



Development of Mathematical Model for Controlling Drilling Parameters with Screw Downhole Motor

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ABSTRACT

Present article results of study on possibility of increasing the efficiency of drilling directional straight sections of wells using screw downhole motors (SDM) with a combined method of drilling with rotation of drilling string (DS). Goal is to ensure steady-state operation of SDM with simultaneous rotation of DS by reducing the amplitude of oscillations with adjusting the parameters of drilling mode on the basis of mathematical modeling for SDM – DS system. Results of experimental study on determination of extreme distribution of lateral and axial oscillations of SDM frame depending on geometrical parameters of gerotor mechanism and modes ensuring stable operation are presented. Approaches to develop a mathematical model and methodology are conceptually outlined that allow determining the range of self-oscillations for SDM – DS system and boundaries of rotational and translational wave perturbations for a heterogeneous rod with an installed SDM at drilling directional straight sections of well. This mathematical model of SDM – DS system's dynamics makes it possible to predict optimal parameters of directional drilling mode that ensure stable operation of borehole assembly.

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NOMENCLATURE

z_1, z_2	Number of rotor / stator teeth	H	Well depth (m)
ω_r	Angular rotation velocity of rotor around its own axis (s^{-1})	λ_{Ln}	Speed of rotation oscillation transmission (m/s)
M_{ind}	Indicator moment (kN·m)	$\varphi_n(s_n, t)$	Angular deflection current cross-section column on according part (rpm)
e	Eccentricity (m)	s_n	Current position cross-section (m)
m	Rotor mass (kg)	$u_n(s_n, t)$	Translational movements of current cross-sections of string in corresponding sections (rad)
ω	Angular velocity (s^{-1})	h	Value of translational movement with transfer
D	Stator diameter at tooth cavities (m)	$f_{\tau_n} \left(\frac{\partial \varphi_n}{\partial t} \right)$	Dissipative term described to resist of drilling string rotation on according part
P_p	Pressure difference (Pa)	$f_{\tau_n} \left(\frac{\partial u_n}{\partial t} \right)$	Dissipative members characterizing the resistance of drilling string translational movement
t	Rotor pitch (m)	n_0	Rotation speed for upper end of string (rad/s, rpm)
A	Amplitude (mm)	G_1, G_2, G_3	Modulus of rigidity material according part (N·m ²)
L_a	Level vibration acceleration	E_1, E_2, E_3	Elastic modulus of materials in corresponding sections under tension or compression (kg/m ²)
L_v	Level vibration speed	J_1, J_2, J_3	Polar moment of inertia cross-section column on according part (m ⁴)
$v_0 = 5 \cdot 10^{-8}$	Backup value vibration speed (m/s)	$M_H(P, n_H)$	Moment of resistance bottom composite rod rotation from side of rock (N·m)
$a_0 = 1 \cdot 10^{-6}$	Backup value vibration acceleration (m/s ²)	P	Axial load on the end of composite rod lower section (N)
a	Medium-square value vibration acceleration (m/s ²)	F_1, F_2, F_3	cross-sectional area of string in corresponding sections (m ²)
v	Medium-square value vibration speed (m/s)	n_H	Rotation speed bottom part composite rod (rpm /s)

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n	Frequency rotation (s^{-1} , rpm, Hz)	$\theta = \frac{G_1 J_1}{G_2 J_2}$ $\theta = \frac{E_1 F_1}{E_2 F_2}$	Coefficient of moment-force ratio of the first and the second sections during rotation / translational movement
f	Medium geometrical frequency octave filter (dB)	$\varepsilon = \frac{G_2 J_2}{G_3 J_3}$ $\varepsilon = \frac{E_1 F_1}{E_2 F_2}$	Coefficient of moment-force ratio of the second and the third sections during rotation / translational movement
M	Resistance moment (kN·m)	μ_1, μ_2, μ_3	Coefficient of dissipation on according part composite rod
n_0	Rotor rotation speed (rpm/s)	k, k_1, k_2	Coefficient spect wave circling indignation on boundary heterogeneous parts composite rod
t	Time (s)	$\Delta M_H = M_H(P, 0) - M_H(p, n_0)$	Margin between evict moment end bottom part and nominee moment that end (N·m)
L_1, L_2, L_3	DS length, drill collar length, length of SDM body and navigation and measuring tool (m)	P_b, P_H	Axial loads according top and bottom boundary auto-oscillation (N)
D_{L1}, D_{L2}, D_{L3}	DS, drill collar, SDM body and navigation and measuring tool external diameters (mm)	n_0^*	Rotor speed rotation, when $P_b = P_H$.
d_{L1}, d_{L2}, d_{L3}	DS, drill collar, SDM body and navigation and measuring tool internal diameters (mm)		

1. INTRODUCTION

Increasing hydrocarbon production due to developing offshore fields, as well as additional development of previously drilled areas is only achieved by construction of complex well profiles which trajectories may contain curved or straight sections of long distances [1]. Implementation of such profiles involves the use of a rotary steerable system (RSS), or SDM as a drill bit (DB) drive. The use of expensive RSS [2] is economically unsuitable. Therefore, in most cases (70-80 %) such wells are drilled by using SDM in Russia.

SDM – hydraulic downhole motor of volumetric type, multiple thread working bodies of which are made according to the scheme of the gerotor planetary mechanism, driven by the energy of the drill fluid.

During drilling of extended directional and horizontal sections of wells using volumetric principle engines, part of axial load on the bit is not transmitted due to frictional force arising between walls of well and drilling tool [3].

To ensure required load on the bit a combined drilling method is used in production process. Distinctive feature of this method is in the joint operation of drilling string (DS) and screw down-hole motor (SDM) [4]. In the process of their joint work, torsional, lateral and axial oscillations can occur depending on type of SDM, its energy characteristics and DS, which acts as an elastic unbalanced rod [5].

It should be noted that SDM that is located in lower part of DS has its own beating of the frame, nature of occurrence of which is associated with work of its power section, represented by a planetary reductor. Moreover, frequency, amplitude and direction of the frame beats depend on design of gerotor mechanism, hydraulic component of drilling mud flow, as well as load on the bit [6].

To determine parameters of well drilling mode by a combined method, it is necessary to develop a technique

that allows providing forecast and control of stable operation of bottom hole assembly (BHA), based on mathematical modeling of elastic properties of DS stress-strain state and characteristics of SDM [7, 8].

Specifics of the installed SDM in the BHA, power section lead to independence of axial and lateral oscillations characteristics from DS and drill bit.

Rotation of long elastic rod with different rigidity is limited by the wellbore wall with alternating stress-deformation state (SDS) causing the occurrence of oscillations. DB amplitude changes are difficult to determine by math methods. In practice to measure DB vibration a three-position accelerometer is placed into telemetric tool, which allows to control the acceleration of BHA. According passport, for geophysical well logging to avoid damage of column elements, the vibration acceleration should be in the range of 30-45 G. Resulting from the large values of vibration acceleration, it is difficult to accurately determine the load on the bit. Maintaining the required vibration acceleration is possible due to control dynamic of system «SDM-DS».

Oscillations in the DB, SDM and DS in critical range of vibration [9], acceleration leads to tool instability, adversely affect the formation wellbore walls, decreased quality of well trajectory control and increased risk of accidents as a result of decoupling in screw connections damages to BHA elements.

The dynamics of the DS is due to mechanical energy transmitted from the top power drive while the SDM is based on the conversion of the energy of the process fluid stream pumped by the drilling pump units.

Scientists from the beginning of 17th century studied the dynamics of DS while drilling. The method for research the elastic-deformed state of a drill string represented by a one-dimensional core was formulated by Euler and continues to improve [10]. The increment of the potential energy of the drilling tool in the form of a mechanical system and deviations thereof from the

balance position are recorded and proved in the form of the theorem by Laplace-Dirichlet. For the considered section of the DS, the method proposed by Leibenzon, which determined the nature of rotation, is known in literature [11]. The definition of the area of stable operation of the DS by the mechanical analogue method was developed by Yunin and Khagai [10].

The great contribution to the design, creation and improvement of gerotor machines, as well as to the research of the working processes of the SDM for drilling and workover, was made by domestic and international scientists [11, 12].

Presently, there is a large amount of information on the research of increasing the operating time of the SDM and on ways to increase the efficiency of the motor [11-14]. Much attention is also paid to search the dynamics of the DS and SDM during drilling of deviated and horizontal wells with geomechanical studies [12, 14-17]. The results of these researches shown that the negative effect of vibration and oscillations on the transmission of axial load on the bit can be reduced due to the operational control of the dynamics of the DS, SDM and DB. Therefore, the main objectives of this work are to increase drilling efficiency and reduce the likelihood of accidents inside the well by controlling the dynamics of the DB-SDM-DS system by optimizing drilling parameters (rotation per minute, mathematical models of axial stress) during drilling deviated and horizontal wells.

To achieve these objectives, following analysis should be conducted:

1. researches of SDM wobbling with different work modes;
2. investigate the oscillation of the DS considering its SDS and drilling parameters;
3. develop technology to regulate torque – power and frequency of DS and SDM.

As a result of the conducted research, a model for regulating and controlling the dynamics of the «SDM-DS» system was developed while drilling directional straight sections of the well, which makes it possible to increase the efficiency of drilling directional straight sections of the well.

2. MATERIALS AND METHODS

Stability of SDM operation is characterized by working mode of power section, in which there is no intensive decrease in rotor rotation frequency with increasing torque on motor shaft.

It is known that axis of rotor rotates around its own axis, and also makes a transferring movement around axis of stator, directed counterclockwise. Moreover, frequency of transferring (planetary) rotation of rotor's

axis relative to stator's axis is higher than rotor rotation frequency around its own axis.

Angular rotation velocity of rotor's axis relative to stator's axis, which determines beat frequency of the frame,

$$\omega_n = z_1 \cdot \omega_r. \quad (1)$$

Motor's frame beats depend on inertial F_{in} and hydraulic F_h forces acting on rotor,

$$F_{in} = mz_2\omega^2 e; \quad (2)$$

$$F_h = M_{ind} e z_1. \quad (3)$$

During engine start, a skew moment arises, causing instability of rotor rolling along stator teeth and leading to additional beating of SDM frame.

Skew moment is:

$$M_n = \frac{P \cdot D \cdot t^2}{4\pi}. \quad (4)$$

Experimental study of motor's frame beats is performed at the test bench (Figure 1) [6]. Bench is equipped with an automatic control system that provides real-time output of SDM main energy characteristics to panel of a personal computer. To study beats of SDM, oscillation sensors are installed on frame.

30 SDM with diameters from 156 to 195 mm were tested to determine their optimal stable operation. As an example, the results of investigations on the energy characteristics of the motor with shortened spindle and adjustable unit curvature – 178M.7/8.37 with synchronous measurement of the wobbling on the body (Figures 2-4). Vibration measurement sensors (you can see on Figure 1: 10, 11 and 12 positions) has been installed on three points of the motor. Two sensors

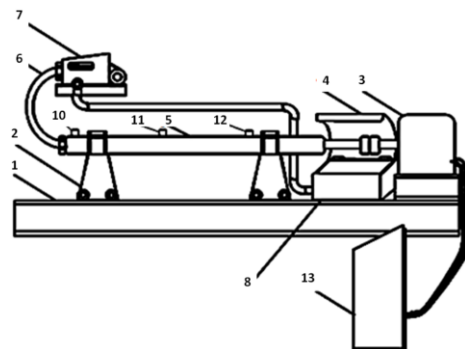


Figure 1. Test bench: 1 – installation platform; 2 – compressor; 3 – electromagnetic powder breaker; 4 – hydro-chisel; 5 – SDM; 6 – pipe line; 7 – pump; 8 – receiver tank; 10, 11, 12 – vibration measurement sensors; 13 – hardware and software complex (information processing module)

$$7.\varphi_1(s_1, t=0) = \frac{f_{\tau_1}(n_0)}{2\lambda_{L_1}^2} \cdot s_1^2 - \left[\frac{f_{\tau_1}(n_0)L_1}{\lambda_{L_1}^2} + \theta \left[\frac{f_{\tau_1}(n_0)L_2}{\lambda_{L_2}^2} + \varepsilon \left(\frac{f_{\tau_1}(n_0)L_3}{\lambda_{L_3}^2} + \frac{M_H(P, n_H)}{G_3 J_3} \right) \right] \right] s_1,$$

$$8.\varphi_2(s_2, t=0) = f_1(L_1) + \frac{f_{\tau_2}(n_0)}{2\lambda_{L_2}^2} \cdot s_2^2 - \left[\frac{f_{\tau_2}(n_0)L_2}{\lambda_{L_2}^2} + \varepsilon \left(\frac{f_{\tau_2}(n_0)L_3}{\lambda_{L_3}^2} + \frac{M_H(P, n_H)}{G_3 J_3} \right) \right] s_2,$$

$$9.\varphi_3(s_3, t=0) = f_1(L_1) + f_2(L_2) + \frac{f_{\tau_3}(n_0)}{2\lambda_{L_3}^2} \cdot s_3^2 - \left(\frac{f_{\tau_3}(n_0)L_3}{\lambda_{L_3}^2} + \frac{M_H(P, n_H)}{G_3 J_3} \right) s_3,$$

$$s_1 \in [0, L_1], s_2 \in [0, L_2], s_3 \in [0, L_3]$$

$$\frac{\partial \varphi_1}{\partial t} = n_0, \frac{\partial \varphi_2}{\partial t} = n_0, \frac{\partial \varphi_3}{\partial t} = n_0.$$

Initial conditions for translational motion at $t = 0$:

$$7.u_1(s_1, t=0) = \frac{f_{\tau_1}(n_0)}{2\chi_{L_1}} \cdot s_1^2 - \left[\frac{f_{\tau_1}(n_0)L_1}{\chi_{L_1}^2} + \theta \left[\frac{f_{\tau_1}(n_0)L_2}{\chi_{L_2}^2} + \varepsilon \left(\frac{f_{\tau_1}(n_0)L_3}{\chi_{L_3}^2} + \frac{P(n_H)}{E_3 F_3} \right) \right] \right] s_1,$$

$$8.u_2(s_2, t=0) = f_1(L_1) + \frac{f_{\tau_2}(n_0)}{2\chi_{L_2}} \cdot s_2^2 - \left[\frac{f_{\tau_2}(n_0)L_2}{\chi_{L_2}^2} + \varepsilon \left(\frac{f_{\tau_2}(n_0)L_3}{\chi_{L_3}^2} + \frac{P(n_H)}{E_3 F_3} \right) \right] s_2,$$

$$9.u_3(s_3, t=0) = f_1(L_1) + f_2(L_2) + \frac{f_{\tau_3}(n_0)}{2\chi_{L_3}} \cdot s_3^2 - \left(\frac{f_{\tau_3}(n_0)L_3}{\chi_{L_3}^2} + \frac{P(n_H)}{E_3 F_3} \right) s_3,$$

$$s_1 \in [0, L_1], s_2 \in [0, L_2], s_3 \in [0, L_3]$$

$$\frac{\partial u_1}{\partial t} = \chi_{L_1} u_1, \frac{\partial u_2}{\partial t} = \chi_{L_2} u_2, \frac{\partial u_3}{\partial t} = \chi_{L_3} u_3,$$

where $n_H = \frac{\partial \varphi_3}{\partial t} \Big|_{s_3=L_3}$ – rotation frequency for end of composite rod lower section.

This problem is most clearly solved for case in which the values of dissipative terms of system are equal to zero. Following equations are used:

$$H = \frac{\lambda_1}{\mu_1} \ln \left\{ \frac{\lambda_1 \left[\frac{\lambda_2}{\mu_2} \ln \left(\frac{\lambda_3 \left[\frac{\lambda_3 \ln \lambda_3 \Delta M_H + G_3 J_3 \cdot n_H}{\lambda_3 \Delta M_H - G_3 J_3 \cdot n_H} \right] + G_2 J_2 \cdot n_0 \right)}{\lambda_3 \left[\frac{\lambda_3 \ln \lambda_3 \Delta M_H + G_3 J_3 \cdot n_H}{\lambda_3 \Delta M_H - G_3 J_3 \cdot n_H} \right] - G_2 J_2 \cdot n_0} \right) + G_1 J_1 \cdot n_0}{\lambda_1 \left[\frac{\lambda_2}{\mu_2} \ln \left(\frac{\lambda_3 \left[\frac{\lambda_3 \ln \lambda_3 \Delta M_H + G_3 J_3 \cdot n_H}{\lambda_3 \Delta M_H - G_3 J_3 \cdot n_H} \right] + G_2 J_2 \cdot n_0 \right)}{\lambda_3 \left[\frac{\lambda_3 \ln \lambda_3 \Delta M_H + G_3 J_3 \cdot n_H}{\lambda_3 \Delta M_H - G_3 J_3 \cdot n_H} \right] - G_2 J_2 \cdot n_0} \right) - G_1 J_1 \cdot n_0} \right\}, \quad (8)$$

$$H \leq \frac{\lambda_1}{\mu_1} \ln \left\{ \frac{\lambda_1 \left| \frac{\lambda_2}{\mu_2} \ln \left(\frac{\lambda_3 \left[\frac{\lambda_3 \ln \lambda_3 \Delta M_H + G_3 J_3 \cdot n_H}{\lambda_3 \Delta M_H - G_3 J_3 \cdot n_H} \right] + G_2 J_2 \right)}{\lambda_3 \left[\frac{\lambda_3 \ln \lambda_3 \Delta M_H + G_3 J_3 \cdot n_H}{\lambda_3 \Delta M_H - G_3 J_3 \cdot n_H} \right] - G_2 J_2} \right) + G_1 J_1 \cdot n_0}{\lambda_1 \left| \frac{\lambda_2}{\mu_2} \ln \left(\frac{\lambda_3 \left[\frac{\lambda_3 \ln \lambda_3 \Delta M_H + G_3 J_3 \cdot n_H}{\lambda_3 \Delta M_H - G_3 J_3 \cdot n_H} \right] + G_2 J_2 \right)}{\lambda_3 \left[\frac{\lambda_3 \ln \lambda_3 \Delta M_H + G_3 J_3 \cdot n_H}{\lambda_3 \Delta M_H - G_3 J_3 \cdot n_H} \right] - G_2 J_2} \right) - G_1 J_1} \right\}$$

That equation defines the occurrence possibility of torsional auto-oscillation in the composite DS during its rotational movement for well deepening purpose.

Equations (8) are found for the special case, when dissipative terms in Equation (7) are equal to zero. Results of Equations (8) are converged with the investigation results. Therefore, it is now possible to determine the value of parameters in the steady work mode of active dynamic system «SDM-DS». Then we have:

$$\left\{ \begin{aligned} P &= \frac{G_3 J_3 (n_0^2) (1 + ke \frac{\mu_3 L_3}{2\lambda_3})}{\lambda_3 \left(1 - ke \frac{\mu_3 L_3}{2\lambda_3} \right)}, \\ P_b &= \frac{1 + ke \frac{\mu_3 L_3}{2\lambda_3} P \cdot \left(ch \left(\frac{\mu_1 L_1}{2\lambda_1} + \frac{\mu_2 L_2}{2\lambda_2} + \frac{\mu_3 L_3}{2\lambda_3} \right) + kch \left(\frac{\mu_1 L_1}{2\lambda_1} - \frac{\mu_2 L_2}{2\lambda_2} - \frac{\mu_3 L_3}{2\lambda_3} \right) \right)}{1 - ke \frac{\mu_3 L_3}{2\lambda_3} n_0 \left(sh \left(\frac{\mu_1 L_1}{2\lambda_1} + \frac{\mu_2 L_2}{2\lambda_2} + \frac{\mu_3 L_3}{2\lambda_3} \right) + ksh \left(\frac{\mu_1 L_1}{2\lambda_1} - \frac{\mu_2 L_2}{2\lambda_2} - \frac{\mu_3 L_3}{2\lambda_3} \right) \right)}, \\ P_H &= \frac{1 + ke \frac{\mu_3 L_3}{2\lambda_3} P \cdot \left(sh \left(\frac{\mu_1 L_1}{2\lambda_1} + \frac{\mu_2 L_2}{2\lambda_2} + \frac{\mu_3 L_3}{2\lambda_3} \right) + ksh \left(\frac{\mu_1 L_1}{2\lambda_1} - \frac{\mu_2 L_2}{2\lambda_2} - \frac{\mu_3 L_3}{2\lambda_3} \right) \right)}{1 - ke \frac{\mu_3 L_3}{2\lambda_3} n_0 \left(ch \left(\frac{\mu_1 L_1}{2\lambda_1} + \frac{\mu_2 L_2}{2\lambda_2} + \frac{\mu_3 L_3}{2\lambda_3} \right) + kch \left(\frac{\mu_1 L_1}{2\lambda_1} - \frac{\mu_2 L_2}{2\lambda_2} - \frac{\mu_3 L_3}{2\lambda_3} \right) \right)}, \\ n_0^* &= \frac{1 - k^2}{sh^2 \left(\frac{\mu_1 L_1}{2\lambda_1} + \frac{\mu_2 L_2}{2\lambda_2} + \frac{\mu_3 L_3}{2\lambda_3} \right) + kch^2 \left(\frac{\mu_1 L_1}{2\lambda_1} - \frac{\mu_2 L_2}{2\lambda_2} - \frac{\mu_3 L_3}{2\lambda_3} \right)}; \end{aligned} \right. \quad (9)$$

Task for case, in which the values of dissipative members of system are equal to zero, and propagation depth of translational oscillations of drilling tool, represented as a composite rod of three heterogeneous sections, is solved by system (8). At the same time G_1, G_2, G_3 are replaced by E_1, E_2, E_3 and J_1, J_2, J_3 by F_1, F_2, F_3 , and also $\lambda_{L_1}, \lambda_{L_2}, \lambda_{L_3}$ propagation velocity of rotational oscillations by propagation velocity of translational oscillations by $\chi_{L_1}, \chi_{L_2}, \chi_{L_3}$ in corresponding sections. Obtained equations determine conditions for occurrence possibility of translational self-oscillations of DS, represented as a composite rod in process of translating to deepen bottomhole of the well [19]. Axial loads on lower end of SDM frame, corresponding to upper and lower boundaries of self-oscillations during translational movement of P_B and P_H , are determined by Equation (9). At the same time propagation velocity of rotational oscillations $\lambda_{L_1}, \lambda_{L_2}, \lambda_{L_3}$ is replaced by $\chi_{L_1}, \chi_{L_2}, \chi_{L_3}$, elastic modulus G_1, G_2, G_3 and polar moments of inertia in cross-section J_1, J_2, J_3 are replaced by E_1, E_2, E_3 and F_1, F_2, F_3 , respectively.

Developed methodology of determining required parameters for drilling mode of inclined sections in a well, ensuring stable operation of BHA, is as follows.

SDM is started and pressure drop is determined during its operation in idle mode. Then, required load on the bit (according to work plan and geological and technical schedule) is created and pressure drop is fixed taking into account loading of gerotor mechanism. On the basis of SDM test bench diagram, optimal range of shaft rotation frequency with corresponding pressure drop is graphically determined. At the same time, maximum allowable decrease in rotation frequency of SDM shaft is noted, which corresponds to optimum amplitudes of frame lateral oscillations.

According to mathematical model developed, boundaries of DS self-oscillation onset are calculated.

After constructing the graphical dependencies, required frequency and load on the bit are determined, at which DS is in permissible range of stable operation. Noting modes of DS stable operation, correlation is made with load on the bit, at which SDM will also be in mode of optimal energy characteristics. If rotation frequency of SDM shaft (according to test bench diagram), determined by pressure drop, has decreased by more than 70 %, load on the bit is reduced. Based on graphical dependences for boundaries range of self-oscillations' onset at given rotation frequencies of DS and load on the bit, rotation frequency of top drive is adjusted to ensure stable operation of the system while maintaining mechanical drilling speed [18].

3. RESULTS AND DISCUSSION

Results of ordeal work process hydro machine with consideration of its vibration acceleration and amplitudes body wobbling on different work mode are illustrated in Figure 3.

The results of investigation on the motor vibration with shortened spindle and adjustable unit curvature – 178M.7 / 8.37 engine body showed that the values of vibration accelerations along the body are different from each other. So, for example, in the upper part of the

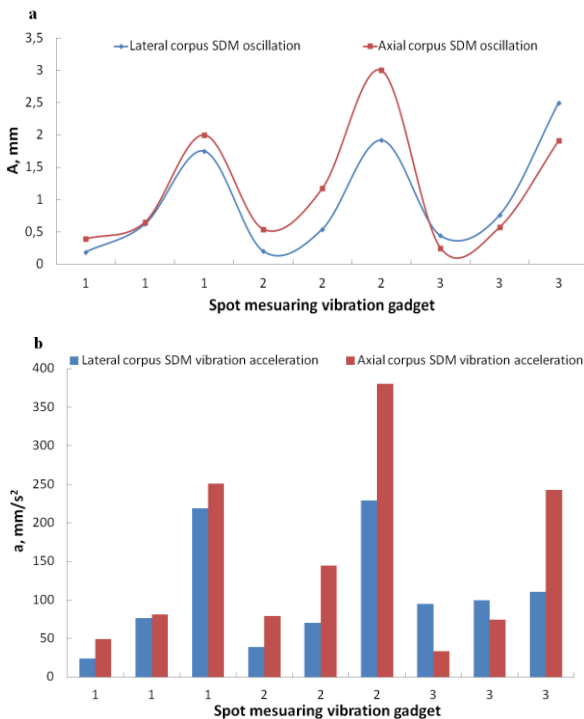


Figure 3. Amplitude (a) and vibration acceleration (b) of frame depending on sensor installation location at the motor's frame: 1 – upper SDM sub; 2 – middle of working bodies' active part of SDM; 3 – upper spindle sub

power section for a shaft rotation frequency of 5 s^{-1} (300 rpm), the values of vibration accelerations of the transverse vibrations of the housing vary from 24 to 219 mm/s^2 , which corresponds to a change in the beat amplitudes from 0, 19 to 1.75 mm, and the values of vibration accelerations of the longitudinal vibrations of the body vary from 49 to 251 mm/s^2 (Figure 3b), which corresponds to a change in the amplitudes of the wobbling from 0.39 to 2 mm (Figure 3a).

The maximum values of the amplitudes and vibration accelerations of the lateral and axial body vibrations are determined at a frequency of 5 s^{-1} (300 rpm) in the middle part of the power section are 1.92 and 3 mm, 229 and 380 mm/s^2 , respectively.

Results of research changes amplitudes lateral and axial oscillation depending on moment shaft of SDM as shown in Figure 4.

Analysis of investigation results showed optimum frequency interval of SDM body beats to be from 35 to 24.5 Hz. Axial and lateral vibrations depend on the moment on shaft SDM. In the regime of motor work, amplitude of lateral body wobbling is 5 mm, while the amplitude of axial oscillations is not more than 3 mm. This is due to the act of skew moments on the working elements. Resistance moment on shaft SDM made descend amplitude lateral vibration corpus to 3.5-4 mm, amplitude axial vibration increases to 7.3-8 mm. Increasing resistance moment from 1 to 4.5 kN·m motor body lateral vibration to 5-6 mm and descend corpus motor axial vibration to 7.3-8 mm. The wobbling frequency is reduced to 24.5 Hz (210 rpm), it is 30% power of idle SDM – optimum exploitation SDM (see Figure 4). With an increase in torque from 4.5 to 9 kN·m, the engine enters the braking (extreme) mode of operation. The wobbling frequency is reduced to 3.5 Hz (30 rpm). As a result, there is an intensive increase in the amplitude of the lateral vibrations of the SDM body from 6 to 10 mm which a corresponding decrease in the amplitude of axial vibrations from 8 to 2 mm.

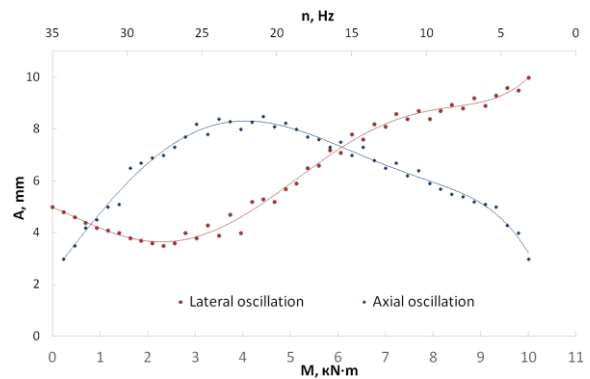


Figure 4. Average amplitude (with respect to the body length) in lateral and axial oscillations in dependence of SDM shaft moment

As a result of calculations based on developed mathematical model (7), range of self-oscillation onset during rotation and translational movement of SDM – DS system was revealed.

Input parameters for calculating rotational and translational movements:

$L_1 = 1800$ m; $L_2 = 190$ m; $L_3 = 10$ m; $J_1 = 5.841 \cdot 10^{-6}$ m⁴; $J_2 = 1.941 \cdot 10^{-6}$ m⁴; $J_3 = 4.928 \cdot 10^{-6}$ m⁴; $k = 0.106$; $G_1 = G_2 = G_3 = 8 \cdot 10^{10}$ Pa; $\lambda_{L1}, \lambda_{L2}, \lambda_{L3} = 3200$ m/s; $n_0 = [0; 7]$ rad/s; $\mu_1 = 0.1, \mu_2 = 0.2, \mu_3 = 0.3$.

$L_1 = 1800$ m; $L_2 = 190$ m; $L_3 = 10$ m; $F_1 = 1.018 \cdot 10^{-3}$ m²; $F_2 = 1.81 \cdot 10^{-3}$ m²; $F_3 = 8.042 \cdot 10^{-4}$ m²; $k = 0.106$; $E_1 = E_2 = E_3 = 2 \cdot 10^{10}$ Pa; $\chi_{L1}, \chi_{L2}, \chi_{L3} = 5320$ m/s; $n_0 = [0; 7]$ rad/s; $\mu_1 = 0.1, \mu_2 = 0.2, \mu_3 = 0.3$.

Results of mathematical modeling are illustrated in Figure 5. Comparison of obtained study results for SDM frame oscillations in bench conditions with calculated values of boundaries of DS self-oscillations allows determining the range of stable operation for SDM – DS system. Values located under the line indicated by lower boundary of P_b self-oscillations mean absence of vibration – uniform translational and rotational movement of tool, between upper P_B and lower P_B boundaries – a temporary stop (jamming), above the upper P_H – braking (no rotation).

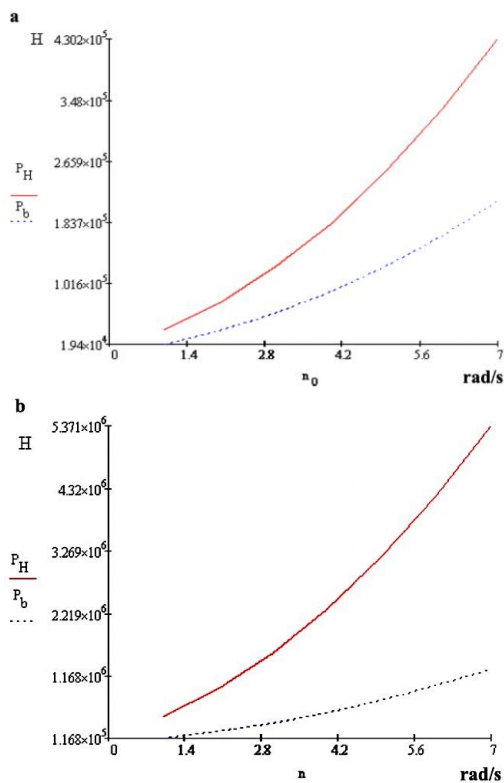


Figure 5. The boundaries of the rotational (a) and translational self-oscillations (b) of the «SDM-DS» system

The results of mathematical modeling are presented in Figure 5 (a and b).

Figure 5(a) shows the boundaries of the self-oscillations of the «SDM-DS» system due to the dynamic axial load resulting from the rotational and translational movement of the system.

Comparison of the obtained research results of the SDM oscillations (Figure 1) with the calculated boundaries values of the DS self-oscillations allow to determine the range of «SDM-DS» system stable work. The values located under the line indicated by the lower boundary of self-oscillations (P_b) means the absence of vibration – uniform translational and rotational movement of the tool, between the upper (P_H) and lower (P_b) boundaries – shut-down (jamming), above the upper (P_H) – braking (deficiency of rotation).

To perform stable operation, the «SDM-DS» system determines the rotor speed and axial load on the end face of the lower part of the composite rod of the represented of SDM, equal to the upper and lower self-oscillation boundaries that are recorded in the system (9).

By knowing the value P_b and P_H and considering the optimum frequency of drill string rotation n_0 , the values of SDM working energy characteristics in the load condition (as shown in Figure 4) can be determined.

4. CONCLUSION

- (1) The data of experimental researches aimed at determining the vibration values of the SDM body showed that a decrease in the rotational speed of not more than 30% of the maximum rotational speed of the shaft in idle mode allows for the stable operation of the «SDM-DS» system. At the same time, the moment in the indicated range varies from 1 to 4.5 kN m, which is sufficient for the implementation of volumetric destruction of rock.
- (2) In the case of operation of the SDM in extreme mode (maximum power mode), a sharp increase in the amplitude of oscillations occurs in the lower part of the SDM, which leads to the appearance of half-waves in the BHA and loss of tool stability.
- (3) The mathematical model has been improved, which allows to determine the diameters of the drilling operational parameters that ensure the BHA stability by controlling the dynamics of the «SDM-DS» system during its joint operation.

The developed methodology and technical recommendations aimed at ensuring stable operation of SDM with simultaneous rotation of drilling string at drilling directional wells are used in the branch of «LUKOIL-Engineering» LLC – «KogalymNIPIneft».

Further research will look at technical solutions that reduce vibrations during drilling. Namely, new models

of screw downhole engines and justification of their technical and technological efficiency.

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Persian Abstract

چکیده

مقاله ارائه شده در مورد احتمال افزایش بهره وری از بخش های مستقیم چاه های حفاری با استفاده از موتورهای سریپیج پیچ (*SDM*) با روش ترکیبی از حفاری با چرخش رشته حفاری (*DS*). هدف حصول اطمینان از عملکرد پایدار *SDM* با چرخش همزمان *DS* با کاهش دامنه نوسانات با تنظیم پارامترهای حالت حفاری بر اساس مدل سازی ریاضی برای سیستم *SDM-DS* است. نتایج مطالعه تجربی در مورد تعیین توزیع اکسترا از نوسانات جانبی و محوری قاب *SDM* بسته به پارامترهای هندسی مکانیسم حرکت و حالت های حصول اطمینان از عملکرد پایدار ارائه شده است. رویکردهای توسعه یک مدل و روش ریاضی به صورت مفهومی بیان شده است که اجازه می دهد طیف وسیعی از نوسانات خود را برای سیستم *SDM-DS* و مرزهای اختلال موج چرخشی و ترجمه ای برای یک میله ناهمگن با یک *SDM* نصب شده در حفاری بخش های مستقیم چاه، مشخص کند. این مدل ریاضی پویایی سیستم *SDM-DS* امکان پیش بینی پارامترهای بهینه از حالت حفاری جهت را فراهم می کند که عملکرد پایدار مونتاژ گمانه را تضمین می کند.