VIBRATIONAL FEEDERS WITH VIBRO-IMPACT ADAPTIVE DRIVE
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ВІБРАЦІЙНІ ЖИВИЛЬНИКИ З ВІБРОУДАРНИМ АДАПТИВНИМ ПРИВОДОМ
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ВИБРАЦИОННЫЕ ПИТАТЕЛИ С ВИБРОУДАРНЫМ АДАПТИВНЫМ ПРИВОДОМ
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Abstract. The object of the research is the vibrational feeders with vibro-impact adaptive drive, the characteristic feature of which is the multifrequency excitation and self-adjusting of the oscillations parameters, including the frequency spectrum, to change the mass of the technological load. The basis of the problem solution is the excitation of vibro-impact oscillations of the working body (tray) of the feeder, at which in its vibration drive the resonant frequency oscillations are realized, the intensity and frequency spectrum of which increase with increasing the technological load on the feeder tray, which distinguishes such vibrators from those used in typical feeders. When increasing the load on the tray from the bulk medium at output from the bunkers or reflacted ore from the chambers of ore blocks, the vibro-impact drive adapts to this by increment of parameters and the oscillations spectrum of the vibro-impact drive impactor, which, respectively, are transmitted to the tray and bulk media. Vibro-impact systems with such vibro-excitement with parameters and dynamic loading, identical with analogues, have, depending on their purpose, from 3 to 8 times less power consumption, and from 3 to 5 times increase in the intensity of oscillations. So when changing the mass of the tray with the attached mass of bulk medium from 300 to 3000 kg, its acceleration increases by 2.4 times, from 60 to 143 m/s^2. At multifrequency excitation, an increase in the level and depth of vibration spreading in an massive of bulk media, the destruction of frozen bulk material, static equilibrium bonds, and reduction the forces of adhesive and friction between particles of bulk material. Thus, the vibro-impact drive of the vibrating feeder is adapted to change the load on the feeder tray from the bulk medium on it. At the same time, due to polyfrequency excitation, the mobility of the technological medium that is being processed is improved, which is especially important for the control of freezing and adhesion of materials in the bunkers, or in the pillar of the reflacted rock mass during its output by the vibrating feeders from the chambers of the wining blocks at the underground mining of minerals. All this compared with the harmonic excitation of a typical feeder tray will lead to an increase in the productivity and efficiency of the vibration output of bulk media from bunkers or chambers of ore blocks.

Keywords: underground mining of minerals, vibrational feeders, bulk medium, vibro-impact adaptive drive, multifrequency excitation, oscillations parameters, frequency spectrum, power consumption.

Introduction. Vibratory feeders have been widely used for the output and loading of various loose media from bunkers, tanks, reflacted ore from chambers of wining blocks, etc. [1-6]. Constructions of vibratory feeders differ in diversity, which is explained by the difference in their technological purpose, the properties of the unloaded loose medium, the conditions of its storage and output. The classification of feeders is carried out according to the following features: the number of oscillating masses, the type of drive, the purpose, the properties to conrtol the oscillation parameters, the method of montage.

According to the number of oscillating masses, vibro-feeders are classified into single, double, and multi-mass systems. Single-mass constructions usually operate in over-resonance mode and require significant excitation efforts to obtain the required amplitudes of oscillation, which is a disadvantage of such systems. Another significant disadvantage of single-mass feeders with the most common debalanced vibration drive is the excitation of significant amplitudes of oscillations during transition the resonance zone during the system startup. This can be the reason for the breakdown of various elements of the construction of the vibration feeder, and leads to the transfer of significant dynamic loads on the foundation and elements of its supporting construction. In addition, to overcome the resonance on its own oscillations frequency of such system during its startup, it is necessary to significantly overvalue the nameplate capacity of the electric motor vibrational generator, in compared with the required capacity for the realization of stationary over-resonance oscillation modes. As a result, the capacity consumption of a electric motor in working modes does not exceed 30-70 % of the nominal, selected under the conditions of system startup and passage the resonance, which reduces system efficiency and power factor of the electric motor. Usually in such systems, using the debalanced drive, the speed of vibration transport of the process medium at a level of 0.25 m/s is provided at an excitation frequency of 1500 rpm and 0.35 m/s, at a frequency of 1000 rpm. The maximum amplitudes of oscillations, respectively, are 5 and 8 mm. Limits of amplitudes and excitation frequencies are due to permissible accelerations, which, in the conditions of durability and reliability of the vibration fedder construction, generally do not exceed 100 m/s^2.

Despite these disadvantages, single-mass vibratory feeders with an inertial debalanced vibrator, due to its simplicity and exploatation reliability, have been widely used, especially in heavy performance in mining at output in the outlets (hollows), and the loading of refined ore from wining blocks chambers at the underground mining of minerals [1, 2]. These vibrofeeders are subject to high requirements for the strength and reliability of the construction, due to they operate in conditions of bulk, under ore heavy load, and explosions at the destruction of oversized bits of the rock mass and the elimination of its freezings in the chamber, as well as significant shock loads from the bites that fall on the feeder tray.

The process of output the refined ore from the chambers through the outlets (hollows) is accompanied by cluster formation and its freezings in the chamber, which significantly reduces the output productivity and worsens the working conditions of miners due to the need to eliminate freezings, in the vast majority, by explosions means. At vibration output, compared with other methods, the statistical number of ore freezings is significantly reduced. This is due to the fact that vibromechanisms carry not only the displacement of the reflacted rock mass for loading, but also provide its excitation in the depth of the array. At the same time, vibration actively influences the parameters of the zone of reflected ore flow due to the destruction of the static equilibrium bonds and the reduction of the adhesive forces, and the friction between the bites of the bulk medium. Limit angles of the leakage of bulk material under the influence of vibration sharply decrease (up to 32-

36°, in comparison with 43-55° natural angles of the slope) [2]. Therefore, vibration improves forced output due to the development of the shape of the loosening in the chamber and increases the speed of displacement the reflected rock mass. In this case, in comparison with other methods of output, vibration increases the speed of ore leakage and flow parameters with constant parameters of outlets - hollows, which sharply increases the throughput of the outlet. This, with the use of vibration output, provides an opportunity to increase the distance between the outlets in the block and outlets in a row, which significantly reduces capital costs for their arrangement [1].

The depth of excitation of the reflected rock mass depends on its physical and mechanical properties and vibration parameters. As a rule, vibration in a lumpfill medium during the excitation of a vibro-tray of a typical feeder, and its maximum acceleration of 100 m/s^2, extends to a distance not exceeding 2-3 m due to the dissipation of energy in the rock massife [1, 2]. In addition, vibration parameters of the vibrating feeder tray with excitation from debalanced vibrators depend on the weight of the bulk material on the tray and decrease with its growth. When operating under a blockage it is possible to completely oscillation stoping of the vibration feeder tray and, accordingly, the ore outut. To restore the feeder operating it is necessary to reduce the load on the working body, which is a rather labor-intensive and dangerous operation for miners. In view of the above, an increase in the depth of excitation of the bulk medium, and the non-interruption of the excitation of the vibration feeder tray under any load from the bulk medium, is important for intensification and increases the output productivity.

Double-mass constructions of vibration feeders due to the possibility of realisation expedient resonance modes are more common in other industries. Resonant oscillation modes allow, on the one hand, to increase the feeder efficiency (resonance machines consume about twice less power than over-resonance ones), on the other hand - to improve the dynamic balance of the system, which enables them to be installed without the foundation. Operating modes of double-mass vibratory feeders are near-resonance (about 0.85-0.95 of the eigenfrequency of the system). With such a pre-resonance setting it is possible to prevent the passage of the resonance zone during the system startup and run-off, as in single-mass systems. With these advantages, the essential disadvantage of such systems is their sensitivity to changing the workload from the technological medium, which leads to the exit the system off the pre-resonance mode and a significant decrease in the oscillation parameters, or to its possible occurrence in non-stationary resonance oscillations. Therefore, to reduce the sensitivity of the machine to the changing of technological load, when using the active mass as a working body, its reactive mass should be significant. The presence of a massive reactive element significantly increases the vibration feeder mass, which is a significant disadvantage of this type dynamic systems.

In view of the above, it is important to create vibration feeders that combine the benefits of single- and double-mass systems, and do not have the above-mentioned disadvantages of these systems.

The purpose of the work is to create vibrational feeders with vibro-impact adaptive drive, which are self-adjusting to change the technological load mass.

From the foregoing it follows that the best for vibration feeders are resonant oscillation modes, which must be resistant to changes in the technological load mass and other parameters of the system. They must also be equipped with the most simple in design and in operation debalance drive, with the electric motor capacity, which does not exceed the power required for the realisation of stationary oscillation modes. For this purpose, a double-mass dynamic vibrating feeder system is proposed, which combines the advantages of double-mass resonant and single-mass over-resonant systems, which has no inherent disadvantages for these systems.

Method. The dynamic scheme of a vibratory feeder with a vibro-impact adaptive drive is shown in Figure 1.

The vibration feeder is a double-mass dynamic system of interconnected with a fixed base using two-side and one-side elastic bonds and damping elements. The system consists of a tray with mass m_1 and a impactor with mass m_2. The tray is installed on a stationary basis by holding elastic bonds with stiffness c_{p10} and viscosity b_{p10}. The impactor is fixed to the feeder tray by holding the elastic bonds with the stiffness c_{p21} and the viscosity b_{p21} and has a one-way and one side elastic constraint with stiffness, respectively, c_{r12} and c_{r21} and viscosity b_{r12} and b_{r21}. The gaps δ_{12} and δ_{21} are set up between the tray and the constraints to facilitate system start-up. The debalanced oscillator is fixed on the impactor, whose axle rotates in a horizontal plane with the debalanced masses m_0, using an asynchronous motor with nominal power N_n. The debalanced masses m_0 are set with an eccentricity r and have a moment of inertia J. Debalanced masses at rotation create a centrifugal force of inertia that excitate small forced fluctuations of the feeder masses. We consider oscillations of the feeder mass only on the central axis. Therefore, the system is characterized by two connected coordinates: the angle - the angle of rotation of the vibration rotor and the oscillating - displacement of the feeder mass along the axis.
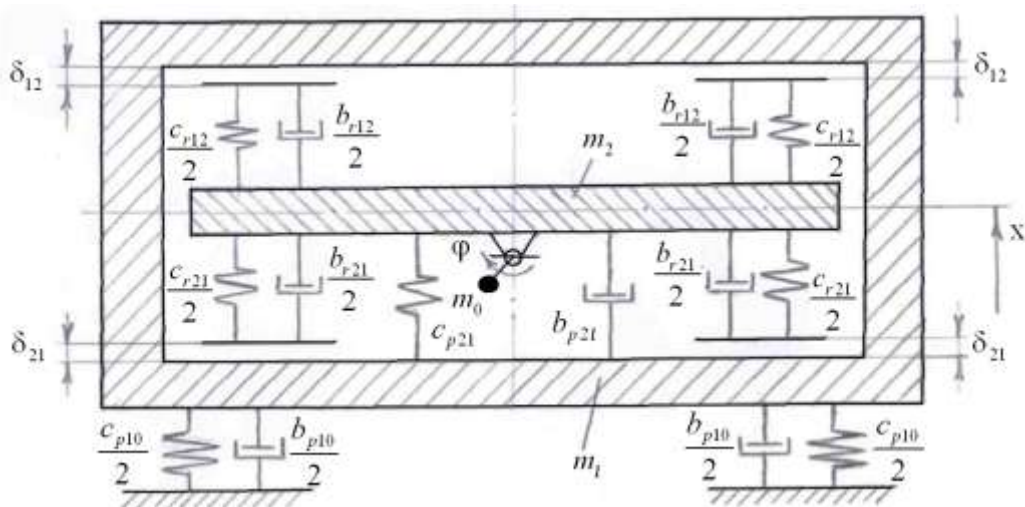

Figure 1 - Estimated dynamic scheme of vibration feeder with vibro-impact adaptive drive of limited capacity

The system of equations, which describes the motion of the dynamic system, shown in Figure 1 has the form

$$\begin{cases} (m_2 + m_0)\ddot{x}_2 + P_2 - P_{01} + P_{02} = m_0 r \ddot{\varphi} \sin\varphi + m_0 r \dot{\varphi}^2 \cos\varphi \\ m_1 \ddot{x}_1 + P_1 - P_2 + P_{01} - P_{02} = 0 \\ J\ddot{\varphi} = m_0 r \ddot{x}_2 \sin\varphi + L(\dot{\varphi}) - R(\dot{\varphi}) + m_0 g r \sin\varphi \end{cases}, \qquad (1)$$

where x_1 - movement of the tray; x_2 - movement of the impactor; P_1 - force in holding elastic bonds of the tray; P_2 - force in holding elastic bonds of the impactor; P_{01}, P_{02} - forces in elastic constraints; φ - angle of debalances rotation; L - moment of rotation on the debalance axle; R - moment of frictional force rotation resistance in bearings; g - free fall acceleration.

Forces in two-side elastic bonds are determined by dependencies

$$P_1 = c_{p10} x_1 + b_{p10} \dot{x}_1 ;$$

$$P_2 = c_{p21}(x_2 - x_1) + b_{p21}(\dot{x}_2 - \dot{x}_1) .$$

Forces in one-side elastic bonds are calculated by formulas

$$P_{01} = (c_{r12}\delta_{12} + b_{r12}\dot{\delta}_{12})H(-\delta_{12}) ;$$

$$P_{02} = (c_{r21}\delta_{21} + b_{r21}\dot{\delta}_{21})H(-\delta_{21}) ,$$

where $H(-\delta_j)$ - the Heviside functions, which are determined as follows

$$H(-\delta_{12}) = \begin{cases} 1, & \delta_{12} > 0 \\ 0, & \delta_{12} \le 0 \end{cases}, \quad H(-\delta_{21}) = \begin{cases} 1, & \delta_{21} > 0 \\ 0, & \delta_{21} \le 0 \end{cases}.$$

Flat gaps (tensions) in one-side elastic bonds are determined by dependencies

$$\delta_{12}(t) = x_2(t) - x_1(t) + \delta_{12} ;$$

$$\delta_{21}(t) = x_1(t) - x_2(t) + \delta_{21} .$$

The moment of rotation on the debalanances axle of the oscillator is calculated by the formula

$$L = \eta u \tilde{M} ,$$

where η - coefficient of effectiveness; u - transmission ratio of the drive; $\tilde{M}$ - moment of rotation on the electric motor axle.

The three-phase asynchronous motors with short-circuited rotor are used as an energy source in vibrating feeders. For such motors, the mechanical characteristic

(the dependence the moment on the axle of rotor sliding) is determined by the Clause formula

$$\tilde{M} = \frac{2M_{cr}}{s/s_{cr} + s_{cr}/s},$$

where M_{cr} - critical moment of the motors; s_{cr} - critical slide of the rotor; s - current slide.

The critical moment of an asynchronous motor is determined by the dependence

$$M_{cr} = \zeta M_n,$$

where ζ - reloading characteristic of the motor, characterizing its ability to short-lived overload; M_n - nominal motor moment.

The nominal moment of an asynchronous motor is determined by the formula

$$M_n = \frac{1000 N_n}{\omega_n},$$

where N_n - nominal motor capacity; ω_n - nominal angular motor speed.

The critical slide of the rotor of the asynchronous motor is determined by the dependence

$$s_{cr} = s_n(\zeta + \sqrt{\zeta^2 - 1}),$$

where s_n - nominal slide, which is determined by the formula

$$s_n = \frac{\omega_c - \omega_n}{\omega_c},$$

where ω_c - the synchronous angular speed of rotor rotation, which is calculated by dependence

$$\omega_c = \frac{2\pi f_c}{p},$$

where f_c - current frequency in the power supply network; p - number of an asynchronous electric motor poles pairs.

Current slide is determined by the formula

$$s = \frac{\omega_c - \omega}{\omega_c},$$

where ω - the current value of the angular speed, which is determined by the formula

$$\omega = u\dot{\varphi}.$$

The moment of resistance to rotation from frictional forces in bearings is determined as

$$R = 0.5\mu Fd \ ,$$

where μ - conditional coefficient of rolling friction; F - centrifugal force of debalance; d - the diameter of the debalanced axle.

The centrifugal force created during the rotation of the vibrator axle with the debalance is determined by the dependence

$$F = m_0 r \dot{\varphi}^2 \ .$$

To solve the equations of motion of system (1) it is necessary to supplement them with initial conditions. We will assume that at the initial moment of time (t=0) the system is at rest, that is

$$\begin{cases} x_1(0) = 0, \ \dot{x}_1(0) = 0; \\ x_2(0) = 0, \ \dot{x}_2(0) = 0; \\ \varphi(0) = 0, \ \dot{\varphi}(0) = 0. \end{cases} \qquad (2)$$

It is difficult to obtain an analytic solution of equations (1), (2), so their solution can be found by methods of numerical integration. The algorithm of the calculation of equations (1), (2) by this method and its program relisation is analogous to the solution and program realization of the mathematical model equations of the vibration polyfrequency sieve with the limited power of its oscillator drive [7].

In the first stage of the computational experiment, a numerical integration of the system of motion equations (1) is performed by means of the calculation algorithm, which is presented in [7]. As a result, the time series of the system mass motion are calculated. At the second stage of the computational experiment, the study of the obtained time series is carried out using:

- autocorrelation functions for determining the period of forced oscillations;
- spectral analysis of motions, speeds and accelerations;
- phase diagrams in "motion-speed" variables;
- dependencies of motions, speeds, accelerations, energies, powers, actions ect. of changes in the parameters of the dynamic system, received by the method of continuation by the investigated parameter, or by the method of discrete switching of this parameter. The algorithm of time series study is presented in [7].

The mathematical model of oscillations of a vibrating feeder with an adaptive vibrator with limited power drive and a calculation algorithm for its numerical researches are realized as a computer program for PC.

Research results and discussion. Researches of the oscillation of the vibrating feeder model by a numerical method were performed at the basic parameters of the feeder, shown in Table 1.

In order to establish the declared advantages of the proposed feeder, comparative studies of vibrating feeders with a vibro-impact drive and a typical feeder, as a single-mass model with harmonic excitation of oscillations, are performed. As the basic parameters of a typical vibration feeder, the parameters of the vibration feeder indicated in Table 1, were accepted. The transformation of the model of a feeder with a vibro-impact drive into a typical model of a single-mass feeder is carried out by assigning zero intervals to one-side elastic bonds and increasing the stiffness of the impactor elastic bonds to a level that eliminates the relative motions of the masses of its oscillatory system.

Table 1 - Basic parameters of a vibrating feeder for research

Parameter	Symbol	Value
Mass of impactor, kg	m_2	160
Mass of feeder tray, kg	m_1	300
Stiffness of two-side elastic bonds between m_1 and the base, kN/m	c_{p10}	1680
Viscosity coefficient of elastic bonds c_{p10}	b_{p10}	2000
Stiffness of two-side elastic bonds between m_1 and m_2, N/m	c_{p21}	278260
Viscosity coefficient of elastic bonds c_{p10}	b_{p21}	2000
Stiffness of the upper constraints, kN/n	c_{r12}	20000
Viscosity coefficient of elastic bonds c_{r12}	b_{r12}	2000
Static gap c_{r12}, m	δ_{12}	0.003
Elasticity of the lower constraints, kN/m	c_{r21}	20000
Viscosity coefficient of elastic bonds c_{r21}	b_{r21}	2000
Static gap c_{r21}, m	δ_{21}	0.003
Type of vibrating oscillator - electromechanical vibrator	*EB100L6U3*	-
Synchronous angular speed of vibrator rotation, rad/s	ω_y	104.67
Nominal angular speed of vibrator rotation, rad/s	ω_v	97.34
Weight of debalances, kg	m_0	12.28
Eccentricity of debalances, m	r	0-0.061
Force of excitation of forced oscillations at synchronous frequency, kN	$m_0 r \omega_c^2$	0-8.2
Static moment, kg·m	$m_0 r$	0-0.75
Nominal power of an asynchronous electric motor, kW	N_n	2.4
Number of pairs of electric motor poles, pairs	p	3
Coefficient of engine capacity to overload	ζ	2.4
Frequency of current in the network, Hz	f_c	50

In Figure 2 and Figure 3 shows fragments of oscillograms of stationary oscillations of the masses of feeders with vibro-impact and harmonic excitation. From the above oscillograms it is seen that the tray and impactor of the feeder with a vibro-impact drive carry out synchronous oscillations that differ in amplitude, and due to the slight divergence in the oscillations phases close to sinphasic. In this case, the amplitudes of the tray vibrations are more than 2.6 times higher then the amplitudes of the impactor oscillations and, accordingly, equal to 4.22 and 1.6 mm.
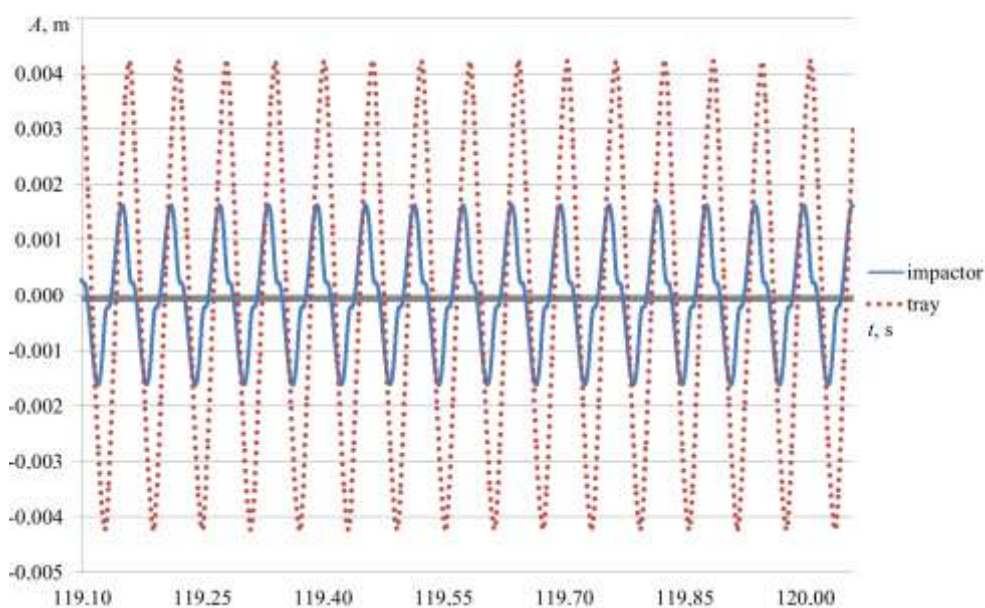
Figure 2 - Oscillograms of stationary oscillations of a feeder with vibro-impact excitation
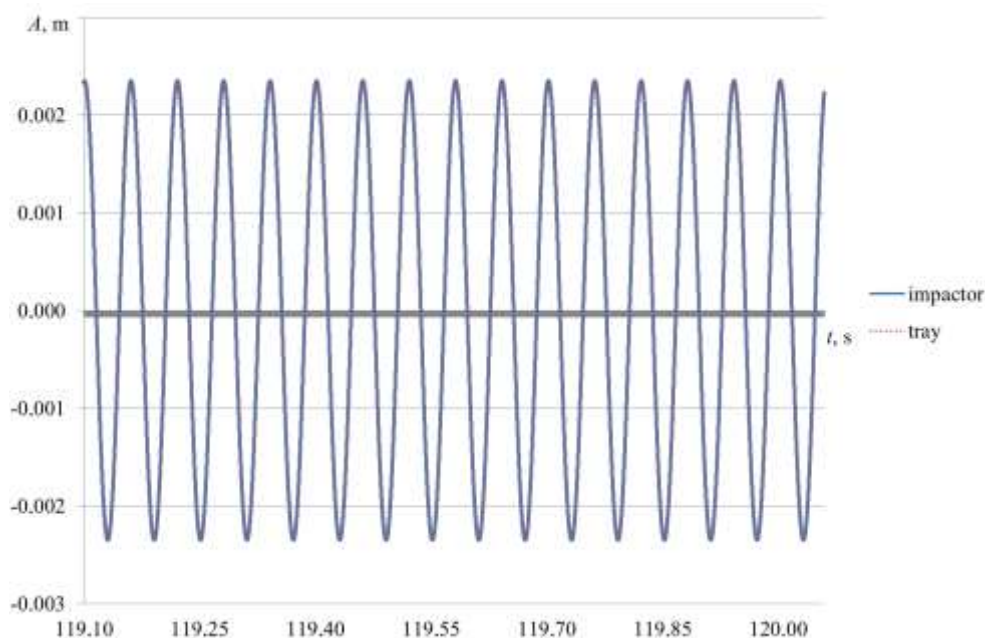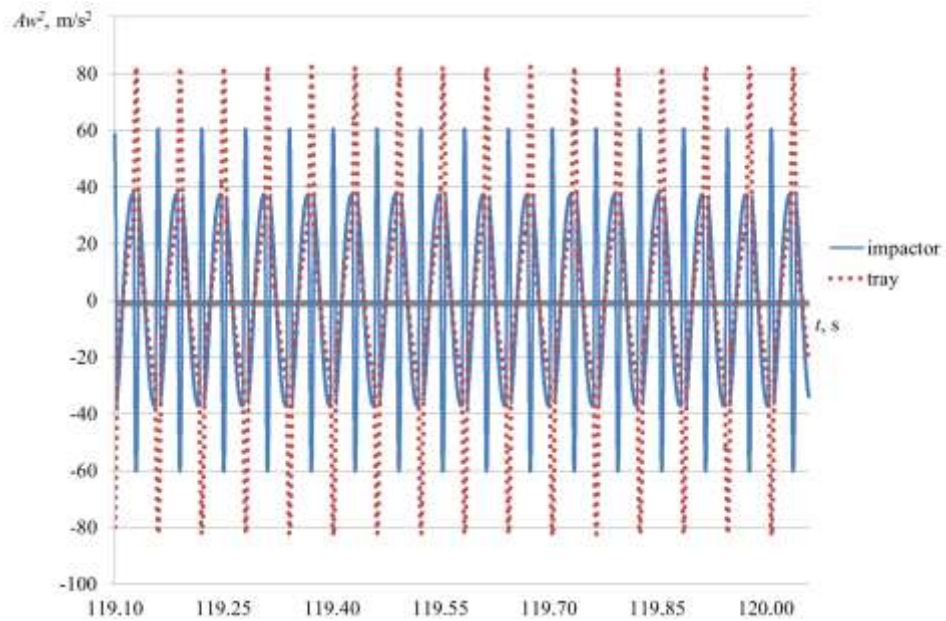

Figure 3 - Oscillograms of stationary oscillations of a feeder with harmonic excitation

The amplitudes of mass accelerations (Fig. 4) differ by 1.37 times and, accordingly, are equal to 82.2 and 60 m/s^2. In a single-mass model, the oscillation amplitude of the tray (Fig. 3) is 2.34 mm, while the accelerations (Fig. 5) is 25.3 m/s^2, which is 2.5 times less than the accelerations of the feeder tray with a vibro-impact drive.

The maximum kinetic energy of the feeder tray oscillation with a vibro-impact drive (Fig. 6) up to 2.42 times higher than the maximum of the kinetic energy of the feeder vibrations with harmonic excitation (Fig. 7), accordingly, having a value of 22.18 and 9.16 J.

Figure 4 - Oscillogram of stationary oscillations acceleration of feeders mass with vibro-impact excitation
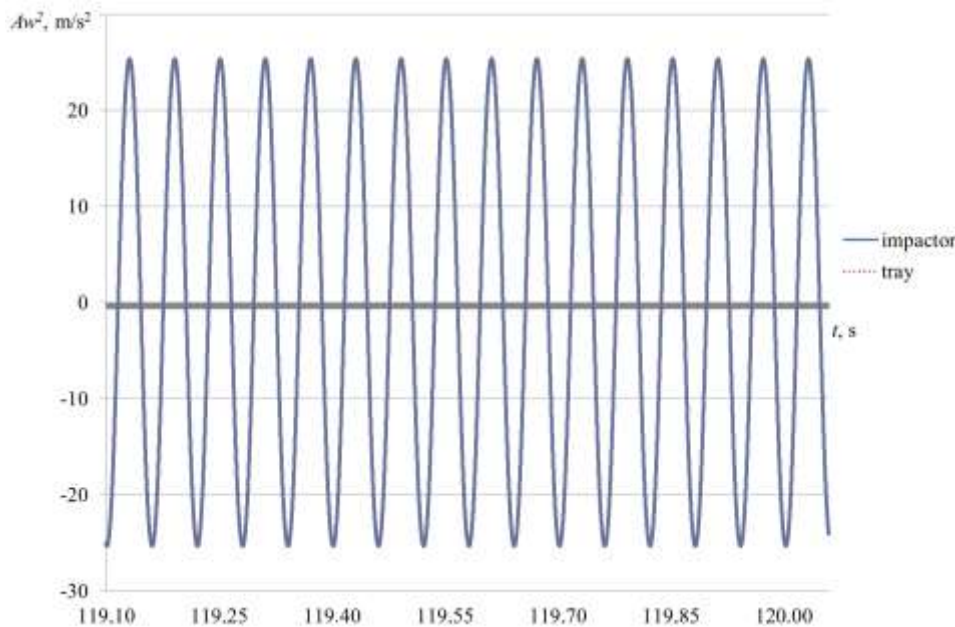

Figure 5 - Oscillogram of stationary oscillations acceleration of a feeder with harmonic excitation

The spectrum of the acceleration of the feeders masses oscillations with vibro-impact and harmonic excitations are significantly differ (Fig. 8 and Fig. 9, accordingly). The vibro-impact fedder spectrum is represented by a discrete set of superharmonic frequencies that are in the frequency of forced oscillations ω of the ratio, which is determined by the dependence $\omega_t = (2k-1)\omega$, where $k = 1, 2, 3, \ldots$. At the same time, in the spectrum of the impactor oscillations, there are harmonics with amplitudes of accelerations that are higher than the amplitude of excitations at the frequency of forced oscillations, that indicating a resonant amplification of the impactor oscillations [8-10]. In feeder oscillations with harmonic excitation (Fig. 9), the harmonics of polyfrequency oscillations are absent.
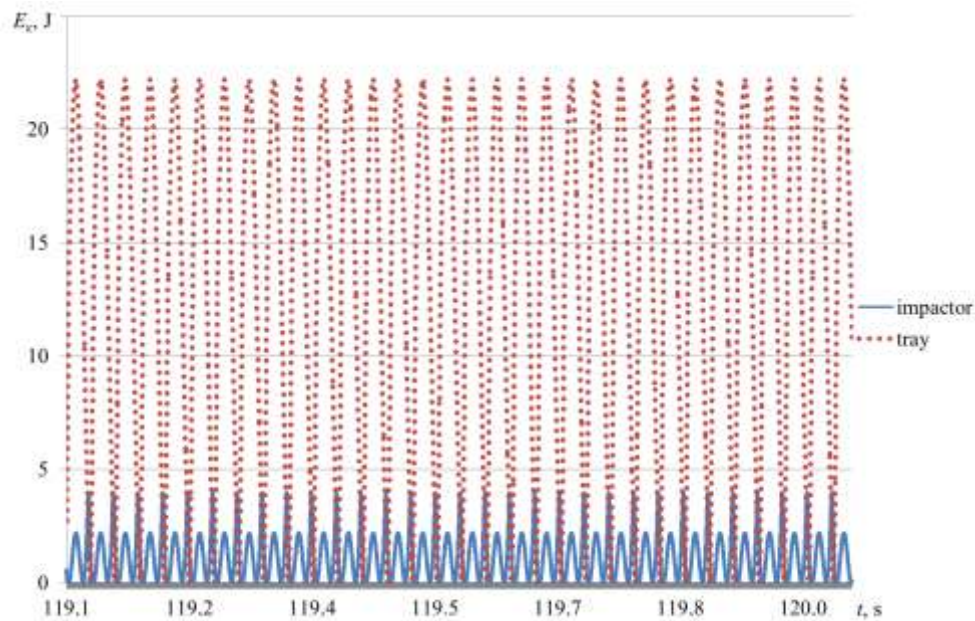
Figure 6 - Change of the kinetic energy of the feeders mass oscillations with vibration vibro-impact excitation for 16 periods of stationary oscillations
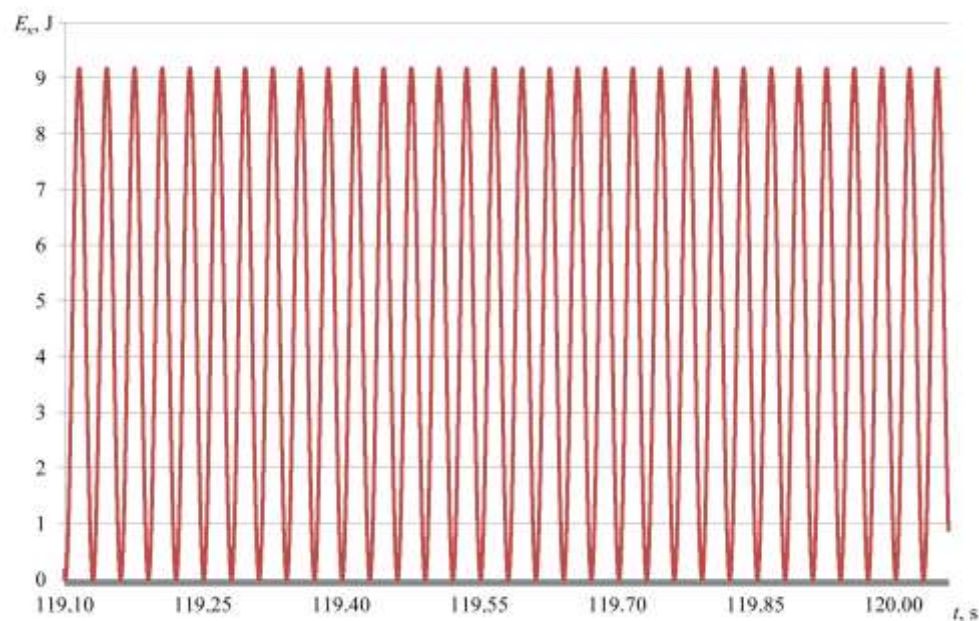

Figure 7 - Changing of the kinetic energy of the oscillations of the feeder with harmonic excitation for 16 periods of stationary oscillations

Comparative study of feeders oscillations with equivalent to the parameters and characteristics with the same parameters of vibrators but with various characteristics of the vibrational influence of drive, shows significant advantages in the effectiveness of oscillations at their excitation by vibro-impact drive. Amplitudes, accelerations, energies of oscillations of the feeder working bodies (trays) at such excitation at times exceed the indices of feeders working bodies oscillations the with harmonic excitation. Of course, this will help increase the efficiency of the output of bulk media from bunkers, or reflacted ore from the chambers of wining blocks at the underground mining of minerals.
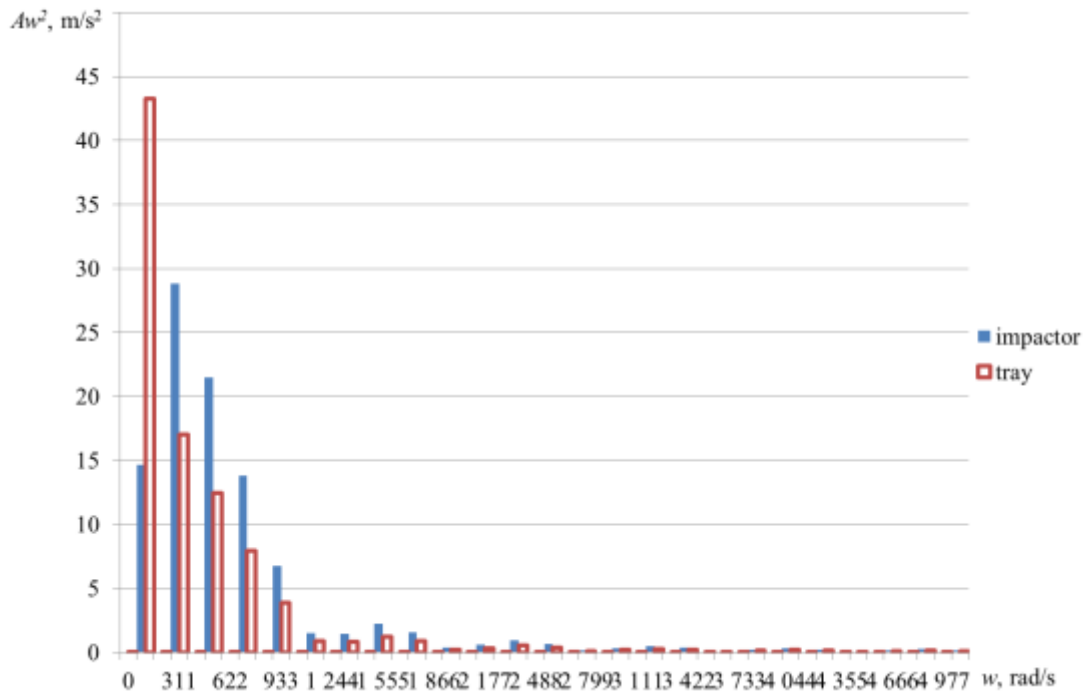
Figure 8 - Spectrogram of acceleration of feeder mass oscillations with vibro-impact excitation
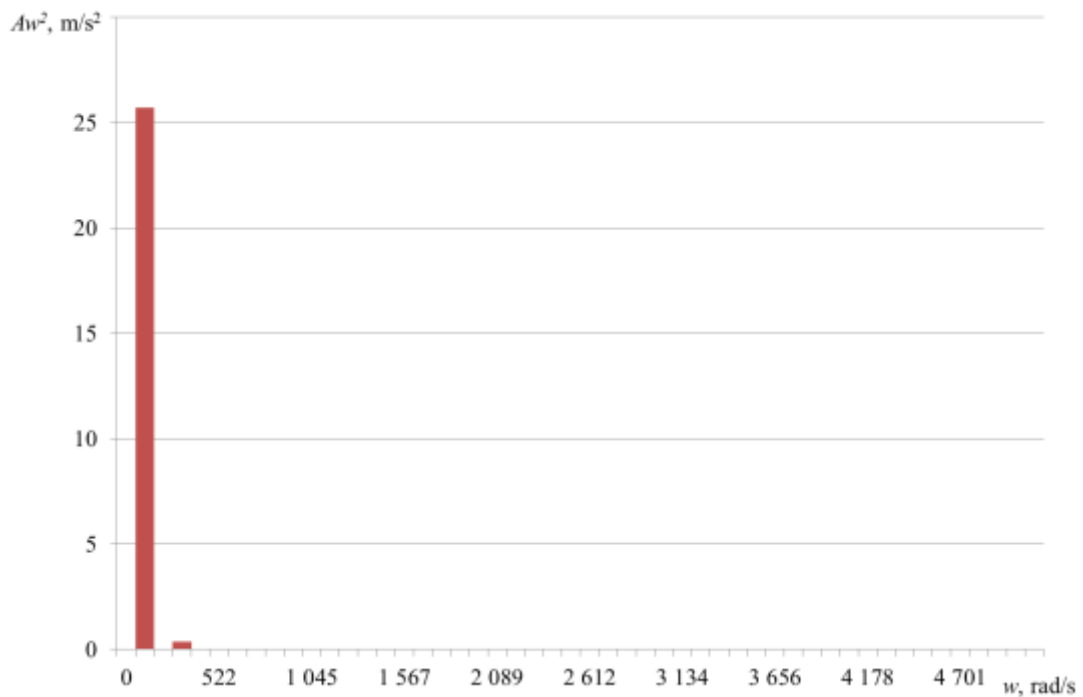

Figure 9 - Spectrogram of accelerating oscillations of the feeder with harmonic excitation

Researches of the effect of increasing the load on feeders from bulk media indicate that the amplitudes of of their working bodies are decrease (Fig. 10 and Fig. 11) and the amplitudes of the feeder impactor oscillations with vibro-impact excitation are significantly increase.

Compared with the oscillations in the basic parameters (Fig. 2), when the mass of the feeder tray increase from the mass of the attached mass of bulk medium up to 3000 kg (Fig. 10), the amplitudes of the impactor oscillations increases more than 2.5 times from 1.6 to 4.14 mm, and its acceleration from 60 m/s^2 (Fig. 4) to 143 m/s^2

(Fig. 12). The phases of oscillations of the masses also change, oscillations become antiphase (Fig. 10). That is, with increasing load on the feeder tray from the attached mass of the bulk medium, the phases of the vibration of the tray and the impactor of the vibro-impact drive are changed from just sinphase (Fig. 2) to the antiphase (Fig. 10). Dynamic gaps to one-side elastic bonds also grow (Fig. 13). The vibro-impact drive of the feeder adapts to the change in the load on the tray on the mass of the bulk medium on it, by increasing the oscillation parameters and the kinetic energy of the impactor from 2.14 (Fig. 14) to 17.4 J (Fig. 14).
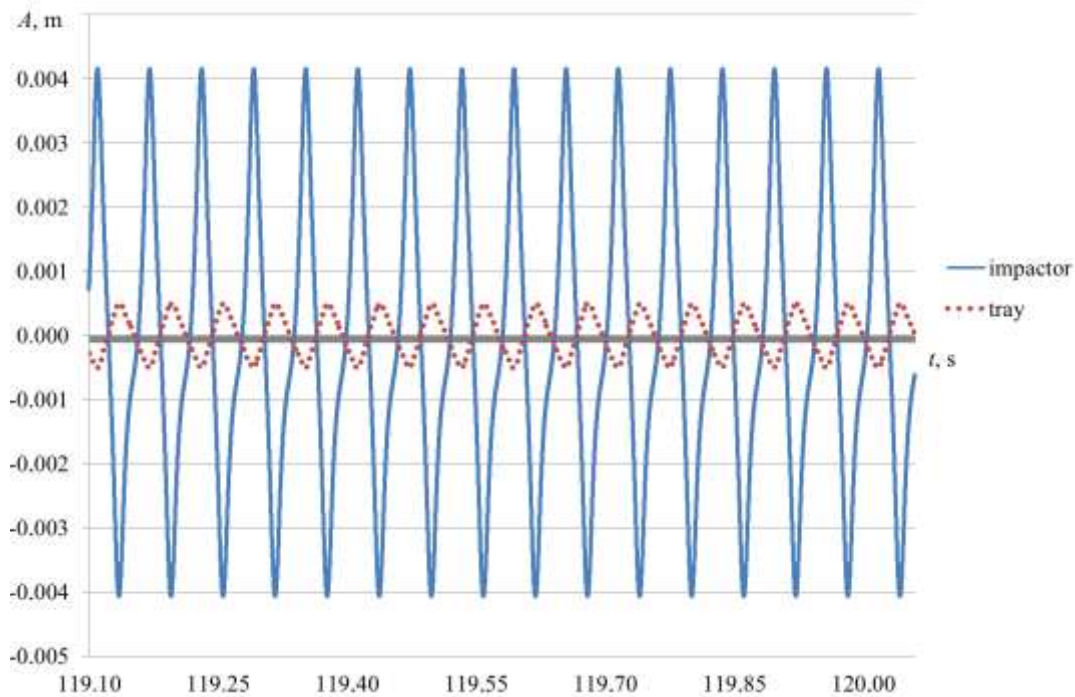

Figure 10 - Oscillogram of stationary oscillations of a feeder masses with vibro-impact excitations at a mass of a tray with an attached mass of bulk medium equal to 3000 kg
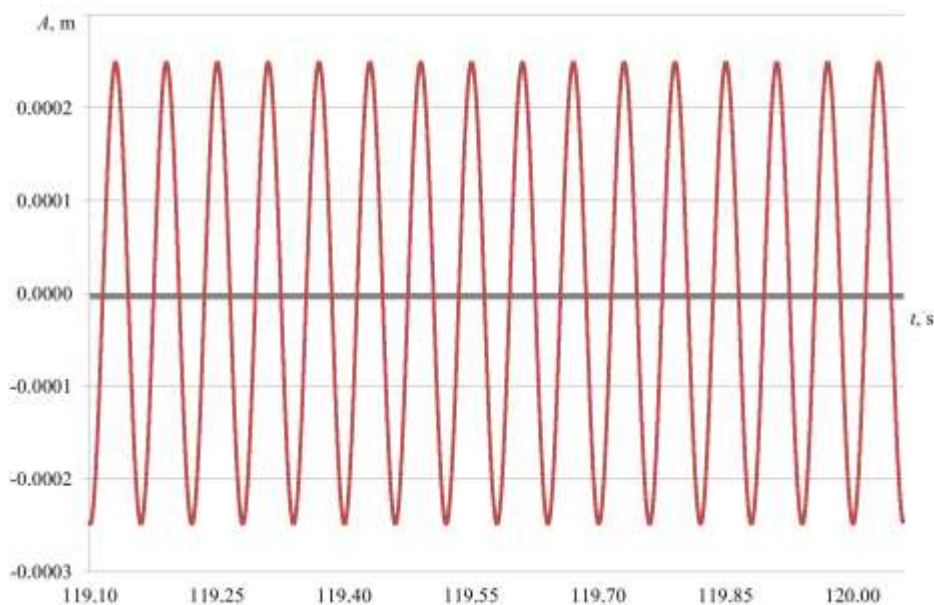

Figure 11 - Oscillogram of stationary oscillations of a feeder with harmonic excitation at its mass with an attached mass of bulk medium equal to 3000 kg

In Figure 15 shown the dependences of changes in mass accelerations of a feeder with an adaptive multi-frequency drive when the mass of the tray increases due to the attached mass of the bulk medium up to 3000 kg, in increments of 300 kg.
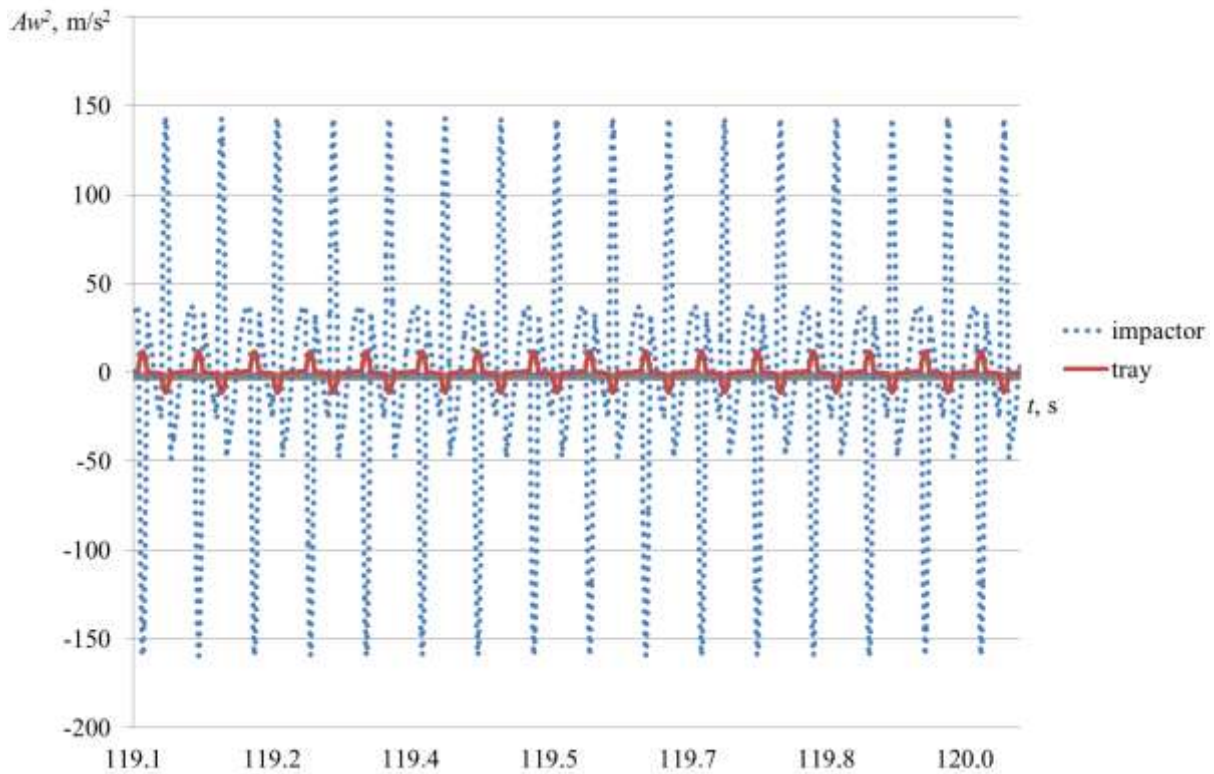

Figure 12 - Oscillogram of accelerated feeder mass with polyfrequency excitation at the mass of the tray with the attached mass of bulk medium equal to 3000 kg
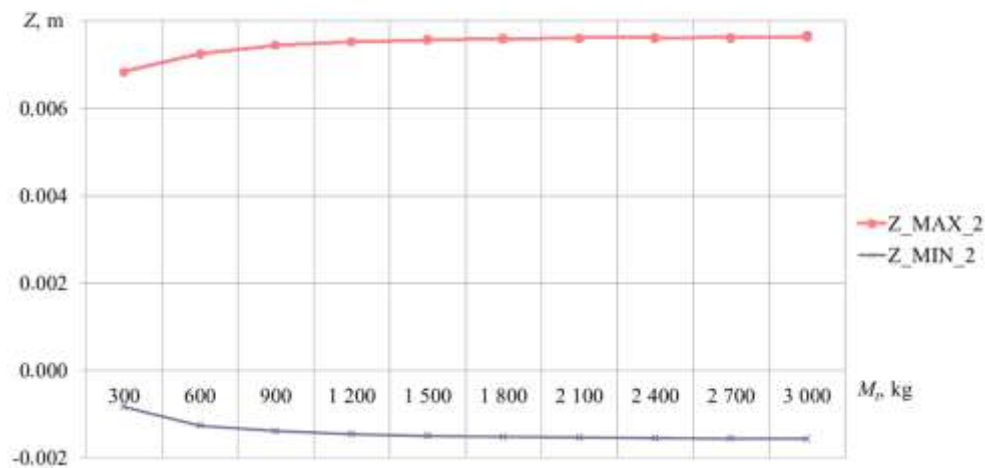

Figure 13 - Dependencies of the growth of dynamic gaps to one-side elastic bonds with increasing load on the feeder tray

Analysis of dependencies on Figure 15 shows that at the interval of mass growth from 300 to 600 kg there is a characteristic point of intersection of dependencies, in which the acceleration of the tray and impactor are aligned and with further growth of mass, the acceleration of the tray become smaller than acceleration of the impactor. It is obvious that the settings of the feeder should be such that the nominal vibrations of the bulk medium the acceleration of the tray were greater than the acceleration of the

impactor, and far from the limits on the conditions of critical permissible loads. In the case of short-term critical loads, when the acceleration of the impactor is maximal, their values should approach the permissible ones. Note, that under any load, due to stationary gaps between the impactor and one-side elastic bonds, the vibro-impact drive will excite the feeder tray and the bulk medium on it. The parameters of this excitation depend on the parameters of interaction between the impactor and the tray.

    With increasing load on the feeder tray with vibro-impact excitation the spectrum of oscillations of the masses also changes (Fig. 16).
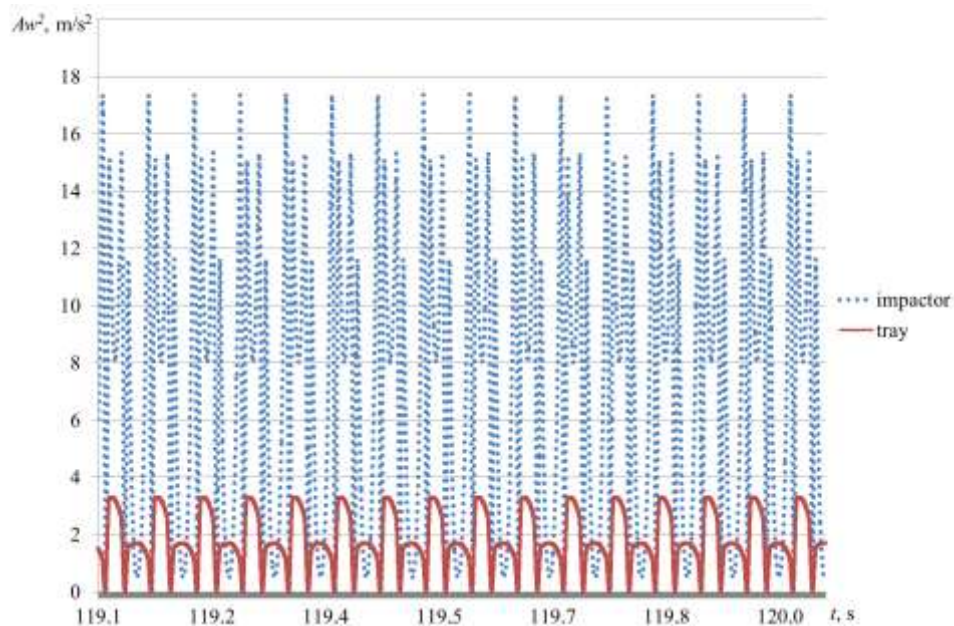

Figure 14 - Changing the kinetic energy of stationary oscillations of the feeder masses with vibro-impact excitation at the mass of the tray with the attached mass of bulk medium equal to 3000 kg

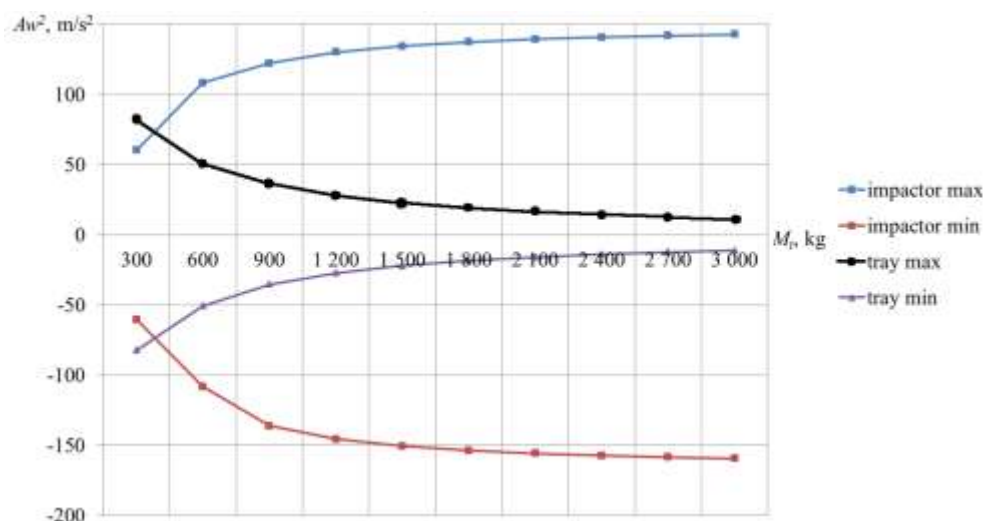Figure 15 - Dependences of mass accelerations of a feeder with an adaptive vibro-impact drive at the load is increased from the attached mass of bulk medium

    In comparison with the spectrum of oscillations acceleration under the basic parameters of the model (Fig. 8), additional harmonics are excited in the oscillations of the feeder with increasing the load. Unlike the spectrum in Figure 8, the spectrum

in Figure 17 is represented by a discrete set of superharmonic frequencies, which are in aliquot ratios to the frequency of forced oscillations ω and are determined by the dependence $\omega_k = k\omega$, where $k = 1, 2, 3, \ldots$. That is, when the load on the feeder tray increases, the spectral density of the frequency spectrum of oscillations increases due to the increase of the number of harmonics generated in the oscillatory feed system. Also, as in Figure 8 in the spectrum of the impactor oscillations in Figure 16 there are harmonics with amplitudes of accelerations which are larger than the amplitude of excitations at the frequency of forced oscillations, which indicates a resonant amplification of the impactor oscillations [8-11].
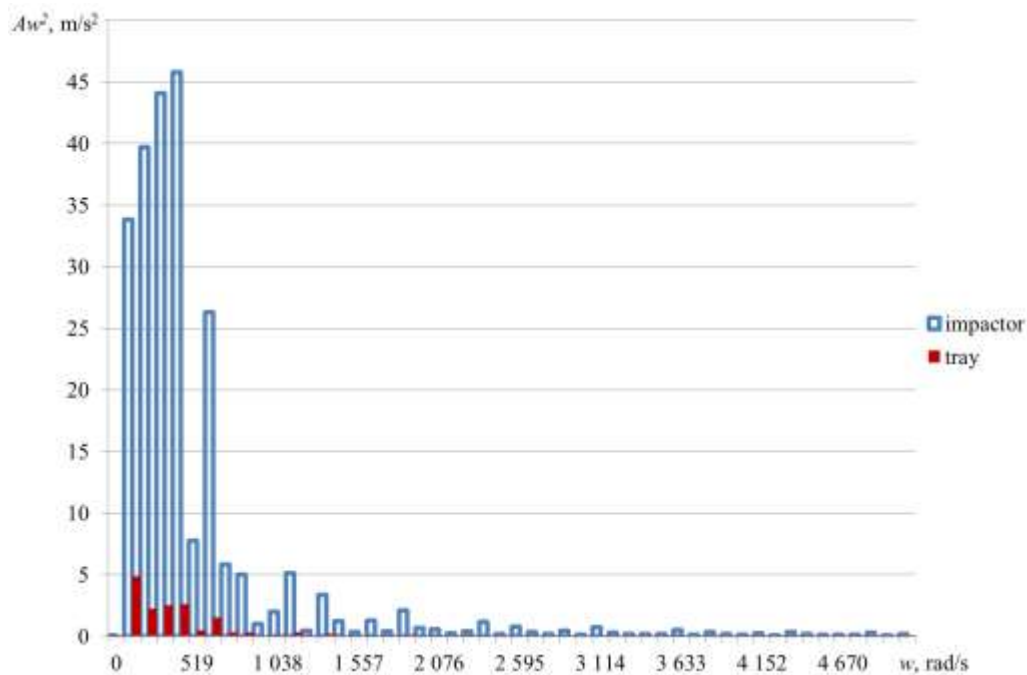

Figure 16 - Spectrogram of the feeder masses oscillations with vibro-impact excitation at the mass of the tray with the attached mass of bulk medium equal to 3000 kg

Researches of the spectrum of oscillations show that in the deformation of one-side elastic bonds, the reaction of their elastic elements is multifrequency, which, accordingly, affects the impactor oscillations and the feeder tray, by changing their speed and acceleration and excitation of additional harmonics of oscillations imposed on the forced oscillations of the feeder. It has been shown that the spectral density of these oscillations and the amplitude of additional excited harmonics depend on the interaction parameters of a feeder tray with one-side elastic elements.

It should be noted that multifrequency oscillations facilitate to the increase of the efficiency of the technological processes of vibration output of bulk media from bunkers, or of wining ore blocks, by increasing the intensity and depth of excitation of bulk media in a bunker or an ore block above the feeder tray and the speed of its motion. The depth of excitation of the massive of the reflacted rock mass in ore block or bulk media in a container or bunker, depends on its physical and mechanical properties and parameters of vibrational action excited by the feeder. Vibration in a lump bulk medium at harmonious excite of vibromotor of a typical feeder and its

maximum accelerations up to 100 m/s^2 extends over a distance not exceeding 2-3 m [11], which is due to the energy dissipation in the massive. Obviously, in order to reduce the number of freezing bulk medium in containers or reflacted ores in the ore block and, accordingly, increase the productivity of the vibration output, it is necessary to ensure an increase in the depth of excitation of the bulk media above the feeder tray. When multifrequency excitation of an massive of bulk media, conditions are created which promote the processes of vibration output. The massive of bulk media in a bunker or an ore block after crushing by an explosion, under the action of gravity, are in an intense state. In this case, the forces of friction, elastic deformation and various forces of elastic linkage (adhesion, electrostatic, coulomb, and others) act on particles or pieces of the medium. For the shift and realisation of relative displacements between particles in an equilibrium intensive massive, it is necessary to provide additional energy to it, for example, due to the vibrational impact on the massive located above the vibrating feeder tray. Obviously, the most favorable conditions, and the smallest amount of required energy entering the massive to overcome the various forces acting on the particles of the bulk medium in the massive, and the realisation of their relative displacements, are those in which the vibrational excitation frequencies coincide with their own oscillation frequencies of particles under the action of these forces. Given the diversity of bonds between particles, the constant and wide change in the parameters and conditions of their interaction, to overcome the forces of cohesion and the relative shift of particles in the massive, it is necessary to excite the multifrequency oscillations of a bulk medium in an massive whose spectra overlap the own frequencies of oscillations of a bulk medium bound by the elastic forces. Vibration frequencies close to the own frequencies of particles bound with the elastic forces, with a minimum of oscillation energy brought to the massive, will contribute to their relative motion and increase the depth of vibration spreading in the massive, to reduce the freezing of bulk media in bunkers or ore blocks when vibration ore output by feeders. Accordingly, the most favorable conditions for the displacement of particles in an massive of bulk media, which are exposed to various elastic forces of contact interaction, and the smallest amount of required energy for this, occures at multifrequency force influence on an massive with a overall and wide spectrum of frequencies that overlaps its own oscillation frequencies of bulk medium particles connected by contact elastic forces. In this regard, the multifrequency oscillation of the vibration feeder tray is conducive for vibrational output of bulk media from bunkers or ore blocks. At multifrequency excitation, an increase in the level and depth of vibration spreading in an massive of bulk media, the destruction of frozen bulk material, static equilibrium bonds, and reduction the forces of adhesive and friction between particles of bulk material.

Conclusions. The results of the researches of the proposed model of the vibration feeder show that at the vibro-impact multifrequency excitation the accelerations of the feeder tray (Fig. 4) in comparison with the oscillations of the typical feeder with harmonic excitations (Fig. 5), with the same basic parameters, increase more than three times. When increasing the load on the tray from the bulk medium at output from the bunkers or reflected ore from the chambers of ore blocks, the vibro-impact

drive adapts to this by increment of parameters and the oscillations spectrum of the vibro-impact drive impactor, which, respectively, are transmitted to the tray and bulk media. So when changing the mass of the tray with the attached mass of bulk medium from 300 to 3000 kg, its acceleration increases by 2.4 times, from 60 to 143 m/s^2 (Fig. 15). At the same time, due to the increase in the number of harmonics excited in the oscillatory system of the feeder as a result of vibration oscillations, the spectral density of the frequency spectrum of oscillations increases, which contributes to the growth of the intensity and depth of the dissemination of vibrational influence in the massive of bulk media above the tray. Thus, the vibro-impact drive of the vibrating feeder is adapted to change the load on the feeder tray from the bulk medium on it. All this compared with the harmonic excitation of a typical feeder tray will lead to an increase in the productivity and efficiency of the vibration output of bulk media from bunkers or chambers of ore blocks in the underground mining of minerals.

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Анотація. Об'єктом дослідження є вібраційні живильники з адаптивним віброударним приводом, характерною особливістю яких є полічастотне збудження та саморегулювання параметрів коливань, включаючи частотний спектр, при зміні маси технологічного навантаження. В основі вирішення задачі лежить збудження віброударних коливань робочого тіла (лотка) живильника, при якому в його вібраційному приводі реалізуються резонансні коливання, інтенсивність і частотний спектр яких збільшуються зі збільшенням технологічного навантаження на лоток подачі, що відрізняє такі вібратори від тих, що використовуються в типових живильниках. При збільшенні навантаження на лоток сипучого середовища при випуску з бункерів або відбитої руди з камер рудних блоків віброударний привід адаптується до цього за рахунок збільшення параметрів та спектра коливань віброударного приводу, які відповідно передаються на лоток та сипуче середовище. Віброударні системи з таким віброзбудженням з параметрами та динамічним навантаженням, в порівнянні з аналогами, мають залежно від свого призначення в 3-8 разів менше енергоспоживання та в 3-5 разів збільшують інтенсивність коливань. Так, при зміні маси лотка з масою сипучого середовища від 300 до 3000 кг його прискорення збільшується в 2,4 рази, з 60 до 143 м/с². При багаточастотному збудженні збільшуються рівень і глибина розповсюдження вібрації в масиві сипучого середовища, руйнування застиглого сипучого матеріалу, статичні рівноважні зв'язки, відбувається зменшення сил адгезії і тертя між частинками сипкого матеріалу. Таким чином, віброударний привід вібраційного живильника адаптований до зміни навантаження на лоток подачі з сипучим середовищем на ньому. У той же час, завдяки збудженню полічастотних коливань, поліпшується рухливість технологічного середовища, яке обробляється, що особливо важливо для контролю замерзання та адгезії матеріалів у бункерах або в стовпі відбитої гірської маси під час її випуску вібраційними живильниками з камер очисних блоків при підземному видобутку корисних копалин. Все це порівняно з гармонічним збудженням типового живильника призведе до підвищення продуктивності та ефективності вібраційного випуску сипучих середовищ із бункерів або камер рудних блоків.

Ключові слова: підземний видобуток корисних копалин, вібраційні живильники, сипке середовище, віброударний адаптивний привід, полічастотне збудження, параметри коливань, частотний сектр, енергоспоживання.

Аннотация. Объектом исследования являются вибрационные питатели с виброударным адаптивным приводом, характерной особенностью которых является поличастотное возбуждение и самонастройка

параметров колебаний, в том числе частотного спектра, при изменении массы технологической нагрузки. В основе решения задачи лежит возбуждение виброударных колебаний рабочего органа (лотка) питателя, при котором в его вибрационном приводе реализуются резонансные колебания, интенсивность и частотный спектр которых возрастают с увеличением технологических нагрузка на лоток подачи, что отличает такие вибраторы от тех, которые используются в типовых питателях. При увеличении нагрузки на лоток от сыпучей среды при выпуске из бункеров или отбитой руды из камер рудных блоков виброударный привод приспосабливается к этому путем адаптации параметров и спектра колебаний вибро-ударного привода, которые соответственно передаются на лоток и сыпучую среду. Виброударные системы с такими вибровозбуждением, параметры и динамическая нагрузка которых идентичны аналогам, имеют в 3–8 раз меньшую потребляемую мощность и в 3–5 раз увеличивают интенсивность колебаний. Так, при изменении массы лотка с присоединяемой массой сыпучей среды от 300 до 3000 кг его ускорение увеличивается в 2,4 раза, с 60 до 143 м/с^2. При многочастотном возбуждении происходит увеличение уровня и глубины распространения вибрации в массиве сыпучих сред, разрушение слипшегося сыпучего материала, уменшаются статические равновесные связи и силы адгезии и трения между частицами сыпучего материала.Таким образом, виброударный привод вибрационного питателя приспособлен к изменению нагрузки на лоток питателя с сыпучей средой на нем. В то же время благодаря поличастотному возбуждению улучшается подвижность обрабатываемой технологической среды, что особенно важно для контроля замерзания и адгезии материалов в бункерах или в столбе налипшей горной массы во время выпуска ее вибропитателями из камер очистных блоков при подземной добыче полезных ископаемых. Все это по сравнению с гармоническим возбуждением типового питателя приведет к увеличению производительности и эффективности вибрационного выпуска сыпучих сред из бункеров или камер рудных блоков.

Ключевые слова: подземная добыча полезных ископаемых, вибрационные питатели, сыпучая среда, виброударный адаптивный привод, поличастотное возбуждение, параметры колебаний, частотный спектр, энергопотребление.