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METHODOLOGY FOR ANALYZING FORCED VIBRATIONS OF A MULTI-FREQUENCY VIBRATING SIEVE FOR CLEANING DRILLING MUD USING THE METHOD OF COMPUTATIONAL EXPERIMENT

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Abstract. To identify and study the characteristics of vibrations of a multi-frequency vibrating sieve for cleaning drilling muds, as a significantly nonlinear dynamic system with a limited source of excitation, it is proposed to use the method of computational experiment. At the first stage, the equations of motion of the dynamic system are numerically integrated using a computational algorithm based on the use of three-layer difference schemes with weights. At the second stage, the obtained time series are studied using autocorrelation functions to determine the period of forced vibrations; spectral analysis of displacements, velocities and accelerations; phase diagrams in the variables "displacement-velocity"; dependences of the amplitudes of displacements, velocities and accelerations on changes in the parameters of the dynamic system, obtained by the method of continuation by parameter. The developed method of spectral analysis of forced vibrations of a multi-frequency vibrating sieve for cleaning drilling muds is based on the assumption that its motions are periodic. If deterministic chaos occurs in the dynamic system under consideration, then the autocorrelation function of the time series of displacements must have a finite carrier, i.e., it must be converted to zero outside a finite time interval. To analyze the process of stabilization of a multi-frequency vibrating sieve's motion and visually detecting attractors, phase diagrams in the variables "displacement-velocity" are used. When studying the dependences of the amplitudes of displacements, velocities and accelerations of a dynamic system on changes in its parameters, the method of continuation by parameter with step-by-step changes in the system parameters was used. The solution obtained in the previous step is chosen as the initial approximation of the solution. A study of a vibroimpact oscillator of a multi-frequency vibrating sieve for cleaning drilling muds was carried out in accordance with the proposed model and methods of analysis of forced vibrations. The vibroimpact modes of oscillations of the oscillator at frequencies exceeding the natural frequency of the generating linear system, at which accelerations are excited in the system, which significantly exceed the acceleration in the linear system, are investigated. Such modes are realized in the resonant area of oscillations of the generating linear system, in the presence of gaps between the mass and one-sided limiters of large amplitudes of oscillations of the linear system in the stationary state of the vibroimpact system.

Keywords: vibrating multi-frequency sieve, drilling muds, cleaning, significantly nonlinear dynamic system, analysis methodology, computational experiment method , vibroimpact oscillation modes.

1. Introduction

Improving the technique and technology for cleaning drilling muds from drilled rock, increasing the speed of well drilling, and improving the quality of drilling mud is an important scientific and applied problem that is of great importance for the oil and gas industry [1-8].

The authors propose the use of drilling mud cleaning technology on a vibrating multi-frequency sieve. The implementation of multi-frequency oscillations and a multiple increase in the acceleration of sieve oscillations, compared to typical vibrating sieves for cleaning drilling muds, will ensure increased productivity and efficiency of drilling mud cleaning on vibrating multi-frequency sieves compared to traditional vibrating sieves with single-frequency excitation [9, 10]. This will allow increasing the permissible drilling speed, which is limited by the degree of cleaning of drilling muds from rock particles, and will contribute to increasing the technical and economic indicators of the drilling process.

A review of modern works in the field of application of vibrating sieves and vibrating impact systems in the processes of fine classification and dehydration of mineral raw materials, including cleaning of drilling muds, has been conducted. The main

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directions of increasing the efficiency of vibration technologies for processing mineral raw materials are the selection and optimization of dynamic parameters of the load of vibration equipment and technological raw materials on the working body, related both to energy-force interactions and to mechanical and structural characteristics of the raw materials. For polydisperse systems, which are the majority of technological materials in the processing of mineral raw materials, the optimal modes of vibration influence lie in the area of resonant multi-frequency excitation and require deep and independent regulation of the parameters of vibration excitation taking into account changes in the values and characteristics of the technological load [11-23]. To date, mathematical models and calculation methods for dynamic, substantially nonlinear vibration systems for fine cleaning of drilling muds, which implement a wide range of frequencies on the working body, have not been developed, and on their basis, the parameters have not been substantiated and industrial samples of such systems have not been developed.

Therefore, the development of a methodology for analyzing forced vibrations of a vibrating multi-frequency sieve for cleaning drilling muds using a computational experiment method for further research into the process and establishing regularities of changes in the power, energy, and operating parameters of a vibrating multi-frequency sieve and substantiating its rational parameters is an urgent scientific task that is of significant importance for the country's oil and gas production industry.

2. Methods

It is difficult to obtain an analytical solution to the Cauchy problem equations describing the motion of a vibrating multi-frequency sieve as a dynamic system with a limited excitation source due to its significant nonlinearity [24]. Therefore, the method of computational experiment is used for identifying and studying the characteristics of oscillations of a vibrating multi-frequency sieve for cleaning drilling muds.

The algorithms and methods for analyzing forced vibrations of multi-frequency vibrating screens using the computational experiment method were developed by the authors [25-27], which were used for studying and substantiating the parameters of such screens [28, 29], vibrating feeders with adaptive drive [10, 30] and mills with vibroimpact excitation for fine and ultrafine grinding of mineral raw materials [31, 32]. In [33], the developed methods and algorithms were adapted to identify and study the characteristics of frictional vibrations in the brake. In this article, the algorithms and methods developed in [25-27] are adapted for the analysis of forced vibrations of a vibrating multi-frequency sieve for cleaning drilling muds.

At the first stage of the computational experiment, the equations of motion of the dynamic system under consideration are numerically integrated using a computational algorithm based on the use of three-layer difference schemes with weights. As a result, time series of displacements of its mass are calculated $\left\{X^k\right\}$ [24].

At the second stage of the computational experiment, the obtained time series are studied using:

- autocorrelation functions to determine the period of forced vibrations;

- spectral analysis of displacements, velocities and accelerations;
- phase diagrams in the variables "displacement-velocity";
- dependences of the amplitudes of displacements, velocities and accelerations on changes in the parameters of the dynamic system under consideration, obtained by the method of continuation by parameter.

3. Theoretical part

3.1. Application of autocorrelation functions to determine the period of forced vibrations.

The dynamic system of a vibrating multi-frequency sieve for cleaning drilling muds is dissipative, since it contains elastic damping elements. Therefore, if the excitation force is periodic, then over time the system's motion stabilizes and also becomes periodic. In this case, in vibroimpact systems, the period of motion can be not only equal to, but also a multiple of the period of change of the vibration force [34]. The period T of rotation of debalances is equal to $2\pi / \omega$, therefore, the period T_1 of the analyzed dynamic system's motion can be equal to $2\pi n / \omega$, $n = 1, 2, \ldots$.

Therefore, the task of studying the operating modes of a vibrating multi-frequency sieve is to find a solution to the nonlinear system of equations of motion [24] that satisfies the periodicity conditions

$$x_n(t) = x_n(t + T_1),\ \dot{x}_n(t) = \dot{x}_n(t + T_1), \tag{1}$$

while the period T_1 of the motion of the dynamic system under consideration is unknown in advance.

Therefore, the task arises of determining the oscillation period of the analyzed dynamic system taking into account the analysis of the periodicity of the time series of its mass displacements $\{X^n\}$. For this purpose, the apparatus of autocorrelation functions is used.

Let the values of a discrete signal (time series $\{x_n\}$) be known, $n = \overline{1, N + M}$, then the discrete autocorrelation function of the signal $\{x_n\}$ is calculated by the formula

$$\psi_m = \frac{1}{N} \sum_{n=1}^{N} x_n \cdot x_{n+m},\ m = \overline{0, M}, \tag{2}$$

where ψ_m, $m = \overline{0, M}$ is the discrete autocorrelation function; N, M are positive integers.

The autocorrelation function serves as a measure of the degree of similarity of a signal with itself in the past. If the time series $\{x_n\}$ is periodic with period K, then its autocorrelation function also has periodicity

$$\psi_m = \psi_{m+K},\ m = \overline{0, M}, \tag{3}$$

while the inequality holds

$$\psi_0 > \psi_m,\ 0 < m < K. \tag{4}$$

In practical calculations, it is convenient to use the scaled autocorrelation function

$$\tilde{\psi}_m = \psi_m / \psi_0,\ m = \overline{0, M}. \tag{5}$$

It is obvious that

$$\tilde{\psi}_0 = 1. \tag{6}$$

Given that the time series of displacements $\{X^n\}$ is an approximate solution to the Cauchy problem of equations describing the motion of vibration multi-frequency sieve as dynamic systems with limited source excitation, the periodicity conditions (2) are approximately fulfilled even for the steady-state motions of the dynamic system. Therefore, for the autocorrelation function, the periodicity condition (3) will also be approximately fulfilled. When using a scaled autocorrelation function, we will assume that its period is equal to K, if

$$\tilde{\psi}_K > 1 - \varepsilon; \tag{7}$$

$$\tilde{\psi}_m < 1 - \varepsilon,\ 0 < m < K, \tag{8}$$

where $\varepsilon > 0$ is a parameter that characterizes the accuracy of fulfilling the periodicity conditions.

On the basis of analysis of the results of numerous computational experiments, parameter ε in practical calculations should be chosen in the range of 0.01-0.1. It should be noted that with increasing level of the dynamic system damping, the value of the parameter ε can be reduced.

If the time series $\{x_n\}$ is aperiodic, its autocorrelation function must have a finite carrier, i.e. it must be zero outside a finite time interval. For finite time intervals of a time series, the aperiodicity criterion can be formulated as follows: for each $\varepsilon > 0$ exists $M(\varepsilon)$ such that

$$|\psi_m| \le \varepsilon \text{ for any } m > M(\varepsilon). \tag{9}$$

Thus, calculating the autocorrelation function for a given time series allows not only to establish whether it is periodic, but also to determine its period in this case. This is important for the study of forced vibrations of vibroimpact systems, because the occurrence of subharmonic oscillations is possible [34].

3.2. Methodology for conducting spectral analysis.

One of the most common methods for studying periodic motions of dynamical systems is spectral analysis. From a mechanical point of view, the decomposition of the studied motion into a Fourier series corresponds to its representation as a set of simple harmonic motions.

Fourier series expansion is applied to both continuous functions and discrete sequences, where they are represented as sums of harmonic functions or complex exponents with frequencies forming an arithmetic progression [35].

Let the time series $\{x_n\}$ is periodic with period K, that is

$$x_{n+K} = x_n \text{ for any } n. \tag{10}$$

Such a time series is completely described by a finite set of numbers, as which we can take an arbitrary fragment of length K, for example $\{x_n\}$, $n = \overline{0, K-1}$. It is known [35] that a real periodic discrete signal (time series) has a periodic discrete spectrum $\{X_n\}$, which has the symmetry property

$$X_{n+K} = X_n \text{ for any } n. \tag{11}$$

$$X_{K-n} = X_n,\ 0 < n < K. \tag{12}$$

In this case, the time series $\{x_n\}$ can be represented as a finite Fourier series in trigonometric form

$$x_n = \sum_{k=0}^{K/2} A_k \cos\frac{2\pi kn}{K} + \sum_{k=0}^{K/2} B_k \sin\frac{2\pi kn}{K}, \tag{13}$$

where A_k, B_k are the coefficients of the Fourier series calculated by the formulas

$$A_k = \frac{2}{N}\sum_{n=0}^{K-1} x_n \cos\frac{2\pi kn}{K},\ k = 1, \ldots, \frac{K}{2} - 1; \tag{14}$$

$$A_k = \frac{1}{N}\sum_{n=0}^{K-1} x_n \cos\frac{2\pi kn}{K},\ k = 0, \frac{K}{2}; \tag{15}$$

$$B_k = \frac{2}{N}\sum_{n=0}^{K-1} x_n \sin\frac{2\pi kn}{K},\ k = 1, \ldots, \frac{K}{2} - 1. \tag{16}$$

The Fourier series (13) can also be represented in the form

$$x_n = \sum_{k=0}^{K/2} C_k \cos\left(\frac{2\pi kn}{K} + \varphi_k\right), \tag{17}$$

where C_k is the amplitude of the k-th harmonic calculated by the formula

$$C_k = \sqrt{A_k^2 + B_k^2},\ k = 0,\ldots,\frac{K}{2}; \tag{18}$$

φ_k is the phase of the k-th harmonic calculated by the formula

$$\varphi_k = arctg\left(-\frac{B_k}{A_k}\right),\ k = 0,\ldots,\frac{K}{2}. \tag{19}$$

The calculation of the spectra of velocities and accelerations can be carried out in two ways. The first is to sequentially differentiate in time the Fourier series for the displacements, which corresponds to (17)

$$\dot{x}_n = \sum_{k=0}^{K/2} \hat{C}_k \sin\left(\frac{2\pi kn}{K} + \varphi_k\right); \tag{20}$$

$$\ddot{x}_n = \sum_{k=0}^{K/2} \hat{\hat{C}}_k \cos\left(\frac{2\pi kn}{K} + \varphi_k\right), \tag{21}$$

where

$$\hat{C}_k = -C_k \frac{2\pi k}{Kh}; \tag{22}$$

$$\hat{\hat{C}}_k = -C_k \left(\frac{2\pi k}{Kh}\right)^2. \tag{23}$$

The second method is to calculate velocities and accelerations based on a time series of displacements. $\{X^n\}$ using difference relations

$$\dot{x}_n = \frac{x_{n+1} - x_{n-1}}{2h}; \tag{24}$$

$$\ddot{x}_n = \frac{x_{n+1} - 2x_n + x_{n-1}}{h^2} \tag{25}$$

and presentation of velocity time series $\{\dot{X}^n\}$ and accelerations $\{\ddot{X}^n\}$ in the form of finite Fourier series

$$\ddot{x}_n = \sum_{k=0}^{K/2} \tilde{C}_k \cos\left(\frac{2\pi kn}{K} + \tilde{\varphi}_k\right); \tag{26}$$

$$\ddot{x}_n = \sum_{k=0}^{K/2} \tilde{\tilde{C}}_k \cos\left(\frac{2\pi kn}{K} + \tilde{\tilde{\varphi}}_k\right), \tag{27}$$

where

$$\tilde{C}_k = \sqrt{\tilde{A}_k^2 + \tilde{B}_k^2},\ k = 0,\ldots,\frac{K}{2}; \tag{28}$$

$$\tilde{\varphi}_k = arctg\left(-\frac{\tilde{B}_k}{\tilde{A}_k}\right),\ k = 0,\ldots,\frac{K}{2}; \tag{29}$$

$$\tilde{A}_k = \frac{2}{N}\sum_{n=0}^{K-1} \dot{x}_n \cos\frac{2\pi kn}{K},\ k = 1,\ldots,\frac{K}{2}-1; \tag{30}$$

$$\tilde{A}_k = \frac{1}{N}\sum_{n=0}^{K-1} \dot{x}_n \cos\frac{2\pi kn}{K},\ k = 0,\frac{K}{2}; \tag{31}$$

$$\tilde{B}_k = \frac{2}{N}\sum_{n=0}^{K-1} \dot{x}_n \sin\frac{2\pi kn}{K},\ k = 1,\ldots,\frac{K}{2}-1; \tag{32}$$

$$\tilde{\tilde{C}}_k = \sqrt{\tilde{\tilde{A}}_k^2 + \tilde{\tilde{B}}_k^2},\ k = 0,\ldots,\frac{K}{2}; \tag{33}$$

$$\tilde{\tilde{\varphi}}_k = arctg\left(-\frac{\tilde{\tilde{B}}_k}{\tilde{\tilde{A}}_k}\right),\ k = 0,\ldots,\frac{K}{2}; \tag{34}$$

$$\tilde{\tilde{A}}_k = \frac{2}{N}\sum_{n=0}^{K-1} \ddot{x}_n \cos\frac{2\pi kn}{K},\ k = 1,\ldots,\frac{K}{2}-1; \tag{35}$$

$$\tilde{\tilde{A}}_k = \frac{1}{N}\sum_{n=0}^{K-1} \ddot{x}_n \cos\frac{2\pi kn}{K},\ k = 0,\frac{K}{2}; \tag{36}$$

$$\tilde{\tilde{B}}_k = \frac{2}{N}\sum_{n=0}^{K-1} \ddot{x}_n \sin\frac{2\pi kn}{K},\ k = 1,\ldots,\frac{K}{2}-1. \tag{37}$$

For a posteriori analysis of the accuracy of the obtained numerical results, a comparison of the acceleration spectra calculated by two methods is carried out.

Taking into account the properties of the spectrum of a discrete periodic time series, the authors used the following method of spectral analysis of forced vibrations of a multi-frequency vibrating sieve for cleaning drilling mud:

1) the integration step over time is calculated

$$h = \frac{T}{N} = \frac{2\pi}{N\omega}, \tag{38}$$

where N is the number of steps in time during the period of rotation of the debalances.

In computational experiments, it was adopted that $N = 200$;

2) a time series of displacements is calculated $\{X^n\}$ masses of the analyzed dynamic system at the time interval $[0, T_0]$, where $T_0 = MT$. In computational experiments it was accepted that $M = 50$;

3) using the final segment of the time series $\{X^n\}$, $n = \overline{(M-K)N, MN}$, autocorrelation functions of mass displacements of the analyzed dynamic system on the segment $[(M-K/2)N, MN]$ are constructed. In computational experiments it was accepted that $K = 32$;

4) using the approximate periodicity conditions of the autocorrelation functions (7), (8), the oscillation period is determined $T_1 = kT$ of the analyzed dynamic system. In the case of subharmonic oscillations $k > 1$. If on the considered finite segment $[(M-K/2)N, MN]$ periodicity conditions (7), (8) are not met, it is necessary to increase the parameter M, which determines the length of the segment $[0, T_0]$, on which the time series of displacements of masses of the analyzed dynamic system are calculated $\{X^n\}$, or increase the parameter K, which determines the maximum permissible oscillation period;

5) using the finite segment of the time series $\{X^n\}$, $n = \overline{(M-K)N, MN}$, the spectrum of mass displacements of the analyzed dynamic system is calculated using formulas (14)–(16). If the spectrum has a limited frequency band, then the condition is satisfied

$$l < kN/2 - s, \tag{39}$$

where $s > 0$ is a parameter that determines the width of the spectrum.

If the condition (39) is satisfied, the discrete Fourier transform allows to restore the original continuous functions of the mass displacements of the analyzed dynamic system. Otherwise, it is necessary to increase the sampling frequency, i.e. reduce the size of the integration step over time h and return to point 2 of the methodology.

The developed method of spectral analysis of forced vibrations of a multi-frequency vibrating sieve for cleaning drilling muds is based on the assumption that its motions are periodic. If the deterministic chaos arises in the dynamic system under consideration, then the autocorrelation function of the time series of displacements $\{X^n\}$ must have a finite carrier, i.e. it must be zero outside a finite time interval. For finite time intervals of a time series, the aperiodicity criterion can be formulated as follows: for each $\varepsilon > 0$ exists $M(\varepsilon)$ such that

$$|\psi_m| \le \varepsilon \text{ for any } m > M(\varepsilon). \tag{40}$$

To study deterministic chaos, computational algorithms proposed in [36] can be used.

3.3. Phase diagrams.

The dynamic system of a vibrating multi-frequency sieve for cleaning drilling muds is described by a nonlinear dissipative non-autonomous system of ordinary differential equations [24]. The motions of dissipative systems can be divided into two classes: transient, non-stationary motions, corresponding to the relaxation process from the initial to the limiting set of states, and the class of stationary, stabilized motions, the phase trajectories of which completely belong to the limiting sets [34, 36]. Important from a physical point of view are the limiting sets that attract - attractors. Over time, an random initial state from some region of attraction G, which includes the attractor G_0, relaxes to G_0. The motion, which corresponds to the phase trajectory in the sphere of attraction, is a transient process. The stabilized motion is characterized by the belonging of the phase trajectories to the invariant limiting set, i.e. the attractor G_0.

To analyze the process of stabilization of the multi-frequency vibrating sieve's motion and visually detect attractors, phase diagrams in the "displacement-velocity" variables are used.

3.4. Construction of amplitude dependences by the method of continuation by parameter.

When studying the dependences of the amplitudes of displacements, velocities, and accelerations of the dynamic system under consideration on changes in its parameters, the method of continuation by parameter with step-by-step changes in the system parameters was used [37]. The solution obtained in the previous step is chosen as the initial approximation.

When performing calculations, the system parameters were changed with a constant step from the initial to the final values specified in the initial data, and then in the reverse direction from the final value to the initial. This approach allows us to obtain, in particular, complete amplitude-frequency dependences for a dynamic system taking into account the presence of unstable branches.

4. Results and discussion

The study of the vibroimpact oscillator of a vibrating multi-frequency sieve for cleaning drilling muds was carried out using the mathematical model developed in

[24] in accordance with the proposed methodology for the analysis of forced vibrations.

Of practical interest are the vibroimpact modes of oscillations of an oscillator at frequencies exceeding the natural frequency of the generating linear system, at which accelerations are excited in the system that significantly exceed the acceleration in the linear system. Such modes are realized in the resonant area of oscillations of the generating linear system, in the presence of gaps δ_1 and δ_2 in the stationary state of the vibroimpact system between the masses m_0 and one-sided limiters of large amplitude oscillations of a linear system [24].

Depending on the values of the oscillator parameters and the initial conditions, two stabilized oscillation modes can be implemented: with impacts on one-sided limiters - as a nonlinear system, and shock-free mode - as a linear system with oscillation amplitudes smaller than the gaps δ_1 and δ_2.

The studies were conducted with the initial parameters given in Table 1.

Table 1 – Parameters of the vibroimpact oscillator

Parameter, units of measurement	Value
Mass m_0, kg	165
Stiffness of two-sided bonds c_0, N/m	1650000
Viscosity of two-sided bonds b_0, kg/s	1000
Stiffness of one-sided bonds $c_1 = c_2$, N/m	17000000
Viscosity of one-sided bonds $b_1 = b_2$, kg/s	1000
Mass of the exciter debalances m, kg	1.0
Eccentricity of the exciter debalances r, m	0.1
Moment of inertia of debalances I, kg m^2	0.01
Efficiency of the transmission drive η, %	100
Drive gear ratio u	1
Engine overload capacity ς	2.4
Rated engine power N_n, kW	1.0
Nominal angular speed of rotation ω_n, `	3-250
Current frequency in networks and power supply f_c, Hz	1-83.3
Number of pole pairs of an induction motor p	2
Conditional coefficient of rolling friction μ	0.001
Diameter of the debalance shaft d, m	0.06

Fig. 1 shows the amplitude-frequency characteristic (AFC) of the oscillator when the synchronous angular speed of rotation of the rotor of the vibration exciter electric motor changes. Vibroimpact mode begins at a frequency of 90 rad/s. The breakdown of the vibroimpact mode occurs later, at a frequency of 190 rad/s, which is associated with the dependence of the external force on the rotation frequency in systems with a limited excitation source.

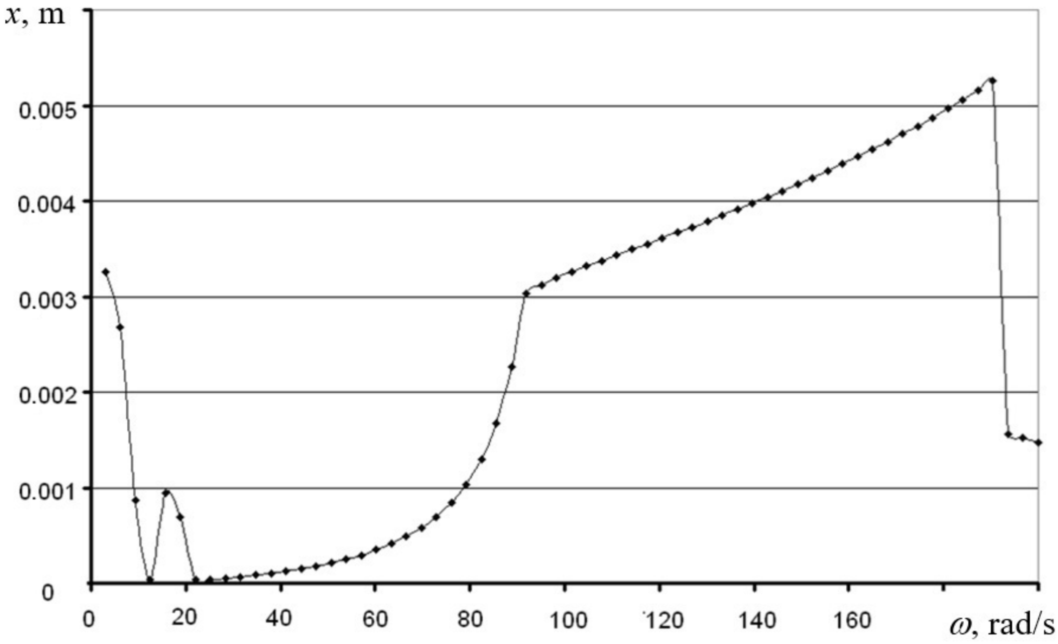


Figure 1 – AFC of the vibroimpact oscillator when changing the synchronous frequency of the vibration exciter electric motor

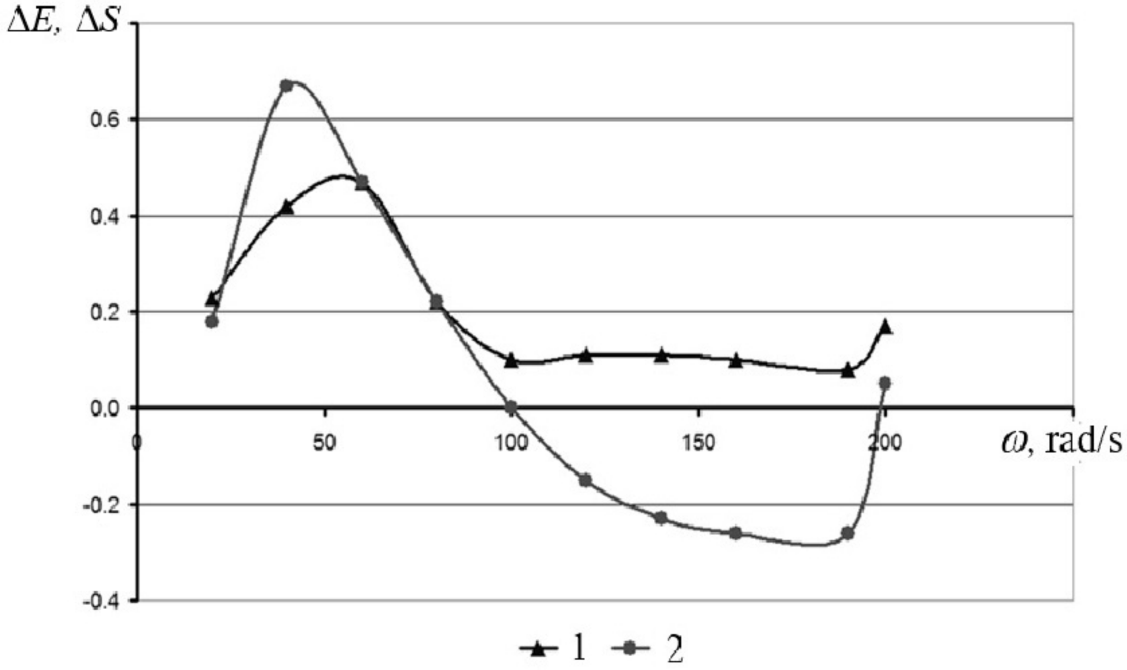
In a system with limited excitation, the kinetic and potential energies at the natural frequency of the linear system coincide during the excitation period. When the oscillations are disrupted, the phases coincide, and energy losses from the action of the external force and resistance forces are completely compensated by the energy of the external source. This is confirmed by the dependences of the ratio of the difference of the energy maxima and their areas to the total energy of the system during the period on the excitation frequency, shown in Fig. 2.

In a system with a limited excitation source, the potential energy maxima exceed the kinetic energy maxima even after the disruption of the vibroimpact oscillations. The minimum difference in energy amplitudes occurs at the frequency of the disruption of the vibroimpact oscillations.

Fig. 3 shows the dependences of the phase shift between the external force and the resistance forces on the nominal excitation frequency for a generating linear and vibroimpact system with an ideal and limited excitation source.

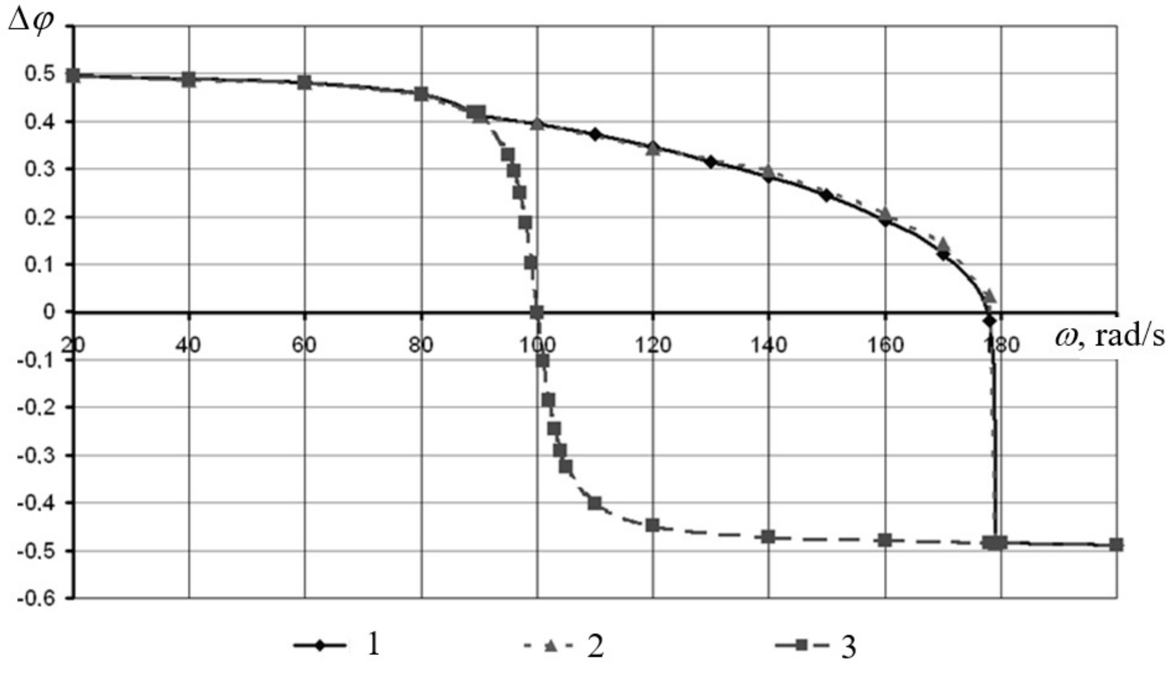
The analysis of the dependencies shows that the phases of the forces coincide for a linear system at the resonant frequency for a vibroimpact system with an ideal and limited excitation source, when the vibroimpact oscillations are disrupted at a frequency of $\omega = 179$ rad/s. Depending on the phase shift of the forces from the nominal excitation frequency for a vibroimpact system with an ideal and limited excitation source, they are identical.

The results of the performed studies show that the oscillation parameters of a vibroimpact oscillator with a limited power of the vibration excitation source, which do not depend on the drive power, completely coincide with the results obtained during the studies of a vibroimpact oscillator with an ideal excitation source. The influence of the vibration exciter power is adequately reflected on the oscillation parameters of the vibroimpact oscillator.



1 – dependence of the difference of energy maxima; 2 – dependence of the difference of energy areas

Figure 2 – Dependence of the relationship between the difference of energy maxima and their areas during the excitation period to the total energy of the system on the synchronous frequency of the vibration exciter electric motor



1 – ideal excitation source, 2 - limited excitation source, 3 – linear

Figure 3 – Dependences of the phase shift between the external force and the resistance forces on the nominal frequency for a linear and vibroimpact system with an ideal and limited excitation source

5. Conclusions

1. It is difficult to obtain an analytical solution of the Cauchy problem for equations, which describe the motion of a vibrating multi-frequency sieve as a dynamic system with a limited excitation source, due to its significant nonlinearity. Therefore, the method of computational experiment is used to identify and study the characteristics of vibrations of a vibrating multi-frequency sieve for cleaning drilling muds.

At the first stage of the computational experiment, the equations of motion of the dynamic system are numerically integrated using a computational algorithm based on the use of three-layer difference schemes with weights. As a result, time series of its mass displacements are calculated.

At the second stage of the computational experiment, the obtained time series are studied using autocorrelation functions to determine the period of forced vibrations; spectral analysis of displacements, velocities and accelerations; phase diagrams in the variables "displacement-velocity"; dependences of the amplitudes of displacements, velocities and accelerations on changes in parameters in the dynamic system obtained by the method of continuation by parameter.

2. The task of studying the operating modes of a vibrating multi-frequency sieve is to find a solution to a nonlinear system of equations of motion that satisfies the periodicity conditions, when the period of motion of the dynamic system under consideration is unknown in advance. Therefore, the task of determining the oscillation period of the analyzed dynamic system arises, taking into account the analysis of the periodicity of the time series of displacements of its masses. For this purpose, an apparatus of autocorrelation functions is used.

3. The developed method of spectral analysis of forced vibrations of a multi-frequency vibrating sieve for cleaning drilling muds is based on the assumption that its motions are periodic. If deterministic chaos occurs in the dynamic system under consideration, then the autocorrelation function of the time series of displacements must have a finite carrier, i.e. turn into zero outside a finite time interval.

4. To analyze the process of stabilization of multi-frequency vibrating sieve's motion and visually detect attractors, phase diagrams in the variables "displacement-velocity" are used. When studying the dependences of the amplitudes of displacements, velocities and accelerations of the dynamic system under consideration on the change of its parameters, the method of continuation by parameter with a step-by-step change of the system parameters was used. The solution obtained in the previous step is chosen as the initial approximation of the solution.

5. The study of the vibroimpact oscillator of a vibrating multi-frequency sieve for cleaning drilling muds was carried out in accordance with the proposed method of analysis of forced vibrations. The vibroimpact oscillation modes of the oscillator at frequencies exceeding the natural frequency of the generating linear system were studied, at which accelerations are excited in the system that significantly exceed the accelerations in the linear system. Such modes are realized in the resonant area of oscillations of the generating linear system, if there are gaps between the mass and one-sided limiters of large amplitudes of oscillations of the linear system in the stationary state of the vibroimpact system.

6. The amplitude-frequency characteristic of the oscillator is constructed when the synchronous angular speed of rotation of the rotor of the vibration exciter electric motor is changed. The vibroimpact mode begins at a frequency of 90 rad/s. The disruption of the vibroimpact mode occurs later, at a frequency of 190 rad/s, which is associated with the dependence of the external force on the frequency of rotation in systems with a limited excitation source.

7. It is shown that in a system with limited excitation, the kinetic and potential energies at the natural frequency of the linear system coincide during the excitation period. When the oscillations are disrupted, the phases coincide, and the energy losses from the action of the external force and resistance forces are completely compensated by the energy of the external source. In a system with a limited excitation source, the maxima of potential energy exceed the maxima of kinetic energy even after the disruption of vibroimpact oscillations. The minimum difference in the amplitudes of the energies occurs at the frequency of the disruption of vibroimpact oscillations.

Conflict of interest

Authors state no conflict of interest.

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МЕТОДИКА АНАЛІЗУ ВИМУШЕНИХ КОЛИВАНЬ ВІБРАЦІЙНОГО ПОЛІЧАСТОТНОГО СИТА ДЛЯ ОЧИЩЕННЯ БУРОВИХ РОЗЧИНІВ МЕТОДОМ ОБЧИСЛЮВАЛЬНОГО ЕКСПЕРИМЕНТУ

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Анотація. Для ідентифікації та дослідження характеристик коливань вібраційного полічастотного сита для очищення бурових розчинів, як суттєво нелінійної динамічної системи з обмеженим джерелом збудження, запропоновано використовувати метод обчислювального експерименту. На першому етапі проводиться чисельне інтегрування рівнянь руху динамічної системи за допомогою обчислювального алгоритму, який засновано на використанні тришарових різницевих схем з вагами. На другому етапі проводиться дослідження одержаних часових рядів з використанням автокореляційних функцій для визначення періоду вимушених коливань; спектрального аналізу переміщень, швидкостей та прискорень; фазових діаграм у змінних «переміщення-швидкість»; залежностей амплітуд переміщень, швидкостей і прискорень від зміни параметрів динамічної системи, одержуваних методом продовження за параметром. Розроблена методика спектрального аналізу вимушених коливань полічастотного вібросита для очищення бурових розчинів ґрунтується на припущенні, що його рухи є періодичними. Якщо в динамічній системі, що розглядається, виникає детермінований хаос, то автокореляційна функція часового ряду переміщень повинна мати кінцевий носій, тобто перетворюватись в нуль поза кінцевим інтервалом часу. Для аналізу процесу встановлення руху полічастотного вібросита та візуального виявлення атракторів використовуються фазові діаграми у змінних «переміщення-швидкість». При дослідженні залежностей амплітуд переміщень, швидкостей і прискорень динамічної системи від зміни її параметрів використовувався метод продовження за параметром при покроковій зміні параметрів системи. Як початкове наближення рішення вибирається рішення, отримане на попередньому кроці. Проведено дослідження віброударного осцилятора вібраційного полічастотного сита для очищення бурових розчинів відповідно до запропонованої моделі та методики аналізу вимушених коливань. Досліджено віброударні режими коливань осцилятора на частотах, що перевищують власну частоту лінійної системи, що породжує, при яких в системі збуджуються прискорення, що істотно перевищують прискорення в лінійній системі. Такі режими реалізуються в зарезонансній області коливань лінійної системи, що породжує, за наявності в стаціонарному стані віброударної системи зазорів між масою і односторонніми обмежувачами великих амплітуд коливань лінійної системи.

Ключові слова: вібраційне полічастотне сито, бурові розчини, очищення, суттєво нелінійна динамічна система, методика аналізу, метод обчислювального експерименту, віброударні режими коливань.