_{0.\text{nom}}) \cdot \frac{p_{\text{p}}}{p_{\text{p.nom}}}, \tag{13}$$

where $\eta_{0.\text{nom}}$ — the nominal volumetric efficiency of the hydraulic pump and hydraulic motor; $p_{\text{p.nom}}$ — the value of the nominal pressure of hydraulic machines; p_{p} — the current value of the pressure on the pump or motor.

Research Results

The proposed mathematical model of the device under consideration provides high-precision theoretical studies on the operational capabilities of rock drilling at the design stage. The calculation of the drilling system, performed using the SimInTech software [10, 11], showed the correctness of this statement.



a) *b)*

Fig. 2. Changing the parameters of the drilling machine operation with a stepwise change in the resistance to rotation of the main working body of the drilling machine from the ground: *a)* — movement of the hydraulic cylinder piston ($x_п$) and its speed ($v_п$); *b)* — torque of resistance to rotation of the hydraulic motor shaft $M_{гм}$

Fig. 2 shows the graphs of change in the parameters of the piston movement of the feed hydraulic cylinder with a stepwise change in the moment of resistance to rotation of the main working body of the drilling machine from the ground.

The analysis of calculation results presented in Fig. 2 shows that with an increase in the moment of resistance to rotation of the working body of the drilling machine (Fig. 2 *b*), the speed of translational movement of the tool decreases and can take negative values (Fig. 2 *a*). This indicates that the system performs the adaptation function. The properties of the system require additional research.

Conclusion. The proposed mathematical model enables to make a preliminary performance assessment at the design stage and select the most rational parameters of the drilling system equipped with an adaptive hydro-mechanical drive under different operating conditions and varying its design properties. The results obtained and their analysis suggest that the proposed method of adapting the main and auxiliary movements of the drive system can be successfully applied in other technological equipment, for example, in deep drilling of multilayer metal workpieces.

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Claimed contributorship

T. A. Khinikadze: basic concept formulation; research objectives and tasks; computational analysis; text preparation; formulation of conclusions. A. T. Rybak: academic advising; analysis of the research results; the text revision; correction of the conclusions. P. I. Popikov: analysis of literature and patent materials, dynamic characteristics of the hydraulic drive.

All authors have read and approved the final manuscript.