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Study on oil pilot circuit of adaptive hydraulic drive of tool advance in mobile drilling machine [*]

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Исследование гидравлического контура управления адаптивного гидропривода подачи инструмента мобильной буровой машины [***]

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Introduction. An adaptive hydraulic drive of the tool advance in a mobile drilling machine is studied on the example of the URB-2.5 installation. A typical technological cycle of the mobile drilling machine is considered; the performance criteria are defined. An original design of the adaptive hydraulic drive is proposed on the basis of the analysis. Adaptation of the hydraulic drive of the tool advance is carried out using an adjustable volumetric hydraulic motor with a hydraulic control circuit under discontinuous loads on the tool during the drilling process.
Materials and Methods. Through a preliminary computational experiment in the Matlab Simulink program, the following parameters of the control loop devices were determined: a hydromechanical sensor and a hydraulically controlled valve, on the basis of which the experimental setup was implemented. The performed multifactor experiment allowed identifying the processes in the original hydraulic control circuit of the hydraulic motor under various modes of tool loading.
Research Results. The kinematic and power characteristics of the hydromechanical system of a mobile drilling rig, the hydraulic control effect on the settings of the hydraulic control circuit devices were obtained and determined. The results enabled to specify the rational ranges of the hydromechanical system operation for a typical work cycle.
Discussion and Conclusions. The results obtained can be used to create hydraulic systems of new drilling machines with various characteristics. The application of the developed techniques of research and processing of their results will reduce the time and costs involved in designing adaptive hydraulic systems for mobile technological machines, creating prototypes and conducting commissioning procedures.

Введение. Статья посвящена исследованию адаптивного гидропривода подачи инструмента мобильной буровой машины на примере установки УРБ-2,5. Рассмотрен типовой технологический цикл мобильной буровой машины, определены критерии функционирования. По результатам анализа предложено оригинальное схемотехническое решение адаптивного гидропривода. Адаптации гидропривода подачи инструмента осуществляется при помощи регулируемого объемного гидродвигателя с контуром гидравлического управления при изменяющейся нагрузке на инструменте в процессе бурения.
Материалы и методы. Предварительным вычислительным экспериментом в программе Matlab Simulink определены параметры устройств контура управления: гидромеханического датчика и гидроуправлямого клапана, на основе которых реализована экспериментальная установка. Выполненный многофакторный эксперимент позволил идентифицировать процессы в оригинальном гидравлическом контуре управления гидромотором при различных режимах нагружения инструмента.
Результаты исследования. Получены и определены кинематические и силовые характеристики гидромеханической системы мобильной буровой установки, влияние гидравлического управления на параметры настройки устройств контура. Результаты позволили определить рациональные диапазоны функционирования гидромеханической системы для типового рабочего цикла.
Обсуждение и заключения. Полученные результаты могут быть использованы при создании гидросистем новых буровых машин с различными характеристиками. Использование разработанных методик исследования и обработки их результатов позволит сократить затраты времени и средств при проектировании адаптивных гидросистем мобильных технологических машин, создании

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опытных образцов и проведении пуско-наладочных работ.

Keywords: adaptive hydraulic drive, mobile drilling unit, drilling technological cycle, hydraulic control circuit, hydromechanical sensor, kinematic and power characteristics.

Ключевые слова: адаптивный гидропривод, мобильная буровая установка, технологический цикл бурения, контур гидравлического управления, гидромеханический датчик, кинематические и силовые характеристики.

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Introduction. Dynamic development of natural resources requires the improvement of existing and the creation of new automated complexes of processing equipment for drilling production with improved mechanical and energy characteristics. Analysis of the known circuit design solutions of hydromechanical systems (HMS) of mobile drilling machines (MDM) identified the key feature of their construction — a multi-engine system, in which it is important to consider the effect of the tool feed drive and the main motion drive during the technological cycle [1–4]. At this, the quality, productivity and safety of operation depend much on how the kinematic and force parameters (V, M, etc.) are matched under changing the tool loads [4, 5]. In view of the above, the work objective was to increase the efficiency of the hydromechanical system of MDM working motions through developing and studying its hydraulic control circuit.

Performance Criteria. In the drilling process, an important criterion is the fulfillment of the basic production requirement — operation matching of the main motion and feed drive, which would ensure a stable tool advance revolution [6].

Using the basic laws of similarity of various process cycles, it is easy to apply the above reasoning to the technique of drilling various wells [7, 8].

The performance ($П_О$) is determined as follows:

$$П_О = 1/Т_{ОЦ} \tag{1}$$

$$П_{СЦ} = П_О \cdot k_ц\, k_ц = L_{СК} / L_{ОЦ} \tag{2}$$

$$Т_{ОЦ} = Т_{МАШ} + Т_{ВС} + Т_{ПЗ} + Т_{ОБСЛ} + Т_{РЕМ}. \tag{3}$$

$$Т_{МАШ} = L_{ОЦ} / V_n;\ V_n = S_o \cdot n_И \tag{4}$$

$$n_И = 1000 \cdot V_{БУР} / 3{,}14 \cdot D_И. \tag{5}$$

$Т_{ОЦ}$ is reference cycle time; $Т_{ПЗ}$ is time of adjustment of the next reference operation cycle (increasing the tool length); $Т_{МАШ}$ is machine time spent on the cutting operation (drilling) when moving the tool by the value of $L_{ОЦ}$ with V_n speed.

When the condition (S_0 = const) is fulfilled, the tool life corresponds to $Т_Н = [Т_{ИН}]$ standard with $D_И$ diameter and $n_И$ frequency of tool rotation. In this case, the number of tool changes throughout $L_{СК} = L_{ОЦ} \cdot k_ц$ drilling depth is reduced.

$Т_{ВС}$ is time spent on installation, commissioning operations, and tool change. It is regulated by the equipment type [7];

$Т_{ОБСЛ}$ is scheduled time of change (replacement) of the tool after the development of its technological stability ($Т_Н$);

$Т_{РЕМ}$ is time for eliminating failures, it is reduced with the increased reliability; $V_{БУР}$, S_o are standards for drilling soils of various categories by σ_i or from the practice of drilling in each region [5].

The tool life and capacity depend directly on the stability of $S_o(t)$ value (working process flow per revolution) [7, 9]. This is achieved in case, when $S_М(t)$ minute feed rate performed by the drive of the MDM dependent feed, decreases synchronously with reducing the tool n(t) rotational speed. When the elastic deformations are not considered in the kinematic tool feed chain, then the feed amount is determined as follows:

$$S_0(t) = \frac{2\pi \cdot v(t)}{\omega(t)} \tag{6}$$

$$S_{\text{M}}(t) = \frac{v(t)}{60} \tag{7}$$

$$\omega(t) = 2\pi \cdot n(t) \tag{8}$$

where $v(t)$ is linear tool advance speed, m/s; $\omega(t) = 2\pi \cdot n(t)$ are angular velocities of tool rotation, rad/s; $n(t)$ is tool rotation frequency, rev/s.

As is well known, the possibilities of rigid stabilization of each of the working motions under the conditions of hydraulic drive application are limited. In the technological machines of this type, the task of adapting a volumetric hydraulic drive is not solved automatically, but is done manually by the operator [3, 10].

Circuit Solution. On the basis of the previously proposed structural scheme [11], as well as the analysis of typical MDM operation cycles, the MDM URB-2.5 schematic hybrid (Fig. 1) is proposed. It considers the behavior and composition of its mechanical subsystem.

The machine power system consists of hydraulic fixed axial-piston pumps (H1, H4), installed on the chassis transfer gear through power take-offs (PTO). The PTO control is electro-pneumatic and it is performed through switches installed in the cab of the auto chassis [10, 12].

The hydraulic control circuit (HCC) of the installation receives hydraulic energy from H2 two-stage pump driven by an electric motor. The electric motor receives power from the auto chassis generator. The first section of H2 directs the hydraulic energy to GMD1 sensor, the second one – to the hydraulically operated valve (HOV).

The operation of both HCC circuits occurs off-load, so overheating of H2 two-stage pump motor is excluded. Each two-stage pump has a safety relief valve with electrical control (KP2, KP3). Pressure is controlled by MN3 and MN5 gauges respectively. HCC has its own closed hydraulic tank (B2) [13]. The main hydraulic tank is B1 tank equipped with F1, F2, F3 drain filters, TO1 and TO2 heat exchangers.

The MDM hydraulic system (HS) is divided into two large circuits. The auxiliary circuit includes an outrigger drive (ГЦ1, ГЦ2), a mast lift-lower drive (ГЦ 3), and a winch drive (ГМ 3). The main contour includes a drive of the tool main motion (rotary table) and a feed drive (Fig. 1).

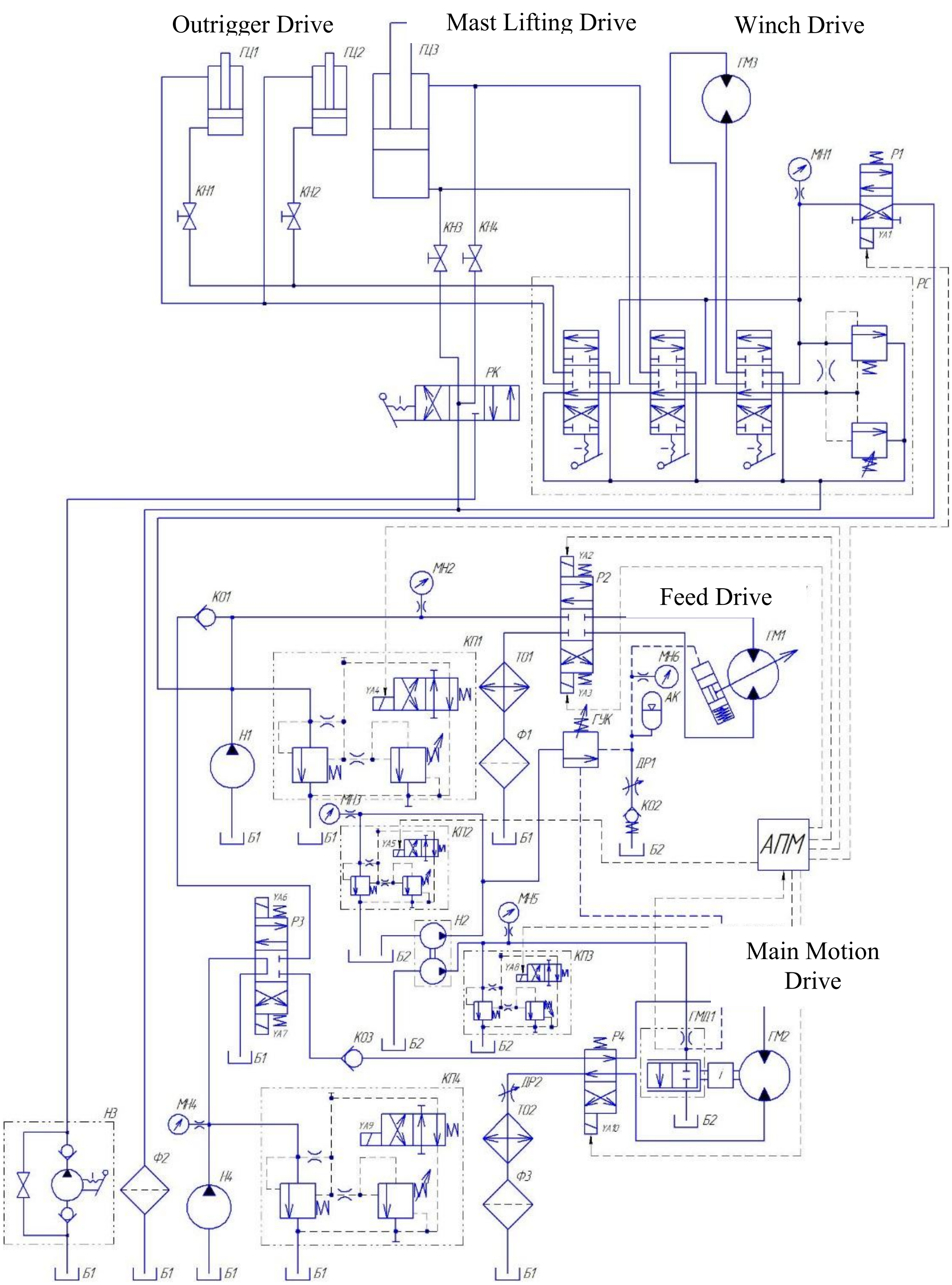

Fig. 1. Hydraulic schematic diagram of MDM URB-2.5

The tool feed drive entering the main circuit receives hydraulic energy from the hydraulic pump (H1). The pump is equipped with an electrically-controlled relief valve (КП1). Operating pressure is monitored by the pressure gauge (MH2). Reverse movement, stop are provided by a three-position electrically-controlled valve (P2) with a closed center. The hydromotor (ГМ1) of the feed drive, adjustable through mechanical transmission, provides a longitudinal feed of the drilling tool. The heat exchanger (TO1) and filter (Ф1) provide filtration and conditioning of the working fluid.

The pump (H4) is rotated in the same way as the pump (H1) (Fig. 1). The valve (P3) initiates start, stop of the main drive or feed rate increase under the rapid traverse of the feed drive. In this case, the upper position of the valve is turned on, and the flow of the pump (H4) enters the drive supply circuit through the check valve (KO1), and is summed with the flow of the pump (H1) providing a quick lowering or lifting of the tool from the well. When the pump (H4) operates on the main drive (P3 in the lower position), the check valve (KO1) prevents the working fluid from being drained out of the supply circuit into the tank (B1) [13].

A similar function is performed by the check valve (КО3) when the pump is operated on the feed drive. The valve (Р4) provides reverse of the drive main motion. Using the choke (ДР2), the tool rotation speed is adjusted (at idle).

The heat exchanger (ТО2) and the drain filter (Ф3) provide filtration and conditioning of the working fluid of the rotational drive circuit.

The HCC circuit includes two original devices – a hydraulically controlled valve (HOV) (4) and a hydraulic multiparameter sensor (HMS) (5) (Fig. 2) [14]. Under the rapid traverse of the feed drive described above, the pump (Н2) (Fig. 1) is automatically shut off by the control system (АПМ).

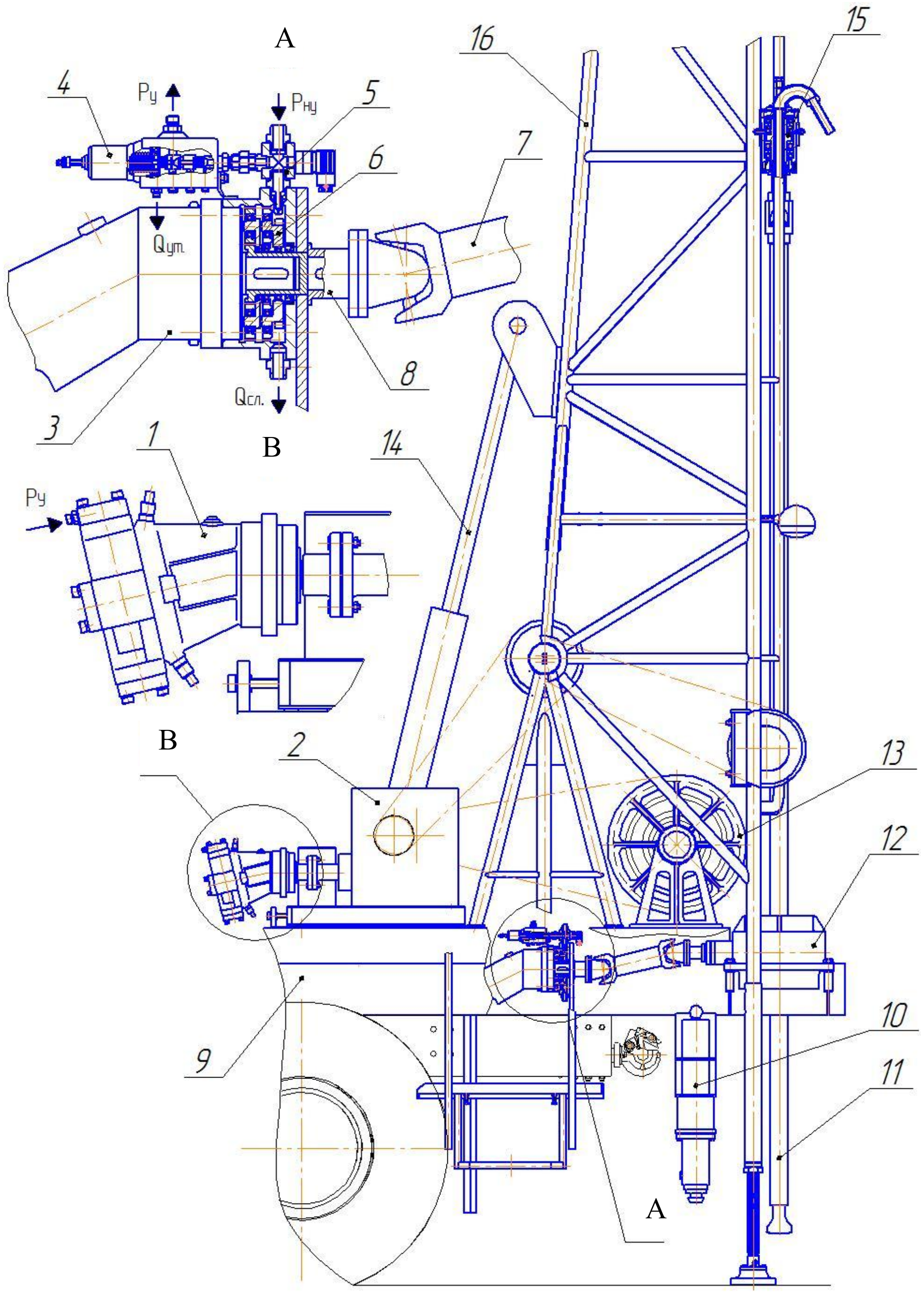

Fig. 2. MDM general view: 1 is adjustable hydraulic motor; 2 is tool feed drive; 3 is uncontrolled hydraulic motor; 4 is HOV valve; 5 is HMS sensor; 6 is gearbox; 7 is driveshaft; 8 is coupling; 9 is auto chassis; 10 is system of outriggers; 11 is tool; 12 is main motion drive; 13 is double drum winch; 14 is hydraulic cylinder of mast lifting (lowering); 15 is swivel; 16 is mast.

The HCC principle of operation (Fig. 2) as part of the MDM hydraulic system is as follows: the shaft (HMS) (5) is mechanically connected to the shaft (ГМ2) through the gearbox (6). Pressure fluctuation at the HMS input (5) is transmitted to the control input of the HOV (4) which is adjusted – by selecting springs – to the mode of operation whereby, at its output, the average operating pressure proportional to the oscillation amplitude and, accordingly, to the rotational speed of the hydromotor shaft (ГМ1) is formed.

With resistance moment increment on the hydromotor shaft (ГМ2), the shaft rotation speed and the oscillation frequency at the HMD input decrease, and the pressure amplitude increases. As it increases, the average value of opening the HOV valve, which supplies a greater amount of fluid from the H2 to the gearbox (ГМ1) of the feed drive, increases. The control pressure (Py) at the HOV input increases, which provides an increase in the hydraulic motor capacity and reduces the feed speed. The mechanism of the HCC contour as a component of the HMS with a dependent tool advance, as well as the operation principle of the (АПМ) unit, is considered in [15].

The accumulator (АК) smoothes the pressure control pulsation (Py) at the hydraulic motor (ГМ1) input control. Using the setting of the chokes (ДР1, ДР2) and the selection of the pressure valve (КО3), the limiting values of the tool feed rate and rotation speed are adjusted, and the required functional relationship between the working motion speeds is formed.

The original HOV differs in function of continuous regulation of the valve flow section. Therefore, it is possible to control the performance of the hydromotor at all speeds.

Experimental Study. To identify the HMS parameters operating under the specified conditions, special bench equipment and accessories are developed [13]. The hydromotor flow rate was determined using a turbine sensor-flow meter connected through the converter board (ЦАП-АЦП). The adjustment ranges of the parameters under the study are shown in Table 1.

Table 1

Parameter ranges under HMS study

No.	Parameter	Designation	Range	Unit of measurement	Control device
1	Nozzle diameter	$d_{сп}$	0.5–1.2	mm	Plug gauge
2	Choke flange	$d_{др}$	0.8–2	mm	Plug gauge
3	Gap between nozzle and modulator	$y_з$	0.2–1	mm	Dial gauge
4	RPM	$n_{гмд}$	5–60	rpm	Rotation velocity sensor
5	Flow through HMS	$Q_{гмд}$	0.5–20	l/min	Flow gage, gage tank
6	Pressure in HMS	$p_{гмд}$	0.5–5	MPa	Pressure gauge
7	Characteristic design factor of HMS flow section	$x_д$	2–8	mm	Slide gage

The traverse speed and acceleration of the shaft (ГМ) (feed drive) was determined by sequential differentiation of motion over time using the following formulas implemented by post-processing of data in the *PowerGraph* program [8, 11, 16]:

$$\omega(t) = \frac{d\varphi}{dt}, \tag{9}$$

$$\varepsilon(t) = \frac{d\omega}{dt}, \tag{10}$$

where ω is angular velocity, rad/s; ε is angular acceleration of (ГМ), rad/s^2.

The results of the obtained experimental data processing through the known methods [17, 18] are presented in Fig. 3–9.

The graph in Fig. 3 explains the pressure fluctuation amplitude response of the HMS under a change in the hydromotor (ГМ1) speed variation within the range of 45–125 rad/s. The data were obtained when testing nozzles with the diameters: $d_{сп}$ = 2, 4, 6 mm.

As a result, the dependence of the pressure change was almost linear in nature, and it fell as the speed increased, which was associated with the non-stationary mode of the working fluid outflow through the flapper-nozzle unit.

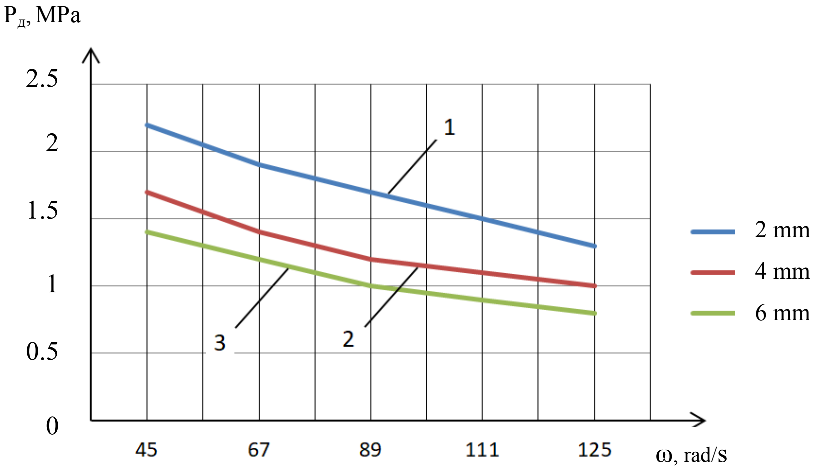

Fig.3. Dependence of pressure fluctuation value ($P_Д$) on ω speed of rotation of hydromotor (ГМ1), approximation: 1 is $P_Д$=0.14ω^2-3ω+24.8; 2 is $P_Д$=0.36ω^2-3.84ω+20.4; 3 is $P_Д$=0.21ω^2-2.8ω+16.6.

The results presented in Fig. 4 show that with an increase in the characteristic structural parameter of the flow section (ГМД $X_Д$) from 2 to 8 mm, the control pressure level changes from 1.9 to 1.4 MPa. This enables to make recommendations for further optimization of the HMS flow path [19], in particular, for some increase in (ХД). However, its further increase is impractical because it increases the geometric dimensions of the modulator disk.

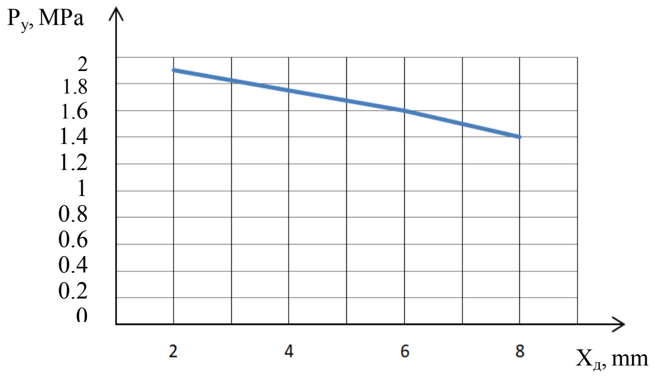

Fig. 4. Dependence control pressure (P_y) variation on effect of characteristic structural parameter of cross section of flow part (ГМД $X_Д$), approximation: $P_У$=1.65$X_Д$+20.75

Fig. 5 shows the effect of the design features of ($X_Д$) on the maximum pressure amplitude ($P_{дmax}$) used for the hydraulic control circuit during advance to the HOV.

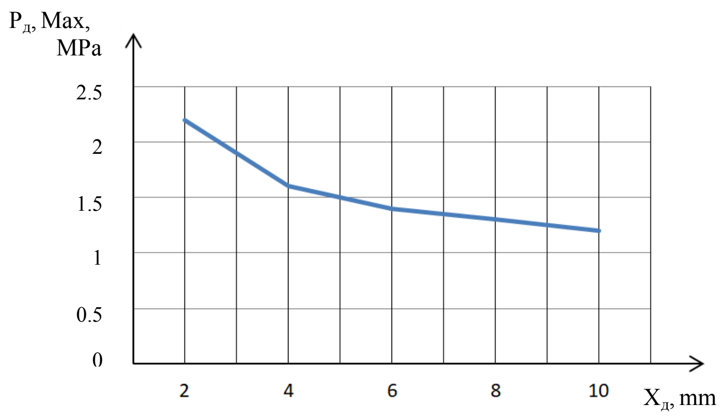

Fig.5. Dependence of change in maximum amplitude of control pressure ($P_{Д.MAX}$) on effect of characteristic design parameter of flow section of (ГМД $X_Д$), approximation: $P_{ДMAX}$=21.46$X_Д^{-0,372}$

During identification, an important step was the determination in the HCC circuit: HMS – HOV, the degree of impact of each control component setting. Thus, the HOV tuning property, under the control loop performance, is its spring constant ($C_{пр}$), the selection of which significantly affects the sensitivity of the control subsystem [20].

The characteristic built in Fig. 6 and approximated by the results of the experiment, is linear, and explains the magnitude of the maximum control pressure drop from 1.7 to 1.4 MPa under changing the HOV spring force.

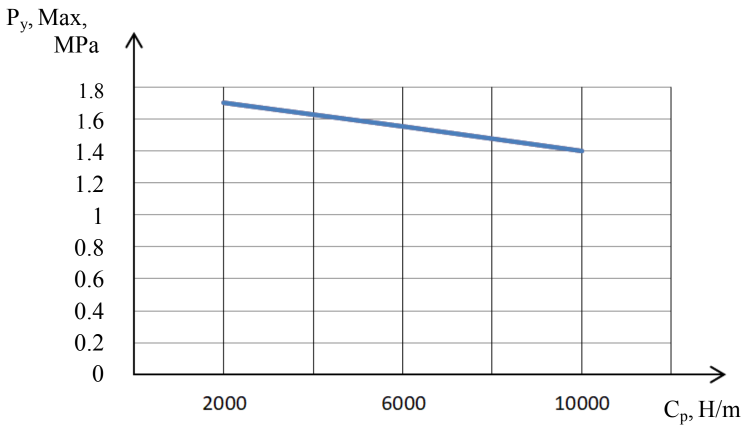

Fig. 6. Dependence of maximum control pressure ($P_{y.max}$) variation on magnitude of HOV spring force, approximation: $P_{YMAX}=1.5C_P+18.5$

The second tuning element in the circuit with the HOV is an adjustable choke in the shunt line (Fig. 1). Its setting parameter is the flow area regulated within the range from 2.5 • 10^{-7} to 7.5 • 10^{-7} m^2. The results of the experiment are shown in Fig. 7.

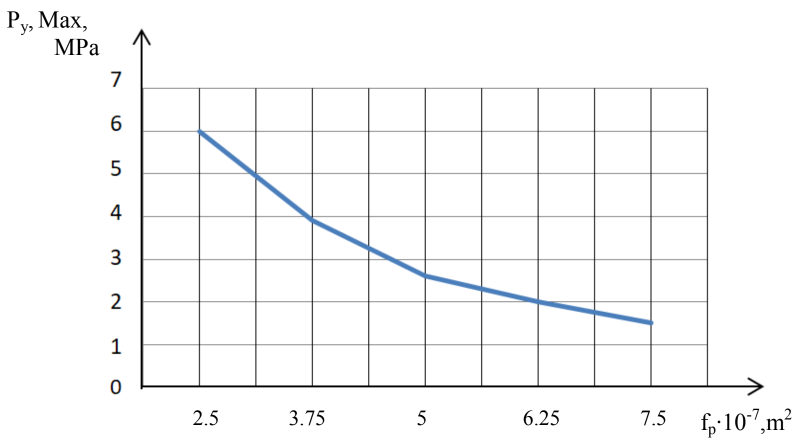

Fig. 7. Dependence of maximum control pressure ($P_{y.max}$) variation on flow area of choke (ДР1), approximation: $P_{YMAX}=1.5f_{ДР}^2-27.6 \cdot f_{ДР}+84.2$

Four operation modes of the tool-drive motor (hydraulic motor (ГМ 1)) under relay loading, and, accordingly, under changing ω angular velocity by: 22 rad/s; 49 rad/s; 71 rad/s and 85 rad/s, are studied. The dependence obtained from the experiment is shown in Fig. 8.

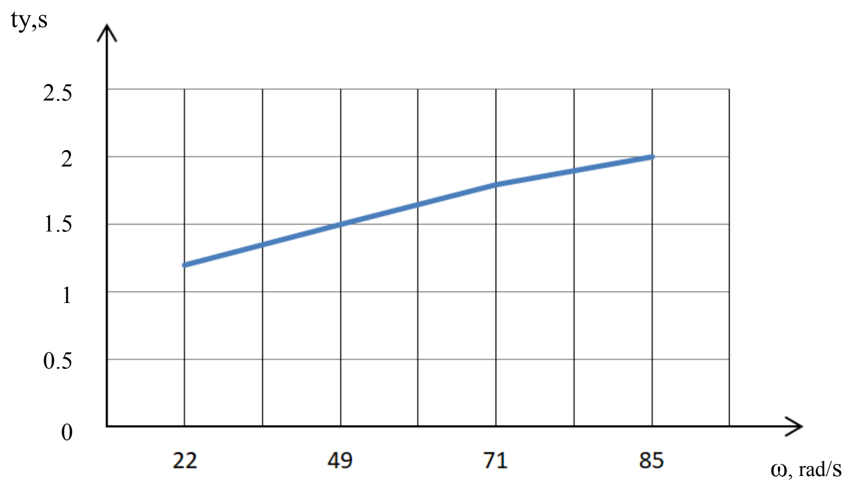

Fig. 8. t_y dependence of HCC response time on ω angular rotation velocity of hydromotor (ГМ1) shaft, approximation:
t_y=-0.002ω^2-0.4ω+0.83

Since HCC is a part of the MDM hydraulic system, its length has a special impact on the quality and time of transient processes [16]. In order to determine the degree of impact on the above parameters, the pressure variation time in the control line with varying of its volume was studied (Fig. 9).

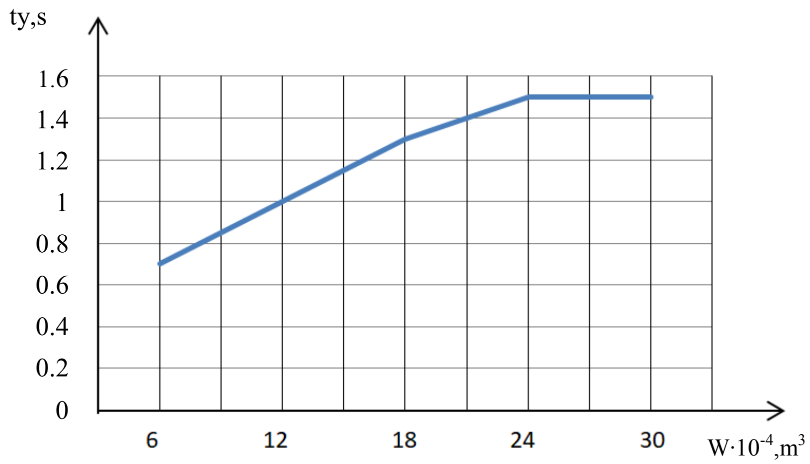

Fig. 9. t_y dependence of HCC response time on W volume in control flow line, approximation:
t_y=-0.02ω^3+0.1ω^2+0.12ω+0.5

The results enable to determine the critical value of the flow line volume: W = 24·10^{-4} м^3, over which the HCC response time does not exceed 1.5 s; that is due to wave processes taking place in the pipeline, as well as to the parameters of the lines themselves [20].

Conclusions. As a result of the research, the requirements were developed, and the HMS generalized structure of the MDM working motions was proposed, which made it possible to increase the efficiency of drilling production through introducing an internal kinematic hydromechanical connection between the main motion and the tool feed movement.

A hydraulic control circuit is developed and implemented on the basis of a multi-parameter sensor and a hydraulically controlled valve, which ensures the coordination of working motions under the discontinuous process duty. Considers the characteristic features of the drilling process, the configuration of the reference operating cycle, which allowed for the development of an automated system for driving work motions, is proposed.

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