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Satbayev University

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Satbayev University

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VIBRATIONS AND STABILITY OF A GEOMETRICALLY NONLINEAR WEIGHTY DRILL STRING FOR DRILLING OIL AND GAS WELLS

Abstract. Heavy weight drill pipe (HWDP) in wells are hollow, weighty rods with stepwise changing physical properties (for example, stiffness), and each link of the string can deform according to geometrically nonlinear laws. They are the most critical part in the drilling process, transmitting power from the drilling rig to the rock failing tool, and are in hydrodynamic and contact interaction with the borehole walls, and are always curved. This occurs due to the curvature of the well itself, and under the action of its own weight, contact forces, as well as centrifugal forces in the case of rotation of the pipe. In this case, the curvature of the HWDP axis can be significantly influenced by the geometric nonlinearity of the deformation of its pipes.

A review of this issue revealed a number of poorly studied problems, which include accounting for both physically and geometrically nonlinear problems, accompanied by various types of complications (loss of stability HDWP, pipe breaks, etc.), as well as other processes in the elements of a dynamic drilling system (DDS).

In this paper, based on the use of modern methods for studying dynamic processes in mechanical systems, a method is proposed for studying longitudinal oscillations of a geometrically nonlinear HWDP of its stability under torsion, taking into account the physical nonlinearity in the process of its deformation. The dependences characterizing this process are found.

Key words: oil and gas wells, heavy weight drill pipes, longitudinal oscillations, stability, geometric nonlinearity, dynamic systems.

Introduction. The dynamics of the HWDP is reflected in analytical works [1-7]. Modern methods of dynamics analysis, in particular, the use of the finite element method (FEM) for the analysis of the dynamics of mechanical systems were considered in [8, 9]. The fundamental problems of the HWDP dynamics in a nonlinear formulation of the problem are originated in works [10-13]. At the same time, the problems of stability loss of rod systems were solved in work [13]. However, as stated above, these fundamental foundations of the dynamics and stability of rod systems were created with some assumptions.

It should be noted that modern works of Kazakhstani scientists are also devoted to analytical studies of drilling systems [14, 15]. They present the results aimed at studying the kinematics of the drilling process to determine the drilling scheme. The analysis of the principles of operation of drilling tools is performed.

Due to the large length of the HWDP in comparison with the transverse dimensions, it is often modeled with a long uniform thin rod, which is a rather rough approximation, since the components of the HWDP are connected by locks, equipped with centralizers and other devices that significantly change the dynamics of the drill pipe. Therefore, theoretically, the column should be considered as a nonlinear mechanical system with an infinite number of degrees of freedom. But here a difficulty arises, associated with the impossibility of analytically studying the dynamics of the operation of such system, and, consequently, identifying its strength, stability, negative or, conversely, positive influence of oscillations and vibrations under dynamic loads during drilling.

The foregoing gives reason to believe that when developing methods for analyzing the oscillation and stability of HWDP, especially in the nonlinear formulation of problems, the results of applied research on the dynamics of drilling systems are also important. The dynamics of HWDP has been studied in the works of Russian researchers [16-25]. These works are undoubtedly sources for a more realistic modeling of the HWDP dynamics. However, these works mainly cover the development of a theory for controlling the dynamics of HWDP associated with increasing the efficiency of the process of drilling deep and directional wells by minimizing torsional and longitudinal low-frequency oscillations of the drilling tool [19, 21-23]. Many of them are devoted to rotational-longitudinal oscillations of the HWDP in order to create its bottom stabilizing assemblies (BSA), which is very important when drilling incliningly directed and horizontal wells with downhole motors, etc.

Aims and objectives. From the above, it follows that the problem of studying the longitudinal oscillations of a geometrically nonlinear HWDP of its torsional stability, considering the physical nonlinearity in the process of its deformation, is an urgent problem and is of great scientific interest.

Results of theoretical research. In order to study the longitudinal oscillations of the drill pipe, considering the physical nonlinearity in the process of its deformation, a model was created where the HWDP is presented in the form of a multi-link long bar performing longitudinal oscillations. After accepting the origin of coordinates in the upper cross section of the pipe and the direction of the *x*-axis vertically downward, the potential and kinetic energies of a physically nonlinear rod were presented in the form [18]:

$$U = \frac{EF}{2}\int_0^l \left((\frac{\partial u}{\partial x})^2 (1 - a_3 (\frac{\partial u}{\partial x})^2 \right) dx, \; T = \rho F \int_0^l \left(\frac{\partial u}{\partial t} \right)^2 dx + \sum_{i=1}^N m_i \left(\frac{\partial u(t, l_i)}{\partial t} \right)^2 \quad (1)$$

where $u(x,t)$ is the longitudinal displacement of the rod; E, ρ is the Young's modulus and the density of the rod material; F and l are the area and length of the rod, respectively; $a_3 = -\frac{2}{9}\frac{3K}{3K+G}\frac{\gamma_2}{G}$, K, G are the moduli of volumetric compression and shear; γ_2 is the coefficient characterizing the geometric nonlinearity of deformation, determined experimentally; m_i is the mass of the lock joint (sleeve) located in the cross section $x = l_i$.

The following boundary value problem is considered:

$$\frac{\partial u}{\partial x} = -\frac{P_0}{EF} \text{ at } x = 0, \; u = u_0(t) \text{ at } x = l$$

where E is the Young's modulus, P_0 is the constant compressive axial force acting on the drill pipe, l is the total length of the drill pipe.

To solve the boundary value problem, the finite element method is applied. The HWDP length was divided into n finite elements (n+1 nodes) of the same length a. It is considered that the lengths of the sections of the elements are related to the value a. The concentrated masses are located at the nodal points, and in the first node there is no mass, and the lowest mass (bit) moves according to a given law $u_o(t)$. Denoting through $q_i(t)$ $(i = 2..n+1)$ $(q_{n+1} = u_0(t))$ displacements of concentrated masses, displacements of drill pipe cross sections (referred to the value a) within each element, is represented in the form of a system of equations:

$$\begin{aligned}
u_{1,2} &= (1 - 4\xi^2)(p - q_2)/3 + N_2(\xi)p, \\
u_{2,3} &= N_1(\xi)q_2 + N_2(\xi)q_3, \\
u_{3,4} &= N_1(\xi)q_3 + N_2(\xi)q_4 \\
u_{i,i+1} &= N_1(\xi)q_i + N_2(\xi)q_{i+1} \\
u_{n,n+1} &= N_1(\xi)q_n + N_2 u_0 \; (2)
\end{aligned}$$

where $\xi = x/a$, $N_1 = 1 - 3\xi + 2\xi^2$, $N_2 = 2\xi^2 - \xi$, $p = P_0 a / EF$, $i = 2..n-1$

substituting $u_{i,j}(\xi, t)$ from (2) into (1) and obtaining expressions for the kinetic and potential energies:

$$U = \frac{EFa}{2}\sum_{i=2}^{n+1}\overline{U}_i\ ,\ T = \frac{F\rho a^3}{2}\sum_{i=2}^{n+1}\overline{T}_i \qquad 3)$$

taking the variables q_i $(i = 2..n)$ as generalized coordinates, the second kind Lagrange equation was formulated [10]:

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{q}_i}\right) - \frac{\partial T}{\partial q_i} = -\frac{\partial U}{\partial q_i} \qquad (4)$$

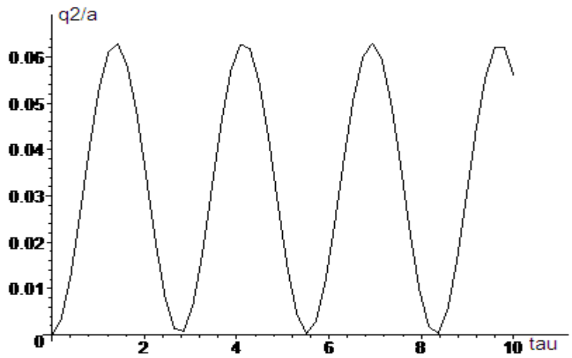
After formulating the expressions for the kinetic and potential energies from (3) to (4) and solving them, a system of nonlinear differential equations is obtained to determine the coordinates of the displacements of the added masses. In particular, in the case when n = 3, it is obtained:

$$27\rho a^2(27\ddot{q}_2 - \ddot{q}_3) = E\{2a_3[9037q_1^3 + q_2^2(837q_3 + 128p) + q_2(621p^2 + 23q_3^2) + (891q_3^3 - 28p^3)] - -378q_2 - 27q_3 + 12p\}, \quad (5)$$

$$\rho a^2(8\ddot{q}_3 - \ddot{q}_2 - \ddot{q}_4) = 2E\{a_3[44q_2^3 + 92q_3q_2^2 + 132q_3^2(q_2 + q_4) + 488q_3^3 + 44q_4^3] - q_2 - q_4 - 14q_4\} \quad (6)$$

$$5\rho a^2(8\ddot{q}_4 - \ddot{q}_3) = E\{15a_3(488q_4^3 + 132q_4^2(q_3 + u0) + 92(q_3^2 + u_0^2) + 44(q_3^3 + u_0^3)] - 50(q_3 + u_0) - 700q_4 + 5\ddot{u}_0\} \quad (7)$$

Figures 1 and 2 show the curves of the dependences of the displacements of concentrated masses q_2/a and q_4/a on the dimensionless time $\tau = ct / a$ ($c = \sqrt{E/\rho}$ is the speed of wave propagation in the links of the pipe) for geometric linear (a_3 = 0) and nonlinear (a_3 = -0.4) deformation.



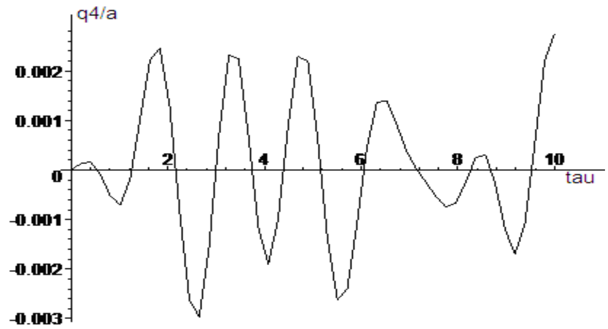
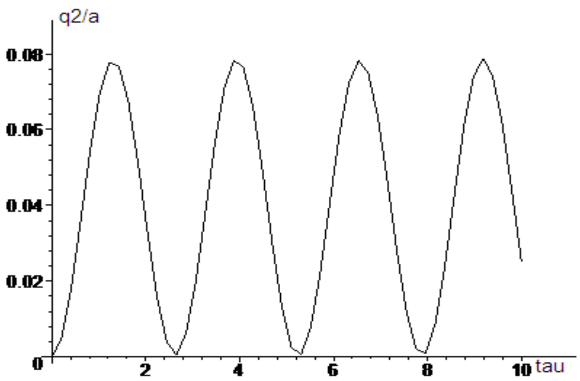


Figure 1 – Curves of dependences of displacements of concentrated masses q_2 / a and q_4 / a on dimensionless time $tau = \tau = ct / a$ at $a_3 = 0$



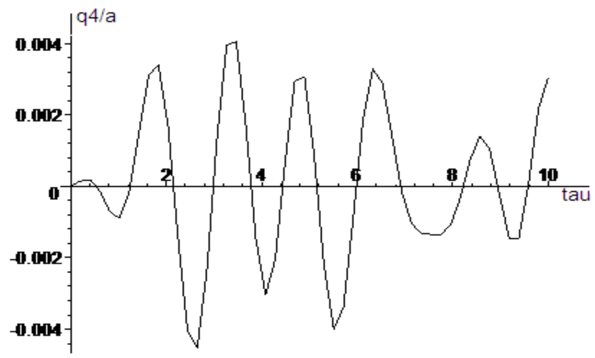


Figure 2 – Curves of dependences of displacements of concentrated masses q_2 / a and q_4 / a on dimensionless time $tau = \tau = ct / a$ at $a_3 = -0.4$

The lower end of the HWDP (bit) moves according to the law $u_0 = v_0 t - j_0 t^2 / 2$ (v_0 и j_0 and the initial velocity and acceleration of the bit deceleration) In calculations it is accepted: $v_0 = 2m/c$, $j_0 = 1m/c^2, c = 4000m/c$, $p = 1$.

From the analysis of the obtained curves, it follows that taking into account the geometric non-linearity leads to an increase in the displacements of the concentrated masses, this regularity is more noticeable for the displacement of the end of the string close to the bit.

Now let us study the torsional stability of a geometrically nonlinear HWDP. In the general case, the problem of the stability of rods under the action of torques is non-conservative, and they must be solved considering the complex deformed state of the rod caused by the action of axial, bending and torsional moments:

$$\left(\frac{M_k}{2EJ}\right)^2 + \frac{P}{EJ} = \frac{\pi^2}{l^2} \tag{8}$$

where M is the torque, P is the compressive force. As can be seen from expression (8), the compressive force leads to a decrease in the value of the critical moment. If the compressive force is known, then from (8) it is possible to determine the critical value of the torque

$$M_k = 2EJ\sqrt{\frac{\pi^2}{l^2} - \frac{P}{EJ}}\,.$$

If we denote by $\varphi(x,t)$ the angle of rotation of the section of the HWDP bar, then the expression of the torque taking into account the nonlinear shear deformation $\frac{\partial \varphi}{\partial x}$ can be expressed by the formula:

$$M = GJ_{p0}\left[1 + \gamma_2 \frac{J_{p2}}{J_{p0}}\left(\frac{\partial \varphi}{\partial x}\right)^2\right]\frac{\partial \varphi}{\partial x}\,, \tag{9}$$

where G is the shear modulus, $J_{po} = \frac{D_o^4 - D_i^4}{32}$ the polar moment of inertia of the bar sections, $J_{p2} = \frac{2}{9}\frac{D_o^6 - D_i^6}{64}$ and γ_2 is the coefficient characterizing the shear nonlinearity.

The total work of deformation is calculated by the formula:

$$U = \frac{\pi}{4} G \frac{D_н^4 - D_в^4}{16} \int_0^l \left(\frac{\partial \varphi}{\partial x}\right)^2 \left[1 + \frac{2\gamma_2}{3}\frac{D_н^2 - D_в^2}{4}\left(\frac{\partial \varphi}{\partial x}\right)^2\right] dx\,. \tag{10}$$

Kinetic Energy of the HWDP rod is equal to:

$$T = \frac{\pi m}{4}\frac{D_н^4 - D_в^4}{16}\int_0^l \left(\frac{\partial \varphi}{\partial t}\right)^2 dx\,, \tag{11}$$

where m is the linear mass of the rod material. A variable moment $\pm M_0(t)$ is applied in the sections $x = 0$ and $x = l$.

Angle of rotation of the rod is presented in the form of

$$\varphi = \frac{M_0 l}{GJ_{p0}\pi}\sin\frac{\pi x}{l} + q(t)\cos\frac{\pi x}{l}\,, \tag{12}$$

For potential and kinetic energy, we obtain the expressions

$$U = \frac{\pi^3}{128} G \frac{D_н^4 - D_в^4}{l} [\overline{M}_0^2 + q^2 + c(\overline{M}_0^4 + 2\overline{M}_0^2 q^2 + q^4)],$$

$$T = \frac{\pi m (D_н^2 - D_в^2) l}{128} \dot{q}^2$$

where $\overline{M}_0 = \dfrac{M_0 l}{G J_{p0} \pi}$, $c = \pi^2 \gamma_2 (D_н^2 - D_в^2) / 8 l^2$.

Taking $q(t)$ as the generalized coordinate, we compose the Lagrange equation of the second kind

$$\ddot{q} + b_1 q + b_2 q^3 = -\ddot{\overline{M}}_0, \tag{13}$$

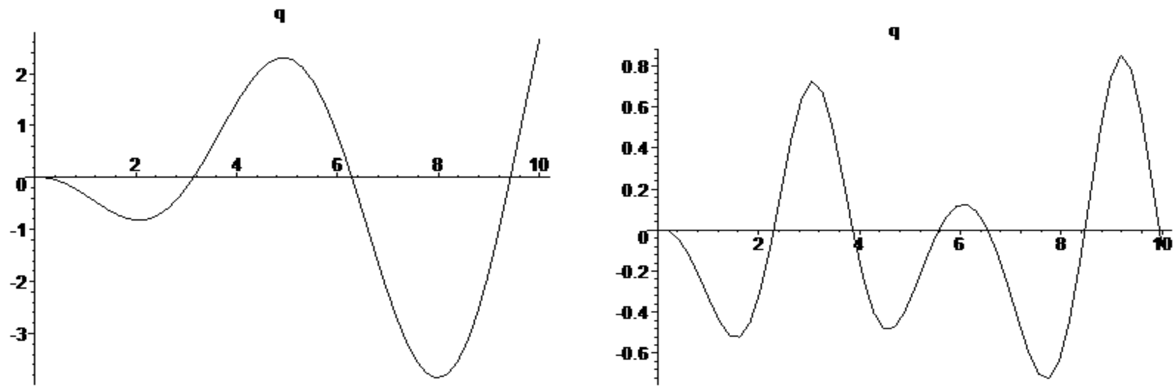
where $b_1 = (1 + 2c\overline{M}_0^2)\omega^2$, $b_2 = 2c\omega^2$, $\omega = \dfrac{G(D_н^2 + D_в^2)\pi^2}{ml^2}$.

The quantity *c* in equation (13) characterizes the nonlinearity of the rod during torsion. The equation is integrated under the following initial conditions $q(0) = 0$, $q(0) = 0$.

Figure 3 shows the graphs of the dependence of the angle of rotation of the bar sections $x = 0$ under the action of the moment $\overline{M}_0$ according to the law $\overline{M}_0 = 1 - \cos\omega_0 t$ for different values of the parameters c and ω_0.

It can be seen that the nonlinearity, depending on the frequency ω_0 of the torque, has a different effect on the torsional oscillations of the rod. At small values of ω_0, nonlinearity in the considered case decreases the oscillation amplitudes.

$\omega_0 = 1 \quad c = 0 \quad c = 1$



$\omega_0 = 10 \quad c = 0 \quad c = 1$

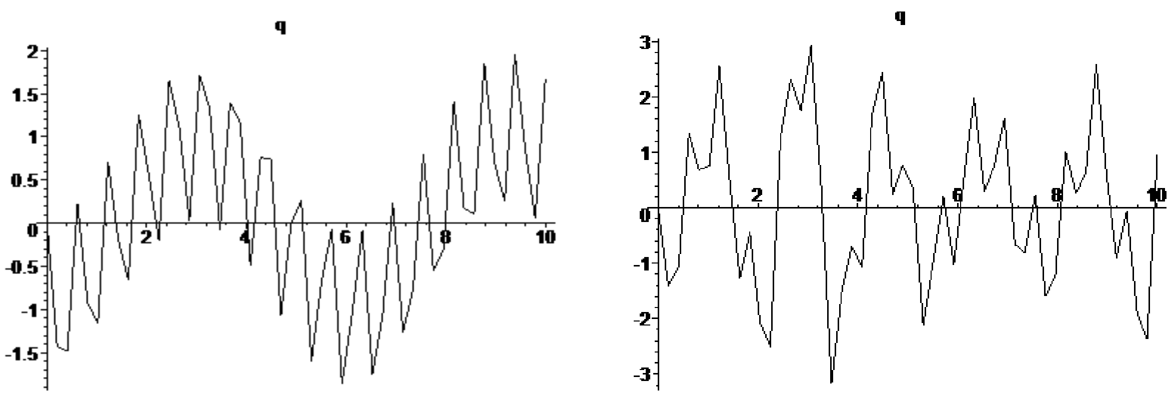


Figure 3 – Torsional vibrations of the rod at different values of the parameters *c* and ω_0

Now let the torsional movement of the HWDP rod occur under the action of a compressive force *P*, we assume that the rod performs transverse oscillations in the plane $y0z$ in two directions. We assume that the transverse motion of the cross-sections of the HWDP rod does not affect its torsional oscillations (figure 4).

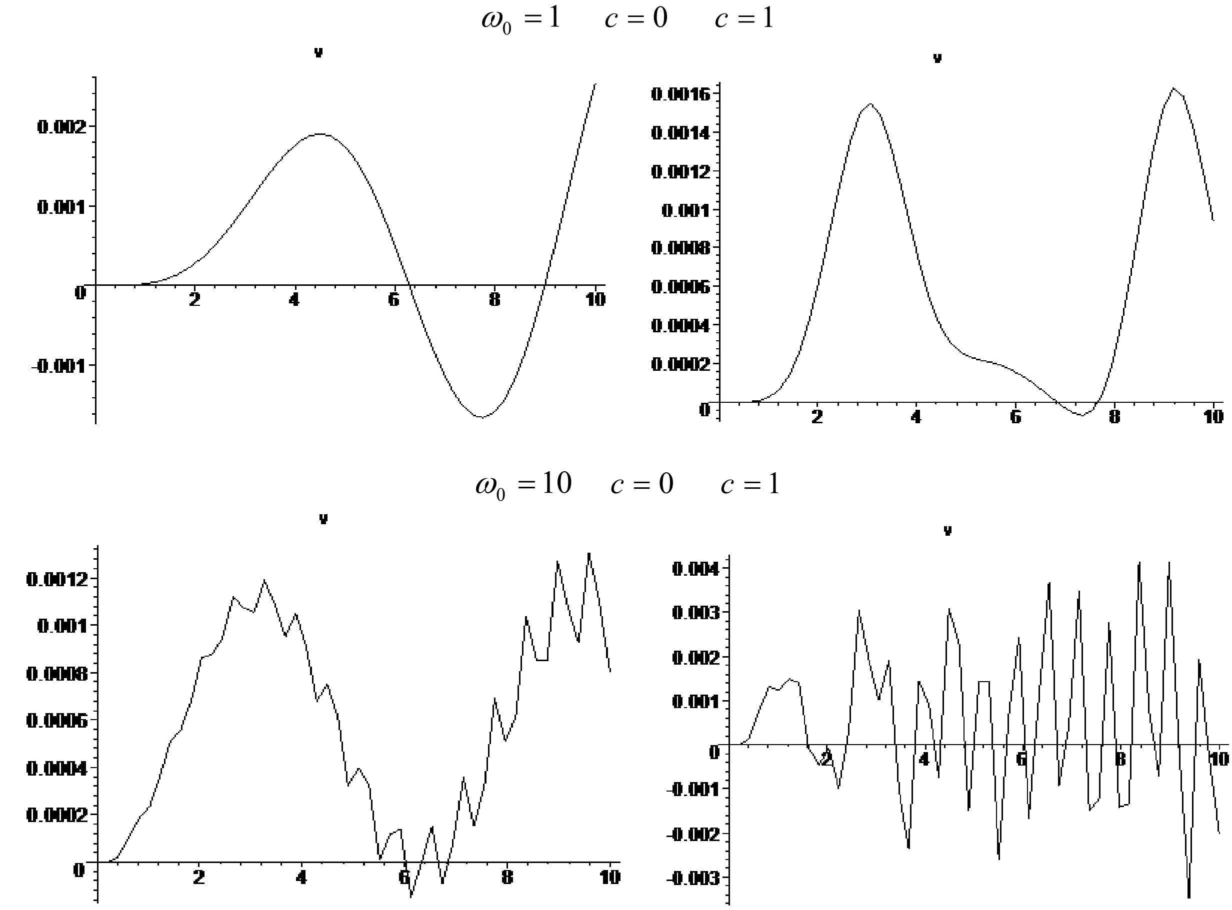


Figure 4 – Transverse oscillations of the rod at different values of the parameters *c* and ω_0

The equations of motion of the rod in the case of geometric linearity are written in the form [1]

$$m\frac{\partial^2 v}{\partial t^2} + EJ\frac{\partial^4 v}{\partial x^4} + Pz_0\frac{\partial^2 \varphi}{\partial x^2} = 0 \tag{14}$$

$$m\frac{\partial^2 w}{\partial t^2} + EJ\frac{\partial^4 w}{\partial x^4} - Pz_0\frac{\partial^2 \varphi}{\partial x^2} = 0 \tag{15}$$

where $v(x,t)$ and $w(x,t)$ are displacements of bar sections along the axes $0y$ and $0z$.

Equations (14) and (15) introducing dimensionless variables and quantities by the formulas $\tau = \frac{t}{l}\sqrt{\frac{EJ}{ml}}$, $\xi = x/l$, $\bar{z}_0 = z_0/l$, $\alpha = \frac{\pi^3 GJ_{p0}}{2EJ}$, $\overline{P} = \frac{Plz_0}{\pi GJ_{p0}}$, considering (12) and bringing to the form

$$\frac{\partial^2 \bar{v}}{\partial \tau^2} + \frac{\partial^4 \bar{v}}{\partial \xi^4} - \alpha\overline{P}[\overline{M}_0(\tau)\sin\pi\xi + q(\tau)\cos\pi\xi] = 0 \tag{16}$$

$$\frac{\partial^2 \overline{w}}{\partial \tau^2} + \frac{\partial^4 \overline{w}}{\partial \xi^4} + \alpha \overline{P}[\overline{M}_0(\tau)\sin \pi\xi + q(\tau)\cos \pi\xi] = 0 \tag{17}$$

Equations (16) and (17) are integrated under the conditions $\overline{v} = 0,\ \overline{w} = 0,\ \frac{\partial \overline{v}}{\partial \tau} = 0,\ \frac{\partial \overline{w}}{\partial \tau} = 0$ at $\tau = 0,\ \overline{v} = 0,\ \overline{w} = 0,\ \frac{\partial^2 \overline{v}}{\partial \xi^2} = 0, \frac{\partial^2 \overline{w}}{\partial \xi^2} = 0$ at $\xi = 0$ и $\xi = 1$, the solution of which is obtained by the Fourier method $\overline{v} = \sum_{n=1}^{\infty} T_n \sin n\pi\xi$, $\overline{w} = -\sum_{n=1}^{\infty} T_n \sin n\pi\xi$ where

$$T_n = \frac{\alpha}{2n\pi}\int_0^{\tau} \overline{P}\sin[n^2\pi^2(\tau - \varsigma)][\overline{M}_0(\varsigma) + q(\varsigma)]d\varsigma .$$

As can be seen from the dependences of the displacement of the midpoint of the elastic line on the dimensionless time τ presented in Figure 4 under the action of the moment $\overline{M}_0$ according to the law $\overline{M}_0 = 1 - \cos \omega_0 t$ for various values of the parameters c and ω_0, consideration of the nonlinearity of the rod during torsion can lead to a decrease in the deflection of the rod at small values of the parameter ω_0. With an increase in this parameter, the rod performs high-frequency oscillations with large amplitudes. The calculations were made taking $\alpha = 0.01$, $\overline{P} = 0.005$.

Conclusion. The proposed method for studying longitudinal oscillations of a geometrically nonlinear HWDP of its torsional stability, considering the physical nonlinearity in the process of its deformation, makes it possible to improve the methods of ensuring the reliability and strength of HWDP at the stage of their design. The established dependences characterizing the stability parameters of HWDP demonstrate the ability to control them by varying their values. This makes it possible to evaluate the most optimal values of the DDS operating parameters for their stable operation, which subsequently ensures their operability during real case.

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МҰНАЙ-ГАЗ ҰҢҒЫМАЛАРЫН ТЕРЕҢДЕТКІШ САЛМАҚТЫ БҰРҒЫЛАУ ТІЗБЕГІНІҢ ГЕОМЕТРИЯЛЫҚ СЫЗЫҚТЫ ЕМЕС ТЕРБЕЛІСТЕРІ МЕН ТҰРАҚТЫЛЫҒЫ

Аннотация. Ұңғымалардағы бұрғылау тізбектері (БТ) – бұл физикалық қасиеттері біртіндеп өзгеретін қуыс салмақты өзектер (мысалы, қаттылығы), бұл жағдайда тізбектің әрбір бөлігі геометриялық сызықты емес заңдарға сәйкес деформациялануы мүмкін. Олар бұрғылау қондырғысынан тау жыныстарын талқандайтын құралға қуат беретін және бұрғылау қабырғаларымен гидродинамикалық, демек жанасу кезінде өзара әрекеттесетін бұрғылау үрдісінің ең маңызды бөлігі болып табылады және әрқашан қисық болады. Бұл ұңғыманың қисаюына байланысты да, өз салмағының, байланыс күштерінің, сондай-ақ тізбек айналған жағдайда ортадан тепкіш күштердің әсерінен болады. Бұл жағдайда тізбек осінің қисаюына тізбек құбырларының деформациясының геометриялық бейсызықтығы айтарлықтай әсер етуі мүмкін.

Бұл мәселені қарау кезінде физикалық және геометриялық сызықтық емес есептердің әр түрлі асқынулар түрлерімен (тізбектердің тұрақтылығының жоғалуы, құбырлардың үзілуі және т.б.), сонымен қатар динамикалық бұрғылау жүйесінің (ДБЖ) элементтеріндегі басқа процестерді есепке алуды қамтитын бірқатар аз зерттелген мәселелер анықталды.

Бұл жұмыста механикалық жүйелердегі динамикалық процестерді зерттеудің заманауи әдістерін қолдану негізінде деформация процесінде физикалық сызықты емес екенін ескере отырып, бұралу кезіндегі

оның тұрақтылығының геометриялық сызықты емес бұрғылаутізбегінің бойлық тербелістерін зерттеу әдісі ұсынылған. Бұл процесті сипаттайтын тәуелділіктер табылды.

Түйін сөздер: мұнай-газ ұңғымалары, бұрғылау тізбектері, бойлық тербелістер, тұрақтылық, геометриялық сызықтық емес, динамикалық жүйелер.

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КОЛЕБАНИЯ И УСТОЙЧИВОСТЬ ГЕОМЕТРИЧЕСКИ НЕЛИНЕЙНОЙ ВЕСОМОЙ БУРИЛЬНОЙ КОЛОННЫ ДЛЯ УГЛУБЛЕНИЯ НЕФТЕГАЗОВЫХ СКВАЖИН

Аннотация. Бурильные колонны (БК) в скважинах представляют собой полые весомые стержни со ступенчато изменяющимися физическими свойствами (например, жесткостью), при этом каждое звено колонны может деформироваться по геометрически нелинейным законам. Они являются наиболее ответственной частью в процессе бурения, передающим звеном мощности от буровой установки до породоразрушающего инструмента и находятся во взаимодействии со стенками скважины как гидродинамическом, так и контактном и всегда искривлены. Это происходит как за счет искривления самой скважины, так и под действием собственного веса, контактных сил, а также центробежных сил в случае вращения колонны. При этом на искривления оси колонны могут существенно влиять геометрическая нелинейность деформирования труб колонны.

Обзор данной проблемы выявил ряд малоизученных задач, к которым относятся вопросы учета как физически, так и геометрически нелинейных задач, сопровождаемых различными видами осложнений (потери устойчивости колонн, разрывы труб и др.), а также другие процессы в элементах бурильной динамической системы (БДС).

В данной работе на основе применения современных методов изучения динамических процессов в механических системах предложена методика исследования продольных колебаний геометрически нелинейной БК ее устойчивости при кручении с учетом физической нелинейности в процессе ее деформирования. Найдены зависимости, характеризующие данный процесс.

Ключевые слова: нефтегазовые скважины, бурильные колонны, продольные колебания, устойчивость, геометрическая нелинейность, динамические системы.

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