Dependent twin-engine hydraulic drive of the drilling rig with a mechno-hydraulic variable pump control circuit

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Abstract. The work is devoted to the study of an adaptive hydraulic drive for tool feed of a mobile drilling rig. The analysis of the ways of regulation of the considered hydraulic systems is carried out, the criteria and parameters of their functioning are determined. As a result of the analysis, an original circuit design of an adaptive hydraulic drive was applied, where the tool is fed using an adjustable volumetric hydraulic motor with a hydraulic control loop, taking into account the changing load on the tool during drilling. By a computational experiment in the Matlab Simulink program, the parameters of the control loop devices are determined: a hydromechanical sensor and a hydraulic valve, on the basis of which the experimental setup is implemented. The performed multifactorial experiment made it possible to identify the processes in the original hydraulic circuit for controlling the hydraulic motor under various modes of tool loading. The results made it possible to determine the rational ranges of functioning of the hydromechanical system for a typical working cycle. The use of the developed research methods and their results makes it possible to reduce the time and money spent in the design of adaptive hydraulic systems of mobile technological machines, in the creation of prototypes and in the commissioning.

1. Introduction
The development of natural resources, the industrial construction of large facilities requires the improvement of existing and the creation of new automated complexes of technological equipment for drilling production with improved mechanical and energy characteristics [1].

When the characteristics of the soil change, the productivity of drilling operations decreases. A solution to this problem can be achieved through the development of adaptive drive systems. The limiting factor is the absence in the operating equipment of a controlled kinematic connection between the working movements of rotation and feed of the drilling tool [2,3].

The construction of adaptive drive systems, as a rule, is carried out by introducing external feedbacks. This forms significant delays in control actions that degrade the quality of drive control [4,5,6].

The use of a hydraulic drive (HP) of dependent tool feed with an internal hydromechanical connection will increase the productivity and efficiency of the hydromechanical system (HMS) of the working movements of machines [1,7].

The basis for increasing the productivity of materials processing is the maintenance of technological parameters of the process and the maximum reduction in the time of setting, auxiliary movements in the working process, as well as preparatory and final procedures organizing processing [2,3,8]. The fulfillment of the basic conditions for increasing the processing productivity with the specified quality is possible by creating new or improving the existing systems of drives of working and auxiliary movements, providing various working cycles of processing [9,11].
The main direction of the search for solutions was the GP of volumetric control, which most fully correspond to the set task \[11,12\].

There are several structures of the volumetric method of speed control \[13\]:
1) Variable pump - unregulated hydraulic motor;
2) Fixed pump - variable motor;
3) Variable pump - variable motor.

When choosing a method for regulating GP, it is necessary to take into account economic indicators. Adjustable hydraulic machines are more expensive than unregulated ones, but due to their higher efficiency they have more comfortable operating costs \[11\].

There is also a volumetric throttle control of the hydraulic motor, which consists in the fact that a hydraulic pump of variable capacity is installed in a constant pressure throttle control system. In such a hydraulic system, there are no losses in the overflow valve \[14, 15\].

The presence of a hydraulically adjustable feed of the drilling tool to the bottom and automatic control of the rotation speed of hydraulic machines allows for an optimal mode of soil drilling. This will significantly affect the technical and economic performance of installations \[8\].

2. Schematic solution

Due to the high power, it is advisable to use volumetric control of both the pump and the engine at the heart of most HMS of drilling technological complexes. However, the installation of hydraulic machines that allow such regulation is not economically feasible. A complex and expensive control system that requires the installation of a large number of additional elements. There are also known studies of GP dependent feed of a tool with the structure of an uncontrolled pump - an adjustable hydraulic motor \[9, 10\]. We will conduct research on the HMS with dependent feed of the tool with the structure of a variable pump - not a variable motor.

![Figure 1. Block diagram of the HMS with dependent feed of the tool with the structure of a variable pump - non-variable motor: PP1, PP2 - power supplies of the power circuit; RR3 - power supply for the control circuit; HD1, HD2, HD3 - valves; HM1, HM2 - hydraulic motors; PPM1, PPM2 - transfer-converting mechanisms; MRI - tool feed mechanism; MGD - main movement mechanism; i1 - mechanical interface; Fн - load force; Мн - torque; APM - hardware and software module; GUK - hydraulically controlled valve; GMD - hydromechanical sensor; HD - battery; TH1, TH2 - chokes; CH1, CH2, CH3 - check valves](image-url)
3. Methods and materials
The design diagram explaining the interconnections of the elements of the schematic diagram and the main parameters adopted when modeling the hydraulic system of the drilling rig is shown in Fig. 2.

![Calculated hydrokinematic drive scheme](image)

**Figure 2.** Calculated hydrokinematic drive scheme

The mechanical subsystem is described by a two-mass model by bringing one part of its components to the driving Y1, and the other to the driven masses Y2 for the feed drive and the driving
Y3, and Y4 to the driven masses to drive the main motion, taking into account the elastic properties of the kinematic chain from the hydraulic motor HM1 and HM2 to the support tool feed [14,15]:

\[ Y_1 \frac{d^2 \phi_1}{dt^2} = M_{HM1} - M_{C1} - M_{1-2}; \]
\[ Y_2 \frac{d^2 \phi_2}{dt^2} = M_{1-2} - M_{C2} - M_{HM1}; \]
\[ Y_3 \frac{d^2 \phi_3}{dt^2} = M_{HM2} - M_{C3} - M_{3-4}; \]
\[ Y_4 \frac{d^2 \phi_4}{dt^2} = M_{3-4} - M_{C4} - M_{HM2}; \]

where \( Y_1, Y_2, Y_3, Y_4 \) – dynamic moment from rotating masses \( Y_1, Y_2, Y_3, Y_4 \);

\[ M_{HM1} = \frac{1}{2 \pi} \cdot q_{HM1} \cdot (p_2 - p_3) \] – torque of the hydraulic motor HM1, N·m;

\[ M_{HM2} = \frac{1}{2 \pi} \cdot q_{HM2} \cdot (p_8 - p_7) \] – torque of the hydraulic motor HM2, N·m;

\[ M_{hi} = M_{hi} (F_n) \] – torque from the load created by the feed force \( F_n \) during drilling, N·m;

\( M_{C1} \) and \( M_{C3} \) – respectively, the torques from the forces of resistance to motion HM1 and HM2 (dry and viscous friction, positional load) determined by the expression, N·m [15,16]:

\[ M_{C1} = M_{T1} + M_{B1} + M_{P1} = M_{T01} \cdot \text{sign} \frac{d \phi_1}{dt} + K_B \cdot \frac{d \phi_1}{dt} + + K_{P1} \cdot \phi_1; \] (5)

\[ M_{C3} = M_{T3} + M_{B3} + M_{P3} = M_{T02} \cdot \text{sign} \frac{d \phi_3}{dt} + K_B \cdot \frac{d \phi_3}{dt} + + K_{P3} \cdot \phi_3; \] (6)

\( M_{C2} \) and \( M_{C4} \) – moment from the forces of resistance to the motion of the driven masses and, determined by the expressions [24]:

\[ M_{C2} = M_{T2} + M_{B2} + M_{P2} = M_{T01} \cdot \text{sign} \frac{d \phi_2}{dt} + M_{BT2} \cdot \frac{d \phi_2}{dt} + + K_{P2} \cdot \phi_2; \] (7)

\[ M_{C4} = M_{T4} + M_{B4} + M_{P4} = M_{T02} \cdot \text{sign} \frac{d \phi_4}{dt} + M_{BT4} \cdot \frac{d \phi_4}{dt} + + K_{P4} \cdot \phi_4; \] (8)

\( M_{i-2} \) и \( M_{3-4} \) – elastic moments of the kinematic connection of masses, and, accordingly, determined by the expressions:

\[ M_{i-2} = h_1 \left( \frac{d \phi_2}{dt} - \frac{d \phi_1}{dt} \right) + C_{ai} \cdot (\phi_2 - \phi_1); \] (9)

\[ M_{3-4} = h_2 \left( \frac{d \phi_4}{dt} - \frac{d \phi_3}{dt} \right) + C_{a2} \cdot (\phi_4 - \phi_3); \] (10)

\( M_{hi} \) – moment from the forces of dry friction, N·m;

\( M_{hi} \) – moment from viscous friction forces, N·m;

\( M_{Pi} \) – moment of forces of positional load HM1 and HM2, N·m;

\( h_1 \) – reduced damping factor, N·m·s/rad;

\( C_{ai} \) – reduced stiffness coefficient, N·m/rad;

\( \frac{d \phi_i}{dt} \) – increment of angular coordinates HM1 and HM2 in time, rad/c.
The mathematical model of the hydraulic power subsystem of the hydraulic feed drive is obtained from the balance equations for the flow rates for the pressure head and drain lines [14,15]:

\[
Q_1 = Q_{HM1} + Q_{Y1} + Q_{P1} + Q_{CG2}; \\
Q_3 = Q_{HM1} - Q_{Y2} + Q_{P1} + Q_{CG3}.
\]  \(11\)

\[
Q_7 = Q_{HM2} - Q_{Y4} + Q_{P2} + Q_{CG7}.
\]  \(14\)

For the hydraulic power subsystem of the hydraulic drive of the main motion, the flow balance equations for the pressure head and drain lines are as follows:

\[
Q_8 = Q_{HM2} + Q_{Y3} + Q_{P2} + Q_{CG8}; \\
Q_7 = Q_{HM2} - Q_{Y4} + Q_{P2} + Q_{CG7}.
\]  \(13\)

The behavior of the hydraulic control subsystem of the feed hydraulic drive describes the flow balance equation in the control line:

\[
Q_{guk} = Q_{TH} + Q_{HD} + Q_{Y, HM} + Q_{CG,Y},
\]  \(15\)

where

\[
Q_1 = \mu_1 \cdot f_{edp1} \cdot \frac{2}{\rho} |p_1 - p_2| \cdot \text{sign}(p_1 - p_2),
\]

\[
Q_3 = \mu_3 \cdot f_{edp3} \cdot \frac{2}{\rho} |p_3 - p_{ca3}| \cdot \text{sign}(p_3 - p_{ca3}).
\]

\[
Q_8 = \mu_8 \cdot f_{edpb} \cdot \frac{2}{\rho} |p_9 - p_8| \cdot \text{sign}(p_9 - p_8),
\]

\[
Q_7 = \mu_7 \cdot f_{edp7} \cdot \frac{2}{\rho} (p_7 - p_{cl7}) - \text{flow rates through HD1 and HD2 valves for pressure and discharge lines, respectively, m}^3/\text{s};
\]

\[
Q_{dp} = \mu_{dp} \cdot f_{dp} \cdot \frac{2}{\rho} |p_y - p_6| \cdot \text{sign}(p_y - p_6) - \text{flow rates through HD1 and HD2 valves for pressure and discharge lines, respectively, m}^3/\text{s};
\]

\[
Q_{KPI} = \mu_{KPI} \cdot \pi \cdot d_{KPI} \cdot x_{KPI} \cdot \frac{2}{\rho} |p_i - p_{y1}| \cdot \text{sign}(p_i - p_{y1}) - \text{the flow rate of the working fluid through the corresponding safety valve, m}^3/\text{s};
\]

\[
Q_{guk} = \mu_{guk} \cdot \pi \cdot d_{guk} \cdot x_{guk} \cdot \frac{2}{\rho} |p_5 - p_3| \cdot \text{sign}(p_5 - p_3) - \text{working fluid flow rate GUK, m}^3/\text{s};
\]

\[
Q_{GMG} = \mu \cdot \pi \cdot d_e \cdot y(\phi) \cdot \frac{2}{\rho} |p_{GMG} - p_{cl.GMD}| \cdot \text{sign}(p_{GMG} - p_{cl.GMD}) - \text{flow rate of working fluid through GMD, m}^3/\text{s};
\]

\[y(\phi) - \text{Is the function of changing the flow area GMD, taking into account the angular speed of rotation of the hydraulic motor shaft HM2, the distance between the nozzle and the disk (modulator) GMD, m. This distance and the value of the design parameter of the section of the flow path GMD;}
\]

\[
Q_{HM1} = q_{HM1} \cdot \frac{\omega_1}{2\pi} \text{ and } Q_{HM2} = q_{HM2} \cdot \frac{\omega_2}{2\pi} - \text{the flow rate of the working fluid for the rotation of the hydraulic motor shaft HM1 and HM2, respectively, m}^3/\text{s};
\]
$$Q_{y1} = k_y \cdot p_1; \ Q_{y2} = k_y \cdot p_3 \ \text{and} \ Q_{y3} = k_y \cdot p_5; \ Q_{y4} = k_y \cdot p_7 \ - \ \text{flow rate for leaks of working fluid in the pressure head and in the drain hydraulic lines for the supply circuit and the main movement, respectively, m}^3/\text{s};$$

$$Q_{p1} = k_{n1} \cdot \sqrt{\frac{2}{\rho}} \cdot |p_2 - p_3| \cdot \text{sign}(p_2 - p_3); \ Q_{p2} = k_{n2} \cdot \sqrt{\frac{2}{\rho}} \cdot |p_8 - p_7| \cdot \text{sign}(p_8 - p_7) \ - \ \text{flow rate for overflows of working fluid for circuits with HM1 and HM2, respectively, m}^3/\text{s};$$

$$Q_{CG1} = \frac{(q / \pi) + W_{sg1}}{E_{cm}} \cdot \frac{dp}{dt} \ - \ \text{the flow rate of the working fluid to compensate for the volumetric deformation of the fluid in the lines of hydraulic motors, m}^3/\text{s};$$

$$Q_{CG,Y} = \frac{W_Y}{E_{cm}} \cdot \frac{dp}{dt} \ - \ \text{the flow rate of the working fluid to compensate for the volumetric deformation in the control line of the tool feed hydraulic motor, m}^3/\text{s};$$

$$Q_{HB} \ - \ \text{the flow rate of the working fluid when moving the movable element of the accumulator, m}^3/\text{s};$$

$$Q_{Y,HP} \ - \ \text{the flow rate of the working fluid when moving the movable element of the hydraulic pump control unit, m}^3/\text{s};$$

$$q_{HP} = q_0 + x_{HP} \cdot k_{HP} \ - \ \text{pump displacement HP1, m}^3/\text{rev};$$

$$k_{HP} = \frac{q_{HP,max} - q_0}{x_{HP,max}} \ - \ \text{coefficient taking into account the kinematics of the mechanism for changing the working volume of the pump HP1, m}^3/\text{rev};$$

$$\mu_i \ - \ \text{flow rate coefficient of hydraulic devices;}$$

$$\chi_i \ - \ \text{movement of elements of hydraulic devices, m;}$$

$$p_i \ - \ \text{pressure at the corresponding point of the hydraulic system, Pa;}$$

$$\omega_1 \ \text{and} \ \omega_3 \ - \ \text{angular speed of rotation of the hydraulic motor shaft HM1 and HM2, respectively, rad/s;}$$

$$\rho \ - \ \text{working fluid density, kg/m}^3;$$

$$k_{pi} \ - \ \text{leakage coefficient, m}^2;$$

$$E_{cm} \ - \ \text{bulk modulus of the working fluid, taking into account the dissolved air, Pa;}$$

$$W_i \ - \ \text{volume of the i-th section of the hydraulic line, m}^3;$$

$$\frac{dp}{dt} \ - \ \text{pressure increment in the concentrated volumes of the hydraulic system, Pa/s.}$$

In the general case, the pressure in all concentrated volumes of the hydraulic system is determined from the flow balance equation:

$$\Sigma Q_{ent} - \Sigma Q_{exit} + Q_{cg} = 0, \quad (16)$$

where

- $\Sigma Q_{ent}$ = the amount of flow rates from fluid inflows into the concentrated volume, m$^3$/s;
- $\Sigma Q_{exit}$ = the amount of costs from the outflow of liquid from the concentrated volume, m$^3$/s;
- $Q_{cg} = \frac{dW}{dt}$ = liquid flow rate during compression (expansion) of liquid, m$^3$/s.

Considering that the compressibility of a fluid is characterized by the compressibility coefficient

$$\beta = -\frac{1}{W} \frac{dW}{dp},$$

and replacing it with the more commonly used bulk modulus of elasticity $E=1/\beta$, the change in pressure over time is determined from the flow balance equation:

$$\frac{dp}{dt} = \left(\Sigma Q_{ent} - \Sigma Q_{exit}\right) \frac{E_{cm}}{W_i}. \quad (17)$$
Then the mathematical model of the hydraulic subsystem will be represented by the following system of equations:

\[
\begin{align*}
\frac{dp_1}{dt} &= (Q_{HP1} - Q_{kp1} - Q_{fedp1}) \cdot \frac{E_{v1}}{W_{eg1}}, \\
\frac{dp_2}{dt} &= (Q_{fedp1} - Q_{HM1} - Q_{fedp2}) \cdot \frac{E_{v2}}{W_{eg2}}, \\
\frac{dp_3}{dt} &= (-Q_{fedp3} + Q_{HM1} + Q_{fedp2}) \cdot \frac{E_{v3}}{W_{eg3}}, \\
\frac{dp_4}{dt} &= (Q_{n2} - Q_{xp3} - Q_{fedp5}) \cdot \frac{E_{v4}}{W_{eg4}}, \\
\frac{dp_5}{dt} &= (Q_{n3} - Q_{xp4} - Q_{GUK}) \cdot \frac{E_{v5}}{W_{eg5}}, \\
\frac{dp_6}{dt} &= (Q_{fedp} - Q_{xp2}) \cdot \frac{E_{v6}}{W_{eg6}}, \\
\frac{dp_7}{dt} &= (Q_{fedp} + Q_{HM2} - Q_{fedp7}) \cdot \frac{E_{v7}}{W_{eg7}}, \\
\frac{dp_8}{dt} &= (Q_{fedp8} + Q_{HM2} - Q_{fedp6}) \cdot \frac{E_{v8}}{W_{eg8}}, \\
\frac{dp_9}{dt} &= (Q_{n4} - Q_{xp5} - Q_{fedp9}) \cdot \frac{E_{v9}}{W_{eg9}}, \\
\frac{dp_{GMD}}{dt} &= (Q_{GUK} - Q_{HB} - Q_{fedp} - Q_{Y,HP}) \cdot \frac{E_{v10}}{W_{eg10}}, \\
\frac{dp_{GUK}}{dt} &= (Q_{fedp5} + Q_{GMD} - Q_{fedp4}) \cdot \frac{E_{v11}}{W_{eg11}, GMD}, \\
\end{align*}
\]

(18)

The movement of the piston \( x_H \), which controls the working volume of the pump \( HP1 \), is described by the equation of motion [15]:

\[
m_n \cdot \frac{d^2 x_H}{dt^2} = p_y \cdot f_n - F_T \cdot \text{sign} \left( \frac{dx_H}{dt} \right) - k_B \cdot \frac{dx_H}{dt} - c_{HM} \cdot (x_{a,H} + x_H),
\]

(19)

where \( m_n \) – reduced mass of the piston controlling the displacement of the pump \( HP1 \), \( kg \);

\( F_T \) – friction force, \( N \);

\( p_y \) – control pressure, \( Pa \);

\( k_B \) – viscous friction coefficient, \( N \cdot s/m \);

\( d \) – piston diameter, \( m \);

\( c_{HM} \) – spring rate, \( N/m \);

\( f_p \) – piston area of the control hydraulic cylinder, \( m^2 \).

Piston movement \( x_{guk} \), controlling the flow path of the GUK valve is described by the equation of motion:

\[
m_{guk} \cdot \frac{d^2 x_{guk}}{dt^2} = p_{guk} \cdot f_{n,guk} - F_{T,guk} \cdot \text{sign} \left( \frac{dx_{guk}}{dt} \right) - k_{B,guk} \cdot \frac{dx_{guk}}{dt} - c_{np,guk} \cdot (x_{0,guk} + x_{guk}) - F_{gd},
\]

(20)

where \( m_{guk} \) – valve spool weight GUK, \( kg \);

\( F_{T,guk} \) – frictional force in the movable joint of the valve GUK, \( N \);

\( p_{guk} \) – valve control pressure GUK, \( Pa \);

\( k_{B,guk} \) – the coefficient of viscous friction in the moving joint of the valve GUK, \( N \cdot s/m \);
The movement of the spool of the corresponding safety valve $x_{KPI}$ is described by the equation of motion:

$$\frac{d^2x_{KPI}}{dt^2} = \frac{p_i}{m_{KPI}} - \frac{f_{KPI}}{m_{KPI}} \cdot \text{sign}\left(\frac{dx_{KPI}}{dt}\right) - \frac{k_B}{m_{KPI}} \cdot \frac{dx_{KPI}}{dt} - \frac{c_{KPI}}{m_{KPI}} \cdot (x_{0,KPI} + x_{KPI}) - \frac{F_{gd,KPI}}{m_{KPI}}, \quad (21)$$

where

- $m_{KPI}$ – spool weight of the corresponding valve, kg;
- $p_i$ – control pressure of the corresponding valve, Pa;
- $k_B$ – coefficient of viscous friction in the movable joint of the corresponding valve, N∙s/m;
- $c_{KPI}$ – spring rate of the corresponding valve, N/m;
- $f_{KPI}$ – spool end area of the corresponding valve, m$^2$;
- $F_{gd,KPI}$ – hydrodynamic force, N.

4. Results and discussing

Simulation of the dynamic hydraulic drive system was carried out in the MatLab Simulink program. Due to the complexity of the implementation and solution of the mathematical model of a hydromechanical two-mass system of each circuit, it was investigated by successively complicating the basic single-mass model, with the adjustment of parameters at each stage of modeling [17, 18, 19].

The computational experiment was carried out with parameters identical to the parameters mobile drilling machine. The experiment was carried out at different loading cycles of the HMS drive (different speeds of rotation of the hydraulic motor shaft of the rotation drive). So, in fig. Figures 3 - 7 show oscillograms of the process of drilling soils with different strengths 1–3-5–7–9 (load) and, accordingly, the changing angular speed of rotation of the hydraulic motor shaft of the tool rotation drive (the time interval from 0 to 2 s corresponds to the angular speed of 125 rad/s; from 2 to 4 s - 111 rad/s; from 4 to 6 s - 45 rad/s; from 6 to 8 s - 111 rad/s; from 8 to 10 s - 67 rad/s. [2,6].

Oscillograms reflect the nature of the change in the main parameters of the hydraulic drive feed depending on the tool loading cycle. When the frequency of rotation of the shaft of the hydraulic motor HM2 changes under the influence of the changing force on the tool, both the frequency and the amplitude of the fluctuations in the pressure of the working fluid at the output of the HMD change [14,15,20].

The lower the rotational speed of the hydraulic motor of the HM2 tool rotation drive, the lower the frequency of pressure fluctuations and the higher the amplitude of the HMD. The amplitude of the oscillation of the control pressure of the main control system affects the amplitude of oscillation of its spool. The higher the amplitude of the control pressure fluctuation, the greater the average value of the displacement of the control valve spool.
Figure 3. Changing the control pressure of the tool feed hydraulic motor in the working cycle

As a result, with a decrease in the rotational speed of the HMD shaft, the average opening area of the main control unit increases, which leads to a greater increase in the volumes of the working fluid in the hydraulic pump control line per unit time $Q_{guk}$, which, in turn, leads to an increase in the control pressure (Fig. 3) [14, 15, 20].

And accordingly, the higher the control pressure of the working volume of the hydraulic pump, the more its cradle displaces and the less its working one (Fig. 4) [1, 14, 35].

Figure 4. Changing the movement of the cradle of the hydraulic motor regulator block tool feed in the working cycle

A decrease in the working volume of the hydraulic pump with a constant rotational speed of the drive shaft leads to a decrease in the angular speed of rotation of the hydraulic motor shaft (see Fig. 5) and, accordingly, to a decrease in the tool feed rate (see Fig. 6).
Figure 5. Changing the angular speed of rotation of the hydraulic motor shaft tool feed drive in the working cycle: ––––– throttle regulation; ——— volumetric regulation

The oscillograms of the angular speed of rotation of the feed hydraulic motor shaft and the output link of the transmission mechanism (see Fig. 5) show what effect the kinematic parameters of the two-mass drive system have on them. The higher the load on the feed drive, the greater the difference between the angular speed of rotation of the feed motor shaft and the output link of the transmission mechanism.

Figure 6. Changing the feed rate of the tool in the working cycle: –—— throttle regulation; ——— volumetric regulation

On the graphs of the tool feed rate (see Fig. 6), the dashed line shows the oscillograms of the tool feed rate with the throttle control method, which was previously used in the MBM. It can be seen from the graph that the range of change in the feed rate with the throttle control method is much smaller, which can lead to tool breakages.

Also, the power consumed by the pump with the throttle control method is much higher (see Fig. 7) and in heavy drilling modes is 2 times higher than the pump power with volumetric control of the feed rate [14,15,20].
Figure 7. Changing the power of the tool feed drive in the working cycle: – – – throttle regulation; –––––– volumetric regulation

Oscillography of the test full-scale experiment was carried out using the PowerGraph program to the DAC/ADC board (E20-10D).

The computational experiment was carried out in the program "Matlab 2017" by various numerical methods of integration (Euler and Runge-Kutta, etc.) [19, 20]. At the same time, similar results were obtained, which confirms the adequacy of the mathematical model.

5. Conclusion

The discrepancies in the drive parameters at the control points during full-scale and computational experiments do not exceed 20%, which is due to the assumptions made in mathematical modeling and the discrepancies between the initial data of the model and the real parameters of the drive. This makes it possible to use the research results for the development and design of hydromechanical systems, hydraulic drilling machines, significantly reducing the time and money spent.

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