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CONTROL SYSTEM FOR DRILLING RIG MACHINE ELECTROHYDRAULIC FEED DRIVE

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The objective of research is to develop a scientifically based approach to the synthesis of a positional control system for the hydraulic drive of a roller cone drilling machine, which ensures increased positioning accuracy, stability of transient processes, and drive efficiency under variable loads typical of the drilling process.

The study has been conducted based on: the application of differential equations of hydraulic drive dynamics (equations describing the movement of the executive hydraulic cylinder, pressure dynamics in cavities, working fluid flow, and elastic properties of the hydraulic line are used); linearization of nonlinear equations (for stability analysis and controller synthesis, the hydraulic system is reduced to a linearized model in the vicinity of the operating point); modeling using transfer functions and block diagrams; Laplace transforms are used in the formation of drive transfer functions and controllers for system analysis in the frequency and time domains; methods of automatic control theory (stability analysis; controller structure selection; synthesis of P-, PI-, PID-controllers for positional control) were applied; optimization synthesis methods for calculating controller parameters that allow setting the desired eigenvalues of the closed system and ensuring the required positioning accuracy of the drive; modeling in MATLAB/Simulink (used to verify the obtained mathematical model and check the operation of the synthesized control algorithm).

Theoretical and experimental research yielded the following scientific and practical results: a mathematical model of the hydraulic drive for a roller cone drilling machine was developed, considering the nonlinear characteristics of hydraulic equipment, the compressibility of the working fluid, hydraulic losses, and the dynamics of the actuating hydraulic cylinder. The model allowed us to investigate the influence of system parameters on positioning accuracy; we analyzed the dynamic properties of the drive and identified the main factors that limit the accuracy and speed of the position control system, in particular: hydraulic delays, force disturbances from the drilling process, and pressure fluctuations in the hydraulic lines; a synthesis of the positional control system was carried out using methods of automatic control theory; based on a linearized model, the choice of the regulator structure was justified and its optimal parameters were determined; an algorithm for positional feed control was proposed, which ensures stable transient processes and high accuracy of the specified position of the actuator under variable loads; a structural diagram of the control system was developed, including a feedback module, corrective links, and an executive hydraulic drive, and its operability was confirmed by modeling; the system was modeled in MATLAB/Simulink, the results of which showed: a reduction in drive positioning error improvement of system stability to disturbances; a reduction in the transition process time; a reduction in feed fluctuations during drilling.

Key words: *electrohydraulic feed drive, position control, control system synthesis, mathematical model, roller cone drilling, drive dynamics, position controller, transient processes.*

Хілов Віктор, Мельничук Аліна, Тропанець Андрій. Система керування електрогідравлічним приводом подачі бурової машини

Метою дослідження є науково обґрунтований підхід до синтезу позиційної системи управління гідравлічним приводом подачі станка шарошечного буріння, який забезпечує підвищення точності позиціонування, стабільність перехідних процесів та ефективність роботи приводу в умовах змінних навантажень, властивих технологічному процесу буріння.

Дослідження здійснено на підставі застосування диференціальних рівняння динаміки гідравлічного привода (використовуються рівняння, що описують рух виконавчого гідроциліндра, динаміку тиску в порожнинах, витрату робочої рідини та пружні властивості гідролінії); лінеаризації нелінійних рівнянь (для аналізу стійкості та синтезу регулятора гідравлічна система наводиться до лінеаризованої моделі в околицях робочої точки); моделювання із застосуванням передавальних функцій та структурних схем;

застосовуються перетворення Лапласа під час формування передавальних функції приводу та регуляторів для аналізу системи в частотній та часовій областях; методи теорії автоматичного керування (аналіз стійкості; вибір структури регулятора; синтез П-, ПІ-, ПІД-регуляторів за позиційного регулювання); оптимізаційні методи синтезу для розрахунку параметрів регулятора, що дають можливість задати бажані власні значення замкнутої системи та забезпечити необхідну точність позиціонування приводу. моделювання в MATLAB/Simulink (застосовується для верифікації отриманої математичної моделі та перевірки роботи синтезованого алгоритму керування).

За результатами виконаних теоретичних та експериментальних досліджень отримано такі наукові та практичні результати: розроблено математичну модель гідравлічного привода подачі станка шарошкового буріння, що враховує нелінійні характеристики гідроапаратури, стисливість робочої рідини, гідравлічні втрати та динаміку виконавчого гідроциліндра. Модель дала можливість дослідити вплив параметрів системи на точність позиціонування; виконано аналіз динамічних властивостей привода та визначено основні фактори, які обмежують точність та швидкодію системи позиційного керування, зокрема гідравлічні затримки, силові збурення від процесу буріння та коливання тиску в гідролініях; проведено синтез позиційної системи управління з використанням методів теорії автоматичного керування; на основі лінеаризованої моделі обґрунтовано вибір структури регулятора та визначено його оптимальні параметри; запропоновано алгоритм позиційного регулювання подачі, який забезпечує стабільні перехідні процеси та високу точність відпрацювання заданого положення виконавчого органу в умовах змінних навантажень; розроблено структурну схему системи управління, що має модуль зворотного зв'язку, коригувальні ланки та виконавчий гідропривід, та підтверджено її працездатність за допомогою моделювання; проведено моделювання роботи системи в MATLAB/Simulink, результати якого показали: зменшення похибки позиціонування привода, покращення стійкості системи до збурень, скорочення часу перехідного процесу, зниження коливань подачі під час буріння.

Ключові слова: *система керування, буровий станок, електрогідравлічний привід подачі.*

Introduction. Roller-cone drilling machines are widely used in the mining industry for drilling holes and drilling holes in rocks of various types. The efficiency of their work is largely determined by the accuracy and stability of the drilling tool feed, while this very parameter affects the energy efficiency of the rock excavation process, the life of the tool and versatility of technological operations. In most industrial designs, the feed functions are implemented using an additional hydraulic drive, which, regardless of high energy consumption and power output, is characterized by the following characteristics: non-linearities, dynamic movements and sensitivity to changes in direction [1].

Traditional hydraulic feed control systems typically use simple control schemes that provide limited ability to accurately position the actuator, especially in conditions of varying rock hardness and operating pressure fluctuations. As a result, the systems lose accuracy, increase transients, and cause oscillations, which reduces drilling productivity and increases wear on drive components.

In this regard, the synthesis of a modern positional control system for the hydraulic feed drive is relevant, which would ensure stable operation of the drive, increased positioning accuracy and resistance to external disturbances. To solve this problem, it is necessary to create an adequate mathematical model of the hydraulic drive, analyze its dynamic properties and develop an effective control law considering nonlinearities and the specifics of the drilling technological process.

The article presents the results of the synthesis of a positional control system for the hydraulic drive of a roller cone drilling machine, provides a mathematical model of the drive, justifies the choice of the regulator structure, and demonstrates the effectiveness of the proposed approach using computer modeling.

Research material and method. The automation of technological processes for SBShS-250N roller cone drilling machines used in explosive drilling operations at Kryvbas quarries requires the development of automated control systems for feed and rotation mechanisms [2].

The feed mechanism is designed to create and regulate the axial load and speed of the bit when the drill rod needs to be moved along the length of the rod.

During drilling, the axial force and speed of the drilling tool on the SBShS-250N machine are controlled by a hydraulic drive using distribution spools and hydraulic throttles installed in the injection and discharge lines. The spools and hydraulic throttles are controlled by electromagnetic coils, in which a proportional dependence of the thrust force on the coil rod on the control current supplied to the coil is implemented [3].

The hydromechanical feed drive with an electrical control system is divided into two parts: the control object and the control device. By the control object we mean the executive hydromechanical device with the observed coordinates: pressure in the cylinders, linear speed of movement, path of movement of the rod. The electrical control device controls the coordinates of the control object. The input of the control system is fed with influences that determine the required amount of movement.

The control device, based on information about the processes in the control object, influences the hydraulic drive in order to transfer the control object from the initial state to the final state, in accordance with the restrictions imposed by the drilling technological process.

The drilling process is characterized by low linear speeds of movement. If it is necessary to build up the rod to the length of the next drilling rod, the rotator carriage rises at a speed that exceeds the drilling speed by three orders of magnitude [4; 5].

To solve the problem of synthesis and analysis of the control system, we will build a mathematical model of a hydromechanical drive, which includes: control signal amplifiers Uv, Un, electromagnets EMv, EMn, hydraulic systems GSv, GSn (where the indices “v” and “n” denote the upper and lower cavities of the cylinders), inertial masses of the piston of the rod Mc, rope-and-pulley system KP, Fig. 1.

Fig. 1 shows: Tv, Tn, T – pressure in the upper and lower cavities of the cylinders, total pressure on the drive piston; Vp, Vc – linear velocities of the piston and rod; Lp, Lc – linear displacements of the piston and rod.

The initial diagram, Fig. 1, corresponds to the mathematical model of the positional system control object, Fig. 2.

The structural diagram, Fig. 2, indicates: Kuv, Kun, Kemv, Kemn, Kgsv, Kgsn – transmission coefficients of amplifiers, electromagnets, hydraulic systems of the upper and lower cavities of the cylinders; Tuv, Tun, Temv, Temn, Tgsv, Tgsn – time constants of amplifiers, electromagnets and hydraulic systems; Mp, Ms – masses of pistons and the rod; C – rope stiffness coefficient; d, β – coefficients that determine the viscous friction force of the pistons in the cylinder and the braking friction force in the rope; Kc – proportionality coefficient of the rock destroyed by drilling; Kp – coupling coefficient between linear velocity and piston movement; S – locking switch; n – rotation speed of the rod.

In the drilling mode, the difference in forces in the upper and lower regions of the cylinders is applied to the inertial mass of the pistons, part of the force is compensated by viscous friction when the pistons move inside the cylinders of the hydraulic system. In the structural diagram, the rope-pulley system is taken into account as a link with finite stiffness. The elastic force in the rope is proportional to the difference in linear velocities of the pistons and the rod, applied to the inertial mass of the rod. The elastic force as drilling is balanced by the force of resistance to rock destruction.

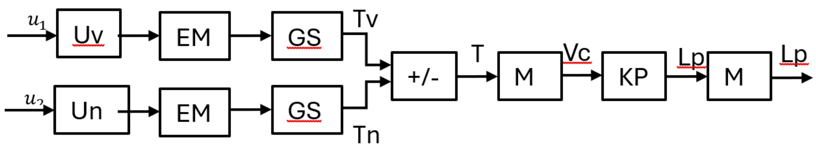

Fig. 1. Functional diagram of the hydromechanical feed drive

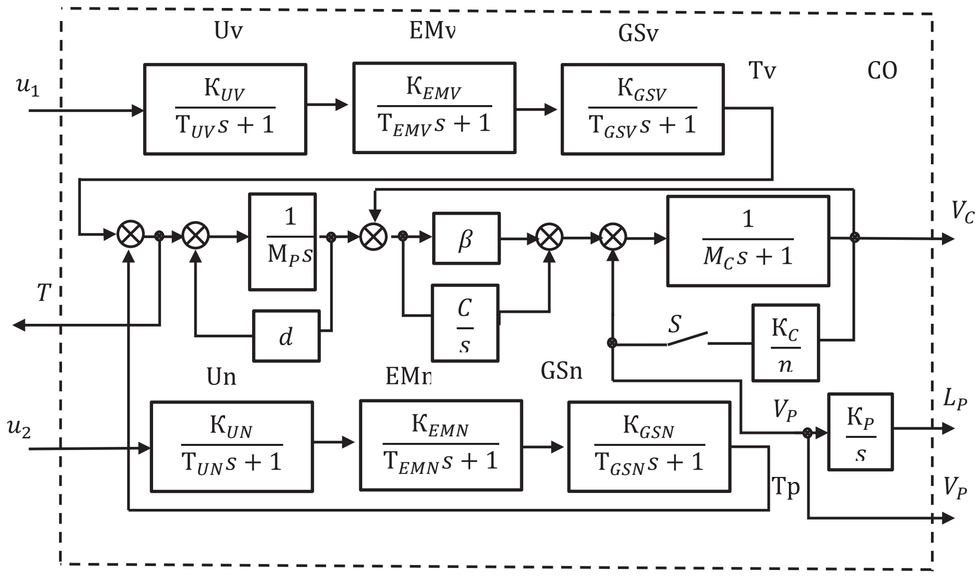

Fig. 2. Structural diagram of the control object

When transporting the rod, there is no force to resist the destruction of the rock. The switch S is open in this mode of operation.

The task of the control system is to move the bit in the rock massif, considering the restrictions, which include limiting the axial forces on the bit, limiting the speed of transporting the rod, limiting the travel path by the length of the rod.

The control object includes three observed coordinates: the difference in forces applied to the pistons T, their linear velocity Vp, and the distance traveled Lp. According to the number of controlled coordinates, the control system must contain three control loops: pressure, linear velocity, and displacement.

Fig. 3 shows a three-loop sequential control system with active correction of the dynamic characteristics of the controlled object. The figure shows: WPP(s), WPS(p), WPD(p) – transfer functions of position, linear piston speed, and pressure controllers; КD, КS, КSP – transfer coefficients of pressure (SD), speed (SS), and position (SP) sensors; TD, TS, TP – time constants of sensor filters; RBP, RBD – referent blocks position and pressure in the delivery line of the hydraulic system; P1, P2, P3, P4 – inverting relay elements.

The need to introduce relay elements is related to the peculiarities of the hydraulic system. The pressure on the piston is formed as the difference in pressure in the upper and lower volumes of the hydraulic cylinders. One of the volumes is connected directly to the pressure line, and the pressure in it is not regulated, while the pressure in the other volume is controlled by a pressure regulator. The latter volume of the hydraulic cylinder is connected to the drain line.

Each loop has a separate controller that is adjusted according to the transfer function of the controlled object. Since the loops of the system are connected in series, their time constants increase from the inner loop to the outer one. The input of each controller receives signals of the setpoint and actual value of the controlled parameter, with the previous controller generating a setpoint signal for the next controller. With such a cascade connection of control amplifiers, it is quite easy to limit the controlled coordinates of the control system [6; 7; 8].

Pressure regulator. The internal control loop in the control system is a loop that controls the pressure difference in the hydraulic cylinders, which is applied to the cylinder pistons. Connecting the pressure difference regulator ensures control of the resulting pressure. Transfer function of the circuit control object.

$$W_{01}(s) = W_{k1}(s)W_{n1}(s) = K_U \frac{K_{EM}}{T_{EM}s+1}\frac{K_{GS}}{T_{GS}s+1}K_D\left(\frac{1}{T_Us+1}\frac{1}{T_Ds+1}\right), \quad (1)$$

where $W_{k1}(s)$, $W_{n1}(s)$ – transfer functions of the compensated and uncompensated parts of the control object (the uncompensated part is indicated in brackets). The indices of the upper- and lower-cylinder volumes are omitted.

Transfer function of the pressure regulator

$$W_{PR}(s) = \frac{1}{sT_{01}W_{k1}(s)} = \frac{(T_{GS}+T_{EM})s+1}{sa_D(T_U+T_D)K_UK_{EM}K_{GS}K_D}, \quad (2)$$

where T_{01} – pressure loop integration time constant; a_D – loop tuning coefficient (a_D = 2 when tuning the loop to the modular optimum).

Transfer function of a closed optimized pressure loop

$$W_{CD}(s) = \frac{1/K_D}{a_D\sigma_D s+1}, \quad (3)$$

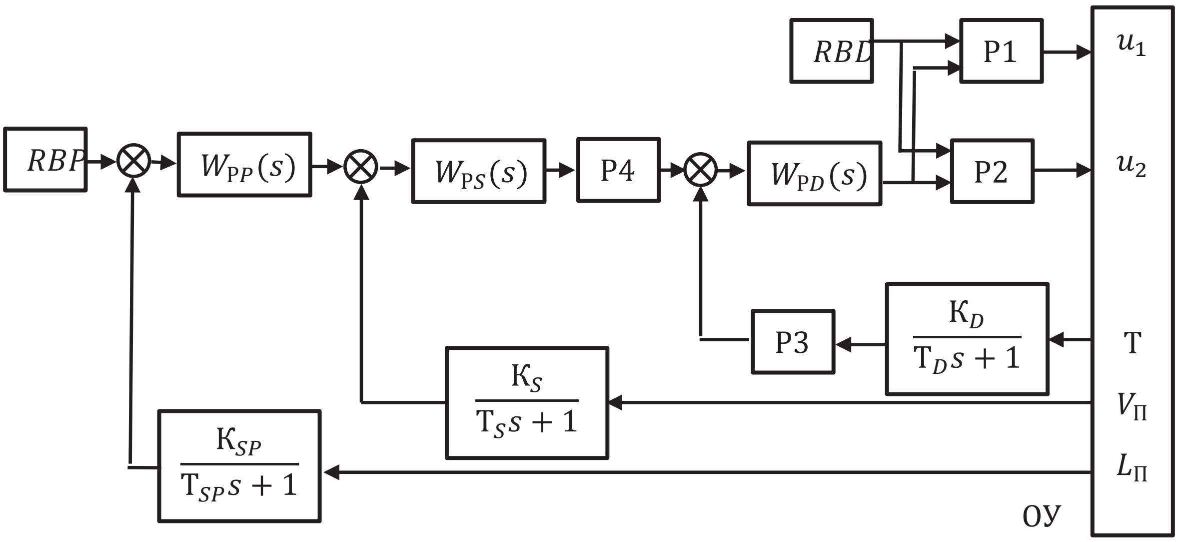

Fig. 3. Structural diagram of the control system

where $\sigma_D = T_U + T_D$ sum of small-time constant pressure loop.

Linear speed controller. External to the pressure difference control loop in the cylinders is the linear speed control loop for the pistons. Transfer function of the linear speed control loop control object

$$W_{02}(s) = W_{k2}(s) W_{n2}(s) = \frac{1}{K_D} K_S \frac{1}{M_P s + d} \left(\frac{1}{a_D \sigma_D s + 1} \frac{1}{T_S s + 1} \right), \quad (4)$$

де $W_{k2}(s), W_{n2}(s)$ – transfer functions of the compensated and uncompensated parts of the control object (the uncompensated part is indicated in brackets).

The object control does not consider the influence of elastic force in the rope-pulley system.

Transfer function of the speed regulator

$$W_{SR}(s) = \frac{1}{sT_{02} W_{k2}(s)} = \frac{K_D d(1 + sM_P / d)}{s a_S (a_D \sigma_D + T_S) K_S}, \quad (5)$$

where T_{02} – integration time constant of the speed loop; a_S – loop tuning coefficient (a_S = 2 when tuned to the modular optimum).

Transfer function of the closed optimized speed loop

$$W_{CS}(s) = \frac{1/K_S}{a_S \sigma_S s + 1}, \quad (6)$$

where $\sigma_S = a_S \sigma_S$ sum of small-time constant speed loop.

To obtain a sign-changing characteristic VP(T) at positive pressures in the upper and lower areas of the hydraulic cylinders, four relay elements P1, P2, P3, P4 are introduced into the control system, which are switched according to the sign of the output voltage of the speed regulator. At a positive output voltage, the relay elements transmit control signals from the input to the output directly, and when the sign of the voltage at the output of the speed regulator changes, the control signals invert the sign to the opposite by switching on the inverter relay elements.

Linear motion controller. The outer contour of the control system is the contour for controlling the movement of the rod. Transfer function of the control object of the movement loop

$$W_{03}(s) = W_{k3}(s) W_{n3}(s) = \frac{K_P K_D}{K_S} \left(\frac{1}{\sigma_S s + 1} \frac{1}{T_P s + 1} \right), \quad (7)$$

where $W_{k3}(s)$, $W_{n3}(s)$ – transfer functions of the compensated and uncompensated parts of the control object (the uncompensated part is indicated in brackets).

Transfer function of the position controller

$$W_{SR}(s) = \frac{1}{sT_{03} W_{k3}(s)} = \frac{K_S}{a_P (a_S \sigma_S + T_P) K_P K_{SP}}, \quad (8)$$

where $a_P = 2$ – position loop tuning coefficient.

To verify the correctness of the technical decisions made and to evaluate the effectiveness of the synthesized hinge position control system, mathematical modeling of the parameters of the SBSh-250MN drilling coil machine was performed (hinge weight 3 t; piston weight 0.9 t; fourfold rope-block system; rope 19.5-G-1-N-O-170 with a diameter of 19.5 mm with stiffness coefficients C = 1000 and dissipative forces β = 100, breaking force 20 t; pressure in the hydraulic system supply system 30 t; viscous friction coefficient of piston movement d = 2.

Based on the initial data of the hydromechanical system, the parameters of the control object were calculated and, according to the given methodology, the transfer functions of the pressure, speed, and position regulators were synthesized.

Mathematical modeling of the hydromechanical drive with an electric positioning system showed a significant impact on the dynamics of the system of elastic forces arising in the rope-pulley system. To eliminate increased variability, negative flexible feedback on the linear speed of the piston was introduced at the input of the speed regulator. This measure effectively eliminated the influence of rope compliance on the dynamics of the entire hydromechanical drive.

Oscillograms of developing pressures in the upper Tv and lower Tn areas of the cylinders, as well as the resulting force applied to the pistons of the T cylinders, are shown in Fig. 4. Fig. 5 shows oscillograms of the linear velocities of the piston VP, the rod VC, and the linear displacement of the LC.

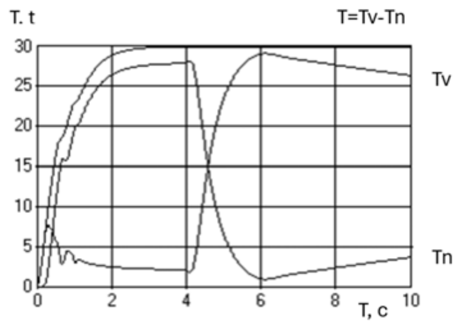

Fig. 4 Oscillograms of pressure changes in hydraulic cylinders

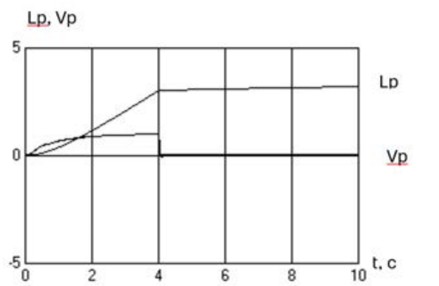

Fig. 5 Oscillograms of piston speed changes (rod), linear displacement of the rod

The process of transporting the rod to the bottom of the hole with a subsequent transition to the drilling process was modeled. From the point of view of the dynamic loads that arise, the selected operating mode is the most unfavorable, since during free movement of the rod, it accumulates a large amount of kinetic energy, which, after the rod hits the bottom, must be effectively damped by a hydromechanical drive with an electric positioning system.

As follows from the curves describing the transition process in the hydromechanical drive, in the process of transporting the pond to the face, acceleration is carried out with a limitation of linear speeds due to the saturation of the position controller. The pressure in the upper and lower cavities of the cylinders differs slightly, which provides the necessary effort to move the inertial masses of the pond and pistons. Some fluctuation in the linear speed of the pistons is explained by the significant influence of the elastic connection of the rope between the inertial masses. When reaching the face pond (t = 4 s), an elastic impact of the bit on the rock being drilled occurs. Fluctuations in the speed of the pond indicate that the pond rebounds from the face after the impact. The linear speeds of the pistons and the pond after reaching the face bit decrease to the drilling speed. The pressure difference in the upper and lower regions of the hydraulic cylinders increases sharply, which ensures the power drilling mode. The maximum pressure on the pond and bit is limited by the control system by controlling whether the maximum output voltage of the linear speed controller is exceeded.

Thus, the synthesized electrical control system for the hydromechanical drive of the rod feed of the roller drilling machine ensures its optimal operating modes with control of the pressure on the face, the speed of the rod movement, and the given path.

Conclusions

1. A mathematical model of the hydraulic feed drive has been developed, which considers the nonlinear characteristics of the hydraulic system, the compressibility of the working fluid, the dynamics of the hydraulic cylinder and the influence of the load from the drilling process. The model provides the ability to adequately reproduce the dynamics of the drive under conditions of variable operating modes.

2. The analysis of the dynamic properties of the system was performed and the key factors limiting the accuracy and speed of positional control were identified, including hydraulic delays, pressure fluctuations, power disturbances and nonlinearity of the flow through the throttling elements.

3. A positional control system for a hydraulic feed drive was synthesized, for which the choice of the regulator structure was justified, and the optimal parameters were determined. The proposed control law provides the necessary indicators of stability, accuracy and quality of transient processes

4. A structural diagram of a control system using position feedback has been developed, which provides compensation for external disturbances and stabilization of the actuator movement during drilling.

5. A computer simulation of the system operation has been carried out in MATLAB/Simulink, the results of which confirm the effectiveness of the developed system. A reduction in positioning error, a reduction in the transient process time, and an improvement in drive stability compared to traditional control schemes have been obtained.

6. The practical significance of the research lies in the possibility of applying the proposed approach in drilling rig automation systems to increase feed accuracy, reduce drilling tool wear, and improve the overall efficiency of drilling operations.

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