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DETERMINATION OF THE CHARACTERISTICS OF DRILL STRING VIBRATIONS DURING THE DRILLING PROCESS

The object of research is vibration processes of a certain origin in the drill string with typical design deviations depending on the mode parameters of drilling. A drill string is an oscillating system with an infinite number of degrees of freedom of a multifactor system. An exhaustive study of oscillatory processes in the drill string is impossible neither analytically nor experimentally, due to the specifics of the hole deepening in various rocks, the design of the well, its shape, etc. Therefore, in practice, they try to solve the problems of the dynamics of the drill string for an idealized system and, while preserving the main oscillatory properties, solve some problems of the rod system. The work carried out was aimed at experimental studies of vibrations of the drill string during the drilling process.

It is shown that the effectiveness of the use of hydrodynamic cavitation requires the development of methods and devices for intensifying the well drilling process. It is proven that the design of the cavitation generator organically fits into the existing well drilling equipment and allows for the intensification of technological processes with lower specific energy consumption. It is found that all oscillatory processes that occur in the drill string are random in nature and must be considered using the mathematical apparatus of the theory of random oscillations.

The study of vibrations during well drilling shows that vibrations can be considered as random stationary processes, since transient modes have a sufficiently short duration for homogeneous rocks with fixed drilling modes. The analysis of the vibrations of the drill string elements based on random oscillations in a number of cases allows to increase the reliability of determining the vibration reliability of the drill string elements. It has been proven that the response of drill string elements to broadband random vibration can be defined as the combined effect of several narrowband random vibrations.

Keywords: *well drilling, drill string, vibration reliability, hydrodynamic cavitation, cavitation generator design.*

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1. Introduction

The vibrations of the drill string, as a mechanical oscillating system, are described by a number of relevant differential equations and are characterized by certain parameters. Harmonic vibrations are determined by three independent parameters: amplitude, frequency and initial phase. In turn, the derived units are vibration velocity and vibration acceleration. The relationship between these values and recommendations for their use in vibrometry are given in a number of works [1–3].

The vibrations of the drill string are its reaction to the action of disturbing forces. The characteristics and magnitude of the reaction depend on many factors, in particular, on the method of drilling, geological and technical conditions of drilling, layout of the column bottom (LCB), etc. [4–6].

A drill string is an oscillating system with an infinite number of degrees of freedom of a multifactor system. An exhaustive study of oscillatory processes in the drill string is impossible neither analytically nor experimentally, due to the specifics of the hole deepening in various rocks, the design of the well, its shape, etc.

Therefore, in practice, they are trying to obtain a solution to the problems of the dynamics of the drill string for an idealized system [6–8] and, while preserving the main oscillatory properties, to solve some problems of the rod system [9–11].

The main features of the vibration of the drill string are determined by the nature of the interaction of its elements with the walls of the well and the bit with the hole, as well as the occurrence of cavitation processes caused by the flow of washing solution through the turbo drill and special devices.

The aim of research is to consider the occurrence of vibration processes of a certain origin in the drill string with typical structural deviations depending on the mode parameters of drilling.

In the future, the vibration velocity amplitude will be used to characterize the vibrations.

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2. Materials and Methods

Studying the course of vibrations over time, in some cases, makes it possible to reveal their structure.

The analysis of spectra and time characteristics of drill string vibrations allows, in the first approximation, to characterize the properties of the latter. When synthesizing vibrations, the task of obtaining specific data on vibrations of one or another LCB is not set. The task is to reproduce the frequency structure of the vibrations of this layout of their generators, related to the drilling conditions, mode parameters and its design features.

The use of deterministic connections of the frequency components of vibrations with the interaction of kinematic pairs in the process of deepening the hole and changing the regime parameters in this geological section lay the methodological foundations for the synthesis of the vibrational state of the drill string with this layout.

The work uses the method of mathematical modeling and experimental studies of the mechanical system. Variants of the occurrence of vibration processes of a certain origin in the drill string with typical structural deviations depending on the mode parameters of drilling were investigated. It is shown that vibrations can be considered as random stationary processes, since the transient modes have a sufficiently short duration for homogeneous rocks with fixed drilling modes. It is proved that the reaction of drill string elements to broadband random vibration can be defined as the total effect of several narrow-band random vibrations. A hydrodynamic generator of oscillations of the washing solution flow and cavitation devices were used for research.

3. Results and Discussion

Vibrations that occur in the drill string are generated by vibration sources, mainly related to the structure of the bottom of the string and the bit with the outcrop rock, as well as the drilling modes.

However, in the future, let's introduce the assumption that when solving the synthesis problem, let's consider that the vibration of any arrangement is complex and is disturbed by independent sources.

In this case, complex vibration can be represented as a sum of terms disturbed by individual sources:

$$X_{\Sigma}(t)=\sum_{i}^{n} x_i(t), \tag{1}$$

where i – the component of vibration displacement, which can be represented in the form of a series.

For convenience, let's divide the frequency components into two groups: the first group is the main components with significant amplitudes; the second group – components with a small amplitude.

Due to the fact that vibrations with a significant amplitude have, as a rule, a relatively low frequency, they are classified as low-frequency. Therefore, with the above, let's introduce the concept of high-frequency and low-frequency vibrations of the drill string.

Vibrations that occur when drilling with downhole motors are, as a rule, of a high-frequency nature. This is due to the frequency of rotation of the spindle and the interaction of the bit with the hole, as well as during the movement of the washing liquid through the turbine, when cavitation phenomena occur, which cause oscillations of the washing solution under the action of water hammers during the destruction of air bubbles. It can be assumed that the phenomenon of cavitation can also occur in screw engines (the phenomenon has not been studied).

In this way, the drilling motor itself is subjected to vibration loads, and its reaction is transmitted to the bit and the drill string.

The vibration of the power unit can be called the general vibration (linear and angular) of the entire assembly. Vibrating transducers for measuring vibrations and assessing the overall vibrations of the drill string were installed on the swivel as a standard measurement point, and on the drill rigs and above the bit (experimental drilling rig of OJSC «Ukrainian Oil and Gas Institute»). Vibrations at the standard point do not correspond to the maximum of torsional and bending vibrations, however, the transformation of these two types of vibrations into longitudinal vibrations of the top of the drill string most accurately determine the nature of the vibration system.

By analogy with general mechanical engineering, let's introduce the classification of vibrations according to the structural units that disturb this type of vibrations. Let's distinguish between «rotor», «bearing», «hydrodynamic» vibrations, as well as «tooth» and «pothole» vibrations adopted in drilling, etc.

Such an approximate classification makes it possible to emphasize the source of vibrations and their causes, as well as to predict, in general terms, the nature of vibrations of the bottom of the drill string depending on its design and the design of the used downhole engine.

The analysis of vibrations of various arrangements of the drill string in the process of deepening the well in connection with the complex root causes of their occurrence is a rather difficult task, and in some cases, it is completely insoluble.

Fig. 1 shows the vibrogram of the top of the drill string when drilling a well in the Stryi sediments, which consist of hard rocks. The used layout of the bottom of the drill string (LCB): bit SZE 290, turbo drill A9Sh, weighted drill pipes (WDP) ∅203 mm – 42 m, steel drill pipes (SDP) – 1423 m, traveling block and swivel – 7.2 kN.

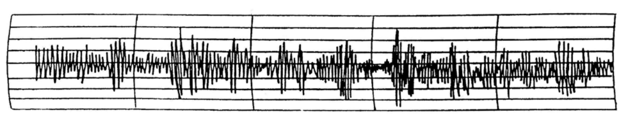

Fig. 1. Vibrogram during drilling with an A9Sh turbodrill in the Stryi deposits: P_{OC}=200 kN, outcrop 1200 m, WDP ∅203 mm – 45 m

Let's observe how the vibrational process of the top of the drill string changes when the axial load on the bit changes and some additional nodes are introduced into the arrangement or design of the downhole engine, which, in general, affect the vibration state of the arrangement and the drill string.

The axial load varied step by step: bit above the hole, 60 kN, 120 kN, 180 kN, 340 kN. At the same time, the vibrations of the top of the drill string were recorded on the magnetic carrier and on the H-375 high-speed recorder using band-pass filters: 0–30 Hz; 30–70 Hz; 30 Hz is an open channel.

Analysis of vibrations in the range of 0–30 Hz showed that this frequency is proportional to the frequency of rotation of the turbodrill rotor. At the same time, an increase in load leads to an increase in frequency and is determined by the following dependence:

$$f = \frac{nk}{60}, \quad (2)$$

where k – the frequency multiplication factor. In our case, it is equal to – 1, 2, 4, 8 – respectively for idle speed of 60 kN; 120 kN; 180 kN; 240 kN.

According to the experimental data, a graph of the dependence of the mechanical speed on the frequency of vertical movements of the bit, obtained by recalculating the data taken from the vibrogram according to dependence (2) and changes in the axial load, is plotted.

The graph in Fig. 2 shows that the change in the rotation frequency of the turbodrill shaft, approximated in the first one, can be described exponentially. However, this exponent has an emission at a load of 120 kN, which can be explained by the occurrence of self-oscillations due to non-linear friction proportional to the speed of rotation of the shaft of the demolition engine. Such phenomena are also present in other experiments. In addition, with such an axial load (120 kN), the vibration level at a frequency above 30 Hz also increases.

From the given data, it can be concluded that the source of the vibrations discussed above is the imbalance of the rotor of the knock-out engine.

The unbalance of the rotor is considered below according to the scheme of Fig. 3.

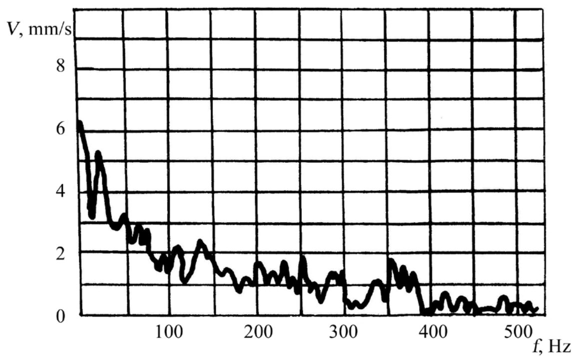

Fig. 2. Graph of the dependence of the bit rotation frequency when recalculating the frequencies taken from the vibrogram

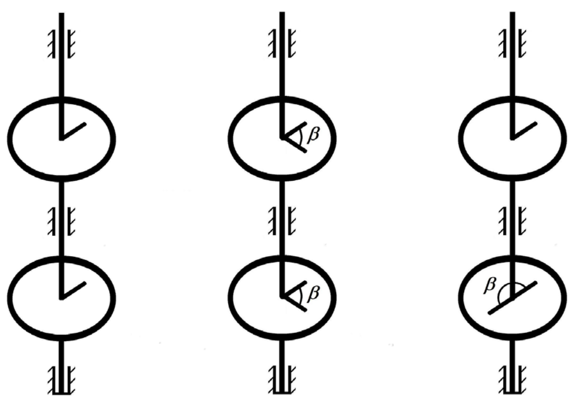

Fig. 3. Schemes of unb alanced rotors (simplified schemes)

The turbine rotor is presented as a long (6–8 m) elastic shaft with a set of turbine disks. The shaft of a single-section turbodrill is a three-support vertical structure in which, during rotation, torsional and bending vibrations occur.

The study of the combined influence of rotor stiffness on bending and torsional vibrations makes it possible to reveal their influence on the dynamic state of the drill string. The insufficiency of the combined influence of torsional and bending vibrations of the rotor of the drilling motor on its stability, taking into account the loading torque, does not allow to explain the occurrence of periodic axial loads on the bit and the drill string, which change in proportion to the angular speed of rotation of the bit. In addition, the recording of the vibrations of the top of the drill string during drilling with downhole engines (for example, a turbo drill) showed that in a certain range of rotation frequencies of the bit (with significant loads), the amplitudes of oscillations increase sharply. Measurements of defects in the casings of downhole engines (turbodrills), the preliminary deflection of the rotor and its imbalance affect the stability of the turbodrill turbine.

The turbine rotor was presented in the form of a simplified equivalent diagram (Fig. 3), where the masses of all turbines are modeled by concentrated masses located between the average radial support and the corresponding axial supports – m_1 and m_2, and their polar moments of inertia $m_1r_1^2$ and $m_2r_2^2$. Here, r_1 and r_2 are the corresponding radii of inertia. The bending stiffness of the turbodrill shaft was expressed through the stiffness coefficients C_{11}, C_{12} and C_{22}, and the torsional stiffness of the shaft between the discs through C_k. The imbalance of both parts of the rotor was expressed through the eccentricity of the disks ε_1 and ε_2. Marked by β – the angle formed by the axial planes of imbalance, which pass through the center of gravity of the discs.

It was assumed that the turbine rotor rotates with an average angular velocity ω. Let's denote by φ_1 and φ_2 the instantaneous angles of rotation of the respective discs. A torque acts on the turbine rotor, which is a function of the angular velocity:

$$M_0 = M_0(\dot{\varphi}),$$

where $\dot{\varphi}$ – the rotation angle of the spindle shaft. Moments due to the viscous friction of the drilling fluid flowing through the turbine, as well as friction in the bearings, act on the entire shaft. These moments depend mainly on the angular velocity $M_1(\dot{\varphi})$ and $M_2(\dot{\varphi})$. The absolute angle of rotation of the disks:

$$\varphi_{1a} = \omega t + \varphi_1; \;\; \varphi_{2a} = \omega t + \varphi_2. \quad (3)$$

The moments acting on the discs are expressed as:

$$M_{1a} = M_0(\omega + \varphi_1) - M_1(\omega + \varphi_2); \quad (4)$$

$$M_{2a} = -M_\partial(\varphi_2) - M_2(\omega + \varphi_2), \quad (5)$$

where $M_\partial(\varphi_2)$ – the moment of resistance transmitted from the bit to the shaft of the drilling motor.

The resulting intense vibrations of the rotors are transmitted to the drill string and the bit through the body of the turbo drill, which, as a rule, has a previous curvature associated with installation work, transportation, etc., which affects the working conditions of the bit on the pits.

Dynamic loads in this case are significant and affect the dynamics of the entire drill string. Fig. 4 shows the records of longitudinal vibrations of the bit and oscillations of the casing of the drilling motor.

Comparison of vibrograms in Fig. 4 shows that the longitudinal amplitude of oscillations of the body of the A9Ш turbo drill is much greater than the amplitude of longitudinal oscillations of the bit.

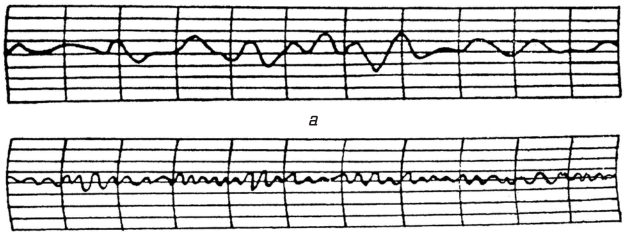

Fig. 4. Vibration records: *a* – longitudinal vibrations of the bit during the drilling process; *b* – longitudinal oscillations of the turbodrill body after three drillings

To increase the drilling performance, a hydrodynamic, high-frequency pulse perforator was used, which made it possible to increase the mechanical drilling speed by 60 % and increase the stability of the bit by 37 % [12]. The principle of operation of the device is based on the use of the phenomenon of cavitation with the use of special devices.

Cavitation is a violation of the integrity of the liquid, which occurs in those areas of the flow of washing liquid, where the pressure decreases and reaches a certain critical value. This process is accompanied by the formation of a large number of bubbles filled with vapors of the washing liquid, which are released from the washing liquid. Falling into the area of reduced pressure, the bubbles increase in size and turn into large bubbles – caverns.

These bubbles are then carried by the flow to the outlet, where the pressure is above the critical one, and are destroyed due to steam condensation. In this way, a limited cavitation zone is created in the flow of the washing solution, which is filled with moving bubbles.

The critical pressure, from the point of view of cavitation, is determined by the physical properties of the flushing solution and, depending on its condition, can vary widely. But when calculating devices for enhancing the effect of cavitation phenomena on the wellbore, it is necessary to take the pressure of the steam-saturated flushing solution as the critical pressure at which cavitation begins. An example is the occurrence of cavitation on a profile surrounded by a liquid stream.

A pothole-shaped and deformed bump is a mechanism for the deflection of streamlines, which causes a decrease in pressure on the convex surface of the profile of the slope or unevenness of the bump. It should be noted that the level of cavitation will depend on the density of the hole ρ_0, the pressure p_0 and the speed of the onrushing liquid flow v_0, the shape of the profile of the deformed hole and the angle of attack.

The instability of the cavitation zone, which is caused by special devices (cavitation nozzles), leads to the appearance of secondary liquid flows and, as a result, to a significant pulsation in the flow, which exert a dynamic effect on the breakout.

The destruction of the hole, which is associated with the «slamming» of cavitation bubbles when they are transferred by a liquid stream to an area with a pressure above the critical one, takes place very quickly and is accompanied by hydraulic shocks. In the final result, cavitation is accompanied by the destruction of the hole. This destruction makes it possible to increase the mechanical speed of drilling.

Cavitation phenomena are enhanced due to the introduction of special generators of hydrodynamic pulses into the layout, such as some of those shown in Fig. 5, 6, the construction of which will be more perfect [13, 14].

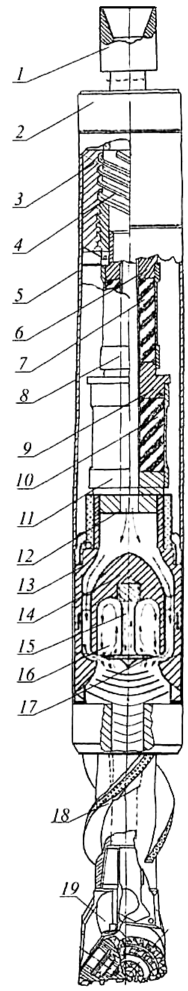

Fig. 5. Drill bit with a high-frequency cavitation vibrator: 1 – converter; 2 – packing nut; 3 – multi-turn nut; 4 – multi-turn non-self-braking screw; 5 – body; 6 – piston; 7 – rubber filler of the first degree; 8 – elastic shell of the first degree; 9 – double piston; 10 – rubber filler of the second degree; 11 – shell of the second degree; 12 – barrel; 13 – hydroacoustic generator; 14 – fairing; 15 – rod resonator; 16 – toroidal resonance chamber; 17 – plate resonator; 18 – spiral expander; 19 – ball bit

A drill bit with a high-frequency cavitation vibrator (Fig. 5) works as follows [13]. The device is installed between the ball bit 19 with a spiral expander 18 and the drive shaft of the drilling motor or weighted drill pipes (WDP) and is connected to them through the adapter 1 and the packing nut 2 with the screw 4. When the drive shaft rotates, the multi-turn non-self-braking screw 4 is screwed into the multi-turn nut 3 and the axial force from it is transmitted through elastic elements. These elements consist of elastic shells of the first 8 and second 11 stages with an elastic filler placed in them, respectively, the first 7 and second 10 stages, connected by a double piston 9 through a hydroacoustic generator 13 (the operation of which will be discussed below) through a spiral expander 18 to a ball bit 19 and intensifies its work. At the same time, depending on the mode of axial loading, one or two elastic elements may be included in order to obtain vibration protection against chisel disturbances. The flow of washing liquid under pressure enters the hydrodynamic generator 13 through the barrel 12 to the fairing 14 and flows around its surface through the annular space and is directed into the toroidal resonance chamber 16 with rod 15 and plate 17 resonators. From the hydrodynamic generator 13, the washing liquid through the channels in the spiral expander 18 and the ball bit 19 is fed to the hole with extraction of acoustic energy. At the same time, energy waves are directed perpendicular to the resonator plane.

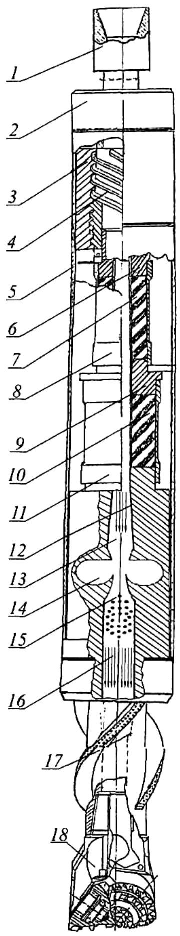

Fig. 6. Drill bit with cavitation chamber:
1 – converter; 2 – packing nut; 3 – multi-turn nut; 4 – multi-turn non-self-braking screw; 5 – body; 6 – piston; 7 – rubber filler of the first degree; 8 – elastic shell of the first degree; 9 – double piston; 10 – rubber filler of the second degree; 11 – shell of the second degree; 12 – barrel; 13 – hydrodynamic emitter of sound vibrations; 14 – toroidal vortex chamber with sharp edges; 15 – output chamber of the hydrodynamic emitter; 16 – output channel; 17 – spiral expander; 18 – ball bit

A drill bit with a cavitation chamber (Fig. 6) works as follows [14]. The device is installed between the ball bit 18 with the spiral expander 17 and the drive shaft of the drilling motor or weighted drill pipes (WDP) and is connected to them through the converter 1 and the packing nut 2 with the screw 4. When the drive shaft rotates, the multi-turn non-self-braking screw 4 is screwed into the multi-turn nut 3 and the axial force from it is transmitted through elastic elements. These elements consist of elastic shells of the first 8 and second 11 stages with an elastic filler placed in them, respectively, the first 7 and second 10 stages, connected by a double piston 9 through a hydrodynamic emitter of sound vibrations 13 (the operation of which will be discussed below) through a spiral expander 17 on ball bit 18 and intensifies its work. At the same time, depending on the mode of axial loading, one or two elastic elements may be included in order to obtain vibration protection against chisel disturbances. The flow of washing liquid enters under pressure into the hydrodynamic emitter of sound vibrations 13 through the barrel 12 and is divided by the sharp edges of the toroidal vortex chamber 14 into parts. One of the parts is directed into the middle of the toroidal vortex chamber 14, thanks to which a rotating jet appears in it, and the other part of the jet into the exit chamber of the hydrodynamic emitter 15, where the self-oscillatory interaction of the jets between themselves at the exit from the toroidal vortex chamber 14 and the main flow generates acoustic vibrations. At a certain intensity of acoustic vibrations, a cavitation mode occurs in the output chamber of the hydrodynamic emitter 15 of the hydrodynamic emitter of sound vibrations 13, the energy of which is directed to the chisel bit 18 through the output channel 16 of the hydrodynamic emitter of sound vibrations 13 and the spiral 5 expander 17.

In addition, the spindle of the punching motor with cavitation and screw amplifiers of the axial load on the bit was developed (Fig. 7) [15].

The spindle of the punching motor with cavitation and screw amplifiers of the axial load on the bit (Fig. 7) works as follows. The device is attached to the turbine using a converter 2. When the turbine rotates, the torque is transmitted through the splined semi-coupling 4 to the shaft 3 and further, through the multi-turn non-self-locking screw 12 and the multi-turn nut 13, shaft 14, the hydroacoustic generator 16 to the bit 21.

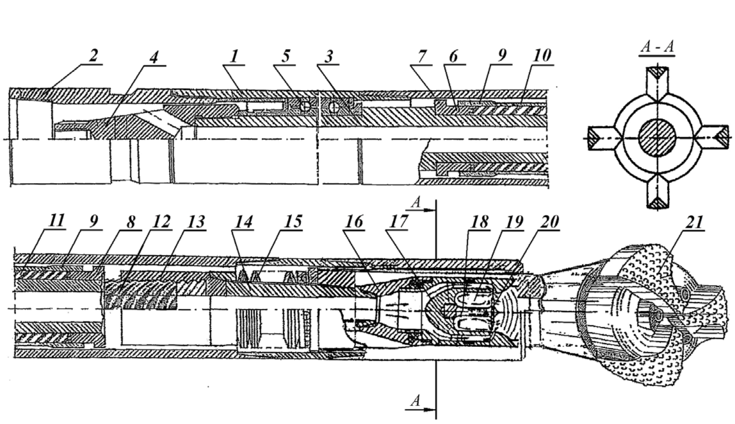

Fig. 7. The spindle of the punching motor with cavitation and screw amplifiers of the axial load on the bit: 1 – body; 2 – converter; 3 – hollow stepped shaft; 4 – splined semi-coupling; 5 – block of support-radial bearings; 6 – the upper piston of the shell shock absorber; 7 – shell shock absorber body; 8 – the lower piston of the shell shock absorber; 9 – sealing rings; 10 – thin-walled elastic shell; 11 – rubber filler; 12 – multi-turn non-self-locking screw; 13 – multi-turn nut; 14 – shaft; 15 – package of disc springs; 16 – hydroacoustic generator; 17 – fairing; 18 – rod resonator; 19 – toroidal resonance chamber; 20 – plate resonator; 21 – ball bit

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When the drive shaft rotates, the multi-turn the non-self-locking screw 12 is screwed into the multi-turn nut 13 and the axial force from it is transmitted through the elastic elements of the plate springs 15 and the hydroacoustic generator 16 (the operation of which was discussed above) to the ball bit 21 and intensifies its operation. From the bit 21, longitudinal vibrations are transmitted through the hydroacoustic generator 16 to the shaft 14, which under the action of the axial load rests against the piston 8, which in turn deforms the rubber filler 11, the pressure of which is transmitted to the elastic shell 10, deforming it radially. At the same time, the longitudinal oscillations are extinguished, reducing the dynamic axial loads on thrust-radial bearings 5 and dynamically unloading them. The flow of washing liquid under pressure enters the hydrodynamic generator 16 to the fairing 17 and flows around its surface through the annular space and is directed into the toroidal resonance chamber 19 with rod 18 and plate 20 resonators. From the hydrodynamic generator 13, the washing liquid through the channels in the ball bit 21 is fed to the punch with extraction of acoustic energy. At the same time, energy waves are directed perpendicular to the resonator plane.

It should be noted that all the above structures are equipped with pulse generators of the flushing solution flow at medium frequencies (Fig. 8) [16].

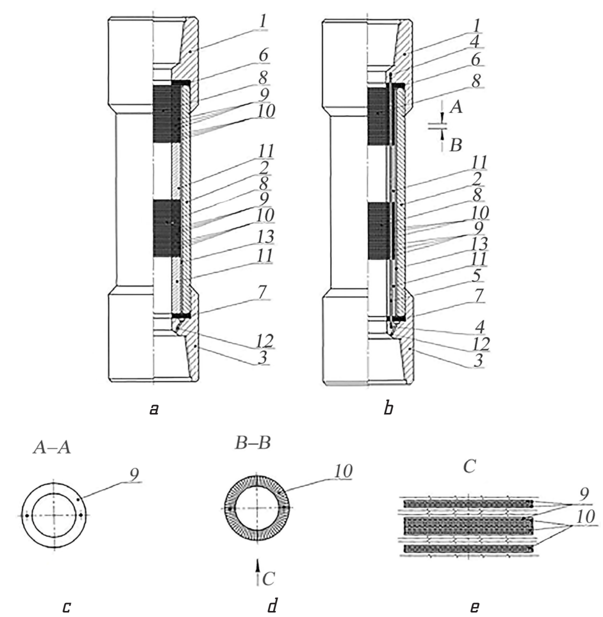

Fig. 8. Hydrodynamic generator of fluctuations in the flow of the washing solution: *a* – general view of the hydrodynamic oscillation generator with a longitudinal section; *b* – variant of the general type of hydrodynamic oscillation generator; *c* – section A–A; *d* – section B–B; *e* – view C of section B–B; 1 – upper coupling adapter; 2 – body; 3 – lower coupling adapter; 4 – holes; 5 – smooth pins; 6 – upper cuff; 7 – lower cuff; 8 – permeable cylinders; 9 – smooth rings; 10 – rings corrugated on the radius; 11 – non-permeable (solid) cylinders; 12 – longitudinal channels; 13 – annular space

The hydrodynamic generator of fluctuations in the flow of the washing solution (Fig. 8) works as follows. In order to carry out the technological process of impacting the near-hole zone of the well, the device is connected to the column of pump-compressor pipes (tubes) through the adapter-coupling 1 and lowered into the well hole. During the technological process, the working solution passes through the permeable cylinders 8, which create hydraulic resistance, which is caused by the combination of smooth rings 9 and radially corrugated rings 10, which form microholes and have a hydraulic connection with the annular space 13 and cause the occurrence of radial component of the flow of the working solution. During the transition of the working solution from the permeable cylinder 8 with significant hydraulic resistance to the impermeable cylinder 11 with a smooth surface, that is, with a small hydraulic resistance, the flow of the working solution undergoes the effect of hydrodynamic resistance. This causes the emergence of a pulsating pulse, the effect of which is transmitted to the near-outbreak zone of the productive layer, improving its collector properties. The degree of compression of the upper cuff 6 and the lower cuff 7 is regulated by the tightening force of the threaded connections of the adapters-couplings 1, 3 with the body 2, which makes it possible to adjust the permeability of the permeable cylinders 8, thereby affecting the change in the hydraulic resistance of the flow of the working solution, which in turn allows to change the frequency of its pulsation.

The effect of cavitation, which is used in the proposed devices, makes it possible to loosen the rock of the excavation, crushes the rock in the working area of the bit and thus facilitates its operation. It makes it possible to increase the stability of the tool and facilitates its operation, and, as a result, increases the stability of the tool and the speed of drilling the well.

Fig. 9 shows the results of statistical processing of the recording of high-frequency vibrations of the top of the drill string when drilling with the А7ПД turbo drill.

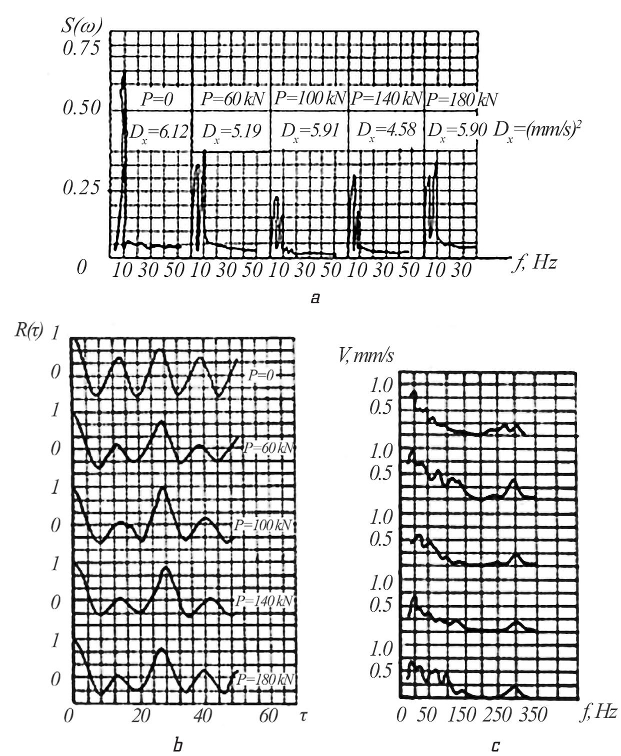

Fig. 9. The results of statistical processing of the recording of high-frequency vibrations of the top of the drill string: *a* – implementation; *b* – normalized spectral density; *c* – normalized autocorrelation function

The cavitation hydraulic vibrator is installed in the drill pipe column in front of the rock-breaking tool or above the column pipe. The analysis of pressure and vibration speed oscillograms during operation of the high-frequency oscillation generator showed that, as a rule, they have the form of narrow-band random vibrations that arise as a response to a wide disturbance with low damping. The average value of the narrow-band vibration frequency can be determined by S. O. Rice formula [17]. Based on the calculated mathematical expectation ω, it is possible to determine the envelope of the narrow-band vibration process of the drilling tool.

A narrow-band random process of vibrations has the appearance of a beat, which is confirmed experimentally.

From the above, it can be concluded that the load on the rock-destructive tool, as a rule, is not periodic and has a random nature of the process. Using the results obtained during the study of the vibration of the tool under the influence of such loads, in order to determine the stresses in the elements of the drill string, it is necessary to use the mathematical apparatus of the theory of random oscillations [18–20].

Hydrodynamic cavitation is characterized by the number of cavitation, which serves as one of the similarity criteria that models the hydrodynamic flow:

$$\chi = 2\left(p_0 + p_H\right) / \rho \upsilon_\infty^2, \tag{6}$$

where p and v – the pressure and velocity of the incoming flushing liquid flow.

The practical significance lies in the fact that the authors proposed a number of constructive solutions for the intensification of the well drilling process using hydrodynamic cavitation. These include the drilling projectile with a high-frequency cavitation vibrator, the drilling projectile with a cavitation chamber, the spindle of the drilling motor with cavitation and screw amplifiers of the axial load on the bit, and the hydrodynamic generator of oscillations of the flushing solution flow described in the work. It should be noted that the above structures are equipped with the latest design solution.

There are no special restrictions for the implementation of research results in order to implement the obtained results in practice. All of the above technical solutions and designs fit organically into the existing drilling equipment and allow for the intensification of technological processes with lower specific energy consumption.

In the future, it is necessary to consider in more detail all oscillatory processes that occur in the drill string and are of a random nature, to consider using the mathematical apparatus of the theory of random oscillations.

In the future, it is necessary to take into account the hydrodynamics and type, as well as the design and parameters of the applied drilling elements for the development of their dynamic models.

4. Conclusions

It is shown that the effectiveness of the use of hydrodynamic cavitation requires the development of methods and devices for intensifying the well drilling process. It has been proven that the design of the cavitation generator organically fits into the existing well drilling equipment and allows for the intensification of technological processes with lower specific energy consumption. It is found that all oscillatory processes that occur in the drill string are random in nature and must be considered using the mathematical apparatus of the theory of random oscillations.

The study of vibrations during well drilling shows that vibrations can be considered as random stationary processes, since transient modes have a sufficiently short duration for homogeneous rocks with fixed drilling modes. Analysis of the vibrations of the drill string elements on the basis of the random fluctuations in a number of cases allows to increase the reliability of determining the vibrational reliability of the drill string elements. It is proven that the response of the drill string elements to broadband random vibration can be determined as the total effect of several narrow-band random vibrations.

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Conflict of interest

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The paper has no associated data.

Use of artificial intelligence

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

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