ФІЗИКА ТВЕРДОГО ТІЛА, ЗБАГАЧЕННЯ КОРИСНИХ КОПАЛИН

UDC 622.74: 621.928.235

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CONDITIONS OF REPLACING A SINGLE-FREQUENCY VIBRO-EXCITER WITH A DUAL-FREQUENCY ONE IN THE FORM OF PASSIVE AUTO-BALANCER

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УМОВИ ЗАМІНИ ОДНОЧАСТОТНОГО ВІБРОЗБУДНИКА НА ДВОЧАСТОТНИЙ У ВИГЛЯДІ ПАСИВНОГО АВТОБАЛАНСИРУ

Purpose. To define conditions of replacing a single-frequency vibro-exciter with a dual-frequency one in the form of a passive auto-balancer in the vibration machine with a translatory rectilinear motion of the box.

Methodology. Elements of the theory of vibration machines and classical mechanics for creation and the analysis of mechanic-mathematical models of the basic and modernized machines are used.

Findings. Parameters of the modernized machine (with the dual-frequency vibro-exciter in the form of the passive auto-balancer) are selected analytically at which the amplitude and frequency of slow vibrations of its box are equal to the amplitude and forced oscillation frequency of the box of the basic machine (with the single-frequency inertial vibro-exciter) with a translatory rectilinear motion of the box.

The results allow calculating parameters of resonant vibration machines with dual-frequency vibro-exciters in the form of passive auto-balancers.

It is established that the frequency of slow vibrations of the modernized machine is influenced by a stiffness coefficient of supports while the amplitude of these vibrations is influenced by a coefficient of forces of viscous resistance of the support.

Modernization of the screen of GIL 42 allowed reducing the total mass of the rotating parts of the vibro-exciter by 3.4 times.

Originality consists in definition of conditions of replacing the single-frequency inertial vibro-exciter in the vibration machine with translatory rectilinear motion of the box with the dual-frequency vibro-exciter in the form of the passive auto-balancer.

Practical value consists in the possibility to design the new power efficient and high-performance vibration machines combining advantages of resonant and dual-frequency vibration machines.

Keywords: *inertial vibro-exciter, dual-frequency vibro-exciter, unbalance, vibration machine, auto-balancer, screen*

Introduction. Power efficient resonant vibration machines are the most perspective among vibration machines [1].

By means of the small drive, the resonant mode of vibrations gives possibility to set in motion boxes of screens, great in the area, with the minimum consumptions of energy [2].

Substantial increase in productivity of vibration machines is provided by using two and more frequency vibro-exciters in them [3]. So, at vibrations of the box (platform, sieve, etc.) with lower frequency the main technology process in the form of separation, boltings, etc. is carried out. Vibrations with higher frequency provide self-cleaning of the box and change of mechanical properties of the processed material that increases productivity of the main technology process [4].

In work [5] influence of multi-frequency vibrations of screen surfaces of vibrating screens with continuous distribution of modes of frequency on efficiency of separation of finely dispersed bulk materials is considered. Resistance of multi-frequency vibrations to change of technology loading and dynamic parameters of the screen is proved.

In work [6] the possibility of forming the significant polyharmonic vibrations by input of the vibration machine in the superharmonic resonance of the second order is established.

The conducted stand research studies completely confirmed availability of essential technology advantages of crushing materials in the biharmonic field of vibrations in comparison with traditional harmonic oscillation [7].

Industrial tests showed considerable advantages of the screen with the biharmonic operating mode compared to the screen with the harmonious mode [8].

The vibration equipment with the biharmonic mode of vibrations allows receiving high technology rates and can be applied to the broad range of fineness and quality of coaly materials for the purpose of receiving commodity products [9].

In work [10] the possibility of simultaneous oscillating, rotary and shock motion is considered. It is offered to use the shock elements allowing one to receive an additional impact on particles of the material which is difficult to yield to screening.

Therefore, creation of the vibration machines integrating advantages of resonant and dual-frequency vibration machines is topical. It will allow receiving power efficient machines with the increased productivity.

The existing ways of excitation of dual-frequency vibrations do not allow arranging automatically the lowest frequency under the resonance frequency of vibrations of the box.

Analysis of the recent research. The authors offered the effective way of excitation of dual-frequency vibrations by means of passive auto-balancers. Their use is based on the specific mode of motion of the rotor with the auto-balancer (biharmonic one), arising at small resistance forces to motion of corrective weights in relation to the rotor. In this mode the rotor rotates with the above resonance frequency, and corrective weights in the auto-balancer cannot catch up with it, practically gather and rotate with the smallest resonance frequency of vibrations of the rotor, rather than adapt to it.

For today, the theory of such vibro-exciters has not been developed, there are no recommendations on design and calculation of general parameters of vibration machines with vibro-exciters in the form of passive auto-balancers.

Unsolved aspects of the problem. In work [11] kinematic schemes of the modernized vibration machines are offered and by means of 3D modeling it is established that within a wide range of parameters of the machine, the auto-balancer works as two separate independent vibro-exciters. The first is formed by corrective weights. It creates slow resonant vibrations of the box for performance of the main technology process. The second vibro-exciter is formed by unbalance mass on the auto-balancer housing. It creates fast vibrations for performance of an auxiliary process (cleaning of the working surface of the sieve, etc.).

The possibility of change of characteristics of dual-frequency vibrations has been established by change of total mass of corrective weights, unbalance mass on the housing of the auto-balancer, and frequency of rotation of the shaft on which the auto-balancer is mounted.

Ranges of parameters which ensure occurrence of dual-frequency vibrations are defined. Assumptions concerning the origins of dual-frequency vibrations are formulated [12].

According to results of 3D modeling, the stand at which the box carries out translational vertical motions is created. The conducted research studies confirmed the possibility of use of the auto-balancer for excitation of dual-frequency vibrations.

Unresolved are questions of selecting parameters of the modernized machine with the dual-frequency vibro-exciter. For this purpose it is necessary to formulate a criterion of replacement of the single-frequency vibro-exciter with a dual-frequency one, to construct Mechanics and Mathematics models of the basic and modernized machines, to investigate these models.

Objectives of the article. The article aims to define conditions of replacing the single-frequency vibro-exciter with a dual-frequency one in the form of the passive auto-balancer in the vibration machine with translatory rectilinear motion of the box.

Presentation of the main research. ***Description of the system.*** Let us consider the basic machine with the single-frequency vibro-exciter (Fig. 1*, a*).

Vertical displacement of the box is determined by the coordinate X. We designate stiffness coefficient of the support as C_b. We designate coefficient of viscosity of the support as H_b. The single-frequency vibro-exciter is used as the source of vibrations. It consists of unbalance mass M_b, rotating at rotating speed of the rotor ω_b. Position of unbalance mass M_b is defined by radius $r_b = r$ and angle $\omega_r t$.

In the modernized machine (Fig. 1, *b*), the dual-frequency vibro-exciter is used. It consists of unbalance mass $\tilde{D}$ on the housing of the auto-balancer and corrective weights of total mass $\tilde{M}$. We denote stiffness coefficient of the support as C and coefficient of viscosity of the support as H.

With the auto-balancer working as a vibro-exciter, its corrector weights almost come together. Therefore, we shall consider them as a single unit. Corrective weights rotate with the lowest frequency of eigentones of the box of the machine ω_{rez}. Their position is defined by radius r_m and angle $\omega_{rez}t$. Unbalance mass $\tilde{D}$ rotates at rotating speed of the rotor ω_r. Its position is defined by radius r_d and angle $\omega_r t$.

We believe that the exciters weight do not significantly affect the total weight M_Σ of the box of basic or modernized machine.

The parameters characterizing dynamics of the basic and modernized machines are specified in Table 1.

Differential equation of motion of the modernized machine in the dimensional and dimensionless form. With use of the theorem of motion of center of mass, it is possible to deduce the following differential equation of translatory motion of the box (without gravity forces)

$$M_\Sigma \ddot{X} + H\dot{X} + CX = r_m \tilde{M} \omega_{rez}^2 \sin \omega_{rez} t + r_d \tilde{D} \omega_r^2 \sin \omega_r t, \quad (1)$$

where points over the coordinate *of X* designate time derivatives, and other designations are interpreted in Table 1.

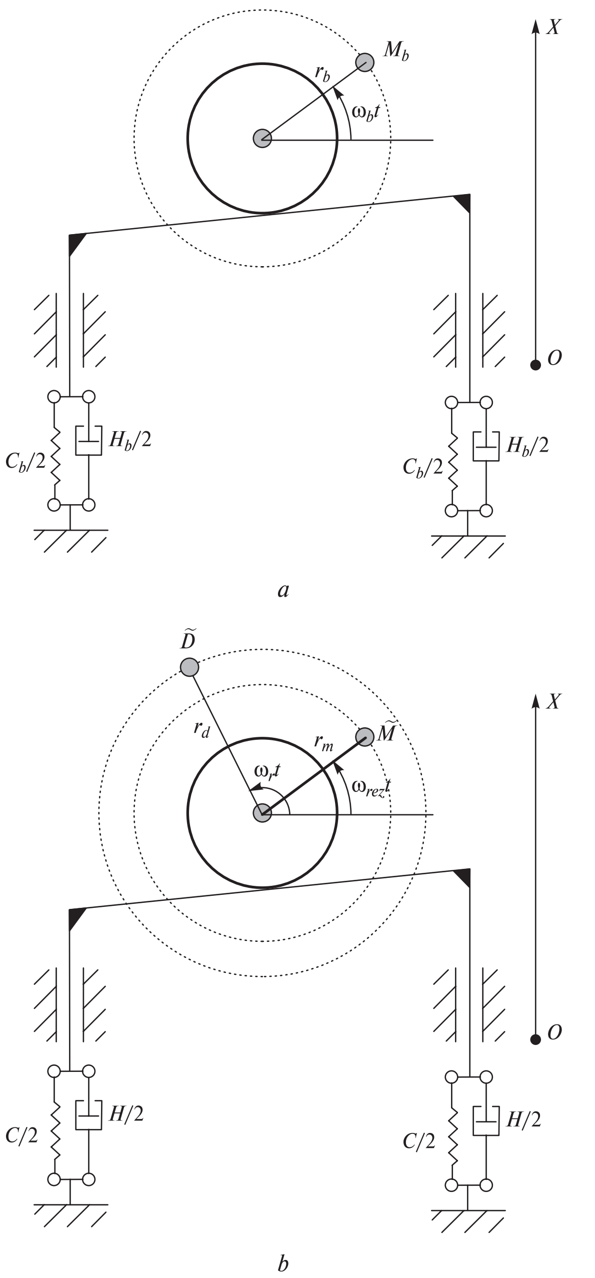

Fig. 1. Scheme of the machine:
a – basic one; b – modernized one

Thus, the dynamics of the modernized machines is influenced by nine dimensional parameters

$$M_\Sigma, H, C, r_m, r_d, \tilde{M}, \omega_{rez}, \tilde{D}, \omega_r.$$

Let us enter the new parameters (specified to the radius *of r* of mass of corrective weights and the unbalance on the auto-balancer housing)

$$M = \frac{r_m}{r}\tilde{M}; \quad D = \frac{r_d}{r}\tilde{D}. \qquad (2)$$

Then the equation (1) will take the form

$$M_\Sigma \ddot{X} + H\dot{X} + CX = r(M\omega_{rez}^2 \sin\omega_{rez} t + D\omega_r^2 \sin\omega_r t). \qquad (3)$$

For further simplification of the form, the differential equation of motion will be written as a dimensionless form. Let us enter characteristic scales: l^* – for distance; t^* – for time. Let us enter new dimensionless variables: ξ – distance; τ – time. Then

Table 1

Machine parameters

Parameter\machine	basic	modernized
Stiffness coefficient of the support	C_b, N/m	C, N/m
Viscosity coefficient of the support	H_b, Nc/m	H, Nc/m
Unbalance mass	M_b, kg	–
Machine box lump	M_Σ, kg	M_Σ, kg
Circular rotating speed of the unbalance/corrective weights	ω_b, rps	$\omega_{rez} = \omega_b$, rps
Circular rotating speed of unbalance mass	–	ω_r, rps
Radius of rotation of the unbalance	$r_b = r$, m	–
Radius of rotation of unbalance mass	–	r_d, m
Distance from the longitudinal axis of the rotor to the center of mass of the corrective weight	–	r_m, m
Unbalance mass	–	$\tilde{D}$, kg
Total mass of corrective weights	–	$\tilde{M}$, kg

$$X = l^*\xi; \quad t = t^*\tau; \quad \dot{X} = \frac{l^*}{t^*}\xi'; \quad \dot{X} = \frac{l^*}{t^*}\xi', \qquad (4)$$

where the stroke behind value means the derivative on dimensionless time.

Substituting (4) in (3), after multiplication by $\dfrac{t^{*2}}{M_\Sigma l^*}$, we will deduce

$$\xi'' + \frac{Ht^*}{M_\Sigma}\xi' + \frac{Ct^{*2}}{M_\Sigma}\xi = \frac{r}{l^*}\left[\frac{M}{M_\Sigma}\omega_{rez}^2 t^{*2}\sin(\omega_{rez}t^*\tau) + \frac{D}{M_\Sigma}\omega_r^2 t^{*2}\sin(\omega_r t^*\tau)\right]. \qquad (5)$$

Let us choose characteristic scales for the purpose of the maximum simplification of the form of differential equation of motion. Let

$$t^* = \sqrt{\frac{M_\Sigma}{C}}; \quad l^* = r\frac{M}{M_\Sigma}\omega_{rez}^2 t^{*2}; \quad \left(\frac{r}{l^*}\frac{M}{M_\Sigma}\omega_{rez}^2 t^{*2} = 1, \quad \frac{r}{l^*} = \frac{M_\Sigma}{M\omega_{rez}^2 t^{*2}}\right). \qquad (6)$$

Then the equation (5) will take the form

$$\xi'' + \frac{H}{\sqrt{M_\Sigma C}}\xi' + \xi = \sin(\omega_{rez}t^*\tau) + \frac{D\omega_r^2}{M\omega_{rez}^2}\sin(\omega_r t^*\tau). \qquad (7)$$

In the vibro-exciter formed by corrective weights the latter ones get stuck at the resonance frequency of vibrations of the box. Therefore, with small forces of viscous resistance in support

$$\omega_{rez} \approx \sqrt{\frac{C}{M_\Sigma}} = \frac{1}{t^*}; \quad \mathrm{x}t^* \approx \frac{1}{\omega_{rez}}. \tag{8}$$

Substituting (8) in (7) we finally deduce such differential equation of motion of the machine in the dimensionless form

$$\xi'' + 2\tilde{h}\xi' + \xi = \sin\tau + \tilde{d}n^2 \sin n\tau, \tag{9}$$

where such dimensionless parameters are entered

$$\tilde{h} = \frac{1}{2}\frac{H}{\sqrt{M_\Sigma C}}; \quad n = \frac{\omega_r}{\omega_{rez}}; \quad \tilde{d} = \frac{D}{M}. \tag{10}$$

Thus, the dynamics of the machine is influenced by three significantly different dimensionless parameters: $\tilde{h}$ characterizes resistance forces in supports; n characterizes the ratio of the frequencies excited by the vibro-exciter; $\tilde{d}$ specifies the ratio of unbalance masses.

The similarity theory allows suggesting that dual-frequency vibrations will only occur within a certain range of the dimensionless parameters.

Steady state motion of the modernized machine. Direct substitution can verify that a particular solution of the differential equation (9), corresponding to the steady state motion of the machine, is the sum of two such components

$$\xi_{ss}(\tau) = \xi_{rez}(\tau) + \xi_r(\tau), \tag{11}$$

where

$$\xi_{rez}(\tau) = -\frac{\cos\tau}{2\tilde{h}}; \tag{12}$$

$$\xi_r(\tau) = -\frac{\tilde{d}n^2}{(n^2-1)^2 + 4h^2n^2}[(n^2-1)\sin n\tau + 2\tilde{h}n\cos n\tau]. \tag{13}$$

The component (12) describes slow vibrations of the box with natural resonance frequency. The component (13) describes fast vibrations of the box with rotating speed of the rotor.

In a dimensional form, the steady state motion of the machine (11) has the form

$$\begin{aligned} X(t) = l^*\xi_{ss}(\tau) = -\frac{Mr\omega_{rez}}{H}\cos(\omega_{rez}t) - \\ -\frac{Dr\omega_r^2(M_\Sigma\omega_r^2 - C)}{H^2\omega_r^2 + (M_\Sigma\omega_r^2 - C)^2}\sin(\omega_r t) - \\ -\frac{DrH\omega_r^3\cos(\omega_r t)}{H^2\omega_r^2 + (M_\Sigma\omega_r^2 - C)^2}. \end{aligned} \tag{14}$$

In case of small forces of viscous resistance in supports, H is relatively small and the law of biharmonic vibrations (14) can be presented approximately in the form

$$\begin{aligned} X(t) \approx -\frac{Mr\omega_{rez}}{H}\cos(\omega_{rez}t) - \frac{Dr\omega_r^2}{M_\Sigma\omega_r^2 - C}\sin(\omega_r t) = \\ = -\frac{\tilde{M}r_m\omega_{rez}}{H}\cos(\omega_{rez}t) - \frac{\tilde{D}r_d\omega_r^2}{M_\Sigma\omega_r^2 - C}\sin(\omega_r t). \end{aligned} \tag{15}$$

It follows from formulas (14, 15) that amplitude of slow vibrations of the box is directly proportional to the total unbalance of corrective weights $\tilde{M}r_m$, and amplitude of fast vibrations of the box is directly proportional to the unbalance $\tilde{D}r_d$ on the auto-balancer housing.

Differential equation of motion of the basic machine in the dimensional and dimensionless forms. Using the theorem of motion of center of mass, it is possible to receive the differential equation of motion of the box of the basic machine in the dimensional form

$$M_\Sigma\ddot{X} + H_b\dot{X} + C_bX = rM_b\omega_b^2\sin(\omega_b t). \tag{16}$$

For reduction of this equation to the dimensionless form we use the same dimensionless variable ξ from (4) and the same scales (6) for distance and time.

Substituting (4) in (16), after multiplication by $\frac{t^{*2}}{M_\Sigma l^*}$, we will deduce

$$\xi'' + \frac{H_bt^*}{M_\Sigma}\xi' + \frac{C_bt^{*2}}{M_\Sigma}\xi = \frac{t^{*2}rM_b\omega_b^2\sin(\omega_bt^*\tau)}{M_\Sigma l^*}. \tag{17}$$

As $\omega_b = \omega_{rez}$, from (8) we will deduce

$$t^* = 1/\omega_b. \tag{18}$$

Let us transform the equation (17). In its left part we substitute (18), we deduce

$$\xi'' + \frac{H_b}{M_\Sigma\omega_b}\xi' + \frac{C_b}{M_\Sigma\omega_b^2}\xi.$$

Let us introduce into consideration the resonance frequency of vibrations of the box of the basic machine

$$\omega_0 = \sqrt{C_b/M_\Sigma}. \tag{19}$$

In the right-hand side of equation (17) we substitute (6), (18), we deduce

$$\frac{rM_b\omega_b^2}{M_\Sigma\omega_b^2}\frac{M_\Sigma}{rM}\sin\tau = \frac{M_b}{M}\sin\tau.$$

Finally we deduce such differential equation of motion of the machine in the dimensionless form

$$\xi'' + 2\mu\xi' + \frac{1}{k^2}\xi = m\sin\tau, \tag{20}$$

where the following dimensionless parameters are entered

$$\mu = \frac{1}{2}\frac{H_b}{M_\Sigma\omega_b}; \quad k = \frac{\omega_b}{\omega_0}; \quad m = \frac{M_b}{M}. \tag{21}$$

Thus, the dynamics of the basic machine is influenced by three dimensionless parameters: μ character-

ISSN 2071-2227, Науковий вісник НГУ, 2017, № 1

izes resistance forces in support; k specifies ratio of frequencies; m characterizes ratio of masses.

Differential equations of motion (9) and (20) describe motions of the box, respectively, of the modernized and basic machines in identical scales of distances and time.

Steady-state motion of the basic machine. Direct substitution can verify that a particular solution of the differential equation (20), corresponding to the steady state motion of the machine, has the form

$$\xi_b(\tau) = -\frac{mk^2[(k^2-1)\sin\tau + 2\mu k^2\cos\tau]}{(k^2-1)^2+4\mu^2k^4}. \qquad (22)$$

Let us enter parameters

$$\Delta = (k^2-1)^2 + 4\mu^2k^4;$$

$$\cos\beta = \frac{k^2-1}{\sqrt{\Delta}}; \quad \sin\beta = \frac{2\mu k^2}{\sqrt{\Delta}}, \qquad (23)$$

where β is an oscillation phase. Then the law of vibrations (22), after transformations, takes the form

$$\xi_b(\tau) = -\frac{mk^2}{\sqrt{\Delta}}\sin(\tau+\beta). \qquad (24)$$

In the dimensional form, the law of vibrations (22) has the form

$$X_b(t) = -\frac{M_b r\omega_b^2}{\Delta_b}\times$$
$$\times[(M_\Sigma\omega_b^2 - C_b)\sin(\omega_b t) + H_b\omega_b\cos(\omega_b t)].$$

In the dimensional form the law of vibrations (24) has the form

$$X_b(t) = -\frac{M_b r\omega_b^2}{\sqrt{\Delta_b}}\sin(\omega_b t+\beta), \qquad (25)$$

where

$$\Delta_b = (M_\Sigma\omega_b^2 - C_b)^2 + H_b^2\omega_b^2;$$

$$\cos\beta = \frac{M_\Sigma\omega_b^2 - C_b}{\sqrt{\Delta_b}}; \quad \sin\beta = \frac{H_b\omega_b}{\sqrt{\Delta_b}}. \qquad (26)$$

At small resistance forces in supports, H_b is relatively small and the law of motion (25) takes the form

$$X_b(t) \approx -\frac{M_b r\omega_b^2}{M_\Sigma\omega_b^2 - C_b}\sin(\omega_b t). \qquad (27)$$

Selection of parameters of the modernized machine. A criterion is proposed according to which the modernized machine is to perform the basic process just as the basic machine does. For this purpose slow vibrations of the box in the modernized machine have to correspond to forced oscillations of the box of the basic machine, both in frequency and in amplitude.

We define stiffness coefficient of supports C of the modernized machine from the condition that the frequency of slow vibrations of its box ω_{rez} equals to frequency ω_b of forced oscillations of the box of the basic machine

$$\omega_b = \omega_{rez} \approx \sqrt{\frac{C}{M_\Sigma}}.$$

From this condition we find

$$C = \omega_b^2 M_\Sigma. \qquad (28)$$

In the basic machine the forced oscillation frequency of the box of the machine is less than the frequency of its eigentones ($\omega_b < \omega_0$). Therefore, rigidity of supports of the modernized machine becomes less than in basic ($C < C_b$).

For further selection of parameters of the modernized machine we use the condition that the amplitude of slow vibrations of the box of the modernized machine has to equal to amplitude of forced oscillations of the box of the basic machine.

Let us equate amplitudes in laws of vibrations (14) and (25)

$$\frac{Mr\omega_{rez}}{H} = \frac{M_b r\omega_b^2}{\sqrt{\Delta_b}}.$$

From where, taking into account that $\omega_{rez} = \omega_b$, we find

$$H = \frac{M}{M_b\omega_b}\sqrt{\Delta_b}, \qquad (29)$$

which is the exact formula for definition of coefficient of viscosity of supports.

As $H_b \approx 0$, it is offered to determine H by the approximate formula

$$H = \frac{M}{M_b\omega_b}(M_\Sigma\omega_b^2 - C_b). \qquad (30)$$

Thus, the frequency of slow vibrations of the modernized machine is regulated by selection of stiffness coefficient of supports C, and the amplitude is regulated by selection of coefficient of forces of viscous resistance H.

It is essential that the selected coefficients of stiffness and viscosity of the modernized machine supports do not depend on the rotor speed. Therefore, there is a possibility of change of this frequency without influencing slow vibrations of the box.

Example of calculations. GIL-42 screen is modernized.

Parameters of the basic machine are:

$M_\Sigma = 2300$ kg; $M_b = 125$ kg; $C_b = 45 \cdot 10^6$ N/m; $\omega_b = 16$ rps; $r_b = r = 0.2$ m.

Parameters of the modernized machine.

Parameters whose value is defined by parameters of the basic machine are:

$M_\Sigma = 2300$ kg; $\omega_{rez} = 16$ rps.

Parameters whose value can be changed are:

$M = 30$ kg; $D = 7$ kg; $r = 0.15$ m; $\omega_P > 16$ rps.

These values are within the range of parameters (at a distance from its boundaries), which ensures that dual-frequency vibrations occur [9]. Therefore, these parameters can be varied within certain limits.

At first we define stiffness coefficient of supports, N/m.

$$C = \omega_b^2 M_\Sigma; \quad C = (16\cdot 2\cdot 3.14)^2 \cdot 2300 = 23.24\cdot 10^6.$$

Having the coefficient of stiffness of supports C we can define coefficient of viscosity of supports H by the approximate formula (30), N · s/m

$$H = \frac{30 \cdot [2300 \cdot (16 \cdot 2 \cdot \pi)^2 - 45 \cdot 10^6]}{125 \cdot (16 \cdot 2 \cdot \pi)} = 51.94 \cdot 10^3.$$

Let us notice that modernization of the machine allowed reducing the total mass of the rotating parts of the vibro-exciter by 3.4 times.

The law of motion of the box of the basic machine (27), m

$$X_b(t) \approx -\frac{125 \cdot 0.15 \cdot (16 \cdot 2 \cdot \pi)^{\cdot 2}}{2300 \cdot (16 \cdot 2 \cdot \pi)^2 - 45\,000\,000} \sin(16 \cdot 2 \cdot \pi \cdot t) = \\ = -8.7 \cdot 10^{-3} \sin(100.53 \cdot t).$$

The law of motion of the box of the modernized machine (15), m

$$X(t) \approx -\frac{30 \cdot 0.15 \cdot 16 \cdot 2 \cdot \pi}{5194000} \cos(16 \cdot 2 \cdot \pi \cdot t) - \\ -\frac{7 \cdot 0.15 \cdot (\omega_r \cdot 2 \cdot \pi)^2}{2300 \cdot (\omega_r \cdot 2 \cdot \pi)^2 - 23240000} \sin(\omega_r \cdot 2 \cdot \pi \cdot t) = \\ = -8.7 \cdot 10^{-3} \cos(100.53 \cdot t) - \\ -\frac{1.96 \cdot 10^{-3} \cdot \omega_r^2}{\omega_r^2 - 256} \sin(6.28 \cdot \omega_r t).$$

As can be seen from the last two equations, the slow vibrations of the box of the modernized machine coincide with the vibrations of the box of the basic machine. From the last equation it is seen that the fast vibrations of the box of the modernized machine can be changed during operation of the machine by changing the rotor speed.

Conclusions and recommendations for further research. The criterion of replacing the single-frequency vibro-exciter with a dual-frequency one is formulated. According to the criterion the modernized machine is to perform the main process as the basic machine does. For that reason, the slow vibrations of the box of modernized machine are to correspond to the vibrations of the box of the basic machine both in amplitude and in frequency.

The simplified Mechanics and Mathematics models of the basic and modernized machines allow determining parameters of the modernized machine which provide this.

It is established that:

- the amplitude of slow vibrations of the box is directly proportional to the total unbalance of corrective weights $\tilde{M}r_m$, while the amplitude of fast vibrations is directly proportional to the unbalance $\tilde{D}r_d$ on the auto-balancer housing;

- frequency of slow vibrations of the modernized machine is regulated by selection of stiffness coefficient of supports C, whereas the amplitude is regulated by selection of coefficient of forces of viscous resistance of supports H.

Modernization of the screen of GIL 42 allowed reducing the total mass of the rotating parts of the vibro-exciter by 3.4 times.

The received findings allow calculating parameters of resonant vibration machines with dual-frequency vibro-exciters in the form of passive auto-balancers.

In the further research it is planned to construct a precise mechanics and mathematical model of the modernized machine to search ranges in the space of dimensionless parameters, providing excitation of dual-frequency vibrations.

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Мета. Визначити умови заміни одночастотного віброзбудника на двочастотний у вигляді пасивного автобалансира у вібромашині з прямолінійним поступальним рухом короба.

Методика. Використання елементів теорії вібромашин і теоретичної механіки для створення й аналізу механіко-математичних моделей базової та модернізованої машин.

Результати. Аналітично підібрані параметри модернізованої машини (із двочастотним віброзбудником у вигляді пасивного автобалансира), за яких амплітуда й частота повільних коливань її короба рівняється амплітуді й частоті вимушених коливань короба базової машини (з одночастотним інерційним віброзбудником) із прямолінійним поступальним рухом короба.

Отримані результати дозволяють розраховувати параметри резонансних вібромашин із двочастотними віброзбудниками у вигляді пасивних автобалансирів.

Установлено, що на частоту повільних коливань модернізованої машини впливає коефіцієнт жорсткості опор, а на амплітуду – коефіцієнт сил в'язкого опору.

Модернізація грохоту ГІЛ 42 дозволила зменшити сумарну масу обертових частин віброзбудника у 3, 4 рази.

Наукова новизна. Полягає у визначенні умов заміни звичайного інерційного віброзбудника у вібромашині з прямолінійним поступальним рухом короба на двочастотний у вигляді пасивного автобалансира.

Практична значимість. Полягає в можливості проектування нових енергоефективних і високопродуктивних вібромашин, що поєднують у собі переваги резонансних і двочастотних вібромашин.

Ключові слова: *інерційний віброзбудник, двочастотний віброзбудник, дебаланс, вібромашина, автобалансир, грохот*

Цель. Определить условия замены одночастотного вибровозбудителя на двухчастотный в виде пассивного автобалансира в вибромашине с прямолинейным поступательным движением короба.

Методика. Используются элементы теории вибромашин и теоретической механики для создания и анализа механико-математических моделей базовой и модернизованной машин.

Результаты. Аналитически подобраны параметры модернизированной машины (с двухчастотным вибровозбудителем в виде пассивного автобалансира), при которых амплитуда и частота медленных колебаний ее короба равняется амплитуде и частоте вынужденных колебаний короба базовой машины (с одночастотным инерционным вибровозбудителем) с прямолинейным поступательным движением короба.

Полученные результаты позволяют рассчитывать параметры резонансных вибромашин с двухчастотными вибровозбудителями в виде пассивных автобалансиров.

Установлено, что на частоту медленных колебаний модернизированной машины влияет коэффициент жесткости опор, а на амплитуду этих колебаний влияет – коэффициент сил вязкого сопротивления опор.

Модернизация грохота ГИЛ 42 позволила уменьшить суммарную массу вращающихся частей вибровозбудителя в 3,4 раза.

Научная новизна. Состоит в определении условий замены одночастотного инерционного вибровозбудителя в вибромашине с прямолинейным поступательным движением короба на двухчастотный вибровозбудитель в виде пассивного автобалансира.

Практическая значимость. Состоит в возможности проектирования новых энергоэффективных и высокопроизводительных вибромашин, совмещающих в себе преимущества резонансных и двухчастотных вибромашин.

Ключевые слова: *инерционный вибровозбудитель, двухчастотный вибровозбудитель, дебаланс, вибромашина, автобалансир, грохот*

Рекомендовано до публікації докт. техн. наук О. С. Ланцем. Дата надходження рукопису 03.02.16.