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CONSIDERATION OF WAYS TO IMPROVE THE EFFICIENCY OF DEEP WELL DRILLING

The object of the research is the composition of the bottom of the drill string when installing screw amplifiers. One of the biggest problems is the place of installation of the screw amplifier in the arrangement of the bottom of the drill string and its influence on the dynamic properties of the rock-crushing tool.

The results of the study of the influence of the use of a screw amplifier on the efficiency of drilling deep wells were obtained. According to the results of research, it was determined in particular that the imposition of vibrations often causes a change in the nature of the interaction of the bit with the hole rock, therefore, the installation of the amplifier at different points of the layout will have different effects on the drilling performance. It is shown that the kinetic energy accumulated by the weighted drill pipes, when the amplifier is installed above the bit, is transmitted through the screw mechanism to the rock-destroying tool (bit) and is an operation that leads to an increase in drilling performance.

Thanks to the conducted research, it has been proven that the property of the drill string elements and the disturbances of the oscillatory processes of various nature acting on them are related to the dynamic loading of its elements due to the installation of the amplifier in different places of the layout of the bottom of the string. This is due to the fact that the proposed screw amplifier is used with a friction fuse, which causes the rotary movement of the screw under the action of the torque and the translational movement of the nut together with the overbit mass attached to the bit.

In most practical cases, especially during deep drilling, bits fail due to the destruction of the rolling bearings of the roller bit, which leads to jamming of the roller bit and leaving them in the holes. For practical use, the proposed design uses a friction fuse, which connects the bit to the drill pipe column through an elastic element. Analytical studies and performed calculations show that this screw amplifier will effectively ensure the destruction of any rocks, especially during overloads without slippage of reaches and be used for the construction of wells in native mining and geological conditions.

Keywords: *well, drilling, dynamic state, screw amplifier, mechanical speed, deepening per run.*

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1. Introduction

Drilling efficiency is affected by mechanical speed and depth per run. Moreover, during the drilling of ultra-deep wells, the last factor is the most important [1–3]. During drilling in the sediments of Stryi Svita, a decrease in the mechanical speed is noted as the depth of the well increases. Since the physico-mechanical properties of the rock packs to be drilled are homogeneous in terms of roughness, lying at different depths and being drilled with the same type of roller bits by the rotary method (the arrangement of the bottom of the column did not change), with unchanged hole washing and constant operating parameters, it can be assumed that that the decrease in the mechanical speed of drilling is affected only by a decrease in the axial load on the bit. And on the horizontal sections of the well trajectory, the reduction of the mechanical speed of drilling is also affected by the reduction of the axial load during its transmission from the mouth to the hole through the horizontal section of the column of drill pipes located in the well [4–6].

To increase the axial load, screw amplifiers are used, which can be used when drilling obliquely directed wells with horizontal sections [7, 8]. But the question arises in which location of the layout to install the device for changing the axial load, taking into account the method and geological and technical conditions of drilling.

The aim of research is to determine the impact on drilling performance of installing axial load amplifiers at various points of the bottom of the drill string. This will make it possible to optimize the well construction process by increasing drilling performance.

2. Materials and Methods

Screw amplifiers include devices in which the energy of the drive (rotor, downhole motor) is transformed into the useful work of destroying the hole rock with the help of a screw mechanism.

Let's consider the influence of the place of installation of the screw amplifier in the arrangement of the bottom of the drill string (DS) on its dynamic properties.

The principle of operation of the device in the case when the screw amplifier is installed above the bit (between the bit and weight drill pipes – WDP), is that the increased torque from the rotor through the drill string is transmitted to the bit to accelerate the WDP to a specified speed. As well as the strengthening of oscillatory processes in order to accumulate the kinetic energy of translational and rotational movements. This energy is used to impact the roller bit tooth (interaction) with the hole, which leads to the hole destruction.

3. Results and Discussions

In order to determine the torque, in the conditions of neglecting the length of the drill pipes, it is proposed to consider that the load on the bit is carried out by a concentrated mass M (part of the WDP weight), connected by an elastic element with stiffness c through c_1 to a screw amplifier with an elastic element with stiffness c_1. In the conditions of the tooth movement of the roller bit along the hole and cutting it into the rock, during the solution of the problem, the complete dependence of the force $P(t)$ is registered. The value of this function for calculating the torque M_t in this case:

$$M_t = P(t) r \sin\alpha. \tag{1}$$

During the interaction of the roller bit tooth with the rock (working cycle), the accumulated kinetic energy, which is amplified by the screw mechanism and used to destroy the hole A_h, is displayed as:

$$A_h \to (T_v + T_\omega) = \left(\frac{mv^2}{2} + \frac{J_1\omega^2}{2}\right) = T_e \to A_h. \tag{2}$$

It should be noted that the energy of rotational motion is $(0.8-0.9)T_e$. A constant ratio of angular and translational speeds is characteristic of a screw mechanism:

$$\omega = \frac{2\pi}{h} v. \tag{3}$$

Therefore, the effective kinetic energy of torsional and longitudinal vibrations of the bit and WDP can be represented as:

$$T_e = \left(\frac{h^2}{4\pi^2} m + J_1\right)\frac{\omega^2}{2} = J\frac{\omega^2}{2}, \tag{4}$$

or

$$T_e = \left(m + \frac{4\pi^2}{h^2} J_1\right)\frac{v^2}{2} = M\frac{v^2}{2}, \tag{5}$$

where J – the reduced moment of WDP inertia (overbit mass):

$$J = m\frac{h^2}{4\pi^2}; \tag{6}$$

M – the overbit reduced mass of the drill string assembly, defined as:

$$M = m + \frac{4\pi^2}{h^2} J_1. \tag{7}$$

In the above expressions, the following is indicated: A_d – drive work; T_v and T_ω – kinetic energy of the translational and rotational movement of the drill string of the overbit masses; m and J_1 – the mass of the translational motion of the DS overbit system and the axial moment of inertia of these masses; h – the screw stroke.

The work of the drive elements is transformed into the kinetic energy of the hole rock destruction.

A change in the direction of movement of the roller bit tooth occurs during the rotation of the roller bit. During the movement of the bit tooth to the hole surface, it is transformed into the energy of the rock destruction due to its cutting into the rock of the hole according to the scheme:

$$A_d \to M_5\varphi_p \to T_e. \tag{8}$$

From the considered principle of action of the screw amplifier, it follows that the strengthening of the rock destruction due to the interaction of the roller bit tooth with the hole occurs due to the increase in the supply of kinetic energy to its contact with the rock (increase in the axial load and torque on the bit).

Let's consider the proposed scheme in more detail (Fig. 1). In this case, let's deal with a two-mass system.

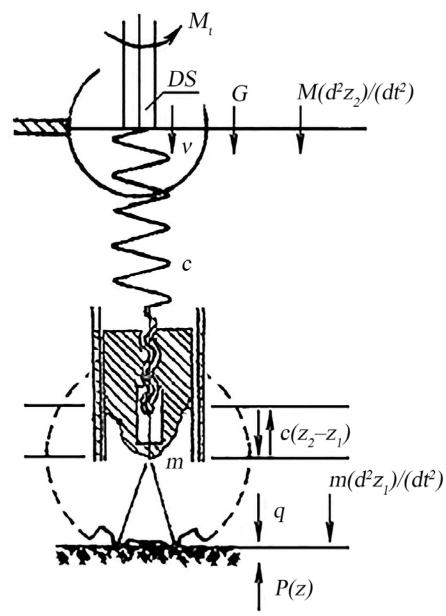

Fig. 1. Two-mass model of a bit with an amplifier and a drive

Mass M corresponds to the distributed mass of the compressed part of the drill pipe string and the parts attached to it, for example, the casing of the downhole motor; m – the mass of the bit with the converter and the shaft of the downhole motor (during drilling with downhole motors). Stiffness c is included between the masses M and m through a helical stiffener with heel stiffness, a downhole motor shaft, and an elastic stiffener element. The equation of mass movement along the z axis forms the following system:

$$\begin{aligned}
&M\frac{d^2 z_2}{dt^2} + G + c(z_2 - z_1) = 0;\\
&m\frac{d^2 z_1}{dt^2} + q - c(z_2 - z_1) - P(z) = 0;\\
&J\frac{d^2\varphi_1}{dt^2} + M_t = 0,
\end{aligned} \tag{9}$$

where z_1 – the mass coordinate m (the mass of the bit with the screw amplifier and the motor shaft); z_2 – the coordinate of the conventional stiffness index c associated with the mass M; $G=Mg$ – the axial load on the bit; $q=mg$ – the weight of the bit, the screw amplifier and the shaft of the downhole motor; $P(z)$ – the force of resistance to the introduction of the tooth into the hole rock, currently:

$$\frac{dz}{dt}>0,\ P(z)z>0;\ z>0,\ P(z)=0.$$

The axis begins on the surface of the hole and is directed upwards.

Supplementary conditions imposed on $P(z)$ state the fact that during the movement of the roller bit tooth from the hole upwards, the force applied to it from the side of the rock is zero. The same is observed when the tooth is above the hole surface, $P(z)$ regardless of its movement.

During the rotation of the screw, considering that the body of the amplifier is relatively stationary (Fig. 2), then all points, except the body of the amplifier, carry out rotational movement around the axis. Thus, by turning the screw by an angle φ, the body with the bit will move translationally by the amount:

$$l=\varphi t/2\pi. \quad (10)$$

By differentiating (10) over time, the function of the movement of the amplifier case and the bit is obtained:

$$v_2=\omega_1 t/2\pi, \quad (11)$$

where $v_2=dl/dt$ – the linear speed of the point moving forward along the axis of the screw; $\omega_1=d\varphi/2$.

The angular velocity of the point of the screw d_2, which lies on the circle of the average diameter of the section, takes the form:

$$v_1=\omega_1 d_2/2. \quad (12)$$

The gear ratio of the helical gear in our case is defined as:

$$i=\frac{v_1}{v_2}=\frac{\pi d_2}{t}=\tan\gamma, \quad (13)$$

where γ – the angle of elevation of the helical line.

When solving system (10), expression (13) must be taken into account.

Taking into account the equation [9], and taking into account the indicators of effective drilling with the use of a screw amplifier – the mechanical speed of drilling v_{mech}, penetration per bit per run H and travel speed v_{run} per run H, then the penetration per run is determined as:

$$H=i\int_0^T v_{mech}(t)\,\mathrm{d}t, \quad (14)$$

where T – the full working time of the bit.

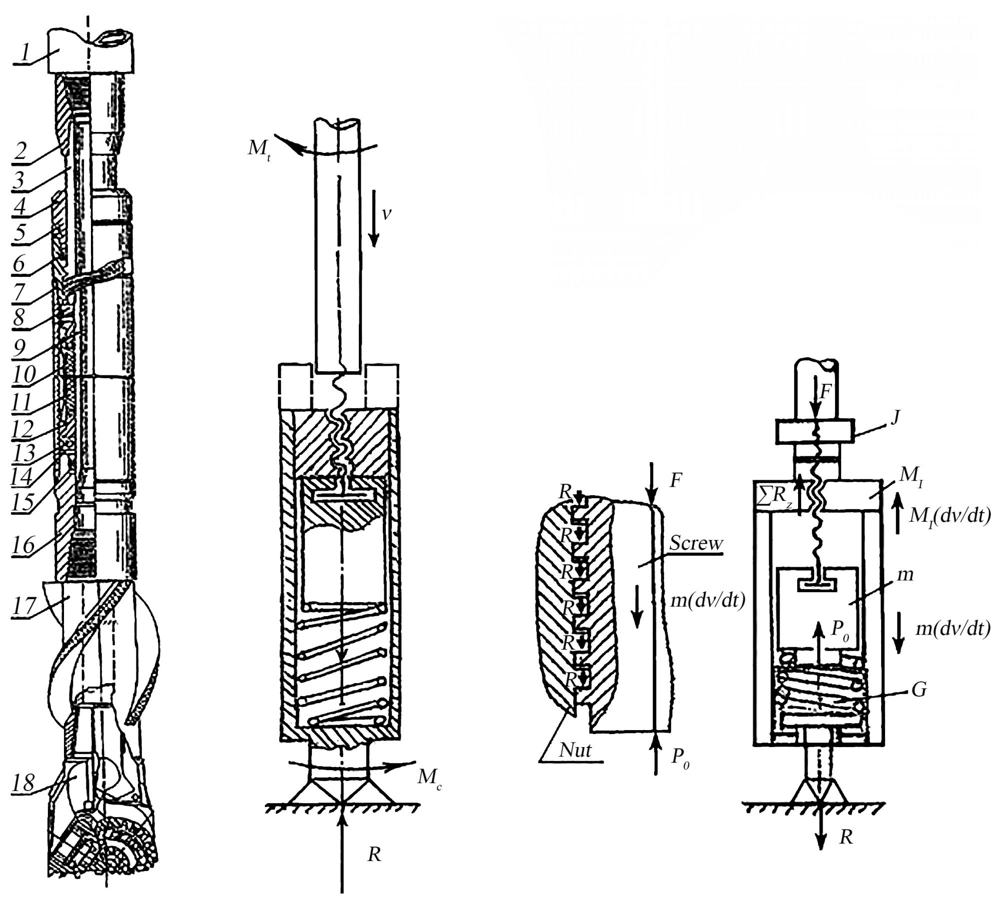

Fig. 2. General view of the amplifier of the axial load of the bit and calculation schemes for determining the main parameters of the amplifier: 1 – drill string; 2 – converter; 3 – screw; 4 – packing nut; 5 – nut; 6 – oil seal; 7 – body; 8 – puck; 9 – barrel; 10 – shell; 11 – filler; 12 – piston; 13 – bearing; 14 – puck; 15 – oil seal; 16 – converter; 17 – blade spiral calibrator (BSC); 18 – bit

The mechanical speed changes (decrease) according to the exponential law:

$$v_{mech} = i v_{0mech} e^{-\alpha t}, \tag{15}$$

where α – the coefficient that depends on the properties of the tooth material, abrasiveness of the rock, etc.; e – the base of the natural logarithm; i – the transmission ratio of the screw.

The effect of rock properties on the stability of the support is due to a change in the dynamic operating mode of the bit. Since in the calculation of the drilling system proposed by us, the $P(t)$ dependence must be used to introduce the appropriate corrections for determining the torque, it can accordingly be used to introduce the appropriate corrections for determining the time T.

Research conducted by Hughes Tool Company (USA) showed a pattern that the total number of rotations that the supports can withstand (motor resource) does not depend on the bit rotation speed [1].

Thus, it is enough to determine the dependence of the stability of the bit's supports on the axial load at some specific speed of rotation for the bit of this type, taking into account the use of an amplifier. And then, based on this, it will not be difficult to determine the time of its operation for an arbitrary speed of rotation.

For changes in the cross-section of the column of technological origin, for example, the presence of a heavy bottom, as calculations show, the work of the bit is affected only by the lower part of the column. Therefore, any column of mixed construction can be replaced by a column of constant section (rod) that corresponds to its lower section.

Since the upper end of the column practically does not affect the operation of the bit, it is possible to consider the upper end fixed.

The effect of rock properties on the stability of the support is determined by the operating mode of the bit. At the same time, penetration per run is proportional to the immersion of the teeth under a given load. The run speed can be calculated depending on:

$$v_{run} = \frac{H}{T + t_{LLO}}, \tag{16}$$

where H – the penetration per run; t_{LLO} – the time spent on lowering and lifting operations (LLO) according to normative data.

To improve the calculation model of the drill string, taking into account the oscillatory processes occurring in it, let's use the provisions [10–12]. First of all, this includes longitudinal and torsional oscillations, which are described by hyperbolic equations.

Damping longitudinal oscillations of the column is determined by the equation:

$$\frac{\partial^2 u}{\partial t^2} + \lambda \frac{\partial u}{\partial t} - a^2 \frac{\partial^2 u}{\partial x^2} = 0, \tag{17}$$

where $u(x,t)$ – the movement of the column point with the x coordinate at the time t; $\lambda = \mu/\rho G$ – the attenuation coefficient; μ – the damping constant; ρ – the mass density of pipe material; G – the cross-sectional area of pipes; α – the speed of longitudinal elastic wave propagation in pipes.

A stress $\left(P(z) - G\right)/A$ acts on the lower end of the column.

It should be noted that we have taken into account only additional power factors, compared to the static load due to the use of an amplifier.

In order to get the possibility of approximating the calculation scheme to reality and taking into account new factors that are characteristic of various designs, first of all, let's take into account the torsional vibrations of the drill string:

$$\frac{\partial^2 \theta}{\partial t^2} + \lambda_1 \frac{\partial \theta}{\partial t} - b^2 \frac{\partial^2 \theta}{\partial x^2} = 0, \tag{18}$$

where $\theta(x,t)$ – the twisting angle of the column section with the coordinate at time t; λ_1 – the vibration damping coefficient; $b^2 = G/\rho$ – the speed of longitudinal elastic wave propagation in pipes; G, ρ – the shear modulus and density of pipe material.

Another improvement of the calculation model is the consideration of some features of the drilling motor design, as well as cross-sections of the drill pipe column. The described system will now consist of a bit, the properties of which are described by $P(z)$, a ball bit, a turbo drill shaft, a support heel, a bit load intensifier, a set of WDPs and a string of drill pipes. The axial load is transmitted to the shaft of the downhole motor through the heel and the damper of the amplifier, which in our case leads to the amplification of the longitudinal vibrations of the column and the shaft.

In this case, the shaft of the downhole motor is connected to the bit rigidly either through a screw amplifier, or it is located above the body of the downhole motor (Fig. 3).

If the heel and the elastic element of the amplifier are made in the form of upper and lower elastic elements that have different stiffness, then the nature of the shock absorbers will be affected by the fluctuations of the lower end of the WDP. In this case, the torsional vibrations of the shaft of the drilling motor are transmitted to the WDP and the drill pipe string.

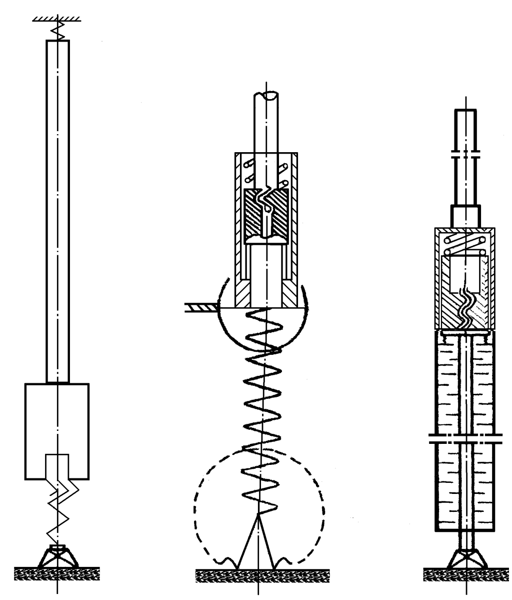

Fig. 3. Schemes for assembling calculation models for different positions of the load amplifier on the bit (three positions of the amplifier are given): *a* – rigid connection of the shaft of the downhole motor with the bit; *b* – the shaft of the downhole motor, connected to the bit through a screw amplifier; *c* – the screw amplifier located above the casing of the downhole motor

For these conditions, when u moves the points of the shaft, the boundary conditions are defined:

$$u=0,\ \frac{\partial u}{\partial x}i=\frac{P(z)-G}{QT}i. \tag{19}$$

If the upper end of the shaft of the downhole motor and the lower end of the WDP are connected by a support heel or a screw amplifier, then their deformations can be related by the following dependencies (forces are transmitted through the support heel):

$$\frac{\partial u}{\partial x}=\frac{c(\bar{u}_0-u_n)}{AE}\ \text{at}\ x=l, \tag{20}$$

$$\frac{\partial \bar{u}}{\partial \bar{x}}=\frac{c(\bar{u}_0-u_n)}{A_1E}\ \text{at}\ \bar{x}=0, \tag{21}$$

where l – the shaft length of the downhole motor; A_1 – the cross-sectional area of the WDP; $\bar{u}_0, u_n$ – the corresponding movement of the weight drill pipe and the upper end of the shaft; c – the stiffness of the booster shock absorber and the heel of the downhole motor.

It is obvious that in the case of a column consisting of sections of different sections, the final value of displacements and longitudinal force for one section will simultaneously be the initial values for the adjacent section.

Based on this, let's introduce additional conditions at the junction of the weighted pipe and the drill pipe string:

$$\bar{u}\ \text{at}\ \bar{x}=l_1\ \text{and is equal to}\ u\ \text{at}\ x=0,$$

$$A_1E\frac{\partial \bar{u}}{\partial x}=A_2E\frac{\partial u}{\partial x}, \tag{22}$$

where A_2 – the cross-sectional area of the drill pipes; l_1 – the WDP length; l_2 – the drill pipes length.

If $x=0$, then the end of the drill pipe column can be considered free, then:

$$\frac{\partial u}{\partial x}=0. \tag{23}$$

Analyzing the law of movement of the bit, it is noted that during the calculation of the square bit, each cycle of calculation coincides with the cycle of interaction with the rock of the next tooth, that is, at the moment of contact of the next tooth, the previous one ceases to interact with the rock.

The time count for each new tooth starts from the beginning. During the movement of the column, the movement time is continuous.

It should be noted that when calculating the displacement of the column, boundary conditions (19) were used, which includes the rock resistance $P(z)$, which is determined according to experimental data.

So far, it has been established that during the exit of the tooth from the contact, the force P will change according to another law:

$$P(z)=P(z')+k(z-z'), \tag{24}$$

where z' – the tooth depth at the previous moment; k – the tangent of the angle of inclination of the elastic part of the experimental curve $P(z)$.

Thus, the tooth will come out of contact with a rapid decrease in force P along the line of the initial part of the curve, Fig. 4 [10].

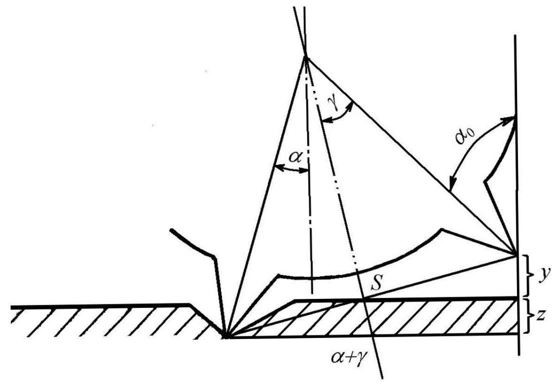

Fig. 4. Scheme for determining the duration of the work cycle of the roller bit tooth [10]

Therefore, in the case of calculation by the grid method, at each given moment of time, the movement of the lower end of the column at the previous moment is known [10].

In most practical cases, especially during deep drilling, bits fail due to the destruction of the rolling bearings of the balls.

Sometimes this destruction of the supports leads to jamming of the roller bits and leaving them on potholes. Therefore, the working time of the bit on the hole is determined by the stability of the supports, when the torque on the bit increases greatly, which can lead to the failure of the column elements. Therefore, it is proposed the design of a screw amplifier with a friction fuse (Fig. 5) [13].

The screw working mechanism creates a constant ratio between the force on the bit and overbit elements (BSC, etc.) and the torque on the screw and the drive shaft. This makes it possible to limit the torque on the screw (DS column) and bit.

In the proposed design, a friction fuse is used, with the help of which the bit is connected to the drill pipe column through an elastic element.

The operation of the friction fuse is due to the continued stationary contact of its friction surfaces under certain drilling modes without overloads and the dynamic nature of these surfaces during overloads, when slippage occurs in a short period of time.

For further explanations, the working conditions [10] are used, taking into account the situation when the pre-tightened fuse does not slip until the dynamic moment corresponds to the value:

$$M_d=\alpha P_d\frac{h}{2\pi}, \tag{25}$$

where α – the overload factor.

At $M_d=M_f$ (the maximum value of the moment on the bit) the fuse will start to operate. In the process of its operation, the friction moment changes from the maximum value $M_f\approx 2M_{st}$ to the static one M_{st}.

Under these conditions, at any moment in time the torque on the screw takes the form:

$$M(t)=M_F\mp A_0e^{-\lambda t}\sin(\omega t+\varphi), \tag{26}$$

where M_F – the average value of the friction moment; A_0 – the initial amplitude; ω – the angular frequency; φ – the phase angle; λ – the damping coefficient of oscillations:

$$\lambda = \frac{\ln A_i / A_{i+1}}{T}, \tag{27}$$

where A_i and A_{i+1} – the amplitudes of the next two oscillations; T – the period of oscillations, $T = 2\pi/\omega$.

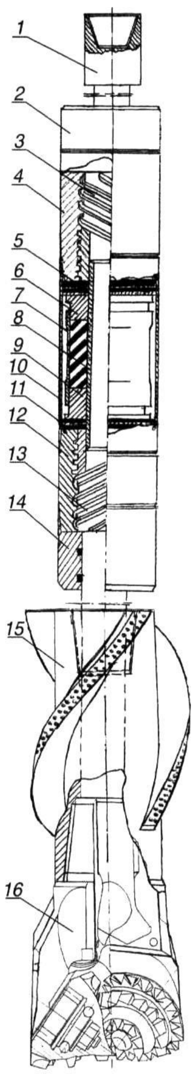

Fig. 5. Screw amplifier with friction fuse: 1 – converter; 2 – packing nut; 3 – propeller shaft; 4 – nut; 5 – friction clutch; 6 – piston; 7 – elastic shell; 8 – filler; 9 – lower piston; 10 – body; 11 – friction clutch; 12 – nut; 13 – lower screw; 14 – packing nut; 15 – BSC expander; 16 – bit

During the acceleration of the bit, a torque is applied to the overbit masses $J_2 + J_3$ of the drill string or the shaft of the downhole motor J_1, which causes elastic torsional deformations of the screw. It is noted that the overbit mass J_2 and J_3 the WDP mass will move together because the fuse is tightened.

Currently, the moment of tightening the fuse is defined as:

$$M_f = zSqf_s r_r = zP_f f_s r_r, \tag{28}$$

where z – the number of friction surfaces; S – the area of the friction surface; q – the specific forces for the friction material, $q_1 \le q \le q_{per}$ (q_{per} – permissible specific force); f_s – the static coefficient of friction; r_r – the reduced radius of the friction surface, $r_r = \dfrac{2r_{ex}^3 - r_{in}^3}{3r_p^2 - r_{in}^2}$; P_t – the tightening force.

We have considered the movement of the bit and the overbit mass until the fuse is turned on, when the elastic moment M_{el} and the moment of resistance M_d act on this mass J_1. The equation of motion is proposed as:

$$-J_1 \frac{\partial^2 \varphi}{\partial t^2} = M_{el} - M_d. \tag{29}$$

The elastic moment is written as:

$$M_{el} = c_\varphi (\varphi_1 - \varphi_2), \tag{30}$$

where c_φ – the torsional rigidity of the WDP with a screw.

When the elastic moment M_{el} acts on the drill string (WDP, the shaft of the downhole motor), the equilibrium equation is written:

$$(J_1 + J_2)\frac{\partial^2 \varphi}{\partial t^2} = -M_{el}. \tag{31}$$

Solving (28)–(31), the equation is obtained:

$$\frac{\partial^2 M_{el}}{\partial t^2} + c_\varphi \frac{J_{23} - J_1}{J_1 J_{23}} M_{el} = \frac{c_\varphi}{J_1} M_d, \tag{32}$$

where $J_{23} = J_2 + J_3$.

The greatest overload occurs during drilling of layers in a pack of rocks with interlayers of increased hardness, or sharp landing of the bit on the hole, therefore the dependence of the moment of resistance on the working time is adopted [1, 14], as:

$$M_d = \alpha P_{ex} \frac{h}{2\pi} \sin \gamma t, \tag{33}$$

where $0 \le t \le t_d$, t_d – the cutting of the tooth into the hole rock or the increase of the load due to the hard landing of the bit on the hole, which leads to the deformation of the system M_d from zero to its maximum value.

After a number of transformations using (32) and (33), it is possible to determine the conditions for the operation of the friction fuse.

The friction fuse must work under the condition $t = t_d$ and $M_{el} = M_f$. Having solved the expressions for M_{el} and (28) together, it is possible to obtain the necessary tightening force of the friction fuse:

$$P_f = \frac{c_\varphi M_m \left(\sin \gamma t_d - \dfrac{\gamma}{\beta} \sin \beta t_d \right)}{J_1 (\beta^2 - \gamma^2) z f_s r_r}, \tag{34}$$

where β – the cyclic frequency that is determined as:

$$\beta = \sqrt{\frac{c_\varphi (J_{23} - J_1)}{J_1 J_{23}}}; \; M_m = \alpha P_{ex} h / 2\pi.$$

This force will be provided by means of a shell shock absorber (an elastic cylindrical shell with a filler in Fig. 5).

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A specific feature of the work of the bit with the use of a screw amplifier with a friction fuse causes the rotary movement of the screw under the action of the torque and the translational movement of the nut together with the overbit mass attached to the bit.

The research results in full are ready for the design of the screw amplifier. And after the production of its layout for use in the process of drilling wells.

In the future, the developed model will be implemented in the deposits of Ukraine, and the proposed calculations will help specialists in choosing the optimal application conditions in order to increase the speed of drilling wells.

4. Conclusions

It has been determined that the imposition of vibrations often causes a change in the nature of the interaction of the bit with the hole rock, therefore, the installation of the amplifier at different points of the layout will have different effects on the drilling performance.

It has been proven that the kinetic energy accumulated by the WDP, when the amplifier is installed above the bit, is transmitted through the screw mechanism to the rock-breaking tool (bit) and is an operation that leads to an increase in drilling performance.

It has been shown that the properties of the drill string elements and the perturbations of oscillatory processes of various nature acting on them are related to the dynamic loading of its elements when the amplifier is installed in different locations of the bottom of the string.

In order to prevent the destruction of the supports and the jamming of the roller bits during the bit operation on the holes with a significant increase in torque, it has been proposed the design of a screw amplifier with a friction fuse. Analytical studies and performed calculations show that this screw amplifier will effectively ensure the destruction of any rocks, especially during overloads without slippage of bits and be used for the construction of wells in native mining and geological conditions.

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Conflict of interest

The authors declare that they have no conflict of interest in relation to this study, including financial, personal, authorship, or any other, that could affect the study and its results presented in this article.

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Data availability

The paper has no associated data.

Use of artificial intelligence

The authors confirm that they did not use artificial intelligence technologies when creating the presented work.

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