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## Azimuth correction system of an inclinometer

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**Abstract.** The paper is devoted to azimuth correction system of an inclinometer used for well drilling in high latitude areas. In these areas, ordinary types of navigation equipment (gyroscopic and magnetic sensors) are ineffective because of small horizontal component of the Earth rotation velocity and significant values of magnetic declination. In the paper, an approach to the correction system study is described. The system consisting of two vibrational sources shifted from the wellhead to a certain distance in the direction of projected drilling and spaced to same distance from the specified direction is characterized.

Special attention is given to investigation of a motion model of a flexible shaft, which is proposed as the vibrational source of the system. On the basis of experiments and calculations, it is shown in the paper that in a complex mechanical system with a flexible shaft of vibrational source, a consistent multi-resonant effect can be observed when the shaft starts rotating.

Moreover, a simulation model and results of estimation of the vibrational source power are presented. A possibility of use a compact “handheld” vibrational source with flexible shaft for azimuth correction system operation is shown

### Introduction

In the modern world, there is a systematic significant increase in hydrocarbons consumption. To meet these needs, oil and gas production in existing fields is correspondingly growing, and new ones are being developed. One of the promising areas for exploration and production of oil and gas is the Arctic region and the shelf of northern seas. However, it should be noted that drilling operations in these areas are carried out in extremely heavy conditions: climatic for specialists and operational for equipment. An example of difficulties with drilling in the Arctic region and in high latitude areas is challenging accurate autonomous reckoning of moving objects location under water and under ground. During the drilling process, it is necessary to precisely determine the position of a drill bit located on the bottom hole assembly (BHA) of the drill string. Besides the drill bit and the downhole motor section, the BHA also includes an inclinometer. It calculates current kinematic parameters and the actual location of the BHA in the earth crust, and transfers the information up to the wellhead to the drilling operators. The inclinometer employs a triad of pendulum compensation accelerometers to determine a zenith angle and an angle of a tool face. These devices have good resolution and high resistance to shocks and vibrations that occur during drilling. In addition to the triad, several different types of azimuthal sensors can be used to determine the azimuth angle of the BHA. Depending on the sensor types, the inclinometers are divided into gyroscopic or ferromagnetic ones. In the former group of inclinometers, gyroscopic angular velocity sensors are used for calculating the azimuth angle, while the latter group uses magnetic sensors. These types of sensors also have good resolution and resistance



to significant external loads. However, in high latitude areas, these sensors are ineffective because of inefficiency of their measurement principles. The gyroscopic sensors measure the horizontal component of the Earth rotation velocity to determine the azimuthal angle. However, when approaching the geographic poles (a latitude above  $70^\circ$ ), the horizontal component of the Earth rotation velocity decreases significantly. As a result, in order to maintain high accuracy of measurements, it is necessary to use gyroscopic sensors with significantly lower inherent instrumental errors. At the same time, the existing overall limitations do not allow calibration of such devices. The specified requirement leads to significant increase in the cost of devices. Magnetic azimuth sensors also become ineffective in high latitude areas because of high values of magnetic declination and mismatch of the geographical and magnetic poles of the Earth. Therefore, standard types of gyroscopic and magnetic inclinometers become either very ineffective or completely useless for determining the precise actual location of BHA in high latitude areas. Based on this, there is an increased interest in alternative methods of navigation when drilling wells.

One of the alternative methods for azimuthal navigation while drilling in high latitude areas is a method of azimuthal correction of inclinometer. This method was implemented in the azimuth correction system of the inclinometer.

### 1. Azimuth correction system of the inclinometer

The method of azimuthal correction of the inclinometer is based on seismic signals propagation through the earth crust and measurement of parameters of these signals. To create a seismic signal, different types of seismic sources can be used. Depending on the source type, the nature and forms of the seismic signals are different. For example, explosive, hydrostatic, electromagnetic, or vibrational sources can be used. Due to the compact size and significant seismic power, it is proposed to create the seismic disturbance of the earth crust with a vibrational source. Vibrational disturbance (seismic signal) from the vibrational source propagates through the earth crust and is measured in a seismic receiver. On a basis of signal processing, mutual azimuthal position of vibrational sources and seismic sensor is calculated [1]. To implement the method of azimuthal correction, it is proposed to use a scheme where the vibrational sources of seismic disturbance are located at points with known distances from the wellhead in the direction to projected drilling. To measure small responses of the earth crust, it is proposed to use the accelerometers of the inclinometer as the seismic sources.

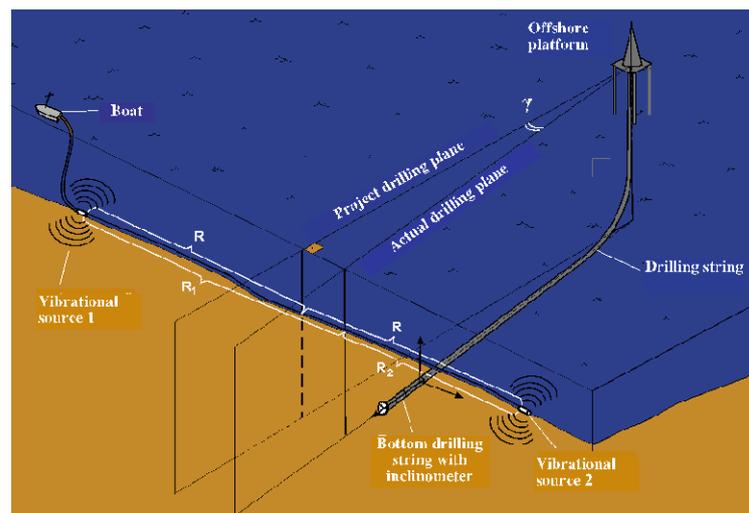


Fig. 1. The scheme of the azimuth correction system of the inclinometer

In Fig. 1, the diagram of the azimuth correction system of the inclinometer is shown. The figure shows the layout of an offshore drilling platform with a drilling string with BHA. The offshore platform can be used for cluster drilling, but for easier understanding the figure shows only one drilling string. BHA of a drilling string contains the inclinometer and can be placed at significant

distances from the platform. The drilling string forms a vertical actual drilling plane. Besides the actual drilling plane, there is a projected drilling plane which coincides with the vertical plane with proper drilling direction. Deviation of the actual drilling plane with BHA from the projected drilling plane illustrates incorrect drilling direction. Moreover, in Fig. 1 there are two vibrational sources 1 and 2 submersed in the sea bottom and distanced from the platform and the wellhead to a known distance in the same direction. Two vibrational sources are mounted into a plastic pipe to provide easy submersing and re-submersing in the bottom. The plastic pipe with the vibrational sources is laid down perpendicularly to the projected drilling plane, and the vibrational sources are placed at equal distances  $R$  in opposite directions from this plane. The distance between two sources is much less than the distance to the estimated position of BHA with inclinometer (for example, 100 m and 1000-1500 m, respectively). Both sources start to generate seismic disturbances which propagate through the earth crust and reach the BHA inclinometer. In case of proper drilling direction, the actual drilling plane coincides with the projected plane, and the distances from each seismic source to the inclinometer are equal (in particular,  $R$  and  $R$ ). If there is an azimuthal deviation of the actual drilling plane from the design plane, these distances will differ from each other (for example,  $R_1$  and  $R_2$ ). By defining the distances from the vibrational sources to the inclinometer, a drill operator can determine the azimuthal deviation angle  $\Delta\gamma$  and correct the drilling direction.

## 2. Seismic sources in the correction system

For choosing a seismic source type, a calculation method for the azimuthal deviation should be presented. There are two methods for determining the azimuthal deviation of the actual drilling plane from the projected one. First of all, the distances  $R_1$  and  $R_2$  from the seismic sources to the BHA can be calculated on the basis of time delays during signal propagation. However, the use of time delays in determining the distances between the seismic sources and the receiver seems to be not efficient because of the necessity to use high-resolution seismic signal and extremely low errors of accelerometers of the inclinometer in short-time measurements. To provide an easier approach to the azimuth correction system, another method to calculate the azimuthal deviation is considered. In this method, the focus is made on signal attenuation during its propagation through the earth crust, and on the measurements of seismic spectra of the signal. In the embodiment of the azimuth correction system, accelerometers are placed on the sea bottom where only small seismic noises occur when the drilling is stopped. Precise accelerometers with high average resolution are used in the inclinometers (Q-flex or Si-flex compensating type) and are capable to distinguish a small seismic signal [2, 3, 4]. Due to the possibility to take measurements for a long (40-60 s) time, there is no need to rely on extremely small-time delays and high-resolution measurements of seismic signals. The seismic spectra are accumulated in the accelerometers of the inclinometer for a stop-drilling time. Therefore, it possible to use low-power vibrational sources in the correction system. The seismic signal is formed as a continuous harmonic vibrational radiation created by a flexible rotating shaft. The design of the vibrational source is shown in Fig. 2 [5].

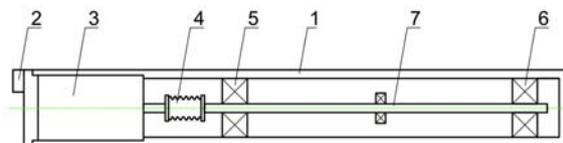


Fig. 2. 1 – vibrational radiator case; 2 – vibrational sensor; 3 – DC motor with control system; 4 – bellows coupling; 5, 6 – spherical bearings; 7 – flexible shaft

When power is applied to the motor, the shaft is driven into rotation. The shaft has a small initial imbalance, and oscillations appear when the angular velocity gets stable. The amplitude of oscillations sharply increases as the frequency approaches the resonance. The frequency of the first form of the shaft bending vibration should be chosen sufficiently low (15-20 Hz), in order to ensure small attenuation of the seismic wave for its propagation through the crust. It is proposed to use two simultaneously operating sources with the same intensity of seismic radiation and different resonance frequencies of the flexible shaft oscillations. The duration of vibrations and the amplitudes of oscillations are the parameters that are strictly controlled by sensors on the cases of vibrational sources.

The sources operate in a mode with continuous vibration for a time period up to 50 seconds. During this time, a data array for measured accelerations is formed in the memory of the inclinometer, and the spectra of received signals are calculated. In these spectra, the effect of accelerometer noise drastically decreases as the measurement time increases. The amplitude of spectrum for each vibrational source is proportional to the time of measurements, the distance between the vibrational source and the inclinometer, and the earth crust physical properties. It is assumed in the method that the physical properties of the earth crust are similar for both signals propagation due to the mutual position of the sources. Thus, the deviation of the drilling assembly's actual position from the projected position is determined by attenuation of seismic signals on their way from the vibrational sources to the inclinometer within equal measurement times and under equal vibration amplitudes of both sources.

### 3. Flexible shaft motion model

The model of the flexible shaft motion was developed to understand the physical processes in the shaft during its rotation. As was mentioned above, the azimuth correction system employs two vibrational sources operating simultaneously. The rotation frequency of the shaft is close to resonant one to get the maximum amplitude of oscillations. However, fine tuning to resonant frequency is challenging because of fast transition of frequency from before-resonant to over-resonant region. This transition will be defined by the quality factor of the mechanical system (combination of mass, rigidity, mechanical and dynamic friction parameters and their nonlinearity), as well as applied control and other external actions. In this case, to provide sustainable rotation of the shaft, it is proposed to use a close pre-resonant area with an amplitude equal to  $0.7...0.8 A_{max}$ , where  $A_{max}$  is the maximum resonant amplitude of oscillations. In real mechanical system of a vibrational source, fine tuning to the shaft oscillations amplitude of  $0.7...0.8 A_{max}$  is challenging because the shaft rotation is affected by the driving motor. Thus, simulation of operation of a source with flexible shaft was performed to determine a control mode of the vibrational source motor.

Computational model of the flexible shaft is shown in Fig. 3. In the initial variant for calculations, the shaft is represented as a point mass  $m_s$  displaced relatively to the rotation axis  $Az$  to a distance  $AC$  (eccentricity  $e_s$ ). The rotation axis  $Az$  is perpendicular to the figure plane and is not shown.

The point A is fixed elastically, with shaft rigidity  $K$  (directional –  $K_x$  and  $K_y$ ), to the zero-point O and might slide along the weightless guide OA. This motion imitates the shaft bending. The torque  $M_{motor}$  and the rotation velocity  $\omega$  about the z-axis are transmitted to the shaft from the motor through elastic safety coupling (Fig. 2) at the point A. Bending stiffness of the coupling is neglected in these calculations. It is assumed that bearings at the point A are absolutely rigid and allow only rotation about the axis passing through the point A perpendicularly to the figure plane. Angular rigidity  $K_\phi$  of the shaft is neglected.

Similar models and computations are presented in [6] and [7]. However, these models do not take into account the dynamic characteristics of the system, namely, the shaft oscillation damping and dependency of DC motor torque on angular velocity. This paper describes the motion of a flexible shaft considering both the damping ratio and the motor torque variability. Equations were written on the basis of the kinetic moment variation theorem for rotation about  $Az$  axis, and D'Alembert principle for linear motions along  $Ox$  and  $Oy$ . The equations have the form:

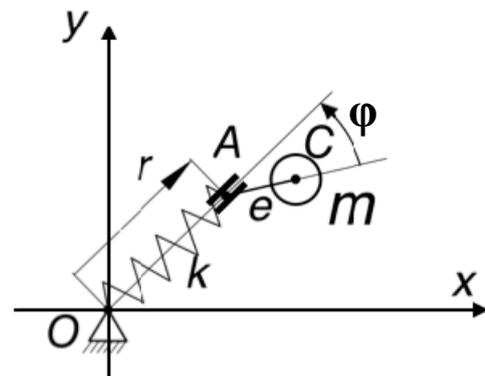


Fig. 3. Initial simulation model of flexible shaft

$$\begin{cases} m_s \cdot \ddot{x} + D_x \cdot \dot{x} + K_x \cdot x = m_s \cdot e_s \cdot \dot{\varphi}^2 \cdot \cos(\varphi); \\ m_s \cdot \ddot{y} + D_y \cdot \dot{y} + K_y \cdot y = m_s \cdot e_s \cdot \dot{\varphi}^2 \cdot \sin(\varphi) - m_s \cdot g; \\ I_z \cdot \ddot{\varphi} + D_\varphi \cdot \dot{\varphi} = M_{motor} - m_s \cdot g \cdot e_s \cdot \cos(\varphi) - m_s \cdot e_s \cdot \dot{x} \cdot \dot{\varphi} \cdot \cos(\varphi) - m_s \cdot e_s \cdot \dot{y} \cdot \dot{\varphi} \cdot \sin(\varphi); \\ M_{motor} = M_0 \cdot (k_1 + k_2 \cdot (1 - e^{-0.01t})) - D_{motor} \cdot \dot{\varphi}; \end{cases}$$

where  $x, y$  are the displacements of the shaft center (p. A) along Ox and Oy;  $\varphi$  is the rotation angle about Oz;  $\ddot{x}, \dot{x}, \ddot{y}, \dot{y}$  are linear accelerations and velocities of the shaft center along the axes Ox and Oy, respectively;  $\ddot{\varphi}, \dot{\varphi} = \omega$  are the angular acceleration and angular velocity of the shaft about Oz;  $m_s$  is the mass of the shaft;  $I_z = \frac{m_s r_s^2}{2} + m_s e_s^2$  is the moment of inertia of the shaft about Oz;  $r_s$  is the radius of the shaft;  $r = \sqrt{x^2 + y^2}$  is the shaft total displacement;  $K_x, K_y$  is linear rigidity of the shaft along Ox and Oy;  $K_\varphi$  is angular rigidity of the shaft;  $D_x, D_y$  is linear damping along Ox and Oy;  $D_\varphi = D_{motor} + 0.01 \cdot (x^2 + y^2)$  is damping in rotation;  $e_s$  is the shaft eccentricity;  $M_0$  is the initial torque of the motor;  $M_{motor}$  is the actual torque in accordance with  $M(\omega)$  characteristic;  $k_1$  and  $k_2$  are the coefficients of the motor torque;  $D_{motor}$  is the initial damping of motor;  $g = 9.81 \text{ m/s}^2$  is gravity acceleration;  $t$  is time; and  $e = 2.71828$ .

The values of the parameters are as follows:  $m_s = 0.8 \text{ kg}$ ;  $r_s = 0.005 \text{ m}$ ;  $e_s = 0.001 \text{ m}$ ;  $M_0 = 310 \text{ Nsm}$ ;  $D_{motor} = 0.001 \text{ Nm/(rad/s)}$ ;  $K_x = K_y = 9600 \text{ N/m}$ ;  $k_1 = 0.1$ ;  $k_2 = 0.9$ .

The initial conditions are as follows:  $x = \dot{x} = y = \dot{y} = 0$ ;  $\varphi = \dot{\varphi} = 0$ .

The equations are solved by numerical integration of a set of nine ordinary differential equations of the first order by means of the function ode45 in Matlab software package. Graphical representations of the solution ( $\omega, x_s, y_s$ ) to the set of differential equations are shown in Fig.4.

Figure 4a shows the dependency of the angular velocity  $\omega, \text{ rad/s}$ , of the shaft on the time, s. Figs. 4b and 4c show the displacements  $x_s, \text{ m}$ , and  $y_s, \text{ m}$ , of the elastically fixed point A from the point O along the axes X and Y, respectively, on the time, s.

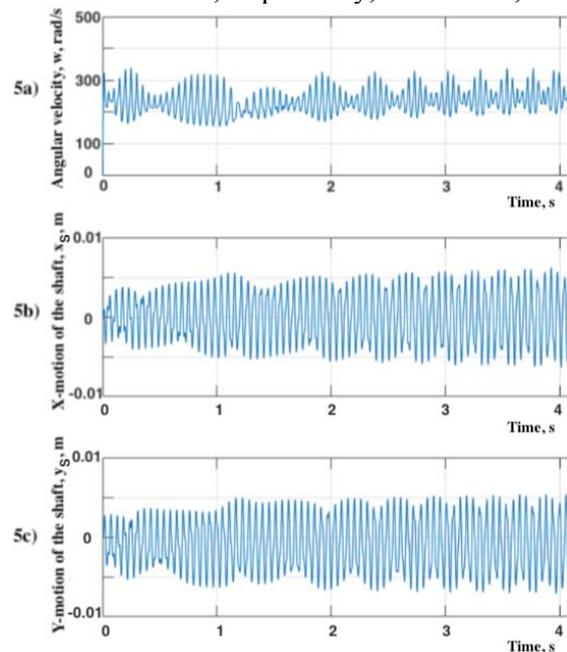


Fig. 4 Angular velocity, x- and y- displacement of the shaft in rapid acceleration mode.

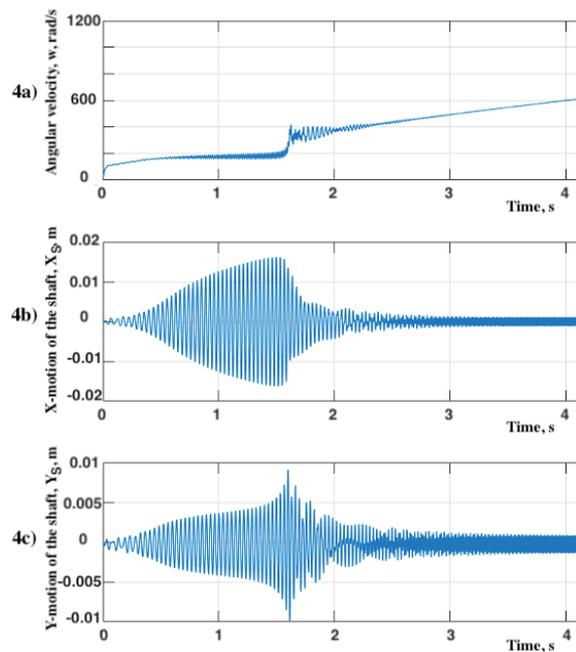


Fig. 5. Angular velocity, x- and y- displacement of the shaft in slow acceleration mode and with unequal stiffnesses  $K_x$  and  $K_y$  of the shaft.

A distinctive part of the calculations is separation of the shaft rotation control options. There are two opposite variants: “slow” and “rapid” accelerations of the shaft. The slow acceleration mode provides gradual (slow) increase of the motor torque  $M_0$ . On the other hand, the rapid mode corresponds to the transmission of the nominal full torque from the motor to the shaft. Separation of these two modes is provided by varying the  $k_1$  and  $k_2$  coefficients in the motor torque equation. The analysis of processes during starting rotation of electric machines shows [8] that in the slow mode of acceleration, the peak value of rotation resistance torque appears near the resonance frequency of rotation. It can be explained by the occurrence of the maximum amplitude of the shaft oscillations according to the amplitude-frequency characteristic. For comparison of two modes, the results of the shaft rapid acceleration modeling are shown in Fig. 5.

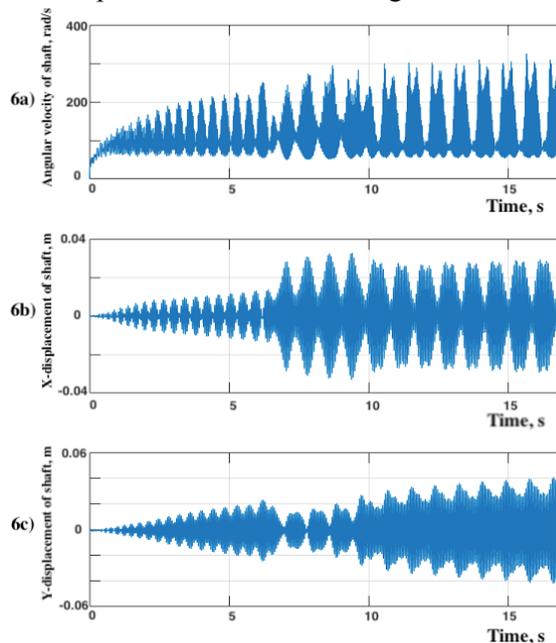


Fig. 6. Angular velocity, x- and y- displacement of the shaft under slow acceleration regime

Figure 5a shows that the angular velocity  $\omega$ , rad/s, instantly increases up to close maximum at rapid acceleration with the torque  $M_0$  value; however, the amplitudes of oscillations along the  $Ox$  and  $Oy$  axes are several times lower than in the slow acceleration mode. Therefore, the rapid acceleration mode is not quite appropriate for the vibrational source of the system. This mode does not provide the maximum amplitude of the shaft oscillations, and significant vibrational disturbance of the earth crust cannot be achieved.

Also, it should be highlighted that the coefficients of the motor torque do not change significantly; new  $k_1 = 0.4$  and  $k_2 = 0.6$ . The ratio of  $k_1$  and  $k_2$  indicates that the motor torque instantly applied onto the shaft is not maximal (but only 0.4 units of the total value  $M_0$ ). This starting value of  $0.4M_0$  in a real mechanical system can be achieved even at a large starting torque of the motor, without any additional control. This is inappropriate for vibrational source control and should be suppressed by matching the inertia parameters of the shaft and the starting torque  $M_0$  of the rotor.

In addition to comparison of slow and rapid modes of the shaft acceleration, during calculations attention was given to the equality of shaft stiffnesses along the axes  $Ox$  and  $Oy$ . Calculations revealed that a small difference in stiffness values causes the multi-resonant effect on oscillations.

Multi-resonance occurring in the slow acceleration mode significantly changes motion parameters of the shaft (Fig. 6). Oscillation forms become similar to the forms under Sommerfeld effect [8]. For example, a model for the results in Fig.6 has the same motor torque coefficients  $k_1$  and  $k_2$  as for slow acceleration mode, and there is a difference in the shaft linear stiffness:  $K_x = 9600$  N/m,  $K_y = 10600$  N/m. As a result, the form and parameters of oscillations change significantly. The amplitude of oscillations is several times greater than the amplitude in the slow acceleration mode, but in steady state, undamping oscillations of the values  $\omega$ ,  $x_s$ ,  $y_s$  appear. These oscillations and this multi-resonant mode are appropriate for the vibrational source of the azimuth correction system; however, the shaft motion becomes hardly predictable because of uncertainty of Zommerfeld effect damping and possible inequality of  $K_x$  and  $K_y$ .

## Conclusions

In the paper, the azimuth correction system has been described. It has been shown that azimuthal angle of BHA and azimuthal deviation of the assembly from the projected drilling direction can be determined by measuring the seismic signals propagated through the earth crust from vibrational

seismic sources to accelerometers of the BHA inclinometer. Two methods of azimuthal position computation have been presented, and the method with the earth crust seismic spectra measurements has been verified as primary. In connection with the primary method for determining the azimuthal deviation, design of the vibrational source and its computation model have been considered. It has been suggested to use a flexible rotating shaft as a vibration source. The computation model created to control the shaft rotation showed that there were different modes of the shaft rotation. Comparison of two variants of motion control demonstrated that the mode with slow acceleration of the shaft is preferable for the vibrational source operation. This mode provides the maximum amplitude of the shaft oscillations and creates a significant vibrational disturbance of the earth crust. However, this mode requires accurate adjustment and control of the motor and the mechanical system of the vibrational source due to the significant impact of non-linearity of the mechanical system on damping, rigidity and motor torque ratio. Small differences in the model coefficients and significant changes in rotation indicate a complex dynamic system with distributed inertial and rigidity parameters. Each parameter should be studied separately, since the mass-inertial characteristics of the flexible shaft have a complicated dependency on the environment and change in time.

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