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Research paper

Dynamic Model of a Drilling Rig with a Top Drive System for Rotary Steerable Systems

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Abstract

The fundamentals and principles of constructing a generalized dynamic model of a drilling rig with a top drive system for rotary steerable systems, which consists of models of individual mechanisms and takes into account many physical processes occurring in a real object, are presented. The most complex element of such a system is the drill string model, which has many degrees of freedom.

Keywords: *rotary steerable systems, drilling rig, drill string, dynamic model, Top Drive System, downhole motor, lift mechanism model, friction forces, drawworks model, regime parameters of drilling process.*

1. Introduction

Currently, drilling installations, including mobile ones the main power unit of which is the top drive system, are widely used in the drilling of deep wells. The top drive system is placed on the crown block of the drawworks, resting on its own guides, and serves to rotate the drill string, as well as to perform a number of auxiliary operations related to the manipulation of the drill pipes and control of the flow of drilling fluid.

During operation of this device, control of tripping is carried out by the operator, who needs to constantly monitor compliance with technological regimes. During drilling, there are dangerous vibrating phenomena [1, 2, 3, 4] affecting mainly the strength and durability of the entire structure. The solution to the first question can be the development of an automated system for managing the drilling process, and the second is the creation of methods to determine dangerous modes at the installation projecting stage. Since these tasks are quite complex, their implementation can be divided into several stages, one of which will be the development of an adequate mathematical model of the machine.

The use of generalized dynamic and mathematical models of a drilling rig with a top drive system implemented as a software product will allow to determine such important parameters as the inherent frequencies of the machine, the load on its elements and the necessary driving forces, as well as to investigate the behavior of the system in various, even unrealistic, operating conditions.

2. Main part

The complete model of a drilling rig with a top drive system is quite complex, so it is divided into several models interacting with each other, as shown in the generalized block diagram in Fig.1. The diagram shows the following model blocks: CU – control unit model; LM – model of the lifting mechanism of the top drive system and the drill string; TDS – model of the mechanism of rotation of the top drive; DS – model of the drill string interacting with the material of the borehole; DM – model of the downhole motor and drilling tools; SM – soil model in the borehole bottom.

The control unit model (CU) is designed to reproduce the logic of the drilling rig, giving the mechanism models appropriate control signals and forming the necessary operating parameters, such as drilling fluid pressure, lowering speed of the talblock, brake action moments. It can be used to simulate real control systems of lifting mechanisms and top drive systems, taking as a basis the information on the loading of elements of the mechanical part.

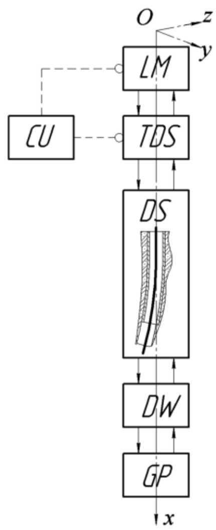

Fig. 1: Generalized model structure of the rig

All other blocks on the generalized scheme are the elements of the mechanical part of the installation, between which direct and reverse interactions are established, shown by arrows.

The LM unit includes models of the construction of a drilling rig, drawworks, and polyspasts (Fig. 2).

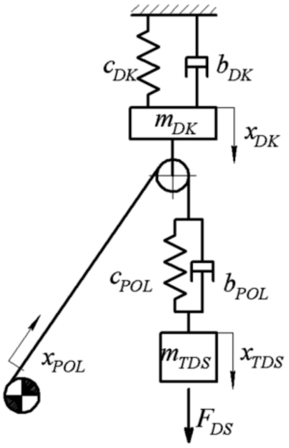

Fig. 2: Dynamic model of structures and mechanisms serving for lifting the TDS

Given the inertial parameters of the drawworks drive, such a system can be mathematically described by the following equations

$$\begin{cases} m_{DK}\cdot\dot{x}_{DK} = \quad x_{DK}\cdot c_{DK}\cdot\dot{x}_{DK}\cdot b_{DK} + \left(x_{DR}\cdot x_{\Pi POL}\cdot x_{DK}\right)c_{POL} + \\ \qquad\qquad +\left(\dot{x}_{DR}\cdot\dot{x}_{POL}\cdot\dot{x}_{DK}\right)b_{POL} + m_{DK}\cdot g \\ m_{TDS}\dot{x}_{TDS} = \left(x_{POL} + x_{DK}\cdot x_{DR}\right)c_{POL} + \\ \qquad\qquad +\left(\dot{x}_{POL} + \dot{x}_{DK}\cdot\dot{x}_{DR}\right)b_{POL} + m_{TDS}\cdot g + F_{DS} \\ J_{DR}\ddot{} = M_{DR}\cdot D_D\dfrac{\left(x_{DR}\cdot x_{POL}\cdot x_{DK}\right)c_{POL} + \left(\dot{x}_{DR}\cdot\dot{x}_{POL}\cdot\dot{x}_{DK}\right)b_{POL}}{2\cdot i_{POL}\cdot i_{DR}\cdot\eta_{DR}} \end{cases} \qquad (1)$$

The first two equations characterize the vertical motion of the given masses of the derrick. m_{DK} and top drive systems m_{TDS} along the corresponding directions of the coordinate axes x_{DK} and x_{TDS}. These movements, as occurs in the lifting mechanism [5], are due to the action of the drawworks on one side and the pressure of the string on the other F_{DS}. The specified action of the drawworks is represented by the coordinate position of the branch of the assembly pulley x_{POL} casting down from the drum of a diameter D_D. Options c_{DK}, c_{POL}, b_{DK}, b_{POL} – stiffness and coefficients of viscous friction of at assembly pulley and a derrick, which can be approximately determined by calculations or full-scale measurements.

The last equation is the equation of motion of the rotating inertial masses of the drawworks drive J_{DR} given its efficiency η_{DR}. The generalized coordinate here is the angle of rotation of the motor shaft φ associated with the coordinate x_{POL} by the expression

$$x_{POL} = D_D\cdot\varphi/\left(2\cdot i_{POL}\cdot i_{DR}\right),$$

where i_{POL} and i_{DR} – the multiplicity of the assembly pulley, and the gear ratio of mechanical gears, respectively.

Modern drawworks of drilling rigs have the electromechanical drive. Therefore, a driving moment is proposed. M_{DR} to be defined as the electromagnetic moment of the induction motor, taking into account the required control parameters [6]

$$M_{DR}(s,f,u,\rho,\tau) = \\ = \frac{3\cdot U_{1N}^2\cdot R_{2N1}\left(1+\alpha\cdot\tau+\rho\right)\cdot u^2\cdot s\cdot p_N}{2\cdot\pi\cdot f_N\cdot R_{1N}^2\cdot\left[\left(\left(1+\alpha\cdot\tau\right)\cdot s+\rho_1\cdot\left(1+\alpha\cdot\tau+\rho\right)\right)^2+\left(\dfrac{X_{KN}\cdot f\cdot s}{R_{1N}\cdot f_N}\right)^2\right]}.$$

Here, the following values are tabular: R_{1N} – rated resistance of the stator winding; R_{2N1} – rated resistance of the rotor winding, related to the stator winding; X_{KN} – rated inductive short circuit resistance; p_N – the number of pairs of poles; α – coefficient of temperature variation of the resistances of the stator and rotor windings (you can change the equation of the electromagnetic moment so as to take into account the difference of the winding materials); U_{1N}, f_N– nominal values of the supply voltage and its frequency. Parameter ρ_1 is defined as $\rho_1 = R_{2N1}/R_{1N}$. Parameter s – slip, determined according to the formula

$$s = 1 - \frac{\varphi\cdot p_N}{2\cdot\pi\cdot f_N}.$$

The control parameters are: $\rho = R_{ADD}/R_{2N}$ – the ratio of additional resistance to the rated value of the resistance of the rotor winding, which is necessary when modeling a motor with a squirrel cage; τ – the magnitude of the temperature rise of the windings of the rate temperature value (assumed to be 20 degrees Celsius); f – frequency of the supply voltage; u – the ratio of the supply voltage to the rated voltage U_{1N}.

The TDS block includes a model of a drive rotating a drill string and a model of interaction of the machine with guides that perceive a torque load (Fig. 3). The mathematical description of these models is reduced to the following equations:

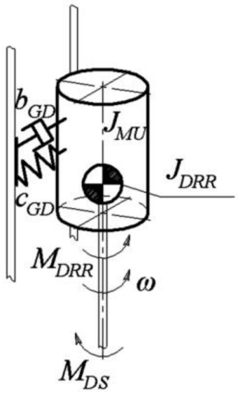

Fig. 3: TDS dynamic model

$$\begin{cases} J_{MU}\cdot\ddot{\varphi}_{MU} = -c_{GD}\cdot\varphi_{MU} - b_{GD}\cdot\dot{\varphi}_{MU} - M_{DS} \\ J_{DRR}\cdot\ddot{\varphi}_{DRR} = M_{DRR}(s,f,u,\rho,\tau)\cdot i_{DRR}/\eta_{DRR} - M_{DS} \\ \varphi_{TDS} = \varphi_{DRR} - \varphi_{MU} \end{cases} \qquad (2)$$

The result of solving the system (2) is the determination of the angle of rotation of the drill string φ_{TDS}. φ_{TDS} resulting from the rotation of its angle drive φ_{DRR} and the vibrations of the engine block itself φ_{MU} protected from rotation by the guides. Strictly speaking, the guide stiffness c_{GD}, is not constant and depends on the geometrical dimensions of the guides, the location of their fastenings on the derrick and the location of the power unit, which can move along the guides relative to the fasteners. Therefore, the definition of this parameter c_{GD}, as well as the viscous friction parameter b_{GD} is complicated by the uncertainity of these factors.

In this regard, we can neglect the oscillations of the mass of the machine unit J_{MU} considering the stiffness c_{NP} as very large, and the oscillations as very small (φ_{MU} = 0). Then the moment of resistance from the string M_{DS} will be fully compensated by the moment of the drive M_{DRR} generated, for example, by an electric induction motor, and by the inertial force, rotating driven to the output TDS shaft mass of the drive J_{DRR}. The torque of the motor in equations (2) is related to the output shaft, taking into account the gear ratio i_{DRR} and drive efficiency η_{DRR}.

The most important and difficult element to implement is the drill string model [7, 8, 9] and its interaction with the wellbore walls, which often consumes the greatest amount of energy generated by the rotation drive [10, 11].

The drill string consists of pipes of small cross-section and has a considerable length, augmenting with increasing of a well length.

Therefore, in the course of its work, its lower part, which rests in the bottom of a well, contracts and loses stability. The action of the rotating drive from the top and the external reverse action of the soil in the bottom from below makes it twist into a spiral. The described situation of a multiaxial deformed state requires an adequate physical and mathematical description.

The drill string model shown in Fig. 4, is a multi-mass system, which takes into account its torsional, flexural and longitudinal deformations under the action of various external influences.

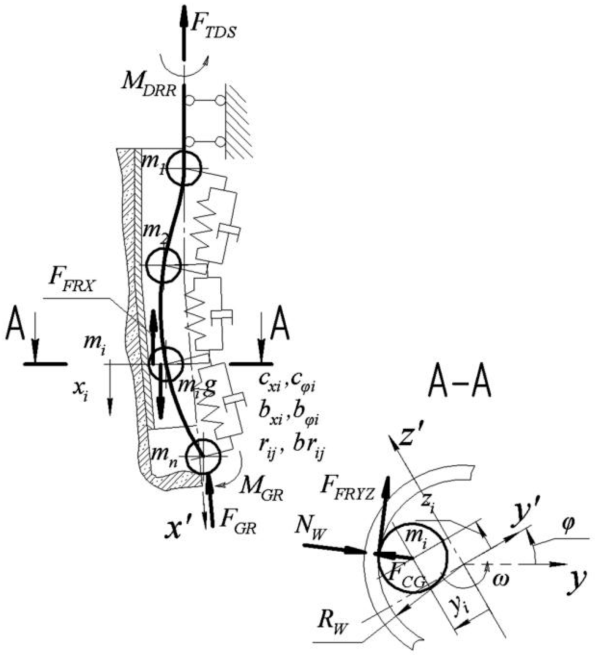

Fig. 4: Drill string dynamic model

The coordinate system is set in such a way that the axis Ox is directed along the axis of the string, which, in general, can be bent in the case of the curvilinearity of the simulated well. The axes Oy' and Oz' – are connected with the drill string fixedly and rotate in the absolute, for this example, coordinate system associated with the TDS model.

The string is divided into «*n*» discrete masses of interconnected elastic-viscous bonds. Each *i*-th discrete mass is characterized by inertial parameters m_i and J_i and radius R_i of a drill pipe, which serves to determine the moment of contact of the string with the wellbore walls. The mass movement is described by 3 coordinates in translational motion. x_i, y_i, z_i in the above coordinate system, characterizing the displacement from the position of elastic equilibrium at the initial moment of time. Angle of rotation φi characterizes the rotation of the coordinate plane Oy'z of each mass around its axis Ox'.

Parameters describing the elastic-viscous properties of the column: c_{xi} – the stiffness of the longitudinal elastic connection; $c_{\varphi i}$ – stiffness of torsional elastic bond; r_{ij} – coefficient of influence

j mass on *i* with transverse deformations, viscous drag coefficients b_{xi}, $b_{\varphi i}$, br_{ij} corresponding in the indicated directions of the system deformations.

Among the external forces impacting the discrete mass of the string, it is necessary to take into account the friction forces arising during the descent of the drill string F_{FRX} and its rotation F_{FRYZ}; wellbore reaction force N_W; centrifugal force F_{CG}, gravitational forces $m_i g$. At last, *n*, a mass characterizing the drilling tool and the bottom-hole motor, also act from the forces of resistance to cutting of the soil M_{GR} and reaction directed along the column F_{GR} equal to the feed power F_S. The first mass of the string, associated with the TDS block, is affected by the torque from M_{GR} and the force holding the string balanced F_{TDS}.

The complexity of building such a model is that these forces will not be stationary in the direction of action on each of the masses in the specified curvilinear rotating coordinate system Ox'y'z', and their magnitude will depend on the position of the system, its speed and medium properties at each moment of simulated time. For example, the magnitude of the friction forces will depend on the force of interaction of the mass under consideration with the wellbore wall, which may be absent if there is no such interaction. A force vector of its own weight $m_i g$ will be projected on all three axes of the coordinate system, and, due to its rotation, the components of the projections on the axes Oy' and Oz' will be variable.

Thus, the system of equations for the drill string must take into account the interrelated movements and oscillations of the drill string in three planes, plus rotation around the longitudinal axis

$$\begin{cases} Ox':\ m_i \cdot \ddot{x}_i = F_{EVXi} + F_{gXi} + F_{FRXi} \\ Oy':\ m_i \cdot \ddot{y}_i = F_{EVYi} + F_{gYi} + F_{AXYi} + N_{WYi} + F_{CGYi} + F_{FRYi} \\ Oz':\ m_i \cdot \ddot{z}_i = F_{EVZi} + F_{gZi} + F_{AXZi} + N_{WZi} + F_{CGZi} + F_{FRZi} \\ Oy'z':\ \left[J_i + m_i \cdot (y_i + z_i)^2\right] \cdot \ddot{\varphi} = M_{EVi} + M_{gi} + M_{FRi} \end{cases} \quad (3)$$

The magnitude of the elasticity and viscosity forces acting between the given masses of parts of the drill string are determined by the corresponding stiffness and viscous resistance coefficients, as well as by the positions of the adjacent masses determining the current system deformation

$$F_{EVXi} = c_{xi-1}(x_{i-1} - x_i) + c_{xi}(x_{i+1} - x_i) + b_{xi-1}(\dot{x}_{i-1} - \dot{x}_i) + b_{xi}(\dot{x}_{i+1} - \dot{x}_i);$$

$$F_{EVYi} = \sum_{j=1}^{n}\left[r_{i,j} \cdot y_i + br_{i,j} \cdot \dot{y}_i\right];$$

$$F_{EVZi} = \sum_{j=1}^{n}\left[r_{i,j} \cdot z_i + br_{i,j} \cdot \dot{z}_i\right];$$

$$M_{EVi} = c_{\varphi i-1}(\varphi_{i-1} - \varphi_i) + c_{\varphi i}(\varphi_{i+1} - \varphi_i) + b_{\varphi i-1}(\dot{\varphi}_{i-1} - \dot{\varphi}_i) + b_{\varphi i}(\dot{\varphi}_{i+1} - \dot{\varphi}_i).$$

The components of gravity, due to the possible different inclination of sections of the well at an angle β_i to the absolute vertical axis and rotation of the plane Oy'z' each ***i***-mass at angle φi, act in in all planes of the coordinate system Ox'y'z' of each given mass separately (Fig. 4):

$$F_{gXi} = m_i \cdot g \cdot \cos\beta_i;$$

$$F_{gYi} = -m_i \cdot g \cdot \sin\beta_i \cdot \cos\varphi_i;$$

$$F_{gZi} = -m_i \cdot g \cdot \sin\beta_i \cdot \sin\varphi_i.$$

The additional torque around the axis Ox' caused by mass displacements in its coordinate system will be determined by the relation

$$M_{gi} = F_{gYi} \cdot z_i - F_{gZi} \cdot y_i.$$

The magnitude of the interaction force of the drill string with the wellbore walls N_W will be determined by the stiffness coefficient of the walls c_W and their deformation Δ_W. The deformation of the wellbore walls can be determined from the geometric ratios

$$\Delta_W = \Delta - \sqrt{y_i^{\,2} + z_i^{\,2}},$$

where Δ is the nominal size of the gap between the drill string and the walls, equal to half the difference of their diameters.

To simulate the free rotation of the string when it does not interact with the walls of the well, the stiffness of the walls must change non-linearly according to the law

$$c_W(\Delta_W) = \begin{cases} = 0 & if\, \Delta_W < 0 \\ = c_W & else \end{cases}.$$

In this way $N_{Wi} = c_W(\Delta_{Wi}) \cdot \Delta_{Wi}$, and taking into account their projections along the axes of coordinates and neglecting the deformation of the wellbore walls, we have

$$N_{WYi} = c_W(\Delta_{Wi}) \cdot \Delta_{Wi} \cdot y_i / \Delta\,;\; N_{WZi} = c_W(\Delta_{Wi}) \cdot \Delta_{Wi} \cdot z_i / \Delta\,.$$

The forces and moments of friction arising due to the above-described interaction are determined by the coefficient of friction μ and the directions of movement of the contacting elements. Given the diameter of the well D_{WL} direction of the axes *i*-th mass and that N_{CTi} less than or equal to zero the expression of friction forces will take the form

$$F_{FRXi} = N_{Wi} \cdot \mu \cdot sign(\dot{x}_i)\,;\; F_{FRYi} = -N_{WZi} \cdot \mu \cdot sign(\dot{\varphi}_i)\,;$$
$$F_{FRZi} = N_{WYi} \cdot \mu \cdot sign(\dot{\varphi}_i)\,;\; M_{FRi} = N_{Wi} \cdot \mu \cdot sign(\dot{\varphi}_i) \cdot D_{WL} / 2\,.$$

The magnitudes of the centrifugal forces, which in this case can be neglected, along the corresponding axes of coordinates are determined by the known relations:

$$F_{CGYi} = m_i \cdot \dot{\varphi}_i^{\,2} \cdot y_i\,;\; F_{CGZi} = m_i \cdot \dot{\varphi}_i^{\,2} \cdot z_i\,.$$

Axial forces F_{AXYi}, F_{AXZi} caused by the curvature of the drill string in the respective planes during its deformation and are a known cause of buckling of the compressed rods. Their value is determined by both the values of the longitudinal forces in the column and its spatial configuration.

$$F_{AXYi} = -\left[F_{EVXi-1} \cdot (y_i - y_{i-1}) + F_{EVXi} \cdot (y_i - y_{i+1})\right] / dx\,;$$
$$F_{AXZi} = -\left[F_{EVXi-1} \cdot (z_i - z_{i-1}) + F_{EVXi} \cdot (z_i - z_{i+1})\right] / dx\,.$$

In the above ratios of the constituent axial forces, the assumption is made that the axial deformations of a string are small x_i compared to the distance between adjacent discrete masses, denoted as *dx*.

From equations (3), a general system of equations of mass of the drill string is compiled. As mentioned earlier, additional forces act on the first and last masses of the system, instead of discrete masses missing next to them: the holding force and the torque of the drive act on the first mass, and the result of the interaction of the working body with the soil in the bottomhole – on the last.

As can be seen from the presented ratios, the efforts caused by deformations in one plane affect the magnitude of other components of the forces acting in other planes, which leads to a complication of the general method of calculation.

Fig. 5 shows the general information flow diagram of the relationship between the blocks of calculation of the components of the deformations of the drill string, built on the general system of equations (3).

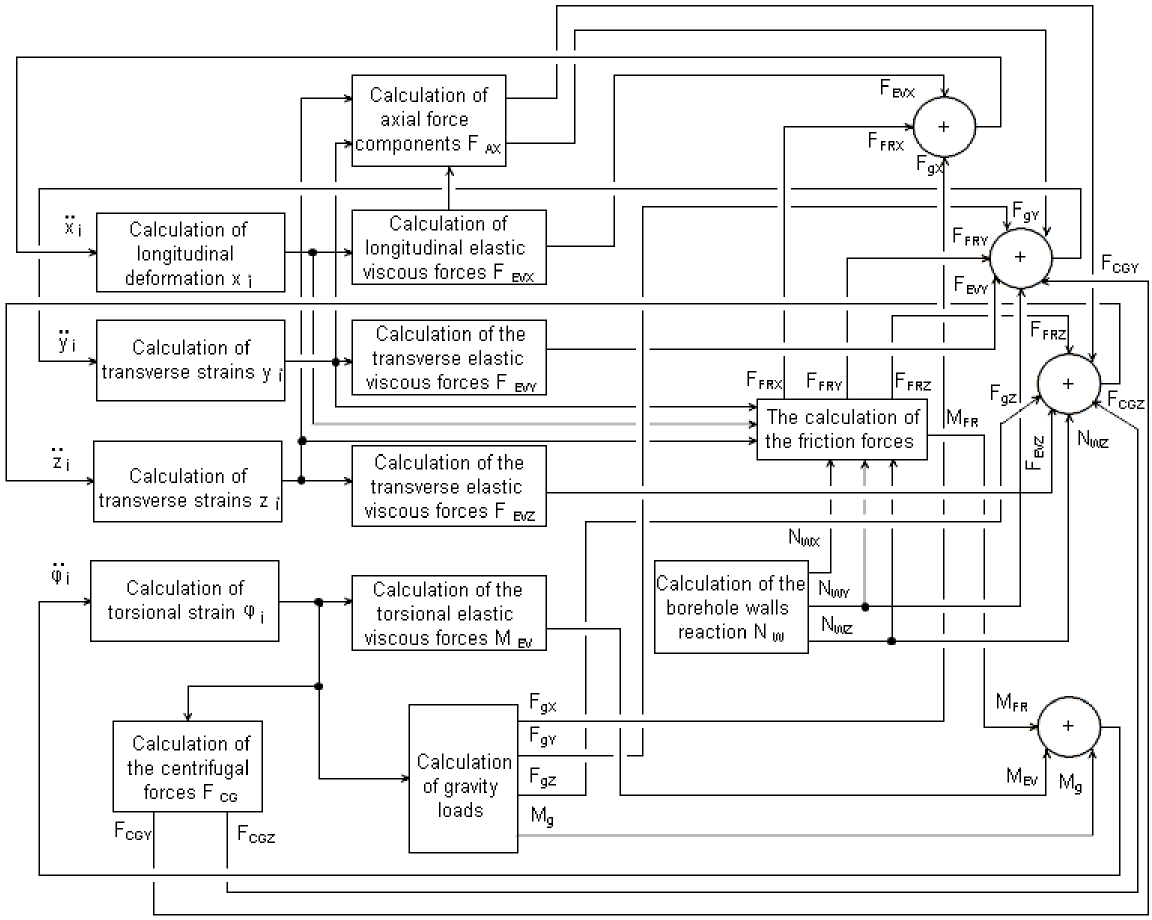

Fig. 5: Diagram of the analytical model of the drill string

Impact on the soil is usually formed at the bottom, at the bottomhole, where the bottom-hole motor is located. It, under the action of the flushing fluid flow, descending through the channel of the drill pipe, rotates the drilling tool and creates the necessary torque to perform the cutting process. The model of a downhole motor must take into account that its mechanical characteristic is not rigid and with the same flow rate of working fluid and different loads the speed of rotation of the output shaft will be different.

For the construction of the mechanical characteristics of a downhole motor, it is possible to use its energy characteristics: the operating range of the flow rate of the washing fluid (from Q_1 to Q_2), corresponding to their idling speed n_{01} and n_{02} and rotation speed n_1 and n_2 at the torque M_{MDPD} corresponding to the maximum allowable differential pressure drop. By interpolating these characteristics, torque can be obtained. M_{DM} from the feed flow Q and current speed n downhole motor output shaft:

$$M_{DM}(Q,n) = \\ = M_{MDPD} \cdot \frac{(n_{01}-n)(Q_2-Q_1)+(n_{01}-n_{02})(Q-Q_1)}{(n_{01}-n_1)(Q_2-Q_1)+(n_2-n_1-n_{02}+n_{01})(Q-Q_1)} \quad (4)$$

The resulting expression reflects the fact that between the speed of rotation and the effective torque of the downhole motor, a linear relationship is assumed, as a first approximation. But for different values of drilling fluid waste, the parameters of this dependence, the slope and the point of intersection of the line with the axes of coordinates, will be different (Fig. 6). Relationship (4) allows building intermediate mechanical characteristics of a downhole motor also assuming a linear relationship between the rotational speed of the output shaft n and expense Q of a drilling fluid.

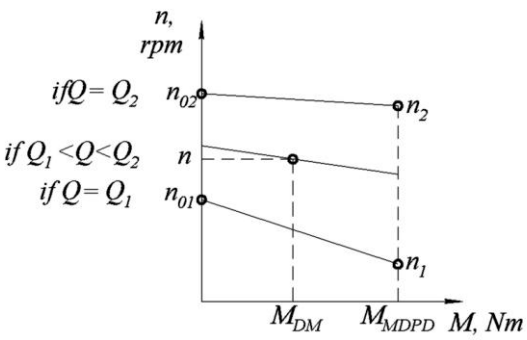

Fig. 6: Linear mechanical characteristic of the downhole motor

The model of a downhole motor can be represented as an inertial model

$$J_{DM} \cdot \ddot{\varphi}_{DM} = M_{DM}(Q, \varphi = \pi \cdot n/30) - M_{GR}(F_{\Pi}),$$

and the inertialess model, assuming that the moment created by the downhole motor is completely transmitted to the bottom of the drill string and is equal to the moment of the destruction of the soil $M_{DM}(Q, n) = M_{GR}(F_S)$. In the latter case, the equation of moments serves to determine the frequency of rotation of the cutting tool, and hence the speed of its deepening in the bottomhole.

It is well known that the moment of resistance to cutting soil M_{GR} depends on feed effort F_S. For their definition, for example, for cutting drilling tools, the known relations are used [12]:

$$M_{GR}(F_S) = 0{,}5 \cdot D \cdot F_Z \cdot Z \cdot (1+\mu_{GR});$$

$$F_S = 0{,}25 \cdot K_d \cdot D \cdot h \cdot \sigma_{M.D.}, \quad (5)$$

the parameters of which are D - diameter of the drilling tool; Z_B – the number of cutting blades; μ_{GR} – coefficient of friction of steel on the ground; K_D – coefficient of blunting of the cutting tool; h – deepening of cutting blades in the ground; σ_{MD} – the ultimate strength of the rock during the mechanical method of drilling.

Drilling rate v depends on the depth of introduction of the tool h, the number of cutting blades Z_{Π} and rotational speed of the drilling tool n:

$$v = h \cdot Z \cdot n.$$

The rotational speed, in this case, will be the sum of the rotational speed of the downhole motor and the rotational speed of the drill string.

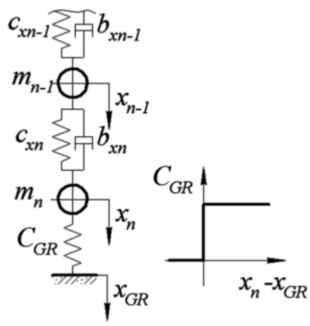

Fig. 7: To the dynamic model of the soil in the bottomhole

To create a model of the interaction of the drilling tool with the ground (Fig. 7), it is proposed to determine from the expression (5) the rigidity of the soil during the destruction of the rock, as the ratio of force to deformation

$$C_{GR} = \frac{F_S}{h} = 0{,}25 \cdot K_d \cdot D \cdot \sigma_{M.D.},$$

then the depth of penetration of the tool into the ground $h = x_n - x_{GR}$ and feed effort

$F_s = C_{GR} \cdot h.$

The position of the bottomhole in this case will be determined by the integral dependence on the speed of drilling v on time

$$x_{GR} = x_{0GR} + Z \cdot \int h^*(t) \cdot n(t)dt \quad (6)$$

However, according to the theory [12], the bulk destruction of the rock begins only when the feed force is greater than the minimum value determined by empirical dependence

$$F_{S\,MIN} = 0{,}5 \cdot \sigma_{M.D.} \cdot S_d,$$

taking into account the coefficient of blunting the edges of the drilling tool S_D. Therefore, when calculating the position of the bottomhole it is necessary to take

$$h^*(t) = \begin{cases} 0 & if\ F_S < F_{S\,MIN} \\ h & else \end{cases}.$$

The equations presented above describe the mathematical component of the blocks of the general model of a drilling rig with an upper drive and a downhole motor. In addition to the mathematical component, to create an adequate model, it is required to determine, with sufficient reliability for scientific or engineering calculations, the resistance coefficients, mass, hardness and other parameters of the system. Many of them are pre-selected from the relevant reference books or determined by static calculation of the simulated structures. For example, to determine the given mass of a derrick, it is required to calculate the eigenfrequency of its construction and equate it with the eigenfrequency corresponding to a single mass system.

The refinement of the adopted coefficients will be facilitated by the modeling of various situations with the subsequent comparison of the results with the data obtained from the monitoring systems of real machines [11]. Such studies will expand the theoretical

knowledge of the drilling processes, creating the necessary engineering techniques.
Since the system described above is quite complex, the definition of the initial conditions also causes known difficulties. Therefore, a phased model calculation is proposed.
At the first stage, all the generalized coordinates, and hence the efforts in the design are considered to be zero. There is no rotation of the drill string and the interaction with the bottomhole during the calculation. As a result, after the attenuation of the oscillations arising during the simulation, the position of the machine will be found, which is taken as the initial conditions at the next stages of the workflow simulations.
Further, various situations can be modeled and investigated: drilling with a downhole motor without rotating the drill pipe, drilling with a downhole motor together with rotating the drill pipe with various operating conditions, depths and well configurations, with changing properties of the drilled rock.
The purpose of the calculations in these situations can be both the specification of the model parameters and the study of possible loads, the identification of the optimal operational parameters of the installation, the theoretical testing of the model of the installation management system before it is realized.
The project is being implemented by Federal State Autonomous Educational Institution for Higher Education "Peter the Great St. Petersburg Polytechnic University" together with an industrial partner of **Specialnoe konstruktorskoe byuro priborov podzemnoj navigacii, AO**. The commissioner is the Ministry of Education and Science of the Russian Federation. Project Theme: “Development and research of the principles of trajectory curvature vector control when drilling small diameter wells using rotary steerable systems” (Agreement No. 14.575.21.0138 of September 26, 2017. UPI RFMEFI57517X0138). This project is being implemented with financial support from the Ministry of Education and Science of the Russian Federation.

3. Conclusion

For the first time, a mathematical model of a drilling rig with a top drive system is presented from a unified position. However, it requires in-depth analysis, improvement and careful determination of the values of the coefficients included in the above equations.
The implementation of such a model will allow it to be used in the optimal design of the mechanical part and the creation of control systems for a drilling rig with the top drive.

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