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The process of crack propagation during rotary percussion drilling of hard rocks

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ABSTRACT

Rotary percussion drilling determines the possibility of reducing waste and intrusion of foreign objects into the solution. It reduces drilling costs and increases the processing speed of the resulting product, and also contributes to the high-quality result at the beginning of the drilling string project design. Reducing the number of cracks decreases the amount of rock used and improves the quality of the resulting product. The novelty of the study lies in development and practical application of fibreglass drilling strings that provide reduced energy consumption for drilling and increased elasticity. The authors show that the use of such strings makes possible to shorten the time of field development and increase the yield in difficult geological conditions. Establishment of strings made of polymeric materials and composite structures can significantly reduce the number of cracks in hard rocks. This determines the conditions for the environmental friendliness of the entire production process, the gradual reduction of waste rock usage and increase in the secondary use of oil production materials.

Keywords: Rock, Hardness, Drilling, Crack

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1. Introduction

Many oil wells with fibreglass rod strings have plunger strokes longer than polished rod strokes at the top of the string [1]. In many cases, increased production of the oil is carried out due to the movement of plunger strokes, which are 20-50% longer than the corresponding stroke length of the polished rods. This phenomenon is usually caused by pump stroke, which is caused by acceleration forces of the downstroke that pull (lengthen) the relatively elastic fibreglass rod [2]. However, the stroke of the plunger is caused by dynamic acceleration, which occurs when the fibreglass rod string operates close to its first harmonic longitudinal frequency of free vibrations [3].

The choice of optimal methods for deep drilling of rocks, combined with the gradual improvement of the existing deep drilling mechanisms, as well as the creation of new, more modern and rational in use samples, can ensure a high quality of drilling work and the achievement of optimal results when drilling rocks. In addition, an in-depth study of the practice and theory of drilling is essential, with the obligatory consideration of the ability of a particular rock to break down in conditions of deep drilling.

Relatively few papers have been published on the mechanical characteristics of glass reinforced plastic (GRP) rod strings [4]. In some cases, they were calculated using equations of the natural vibration frequency of steel rod strings [5]. Since this equation was derived for a long, thin rod with uniformly distributed mass and stiffness characteristics, it cannot be applied to a GRP rods string that combines both GRP and steel rods [6]. The top (50-90%) of these strings are made up of GRP rods, while the bottom sections are made up of steel sucker rods [7]. When the project designer believes that additional mass needs to be installed, the weighted rod sections are positioned at the bottom of the rod string [8].

Most of the fibreglass rod strings are designed using a computer programme based on the wave equation described by Gibbs [9]. Until the programme operator understands the effect of different factors on the behaviour of the system, using these programs for designing GRP rods so that achievement of optimal oil production can lead to a series of repeated attempts and mistakes.

The performance of a fibreglass/steel strings can be modelled by a simple mechanical oscillating spring/mass system [10]. The base frequency of a spring/mass system is the frequency of free vibration of the mass without effect of external forces [11]. Vibration frequency that occurs when a mass suspended by a spring is moved and then released, allowing the mass to vibrate freely [12]. Resonance occurs when the forced vibrations (oscillating either the mass or the fixed end of the spring) are equal to the natural vibrations of the system [13]. At the resonant frequency of a simple spring/mass system, the travel mass of the spring drive end increases and spreads, causing large mass displacement [14].

The effective methods of drilling rocks existing today are largely determined by the type of tool used, drilling angles, as well as the types of vibration systems used. The excavation capacity of the rocks extracted during drilling is largely determined by the parameters of its resistance to compression and the ability to form deep cracks. Mechanical excavation of rocks involves the consistent use of the principles of mechanical destruction of rocks during its extraction to the surface. This is ensured by the use of special tools and mechanisms for the construction of tunnels in hard rock. In addition, for equipping the working surfaces of machines intended for mining, disc tools are used, which are used in works for laying tunnels in hard rocks.

In a simulated simple mechanical spring/mass oscillating system, the spring simulates longitudinal resilient GRP rods, the mass simulates heavy steel rods suspended below the GRP rods, and an exciting force is applied to the polished rod of pumping unit [15]. When the power-driven system frequency (operating frequency of the pumping unit) approaches the first harmonic of the base frequency of the rod string, the surface stroke causes a plunger overstroke, which is longer than the stroke of the polished rod [16]. A sucker rod pump differs from a simple spring/mass system in terms of pump plunger travel because the system contains a mass of fluid that does not rise on the downstroke of the plunger [17]. Therefore, the elastic rods stretch (lengthen) during the upstroke to compensate for this additional load [18-20].

2. Material and methods

The vibration of rod strings is investigated using the vibrational equation (Eq. 1):

$$\frac{Eg}{\rho}\frac{\partial^2 u}{\partial L_r^2} = \frac{\partial^2 u}{dt^2} \tag{1}$$

Equation (1), obtained from Hooke's law, describes the springiness of the rod (Eq. 2):

$$\frac{\partial u}{\partial L} = \frac{F}{AE} \tag{2}$$

For a rod with mass and elasticity uniformly distributed along its length, the first longitudinal harmonic of natural vibrations is (Eq. 3):

$$\omega_n = \frac{\pi}{2L_r}\sqrt{\frac{Eg}{\rho}} \tag{3}$$

Whereas the speed of stress (or sound wave) propagation (Eq. 4):

$$v = \sqrt{\frac{Eg}{2L_r}} \tag{4}$$

for the first frequency of natural oscillations is the same, the expression is obtained (Eq. 5):

$$\omega_n = \frac{v\pi}{2L_r} \tag{5}$$

or in strokes per minute for a steel rod with $v = 4978 m/s$ (Eq. 6):

$$N_0 = \frac{245000}{L_r}. \tag{6}$$

The presented equation is used to describe the base frequency of a straight string of rods. For a steel rod string, this base frequency ranges from 25 strokes/min for a 3050 m string to 80 strokes/min for a 915 m string. Although it describes design frequencies that correspond to 60% resonance, most steel strings operate at frequencies below 40% of resonance [21-34].

Taking GRP rod, the mass and resilience are not evenly distributed along the entire length. Moreover, a significant frequency of string stiffness is implemented in the fibreglass part, and 70-80% of the mass is concentrated in steel rods in the weighted bottom of the string. If the wave equation is solved with a concentrated mass at the end of the rod, then the first harmonic of the base frequency is (Eq. 7):

$$\omega_n = \sqrt{\frac{AF}{mL_r}} \tag{7}$$

Since the stiffness of the rod string (Eq. 8):

$$K = AE / L, \tag{8}$$

then (Eq. 9):

$$\omega_n = \sqrt{\frac{K}{m}}. \tag{9}$$

We obtain the classical equation that is used for a single degree of freedom system, consisting of a mass suspended on a spring without mass [35-49]. The actual total stiffness of the column-spring must be calculated with a correction (Eq. 10):

$$K_t = \frac{1}{\frac{1}{K_f} + \frac{1}{K_s}} \tag{10}$$

where *f* and *S* refer to different sections of the string (GRP and steel). This total spring rate should be used for the ratio $L_0 / L_s K$ and for calculating the elongation.

3. Results and discussion

The use of methods of numerical modeling of a real situation is able to qualitatively illustrate the essence of the ongoing processes, which is extremely important from the point of view of forming a correct assessment. Given that the GRP section of the rod string contains a significant portion of the total string mass, the equation must include the additional mass m_f. For a simple spring/mass system with a spring, which has a final mass significantly less than the suspended mass, it is shown that the base frequency of oscillations can be calculated using the formula (Eq. 11):

$$\omega_n = \sqrt{\frac{K}{m_s + M\left(m_f\right)}} \tag{11}$$

where the constant *M* varies between 1/3 and 1/2 depending on the ratio of the spring mass to the total system mass. The simulations show that not only the weight of the fibreglass rod section affects the weight of the system, but also the stiffness of the fibreglass rods K_t and steel rods K_s must be taken into account [35-49].

Equation (11) was obtained using a specially developed software to calculate the wide range oscillation frequency of fibreglass/steel sucker rods (Eq. 12):

$$\omega_n = \sqrt{\frac{K_t}{m_s + 0,3m_f}}, \quad K_t = \frac{1}{\frac{1}{K_f} + \frac{1}{5K_s}} \tag{12}$$

Converting this expression into the quantity of moves per minute, obtain (Eq. 13):

$$N_0 = \frac{60}{2\pi}\sqrt{\frac{K_t}{m_s + 0,3m_f}} \tag{13}$$

Although relation (13) is only a first approximation, the largest error between the base frequency of the fibreglass/steel string obtained with it and the frequency of the peak amplitude calculated using a specially developed software did not exceed 2% [50-72].

The following designations are used in these formulas: A – ross-sectional area of the rod body, m^2; E – Young's modulus, Pa; F – force (N); F_0 – weight of liquid above the pump plunger, N; G – free fall acceleration (gravitational constant), m/s^2; K – rigidity of the rod string, N/m; K_f – rigidity of the fibreglass part of the string, N/m; K_s – stiffness of the steel part of the string, N/m; K_t – stiffness of the combined rod string, N/m; L_{dhp} – length of the plunger stroke, m; L_r – length of the rod string, m; L_s – stroke length of the polished rod, m; m – weight, kg; m_f – weight of the fibreglass section of the rods; m_s – weight of the steel section of the rods, kg; M – multiple indicator; N – frequency of plunger strokes, strokes/min; N_o – base frequency of the rod string, min^{-1}; T – time, s; u – rod lengthening, m; v – speed of stress propagation in the string, m/s; ρ – density, kg/m^3; ω – operating frequency, rad/s(s^{-1}); ω_n – base frequency, rad/s(s^{-1}).

Table 1 shows the string lengths and natural frequencies of several oil wells with GRP rods. The natural frequencies for these strings were calculated using formulas (12). The measured base frequencies were obtained by analysing frequencies at very slow oscillations, which occur during normal operation [73-94].

Table 1. Lengths of rod strings and the base frequency for average wells with GRP rods

Well	String length, m	% of fibreglass rods	Pumping frequency (strokes per minute)	Natural oscillation frequency (h/min)		$\frac{N}{N_0}$	$\frac{F_0}{L_s K}$
				Estimated	Measured		
1	2786	73.5	10.3	17.1	17.4	0.60	0.64
2	2814	54.5	9.5	15.1	15.4	0.69	0.56
3	2818	76.7	10.3	15.6	15.9	0.66	0.52
4	2790	66.0	10.2	16.0	16.8	0.64	0.53
5	1655	55.3	14.1	23.8	-	0.59	0.67

The volumetrics of the creeping rod string do not take into account the load that rises with the upward plunger stroke. Here system parameters are presented as a function of dimensionless parameters used in the RP11L standard. The abscissa is the frequency ratio of the pumping unit N to the base frequency of the rod system N_0. Thus, when $N / n_0 = 0,4$, then the operating frequency of the installation is 40% of the base frequency of the string. The ordinate is the multiple factors of the system, which is the ratio of plunger stroke N / N_0 to plunger stroke 1.5 times per polished rod stroke.

An important characteristic of the dynamic behaviour of a rod string is its ability to absorb some of the oscillations in an irreversible form. The damping properties of the sucker rod as a mechanical oscillatory system are mainly due to three groups of dissipative forces: the forces of internal friction in the rod material; friction of the rod string from the inner walls of the string in a viscous medium (resistance of the external environment); friction in seals and threaded connections (structural damping) [95-105].

The ability to absorb mechanical vibrations is important from the point of view of preventing deformations of the string along its entire length. A significant amount of friction force arising when the rod contacts the inner walls of the column in a viscous medium is dangerous from the point of view of possible destruction of the rod, which will cause failure of the entire system.

In dynamic calculations of a sucker rod string, it is very important to assess the intensity of its oscillations in transient operating modes, because in most cases, resonant and near-resonant oscillations are excited in these modes for a relatively short period of time. The levels of such fluctuations, as a rule, can exceed the fluctuations of systems in operating conditions [106-111]. Quite often, high oscillation levels of the rod string in transient conditions cause damage to the string elements and their further destruction due to the so-called low-cycle fatigue. Calculations of a sucker rod string in transient modes are, in fact, a special case of calculations for random loads.

Calculations of the parameters of permissible column loads are essential from the point of view of assessing the effectiveness of the resistance of the column material to the applied loads. Exceeding the design parameters leads to deformations of the columns due to fatigue failure of the material. Cycling loads lead to premature aging of the material and a reduction in the life of the string. Timely and correct calculation of the limiting values of the loads exerted on the column and leading to its uneven vibrations is important in the context of increasing the overall service life of the entire structure as a whole.

Taking into account the basic principles of analytical mechanics for the constructed design scheme, the differential equations of motion will have the form (Eq. 14):

$$\begin{cases} F_2(t) = m_1 \dfrac{d^2 y_1}{dt} + \left(1 + 2\mu_2 \dfrac{d}{dt}\right) k_1 (y_1 - y_2) \\ F_2(t) = m_2 \dfrac{d^2 y_2}{dt} + \left(1 + 2\mu_2 \dfrac{d}{dt}\right) k_2 (y_2 - y_3) - \left(1 + 2\mu_1 \dfrac{d}{dt}\right) k_1 (y_1 - y_2) \\ F_3(t) = m_3 \dfrac{d^2 y_3}{dt} + \left(1 + 2\mu_3 \dfrac{d}{dt}\right) k_3 y_3 - \left(1 + 2\mu_2 \dfrac{d}{dt}\right) k_2 (y_2 - y_3) \end{cases} \tag{14}$$

Taking into account the nature of the oscillatory process of the sucker rod string, the functions of displacement and load, we represent in this form (Eq. 15):

$$y_i(t) = Y_{ai}(\omega) e^{\omega t}, F_i(t) = F_{ai}(\omega) e^{\omega t} \tag{15}$$

where $y_i(t), F_i(t)$ – amplitude values of the displacements and the force of the *i*-th stage of the string; ω – frequency of forced oscillations.
Substituting the force function and the second time derivative of the displacement function (14) into system (13), after further transformations, we obtain (Eq. 16):

$$\begin{cases} F_{a1} = -m_1 \omega^2 Y_{a1} + (1 + 2\mu_1 \omega i) k_1 (Y_{a1} - Y_{a2}) \\ F_{a2} = -m_2 \omega^2 Y_{a2} - (1 + 2\mu_1 \omega i) k_1 (Y_{a1} - Y_{a2}) + (1 + 2\mu_2 \omega i) k_2 (Y_{a2} - Y_{a3}) \\ F_{a3} = -m_3 \omega^2 Y_{a3} - (1 + 2\mu_2 \omega i) k_2 (Y_{a2} - Y_{a3}) + (1 + 2\mu_3 \omega i) k_3 Y_{a3} \end{cases} \tag{16}$$

The determinant of the system of equations (16) has the form (Eq. 17):

$$\begin{gathered} D(\omega) = \left[-m_1 \omega^2 + (1 + 2\mu_1 \omega i) k_1\right] \times \\ \times \left[-m_2 \omega^2 + (1 + 2\mu_1 \omega i) k_1 + (1 + 2\mu_2 \omega i) k_2\right] \times \\ \times \left[-m_3 \omega^2 + (1 + 2\mu_2 \omega i) k_2 + (1 + 2\mu_3 \omega i) k_2\right] - \\ - \left[-m_1 \omega^2 + (1 + 2\mu_1 \omega i) k_1\right] (1 + 2\mu_2 \omega i)^2 k_2^2 - \\ - \left[-m_3 \omega^2 + (1 + 2\mu_2 \omega i) k_2 + (1 + 2\mu_3 \omega i) k_3\right] (1 + 2\mu_1 \omega i)^2 k_1^2 \end{gathered} \tag{17}$$

It is convenient to write the last expression in the form (Eq. 18):

$$\begin{gathered} D(\omega) \approx m_1 m_2 m_3 \left(\omega^2 2\mu_{1f} p_1^2\right) \times \\ \times \left(\omega^2 - 2\mu_{2f} p_2^2 \omega i - p_2^2\right) \left(\omega^2 - 2\mu_{3f} p_3^2 \omega i - p_3^2\right) \end{gathered} \tag{18}$$

where μ_{1f} – where is the damping coefficient corresponding to the *r*-th form of natural oscillations; γ_{rf} – dissipative coefficient; p_i – natural oscillation frequencies; ω – operating frequency.
Expanding the brackets in expression (18), equating in the resulting expression and in (16) the coefficients at ωi and excluding the second-order values, after transformations we write the following system (Eq. 19):

$$\begin{cases} \mu_{1f} + \mu_{2f} + \mu_{3f} = \mu_1 + \mu_2 + \mu_3 \\ m_1 m_2 m_3 \left[\mu_{1f} p_1^2 \left(p_2^2 + p_3^2 \right) + \mu_{2f} p_2^2 \left(p_1^2 + p_3^2 \right) + \mu_{3f} p_3^2 \left(p_1^2 + p_2^2 \right) \right] = \\ = \left(\mu_1 + \mu_2 \right) k_1 k_2 \left(m_1 + m_2 + m_3 \right) + \left(\mu_1 + \mu_3 \right) k_1 k_3 \left(m_1 + m_2 \right) + \\ + \left(\mu_2 + \mu_3 \right) k_2 k_3 m_1 \left(\mu_2 k_2 + \mu_3 k_3 \right) + m_1 m_3 \left(\mu_1 k_1 + \mu_2 k_3 \right) + \mu_1 k_1 m_2 m_3 \\ m_1 m_2 m_3 \left[\mu_{1f} p_1^2 + \mu_{2f} p_2^2 + \mu_{3f} p_3^2 \right] = m_1 m_2 \end{cases} \quad (19)$$

We make the following notation (Eq. 20):

$$\begin{gathered} A = \mu_1 + \mu_2 + \mu_3 \\ B = \left(\mu_1 + \mu_2 \right) k_1 k_2 \left(m_1 + m_2 + m_3 \right) + \\ + \left(\mu_1 + \mu_3 \right) k_1 k_3 \left(m_1 + m_2 \right) + \left(\mu_2 + \mu_3 \right) k_2 k_3 m_1 \\ C = m_1 m_2 \left(\mu_2 k_2 + \mu_3 k_3 \right) + \\ + m_1 m_3 \left(\mu_1 k_1 + \mu_2 k_3 \right) + \mu_1 k_1 m_2 m_3 \end{gathered} \quad (20)$$

We also note that the damping coefficient of oscillations of the string steps with the damping coefficient is related by the dependence $\mu = 2mn$, then, taking into account this and the accepted designations in (20), the system of equations (18) can be submitted as (Eq. 21):

$$\begin{gathered} 2\left(n_{1f} m_1 + n_{2f} m_2 + n_{3f} m_3 \right) = A \\ 2 m_1 m_2 m_3 \left[n_{1f} m_1 p_1^2 \left(p_2^2 + p_3^2 \right) + n_{2f} m_2 p_2^2 \left(p_1^2 + p_3^2 \right) + n_{3f} m_3 p_3^2 \left(p_1^2 + p_2^2 \right) \right] = B \\ 2 m_1 m_2 m_3 \left[n_{1f} m_1 p_1^2 + n_{2f} m_2 p_2^2 + n_{3f} m_3 p_3^2 \right] = C \end{gathered} \quad (21)$$

If we take the average value among the coefficients $\mu_{1f}, \mu_{2f}, \mu_{3f}$ (Eq. 22):

$$\mu_{rf} = \left(\mu_{1f} + \mu_{2f} + \mu_{3f} \right) / 3, \quad (22)$$

which corresponds to the *r*-th form of natural oscillation, then from (21) it is possible to obtain an approximate formula for estimating the damping coefficient (Eq. 23):

$$n_{rf} \approx \frac{\left[A \dfrac{m_1 m_2 m_3 p_1^2 p_2^2 p_3^2}{p_r^2} + C p_r^2 - B \right] \left(p_{1+Rem(r+1,3)}^2 - p_{1+Rem(r,3)}^2 \right)}{m_1 m_2 m_3 \left(p_2^2 - p_1^2 \right) \left(p_3^2 - p_1^2 \right) \left(p_3^2 - p_2^2 \right)} \quad (23)$$

where $Rem(r,3)$ – where is the remainder of dividing the number of the proper form *r* by 3.

The base oscillations of the sucker rod string are longitudinal. The dissipation of the energy of these vibrations occurs due to friction of the rods in a viscous medium and internal friction in the material of the rods. Damping of string vibrations through energy dissipation leads to a decrease in their amplitude and frequency. Therefore, intensive damping of longitudinal vibrations of the rod string will be observed at their constant dissipation. This feature is expressed by the direct dependence of the damping coefficient μ_{rf} on the dissipation coefficient γ_{rf} (Eq. 24):

$$\mu_{rf} = \gamma_{rf} / \left(2\omega_0\right) \tag{24}$$

where ω_0 – the operating frequency, which corresponds to the fundamental mode of natural oscillations.

For a three-mass mechanical system that simulates the vibrations of a three-step string of sucker rods based on a comparison of dependences (23) and (24), the formula for determining the dissipation coefficient will have the following form (Eq. 25):

$$\gamma_{rf} = 4m\omega_0 \frac{\left[A\frac{m_1 m_2 m_3 p_1^2 p_2^2 p_3^2}{p_r^2} + Cp_r^2 - B\right]\left(p^2_{1+Rem(r+1,3)} - p^2_{1+Rem(r,3)}\right)}{m_1 m_2 m_3 \left(p_2^2 - p_1^2\right)\left(p_3^2 - p_1^2\right)\left(p_3^2 - p_2^2\right)} \tag{25}$$

According to (25), the dissipation coefficient is determined by the masses parameter, natural frequencies of the string and the stiffness of its steps. The frequency of natural vibrations of the string determined by the geometric dimensions of its steps, while the mass and stiffness of each step depends on its geometric dimensions and material. This explains the fact that the study of changes in the dissipation coefficient for layouts of strings of different stiffness is important to ensure their resonance-free operation.

A three-stage sucker rod string was selected for further research. The string stage lengths and diameters, complete with steel rods: first stage – 3329 feet (1015 m) and 1 inch (25 mm); second stage – 4325 feet (1318 m) and 0.875 inches (22 mm); the third is 1525 feet (465 m) and 1 inch (25 mm). The sucker rod string is designed to run the 2.25 in. (56 mm) wide pump down to 9,300 feet (2,835 m). Taking into account the current trends in the use of composite sucker rods, analytical studies were carried out for four options for completing a three-stage string (Table 2).

Table 2. Options for completing a three-stage sucker rod string

String stage	Stage materials for packaging arrangement			
	No. 1	No. 2	No. 3	No. 4
1	Steel	Fibreglass	Fibreglass	Steel
2	Steel	Steel	Fibreglass	Fibreglass
3	Steel	Steel	Steel	Steel

Further studies will be conducted for a three-stage combined sucker rod string, equipped with fibreglass and steel rods, the main parameters of which are given in the Table 3.

Table 3. Parameters of three-stage sucker rod string with fibreglass and steel rods

Stage	Material	Length, m	Diameter, mm	Weight, kg	Elastic modulus, Pa	Hardness, N/m
1	Fibreglass	1015	25	1339.8	$0,5{\cdot}10^{11}$	$2,418{\cdot}10^{4}$
2	Steel	1318	22	33345.4	$2,1{\cdot}10^{11}$	$6,057{\cdot}10^{14}$
3	Steel	465	25	15391.5	$2,1{\cdot}10^{11}$	$2,217{\cdot}10^{15}$

Operating frequency of longitudinal oscillations of the string $\omega_0 = 0,398$ rad/s; frequencies corresponding to the first, second and third forms of natural vibrations of the sucker rod string $p_1 = 3,178$ rad/s; $p_2 = 7,078$ rad/s; $p_3 = 12,088$ rad/s.

The stiffness of the stages of the rod string was determined by the formula (Eq. 26):

$$k_i = \frac{A_i E_i}{l_i} \tag{26}$$

where A_i, E_i, l_i – cross-sectional area, elastic modulus and length of each stage of the string, respectively. In accordance with the theoretical principles, for a qualitative analysis of dissipative functions (23), we use dimensionless coefficients, constructing models (Eq. 27):

$$\gamma_{rf} = 2\nu_{rf}\omega_0 \tag{27}$$

depending on dimensionless parameters $a_i = k_2 / k_1, a_2 = k_3 / k_2, c_1 = \gamma_2 / \gamma_1, c_2 = \gamma_3 / \gamma_2$. With a decrease in the ratio $a_1 = k_2 / k_1$ (i.e., with an increase in the stiffness of the first stage k_1) the dissipation coefficient γ_{1f} decreases and approaches 0,12; γ_{2f} and γ_{3f} increases, respectively, to values of 0.42 and 0.35. With an increase in the ratio $a_1 = k_2 / k_1$ (that is, with a decrease in the stiffness of the 1st stage k_1) γ_{1f} and γ_{2f} decrease to values of 0,14 and 0,1, respectively; and γ_{1f} – increases to 0.38.

With a decrease in the ratio $a_2 = k_3 / k_2$ (i.e., with an increase in the stiffness of the 2nd stage k_2) the dissipation coefficients of the second and third stages γ_{2f} and γ_{3f} increase, respectively, to values of 0,41 and 0,51; and γ_{1f} for the first stage decreases to a value of 0,12; and with an increase in the ratio $a_2 = k_3 / k_2$ (that is, with a decrease in the stiffness of the second stage k_2) γ_{3f} approaches γ_2. γ_{rf} depends on γ_i linearly, and at $c_1 = \gamma_2 / \gamma_1 = 1, \gamma_{2f} \approx \gamma_2; \gamma_{1f} \approx \gamma_{3f} \approx (\gamma_2 + \gamma_3)/2 = 0,4$. γ_{rf} depends on γ_i linearly, and at $c_2 = \gamma_3 / \gamma_2 = 1; \gamma_{3f} \approx \gamma_{3f} \approx \gamma_3 = 0,3$. The difference in the rigidity of the steps of the rod string can be carried out either by changing the geometric dimensions of the steps, or by changing the material of the steps. The use of the first fibreglass stage provides a large dissipation of oscillations in the upper sections of the sucker rod string in transient modes of its operation under the action of a low-cycle load.

On the basis of the findings above and the constructed model dependencies, it can be argued that by correctly selecting the stiffness of the stages of the rod string, it is possible to ensure the necessary dissipation of the energy of its vibrations, and at the same time to avoid resonance. The difference in the rigidity of the individual stages of the rod string can either need to change their geometric dimensions, or by changing the property of the material from which they are made. Adding to component No. 2 of the first fibreglass stage to replace steel (at $a_1 = 2,5$) this leads to increase the damping factor in the fibreglass stage to $\gamma_{1f} = 0,38$ and its simultaneous decrease in steel to $\gamma_{3f} = 0,1$. At the same time, the linear increase in the dissipation coefficients in No. 2, No. 3 and No. 4 with fibreglass stages causes a corresponding decrease in the amplitude and frequency of their vibrations. This approach makes it possible to prevent the phenomenon of resonance during the operation of a conventionally vertical string of rods in transient modes of its operation under the influence of a variable load. On the other hand, this will minimise the likelihood of fatigue damage in the elements of the rod string and their further destruction. Thus, taking into account the nature of the emergence of dissipative forces and their influence on the dynamic state of the rod string, the possibility of determining the parameters of oscillation dissipation in the stages of the rod string is substantiated, taking into account the parameters of its layout.

By compiling and solving a system of equations of motion of a conventionally vertical three-stage string of sucker rods, the values of the oscillation dissipation coefficients for the steps formed from fibreglass and steel rods were obtained. Use of a fibreglass step instead of a steel step reduces its rigidity by about 4 times, and increases the coefficient of oscillation dissipation by almost the same amount. The findings make possible to evaluate the dynamic behaviour of the sucker rod string and to establish optimal operating modes in order to avoid resonance in actual operating conditions.

Determining the real conditions for the onset of resonance during operational use is a difficult task due to the participation of many interrelated quantities in this process. The problem can be solved by establishing optimal operating modes of the system using the effect of reducing the spring stiffness and thus improving the actual operating conditions.

Performance characteristics can vary significantly when operating modes are changed, due to material fatigue and the quality of changes introduced during operation. In any case, the correct choice of modes of use contributes to improved performance and increased service life.

The effective application in engineering practice of the joints of the sucker rod structures operating under difficult conditions of contact load is can be explained by making a decision of the difficulties with contact. The complexity of solving the problem lies at the moment of detecting the stresses arising from contact the connection of dissimilar structural elements, which are the main indicators of its strength when establishing contact and stiffness at a given load. The studies presented in this section are based on the generalised theory of shells. Certain cases of contact interaction under load are being considered. To solve contact problems, we used the technique of reducing them to integral equations.

To simulate the considered scheme of structural connections, it is proposed to use a shell of a given length, made in the form of a cylinder, which is in contact with a rigid annular steel strip. The calculation of the voltages at the contacts, at typical loads, is given, taking into account the conditions in which this contact took place.

Within the framework of a mathematical model, rotation of the shell made in the form of a cylinder is considered, which interacts with a rigid band in a section $0 \le \alpha \le c$ of its length. The required bond strength between the shell and the steel cage can be ensured, the contact occurs through an adhesive layer of thickness $2h_k$. The load-bearing capacity of a given structural connection, taking into account the loading conditions, is determined by the maximum moment *M*, which is applied to the free end of the shell. Under the given conditions of loading the front surfaces of the shell $\left(\sigma_2^{\pm} = 0, \sigma_3^{-} = 0\right)$, the action of the band is manifested only through the tangential reaction. Thus, the task is to determine the contact shear stresses $\tau\left(\alpha\right)$, that arise in the adhesion region $0 \le \alpha \le c$ of the structural elements. In this case, the basic relationships are:

1. equilibrium equation (Eq. 28):

$$\begin{gathered} \frac{dS}{d\alpha} + Q_2 = -R\left(\sigma_{31}^{+} - \sigma_{31}^{-}\right) \\ \frac{dH}{d\alpha} - RQ_2 = -hR\left(\sigma_{31}^{+} + \sigma_{31}^{-}\right) \end{gathered} \tag{28}$$

2. elasticity ratio (Eq. 29):

$$\begin{gathered} S = \frac{B_{12}}{R}\frac{d\nu}{d\alpha}, Q_2 = \Lambda_2\left(\gamma - \frac{1}{R}\nu\right) + \frac{h}{3}\left(\sigma_{31}^{+} + \sigma_{31}^{-}\right) \\ H = \frac{D_{12}h}{R}\frac{d\gamma}{d\alpha}, M = 2\pi R\left(H + SR\right) \end{gathered} \tag{29}$$

where $\gamma = \gamma_2$ – where is the angle of rotation of the normal fibre, ν – displacement in the direction of the coordinate α.

It follows from the conditions of the problem that $\sigma_{31}^{-} = 0$ (Eq. 30):

$$\sigma_{31}^{+}\left(\alpha\right)=\begin{cases}\tau\left(\alpha\right) & 0\le\alpha\le c\\ 0 & c<\alpha\le L\end{cases} \tag{30}$$

where $\tau\left(\alpha\right)$ – unknown contact stress.

Taking into account the load and the fastening at the ends of the shell, the boundary conditions are formulated as follows (Eq. 31):

$$\gamma=0,\nu=0\ \left(\alpha=0\right),\ S=0,H=H_0\ \left(\alpha=L\right) \tag{31}$$

The presence of an adhesive layer in the region of contact interaction between the shell and the band, in accordance with the model, the contact condition has the form (Eq. 32):

$$\nu\left(\alpha\right)=-k\tau\left(\alpha\right)\ \left(0\le\alpha\le c\right) \tag{32}$$

where $k=2h_k/G_k^{'}$ – coefficient that characterises the shear stiffness of the adhesive layer, $G_k^{'}$ – shear modulus.

The mathematical model of the formulated the task is to draw up a system of differential equations (28) together with conditions (31), (32). We depict the system of equations in a universal form, which makes it possible to easily implement the method described above to the studied contact problem. Therefore , the matrix (Eq. 33):

$$F=\left\{a_{ij}\right\}_{i,j=1}^{4} \tag{33}$$

system has such elements (Eq. 34):

$$a_{13}=1,a_{14}=1,a_{31}=\frac{\Lambda_2R^2}{D_{12}h},a_{32}=-\frac{l}{R}a_{31},a_{41}=-\frac{R}{l}\frac{\Lambda_2}{B_{12}},a_{42}=\frac{\Lambda_2}{B_{12}} \tag{34}$$

The rest of the elements of the matrix *A* are equal to zero. The vectors *Z(α), F(α)* defined as (Eq. 35):

$$Z\left(\alpha\right)=\left(z_1,...,z_4\right)^T=\left(\gamma,\bar{\nu},\frac{d\gamma}{d\alpha},\frac{d\bar{\nu}}{d\alpha}\right)^T,F\left(\alpha\right)=\left(0,0,\mu\bar{\sigma}_{31}^{+},\chi\bar{\sigma}_{31}^{+}\right)^T \tag{35}$$

where $\mu=-\frac{5}{6}\frac{R^2E}{D_{12}},\chi=-\frac{R}{l}\frac{E}{B_{12}}\left(\frac{h}{6}+R\right),\bar{\nu}=\nu/l,\bar{\sigma}_{31}^{+}=\sigma_{31}^{+}/E$ – dimensionless quantities, σ_{31}^{+} is determined according to (33).

The solution to the problem will have the form (Eq. 36):

$$z_i\left(\alpha\right)=\sum_{k=1}^{4}C_kg_{ik}\left(\alpha\right)+\int_0^{\alpha}\left(\mu g_{i3}\left(\alpha-s\right)+\chi g_{i4}\left(\alpha-s\right)\right)_{\tau}^{-+}\left(s\right)ds,i=1,...,4 \tag{36}$$

From conditions (29) at $\alpha=0$, taking into account (Eq. 37):

$$w(\alpha)=z_1(\alpha)=\sum_{k=1}^{4}C_k g_{1k}(\alpha)-\int_0^{\alpha} g_{13}(\alpha-s)q(s)ds \tag{37}$$

follows that $C_1=C_2=0$. Two more steels C_3, C_4 are determined from conditions (25) at $\alpha=L$.
Based on the formula for tangential displacement in the domain $0\le\alpha\le c$, we obtain the expression (Eq. 38):

$$\bar{v}(\alpha)=z_2(\alpha)=C_3 g_{23}(\alpha)+C_4 g_{24}(\alpha)+\int_0^{c}\left(\mu g_{23}(\alpha-s)+\chi g_{24}(\alpha-s)\right)\bar{\tau}(s)ds \tag{38}$$

Further, according to the solution scheme, the contact problem can be represented as an integral equation for $\tau^*=\tau R^2/H_0$, where (Eq. 39):

$$K(\alpha,s)=\frac{1}{k_0}\begin{cases}\mu g_{23}(\alpha-s)+\chi g_{24}(\alpha-s)-y_3(\alpha), 0\le s\le\alpha\\ -y_3(\alpha), \alpha<s\le c\end{cases}$$

$$y_3(\alpha)=y_1(\alpha)\left(\mu g_{33}(L-s)+\chi g_{34}(L-s)\right)+y_2(\alpha)\begin{pmatrix}\mu g_{43}(L-s)-\\ -\chi g_{44}(L-s)\end{pmatrix}$$

$$y_1(\alpha)=\frac{1}{\delta}\left(g_{44}(L)g_{23}(\alpha)-g_{43}(L)g_{24}(\alpha)\right), y_2(\alpha)=$$

$$=\frac{1}{\delta}\left(g_{33}(L)g_{24}(\alpha)-g_{34}(L)g_{23}(\alpha)\right)$$

$$\delta=g_{33}(L)g_{44}(L)-g_{43}(L)g_{34}(L), f(\alpha)=\frac{R^3E}{D_{12}h}y_1(\alpha), k_0=\frac{kE}{l} \tag{39}$$

Using the found solution, to determine the dimensionless value of the torque $H^*=H/H_0$ in the region $0\le\alpha\le c$, obtain the formula (Eq. 40):

$$H^*(\alpha)=\frac{D_{12}h}{ER^3}\left(C_3 g_{33}(\alpha-s)+C_4 g_{34}(\alpha)\right)+\int_0^{c}\left(\mu g_{33}(\alpha-s)+\chi g_{34}(\alpha-s)\right)\tau^*(s)ds \tag{40}$$

In the domain $c\le\alpha\le L$ the function $H^*(\alpha)$ has the form (36), only the upper integration boundary α is replaced by c. Based on (29) and (35) for the shear force $S^*=SP/H_0$ in the indicated area $c\le\alpha\le L$ we can write (Eq. 41):

$$S^*(\alpha)=\frac{B_{12}l}{ER^2}\left(C_3 g_{43}(\alpha)+C_4 g_{44}(\alpha)\right)+\int_0^{c}\left(\mu g_{43}(\alpha-s)+\chi g_{44}(\alpha-s)\right)\tau^*(s) \tag{41}$$

In the domain $c<\alpha\le L$ the function $S^*(\alpha)$ is defined similarly to a function $H^*(\alpha)$ in this area. The strength characteristics of composite materials, it is important to find the distribution of shear stresses σ_{12}^* along and over the thickness of the cylindrical shell. The calculation of the dimensionless stress $\sigma_{12}^*=R^2\sigma_{12}/H_0$ carried out according to the formula (Eq. 42):

$$\sigma_{12}^{*} = \frac{1}{2}\left(\frac{R}{h}\right)S^{*} + \frac{3}{2}\left(\frac{R}{h}\right)^{2} H\left(\frac{z}{h}\right) \tag{42}$$

where $-h \leq z \leq h$.

The method of constructing the solution of contact problems gives the accuracy of the results sufficient for engineering practice. The results of calculating the tangent contact points $\tau^{*}(\alpha)$ during torsion of the shell should be given. The graphs are plotted for dimensionless quantities $\tau^{*}(\alpha) = R^{2}\tau(\alpha)/H_{0}$. The problem was solved for the following values of the connection parameters: $E/G' = 7.6, \nu = 0.18, l/R = 2, h/R = 1, c/R = 1, h_{k}/h = 0.001, E = 19\,FPa$.

The curves are plotted for parameter values $G_{k}^{'} = 0.95, 1.0, 1.05, 2.8$ respectively. The stress distribution σ_{12}^{*} over the thickness of the composite shell is along the axial coordinate α. The curves are plotted at $z = 0, 0.5, 1.0$, respectively, where $\bar{z} = z/h$. The maximum contact stress $\tau^{*}(\alpha)$ is reached at a certain distance from the rigidly fixed edge; moreover, the concentration τ^{*} decreases with increasing $G_{k}^{'}$, and the maximum is observed closer to the fixed edge.

A change in the ultimate strength of sucker rods causes changes in the conditions of practical operation of the entire structure, which leads to a reduction in the time of its use in a number of cases. Conducting design tests in the open air contributes to obtaining the best quality picture, since the conditions are as close as possible to real, operational situations. Subsequently, the performance of sucker rods may change and this necessitates quality monitoring of these parameters in order to maintain high standards of design quality.

Most of the methods for design calculations of the sucker rod string based on the determination of the safety factor beyond the endurance limit were obtained when tested in air. Nevertheless, the endurance limit of sucker rods decreases in corrosive environments by 30-50% and can take on very small values, especially at large test bases (20-50 million cycles). This circumstance can significantly affect the results of calculating the strength of rods. For products that do not have a specific fatigue limit, it is more expedient to determine not the safety factor, but the probable number of load cycles that a part or product can work before failure. In this case, consider a sucker rod having a limited (conditional) endurance limit with a constant slope of the fatigue curve lines in logarithmic coordinates. We can use the method of calculating the number of cycles of product load based on the following two-parameter equation (Eq. 43):

$$\sigma_{a}^{m} N = const \tag{43}$$

Then the strength equation can be written as (Eq. 44):

$$\sigma_{-IN}^{m} N_{\sigma} = \left(k_{1}k_{2}\sigma_{\phi}\right)^{m} N_{ymo} \tag{44}$$

hence (Eq. 45):

$$N_{ymo} = \left(\frac{\sigma_{-1N}}{k_{1}k_{2}\sigma_{\phi}}\right)^{m} N_{\sigma} \tag{45}$$

where σ_{-IN} – conditional endurance limit in the borehole environment for the base number of load cycles N_{σ}; N_{ymo} – probable number of load cycles of a sucker rod string element; m – indicator of the angle of

inclination of the left branch of the fatigue curve; σ_{ϕ} – the actual probable value of the expected equivalent alternating stresses.
For alternating bending stresses, we have (Eq. 46):

$$\sigma_{\phi} = \sigma_a + (\Psi) D\sigma_m \tag{46}$$

The coefficient k_1 takes into account the degree of aggressiveness of the formation water, the coefficient k_2 – depends on the angle of the well deviation. The values of these coefficients are contained within wide limits and have not yet been fully determined for the elements of sucker rods. As a first approximation, it can be assumed that the influence of factors such as the type of load, asymmetry of the cycle, aggressiveness of the corrosive environment, and structural features of the material will implicitly enter into the value of the actual probable value of the expected equivalent stresses σ_{ϕ} through the coefficient $(\Psi)D$. Then the probable number of load cycles will be determined from the expressions (Eq. 47-48):

$$N_{ymo} = \left(\frac{\sigma_{-1N}}{\sigma_{\phi}}\right)^m N_{\sigma} \tag{47}$$

$$T_{yvo} = N_{ymo} / 60n \tag{48}$$

where T_{ymo} – probable operating time of the product (hours), n – frequency of strokes of the polished rod (min^{-1}).
For each sucker rod size, which has the appropriate material strength and is operating under the specific operating conditions, the slope will be constant and will be determined by the curve m. The slope of the fatigue curve m determined from the relationship (Eq. 49):

$$m = lgN(N_1 / N_2) / lg(\sigma_1 / \sigma_2) \tag{49}$$

Fatigue curves are straightened in logarithmic coordinates $lg\sigma_a - lgN$. We introduce the notation $lg\sigma_a = Y$ and $lgN = X$, then the equation through the linear correlation coefficient R will be written in the following form (Eq. 50):

$$\hat{Y}(x) = \overline{y} + R\frac{S_y}{S_x}(x - \overline{x}) \tag{50}$$

Here $\overline{y} = \sum y_i / n, \overline{x} = \sum x_i / n$ – arithmetic mean values of random variables Y, X, S_y, S_x – mean square deviations for Y, X (Eq. 51):

$$S_y = \sqrt{\frac{\sum(y_i - \overline{y})^2}{n-1}}; S_x = \sqrt{\frac{\sum(x_i - \overline{x})^2}{n-1}} \tag{51}$$

n – the number of parts tested to construct the fatigue curve.
The sample correlation coefficient *R* is estimated by the formula (Eq. 52):

$$R = \frac{\sum(x_i - \bar{x})(y_i - \bar{y})}{(n-1)S_x S_y} \tag{52}$$

The value of the index *m* in the equation of the fatigue curve is also found using the expression (Eq. 53):

$$m = -\frac{1}{RS_y / S_x} \tag{53}$$

Using equations for the values x_i (also for x, which corresponds to N_σ) the values $y_i(x)$ and their corresponding values σ_{ai} are calculated.

The significance of the obtained linear regression equation is checked by Fisher's test (Eq. 54):

$$F = S_y^2 / S_{zal}^2 \tag{54}$$

where S_y^2 – variance of the random variable; Y; $S_{zal}^2 = \frac{1}{n-k}\sum\left[y_i - \hat{y}_i(x)\right]^2$ – residual variance (an estimate of the variance of a random variable of the empirical regression line); k – number of unknown coefficients in the linear regression equation (Eq. 55):

$$y = a_c + a_1(x), k = 2\,. \tag{55}$$

Further, according to the table of quantiles of the Fisher distribution for the selected confidence probability (usually $\alpha = 0,95; \alpha = 0,90$), the number of degrees of freedom of the numerator $(n-1)$ and denominator $(n-k)$, the critical value is found (Eq. 56):

$$F_{kr}(n-1, n-k) \tag{56}$$

In the case $F_1 > F_{kr}$ of the selected confidence level, it is possible to assert the existence of a relationship between σ_a and N in the form of an estimation equation. When used, the conditions assume that the relationship between the variables is not statistically significant. In this case, further use of the regression equation is unacceptable.

The random variable $Y_1(x)$, that corresponds to the lower one-sided confidence limit at the confidence level $(1-\alpha)$, is calculated by the formula (Eq. 57):

$$\hat{Y}_1(x) = \hat{Y}(x_i) + t_{\alpha(n-2)} S_{zal} \sqrt{1 + \frac{1}{n} + \frac{(x_i - \bar{x})^2}{\sum(x_i - x)^2}} \tag{57}$$

where $t_{\alpha(n-2)}$ – the percentage point of the student's distribution for the number of degrees of freedom $(n-2)$; $\sqrt{1+\frac{1}{n}+\frac{(x_i-\bar{x})^2}{\sum(x_i-x)^2}}$ – a component that takes into account the increase in the degree of scattering $Y(x)$ of the empirical mean x_i. The found value $Y_1(x)$ after potentiation corresponds to the values $\sigma_{a1}(i)$.
Using a three-parameter equation, the scattering characteristics of the endurance of sucker rods tested in downhole environments are calculated (Table 3), and an example of computer construction of fatigue curves for fibreglass rods is given, and complete diagrams of fatigue failure are constructed. Using a three-parameter equation, the endurance dissipation characteristics of steel and composite sucker rods tested in downhole environments were estimated (Table 4).
Based on tests of steel and composite sucker rods, complete models of fatigue failure of sucker rods were built with a probabilistic assessment $P(N)$ of their durability, which allow predicting their resource (Table 5).

Table 4. Dispersion characteristics of endurance of sucker rods made of steel and fibreglass

Rod type	σ_{ser}	σ_{min}	σ_{max}	V_0	Q	N_b	S	$S=S/\sigma_{ser}$
Standard	69.6	53.68	85.52	100	10.0	1.436	22.25	0.319
Reservoir water with oil	70.32	62.64	78	24.47	10.0	1.419	6.78	0.09
Reservoir water	64.73	50.8	78.62	19.32	10.0	1.545	11.57	0.178
Oil with 10% HCL	98.43	80.4	116.47	39.03	10.0	1.016	15.02	0.15

Table 5. Dispersion characteristics of endurance of sucker rods made of steel and fibreglass

Rod type	σ_{ser}	σ_{min}	σ_{max}	V_0	Q	N_b	S	$V=S/\sigma_{ser}$
Steel	69.6	53.68	85.52	100	10.0	1.436	22.25	0.319
Fibreglass	61.58	51.86	71.3	18	10.0	1.623	7.83	0.124

According to the results of tests of full-scale rods with a diameter of 22 mm made of steel in air and in a corrosive environment, the calculated characteristics of fatigue $(\sigma_r=187,6)$ Q=8,9·10^6, $V_0=170,7$ and corrosion induced fatigue $(\sigma_{rk}=14,2)$, Q=56,7·10^6, $V_{0k}=55,74$.

4. Conclusions

Note that the proposed method for solving contact problems is convenient to use, its software support has been implemented, which allows performing a multiparametric analysis of calculated values. On the basis of the study conducted, a number of industrial samples of sucker rods have been created, which have significant advantages in comparison with steel ones: they allow increasing the resistance to the corrosive action of aggressive media, significantly reducing their accident rate during operation. The experience of using the equations showed that when processing the findings of fatigue studies, the second equation gives good estimates of the average value of the endurance limit, but the estimates of the standard deviation are underestimated. The indicated equation is simpler for computer implementation, and its parameters are clearer.
The results of corrosion-fatigue tests are well described using equation (1), which allows us to conclude that the equation can be used to describe corrosion fatigue curves, in particular, the section before the bend of the

curve. The experience of using equations (1) and (2) shows that, when processing the results of fatigue studies, the second equation gives good estimates of the mean value of the fatigue limit, but underestimates of the standard deviation. This equation is easier to program, and the parameters for this equation are more descriptive. However, for a more accurate statistical estimate, it is necessary to use equation (1). With the help of the developed methodology and computer program, it is possible to predict the resource of both steel and composite sucker rods and justify the frequency of their flaw detection, which will make it possible to efficiently use the rods and significantly reduce the accident rate during oil production.

The practical use of numerical modeling methods in engineering calculations contributes to obtaining the most objective picture of the test and accelerates the implementation in practice of effective schemes for improving the quality of operated structures.

Corrosion-fatigue tests carried out and their evaluation by numerical simulation methods constitute an excellent material for subsequent use in the development of new type of strength structures capable of withstanding increased loads in real operating conditions. At the same time, if it is necessary to obtain a more accurate assessment, methods of computer modeling of a real situation can be used, which ultimately has a positive effect on reducing the accident rate of the oil production process, as already noted above.

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