UDC 539

DETERMINATION OF ENERGY PARAMETERS OF VIBRATING MACHINES FOR COMPACTION AND FORMATION OF CONCRETE PRODUCTS ACCORDING TO DIFFERENT POWER FORM OF LOAD

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Abstract. The paper investigates and determines the parameters and energy indicators of vibrating machines with harmonic and vibroshock modes of motion for compaction of concrete mixtures. The equations of motion of the vibration system "working body of the machine - compaction medium" are made on the basis of a hybrid discrete-continuum model, which adequately describes the real process of compaction of the mixture. Calculations were made on the basis of the obtained analytical dependencies to determine the amplitudes of oscillations and the oscillation energy of a two-seater The systems made it possible to assess their changes in different modes of operation. Thus, the first resonance was recorded at a frequency of 5 Hz, and the second at a frequency of 35 Hz. For the action of an external force on the first mass, the resonance mode is determined by the parameters of the second mass, including the energy dissipation coefficient and the ratio between these masses. The conditions for the influence of dissipation in the resonance mode at a frequency close to the partial natural frequency of the mass on which the external force acts have been determined. The energy parameters of the vibrating percussion system have been investigated and determined. The equations of motion of a vibro-percussion installation as a model with discrete-continuous parameters are given and solved.

Amplitude and skeletal characteristics are given, the influence of parameters on their change is determined, and two ways of implementing the resonance mode are revealed: by changing the frequency of the harmonic force, and at the same time, the possibilities of controlling the movement of the installation by changing the frequency and the value of compression of the limiter by a constant force are determined. Formulas for determining the energy parameters of a vibro-percussion installation, which take into account the discrete parameters of the machine and the distributed parameters of the concrete mixture, have been obtained.

Keywords: discrete-continuous model, two-seat vibration unit, vibration shock unit, compaction, mixture, energy, resonance, parameters, amplitude, oscillation frequency.

Entry

Vibrating machines are widely used in various industries. Their effectiveness is especially important in road construction in the construction of asphalt and concrete roads and in the construction industry in the formation of concrete and reinforced concrete products. The vibration effect on the mixture is of great practical importance and is the basis of all modern technology of sewing mixtures. The essence of the vibration action lies in the fact that during oscillations, the mixture acquires the properties of fluidity due to the violation of the bonds between the particles. Particles that receive increased mobility are mixed and, under the influence of weight forces, tend to take a more stable position. At the same time, the air between the particles is squeezed up and the mixture, in the end, is significantly sewn. The process of vibration coating of the mixture is complex and takes place in several stages: restacking of components with intensive air squeezing, particle convergence and final air squeezing, as well as possible additional squeezing due to some additional, for example, static pressure. The complexity of the processes taking place in the mixture, differences in the views of scientists and engineers on the compaction process - these are the reasons for the different approach and the lack of a generally accepted model for calculating the main design and technological parameters. Taking into account the modern requirements for energy reduction, an urgent task is to improve the methods for calculating the parameters of vibration technology, the use of which will make it possible to minimize the energy for the flow of the technological process. The paper discusses the dynamics of vibration machines and the determination of energy indicators on the basis of the classical theory of mechanical oscillations and the theory of continuums.

Analysis of the latest research and publications

Among the works devoted to the determination of the parameters of vibration machines, it is worth noting the following works. In the paper [1], a discrete model was used in the study of the vibration platform with spatial oscillations. In the paper [2], a discrete model was used in the study of sorting processes by a vibrating screen. In paper [3], when modeling a vibrating mixer, a discrete model is given. The obtained research results within the framework of the discrete model are valid only within the framework of the conducted research. The paper [4] describes the creation of a mathematical model of a shock-vibration platform, where the mode of asymmetric oscillations is implemented, in which the upper and lower acceleration of the form with concrete have different values. A discrete-continuum model of the vibration system is proposed in [5]. The developed model for determining the parameters of vibration machines is used in the work [6]. The monograph [7] describes the dynamics and determination of the parameters of vibration processes of grinding, sorting, compaction and reliability of vibration machine elements. Based on the analysis of energy dissipation methods given in [8], the following dependencies can be identified for determining energy or power. Thus, power is determined by the empirical formula:

$$P = b_n S X_0^2 \omega^2, \tag{1}$$

where b_n - specific coefficient of resistance, which refers to the unit of active area of the vibration compaction body; S is the active area of vibration compaction of the mixture; X_0, ω is the amplitude and frequency of oscillations, respectively.

Empirical also includes the formula for determining the power of the vibration pad drive:

$$P = \alpha_M m' P_n, \tag{2}$$

where α_M - empirical coefficient; m' - reduced mass of concrete; P_n power-to-weight ratio.

The viscous friction hypothesis is sometimes used to determine power:

$$P = k_{on} \frac{X_0^2 \omega^2}{2}, \tag{3}$$

where k_{on} - empirical coefficient of resistance to oscillations.

Consequently, the existing methods for determining the oscillation power of the "vibrator-medium" system are very diverse. This is due to the complexity of the "vibrator-medium" system, the lack of a generally accepted theory describing the mechanics of the medium. Hence the tendency to simplify the system, to take into account only certain forces "found mainly empirically" and to neglect other forces, which in the general case leads to a significant deviation of the calculated data from their physical values. It is safe to say that it is necessary to expand our understanding of energy dissipation through a more detailed study of such processes.

Objective

The aim of the study is to determine the energy indicators on the basis of a discrete-continual model of the vibration system, taking into account the dissipative properties of the sealing medium.
To achieve the goal, the following tasks have been formulated and solved:

- research and determination of energy parameters of a vibration unit with a harmonious load change;
- research and determination of energy parameters of the vibration percussion system.

Research and determination of energy parameters of a vibration unit with a harmonious change of external force

As a design model of a vibration unit with a harmonious load change, we take a two-seater one (Fig. 1). Here m_1 is the mass of the working body; m_2 is the mass, which includes the mass of the mold with the concrete mixture; c_1, b_1 – coefficients of stiffness and dissipation of the supporting elements of the installation; c_2, b_2 are the stiffness and dissipation coefficients connecting the masses m_1 and m_2; F_0 is the amplitude of the external force of the vibration exciter. The difference between this scheme and the classical one is that the mass m_2 according to the discrete form of notation takes into account, in addition to the mass of the form *mf,* the mass of the concrete mixture not in the form of a discrete coefficient, as in dependence (2), but by the coefficient α_n, which takes into account the wave phenomena in the mixture according to the method given in the paper [6]:

$$m_2 = m_a + \alpha_n m_б, \tag{4}$$

where

$$\alpha_n = \frac{\alpha sh2\alpha h + \beta \sin 2\beta h}{h\left(\alpha^2+\beta^2\right)\left[ch2\alpha h + \cos 2\beta h\right]}. \tag{5}$$

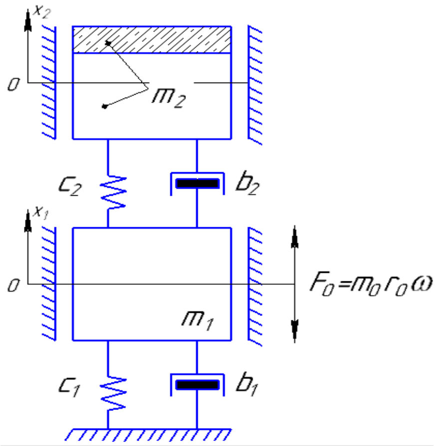


Fig. 1. Design diagram of a two-seater vibration unit

In formula (5): α and β - coefficients that take into account the effect of energy dissipation on the change in the shape and degree of wave attenuation in the concrete mixture, and h - the height of the column of the concrete mixture;

$$\alpha = \mu\frac{\omega}{c_в}; \quad \beta = \nu\frac{\omega}{c_в}, \tag{6}$$

where

$$\mu = \sqrt{\frac{\sqrt{1+\gamma^2}-1}{2(1+\gamma^2)}}, \quad \nu = \sqrt{\frac{\sqrt{1+\gamma^2}+1}{2(1+\gamma^2)}}.$$

The α factor determines the attenuation of the wave propagating in the layer of the medium, and the β factor determines the attenuation of the length of the same wave. Indeed, if there is no resistance (γ=0), then $\alpha = 0$, $\beta = \frac{\omega}{c_в}$ that is, the wave propagates without attenuation under this condition.

Differential equations of motion are composed by the method of equilibrium of forces, [5] which is based on the use of the d'Alembert principle, according to which, along with external forces, the inertial forces of masses m_1 and m_2 are also taken into account. Omitting simple transformations, we get the equations of motion:

$$m_1\ddot{x}_1 + b_1\left(\dot{x}_1 - \dot{x}_2\right) + c_1\left(x_1 - x_2\right) = F_0 e^{i\omega t},$$
$$m_2\ddot{x}_2 - b_1\left(\dot{x}_1 - \dot{x}_2\right) - c_1\left(x_1 - x_2\right) + b_2 x_2 + c_2 x_2 = 0. \tag{7}$$

The obtained equations (7), for a convenient solution, are reduced to the dimensionless form:

$$x_1 = \frac{m_0 r\omega^2}{m_2\omega_0^2}\xi_{11}, \quad x_2 = \frac{m_0 r\omega^2}{m_2\omega_0^2}\xi_{12}. \tag{8}$$

And having carried out the appropriate transformations, we get:

$$\chi\gamma^2\xi_{11}'' + \frac{\chi}{1+\chi}\gamma\beta_1\left(\xi_{I_1}' - \xi_{I_2}'\right) + \frac{\chi}{1+\chi}\left(\xi_{I_1} - \xi_{I_2}\right) = e^{i\tau},$$

$$\gamma^2\xi_{I_2}'' = \frac{\chi}{1+\chi}\gamma\beta_1\left(\xi_{I_1}' - \xi_{I_2}'\right) + \gamma\beta_2\xi_{I_2}' - \frac{\chi}{1+\chi}\left(\xi_{I_1} - \xi_{I_2}\right) + \gamma_2^2\beta_2\xi_{I_2}^* = 0. \tag{9}$$

The solution of equations (9) obtained dependencies for determining the dimensionless amplitude of mass displacements m_1:

$$\xi_{I_{1a}} = \frac{m_2 x_{I_0}}{m_0 r} = \gamma^2\sqrt{\frac{M^2+N^2}{H^2+Y^2}}, \tag{10}$$

where such designations are adopted

$$H = \frac{\chi}{1+\chi}\gamma_2^2 - \chi\gamma^2 - \frac{\chi}{1+\chi}\gamma^2\beta_1\beta_2 - \chi\gamma^2\gamma_2^2 + \chi\gamma^2,$$

$$Y = \frac{\chi}{1+\chi}\gamma\beta_2 + \frac{\chi}{1+\chi}\gamma_2^2\gamma\beta_1 - \chi\gamma^3\beta_1 - \chi\gamma^3\beta_2,$$

$$M = \gamma_2^2 - \gamma^2 + \frac{x}{1+x}, \quad N = \gamma\beta_2^2 + \frac{x}{1+x}\gamma\beta_1. \tag{11}$$

Dependence for determining the dimensionless amplitude of mass displacements m_2:

$$\xi_{I_{2a}} = \frac{m_2 x_2}{m_0 r} = \frac{x}{1+x}\gamma^2\sqrt{\frac{\gamma^2\beta_1^2+1}{H^2+Y^2}}, \tag{12}$$

dimensionless relative amplitudes of mass displacements m_1 and m_2:

$$\xi_{I_a} = \frac{m_2 x_{01}}{m_0 r} - \sqrt{\frac{\left(\frac{x}{1+x}-k\right)^2 + \left(\frac{x}{1+x}\gamma\beta_1 - N\right)^2}{H^2 - Y^2}}. \tag{13}$$

The oscillation amplitudes x_{01} and x_{02} will be at their maximum in the system's resonant mode. At resonance, the maximum oscillation amplitudes will reach the level $b_1 = b_2 = 0$. In this case, the oscillation amplitudes become infinite. For $b_2 = \infty$ the oscillation amplitude is determined solely by the value of b_1, and in the case of $b_2 = \infty$, $b_1 = 0$ it also becomes infinite. This occurs because, at the system becomes a single-mass system with mass $M = m_1 + m_2$. The work of the dissipative force in both cases (for $b_1 = 0$) does not manifest, as in the first case, the dissipative force equals zero, and in the second, there is no relative displacement of the masses From this, it follows that at the value $0 < b_2 < \infty$, the work of the dissipative force will be maximal, and the resonant oscillation amplitude of the second mass will take its minimum value. In the system's resonant modes, the energy parameters are determined as follows: the work performed by the external harmonic force is equal to the work done on the first mass:

$$A_F = \pi F_0 X_{01}\sin\varphi_1. \tag{14}$$

At resonant frequencies ω_{01} and ω_{02}:

$$\left|\sin\varphi_{1\omega_{01}}\right| = \left|\sin\varphi_{1\omega_{02}}\right| = 1.$$

The energy dissipated in the system is

$$\Delta E = \pi\omega\left[b_1 x_1^2 + b_2(x_2 - x_1)^2\right]. \tag{15}$$

Equating expressions (14) and (15) in the resonance mode, we get the formula:

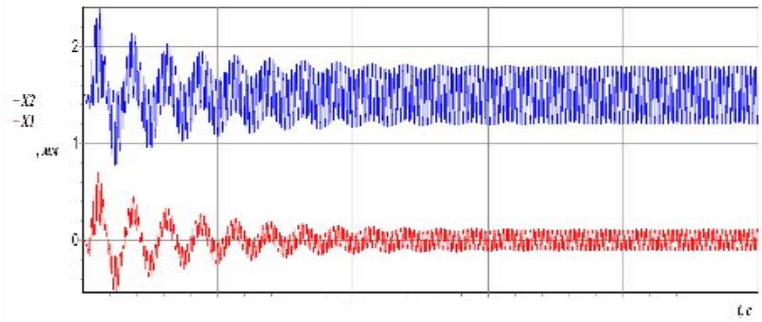
$$x_{01} = \frac{F_0}{b_1\omega_p + b_2\omega_p\left((x_{02}/x_{01}) - 1\right)^2}, \tag{16}$$

which will be maximum at $x_{01} = x_{02}$, in the case of infinitely large values of stiffness C_2 or damping b_2, equation (16) takes the form:

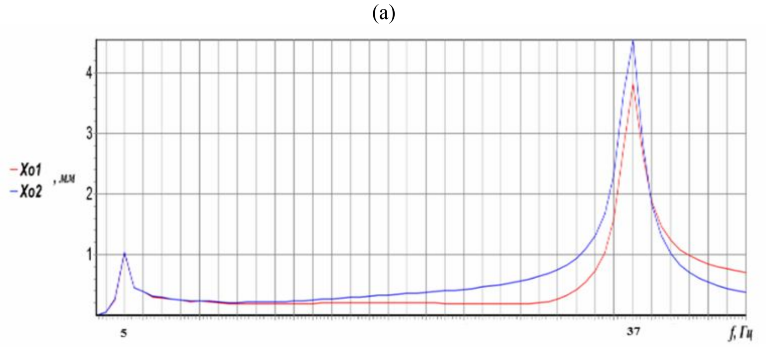
$$x_{01p} = F_0 / b_1\omega_p. \tag{17}$$

For a two-seat system with small dissipation, in the case of a force acting on one of the masses (for example, m_1) for a given amplitude of oscillations of this mass, the ratio of the amplitudes of oscillations in the resonance mode is determined only by the parameters of the second mass and the ratio between the masses and the coefficient b_2. For a resonance at a frequency close to the partial frequency of the mass m_1 on which the force F_0 acts, the difference between the inertial force acting on the second mass m_2 and the elastic force is not zero. Therefore, the effect of dissipation is negligible. In the case of resonance at the frequency caused by the second mass m_2, the role of scattering forces increases. The damping associated with the mass m_1 has a significant effect on the frequency of the first resonance ω_{01}. But in terms of frequency ω_{02}, its effect is not significant. Applying the above dependencies, an algorithm for calculating on a PC was compiled. As an example, the movement of masses m_1 and m_2 (Fig. 2, a), calculations of energy indicators of a vibration unit and graphs of amplitude and frequency characteristics of a vibration unit are constructed (Fig. 2, b).

In steady-state oscillations, the average energy circulating in the system is conserved. Each period, the source replenishes energy expenditure according to formula (15) for a steady state, the energy dissipation must be equal to the work done during that period by an external force F_0 applied to the system.



(a)

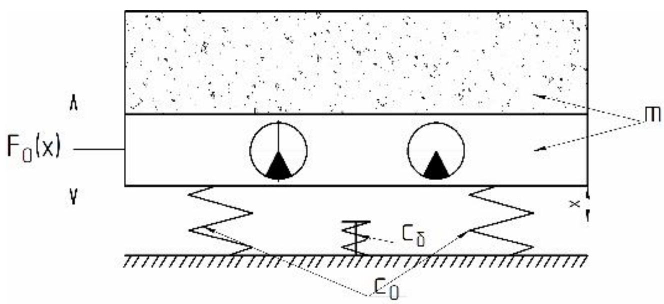


(b)

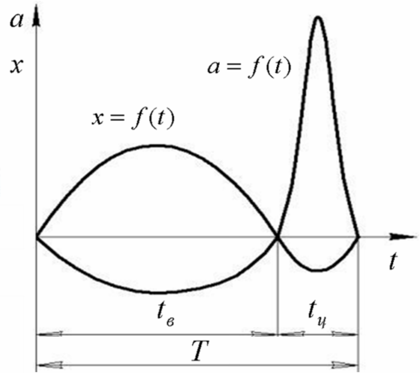
Fig. 2. Amplitudes of mass oscillations m_1 and m_2 and graph of the amplitude and frequency response of the vibration installation

Research and determination of energy parameters of a vibration percussion installation

The design scheme of the vibration percussion system is reduced to a single one by reducing the two-seat circuit (Fig. 3).



(a)



(b)

Fig. 3. Vibration shock installation: (a) - design model;
(b) - change of displacement and acceleration during the period of oscillation

Here m is the mass of the working body, which includes the mass of the mold with the concrete mixture; c_0 – stiffness coefficient of the supporting elements of the installation; $c_б$ is the stiffness coefficient of the mass oscillation limiter m; F_0 is the amplitude of the external force of the vibration exciter; x - displacement; a - acceleration. The period of oscillation T will consist of two parts – the detached motion of the mass in time $t_в$ and the motion in contact $t_к$: $T = t_в + t_к$. Since the stiffness coefficient of vibration isolating supports: $c_б > c_0$ (usually $c/c_0 \approx 7\ldots10$), the half-span of the oscillations will differ significantly from each other: $x_в > x_к$. Accordingly, accelerations, as the second derivatives of displacement, will have a significant difference from each other $a_к > a_в$ (usually $a_к/a_в = 3...5$, see Fig. 3, b). This asymmetry of accelerations is the main criterion for the effectiveness of the use of vibration shock modes in machines for compaction of building mixtures, since the significant forces that arise at the moment of impact coincide with the weight forces of the treated medium, which leads to auxiliary compression forces, and hence compaction of the mixture. In contrast to the dependence (5), which takes into account the wave phenomena in the mixture in the harmonic mode, the coefficient α_n for the vibration shock action will have a different form:

at $0 \le t \le t_в$

$$\alpha_n = \frac{(\alpha \mathrm{sh} 2\alpha h + \beta \sin 2\beta_1 h)\tau_1}{h(\alpha^2 + \beta^2)(\mathrm{ch} 2\alpha h + \cos 2\beta h)(\tau_1 + \tau_2)}, \tag{18}$$

at $t_к \le t < T$

$$\alpha_n = \frac{(\alpha \mathrm{sh} 2\alpha h + \beta \sin 2\beta h)\tau_2}{h(\alpha^2 + \beta^2)(\mathrm{ch} 2\alpha h + \cos 2\beta h)(\tau_1 + \tau_2)}. \tag{19}$$

That is, in the equations of motion, the effect of the concrete mixture on the dynamics of the vibration unit at each stage will be different.

Comparing dependencies (18) and (19) with (5), when $\tau_1 = \tau_2$ (the symmetric law of motion), the dependencies for determining reactive resistance coincide. The setup is under the influence of a constant force Q, the directional harmonic force $F(t)$ and the reaction of an elastic limiter $F_{np} = c_0 x + \mathrm{sign}\dot{x} R(x)$, that is, we provide for the consideration and dissipation of energy. Provided that the elastic coefficient of the supports c_0 is calculated based on the vibration isolation condition, we assume its effect is absent, acting on the mass mmm during its contact intervals with the elastic limiter. Then, the differential equations describing the motion of the system at the stages of contact with the limiter and separation from it are as follows:

$$m\ddot{x} = F_{cm} - c_e x - \mathrm{sign}\dot{x} R(x) + F_0(t) \text{ at } x \geqslant 0,$$
$$m\ddot{x} = F_{cm} + F_0(t) \qquad \text{at } x \leqslant 0$$

or

$$\ddot{x} + \lambda^2 x = \lambda^2 \Delta - m^{-1} \mathrm{sign}\dot{x} R(x) - m^{-1} F(t), \tag{20}$$

$$\ddot{x} = \lambda^2 \Delta + m^{-1} F(t), \tag{21}$$

where is $\Delta = F_{cm}/c_б$ the amount of compression of the limiter by a constant force Q; $\lambda = \sqrt{c_б/m}$ - cyclic frequency of natural oscillations of the installation in contact with the limiter; m - mass of the installation.

To find the equations of the skeletal lines of a vibration installation, consider its free oscillations in the absence of the inelastic resistance of the limiter. At the same time, the differential equations of motion of the vibration installation are presented in the following form:

$$\ddot{x} + \lambda^2 x = \lambda^2 \Delta \quad \text{at } x \geqslant 0, \tag{22}$$

$$\ddot{x} = \lambda^2 \Delta \qquad \text{at } x \leqslant 0. \tag{23}$$

For convenience, we denote $2t_1$ the duration of the movement of the mass of the installation in contact with the limiter for one period T of its free oscillations and through $2t_2$ the duration of its movement in isolation from the limiter.

The periodic solution of differential equations (22) and (23) is written as follows:

$$x=-\frac{\Delta}{\cos\lambda t_1}\left[\cos\lambda(t-t_1)-\cos\lambda t_1\right]\text{ at }0\leqslant t\leqslant 2t_1, \quad (24)$$

$$x=\lambda^2\Delta\left[(t_*-t_2)^2/2-t_2^2/2\right]\text{ at }0\leqslant t_*\leqslant 2t_2. \quad (25)$$

From the equality of the velocities of the vibration installation at the moment of its transition from movement in contact with the limiter to movement in isolation from it, we get

$$\lambda t_2=-\tan\lambda t_1. \quad (26)$$

Let's make an obvious equation

$$T=2\pi/\omega_0=2t_1+2t_2, \quad (27)$$

where ω_0 is the conditional cyclic frequency of natural oscillations of the vibrator with its separation from the limiter. From formula (27) we get

$$\mu=\frac{\lambda}{\omega_0}=\frac{\lambda t_1-\tan\lambda t_1}{\pi}. \quad (28)$$

The parameter μ is equal to the ratio of the period of oscillation of the mass of the vibration unit in the absence of contact with the limiter to the period of its natural oscillations in the presence of contact with the limiter. Equation (28) allows us to find λt_1 at any point in time $1\leqslant\mu<\infty$.

Now let's derive the formulas that characterize movements with a break from the limiter. On the basis of the equations of motion (25) and (26) we obtain

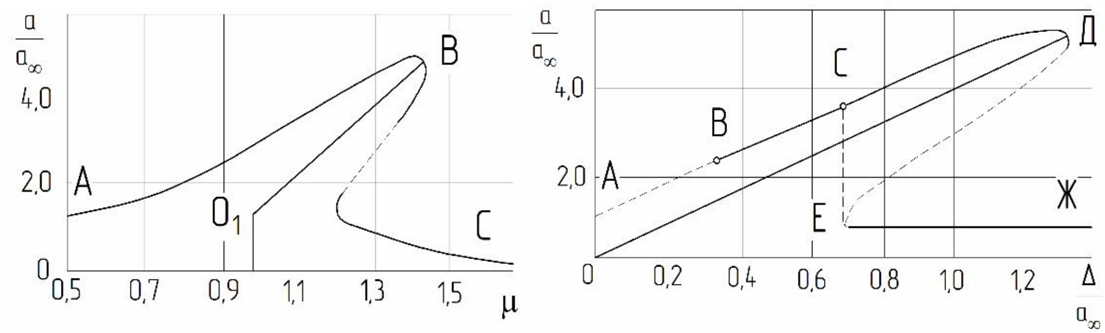
$$a_\kappa/a=k_0(\mu), \quad (29)$$

$$a=4k_0^{-2}(\mu)\Delta, \quad (30)$$

where a_κ is the maximum compression of the limiter; a is the amplitude of oscillations of the vibration installation, which is equal to half of the range of oscillations.

$$k_0(\mu)=\frac{4\cos\lambda t_1}{\cos\lambda t_1-1}=2\left(1-\cot g^2\frac{\lambda t_1}{2}\right). \quad (31)$$

If in formula (30) we take $\Delta=$ const , then we get the first skeletal line of the vibration installation, which corresponds to this Δ . The first skeletal line is important because in the vicinity of this line there will be the first amplitude curve of the vibration installation, which reflects the dependence of the amplitude of its forced oscillations on the period of harmonic force. If we assume in formula (30) that $\mu=$ const , we get the second skeletal line of the vibration installation. In the vicinity of this line there will be the second amplitude of the vibration curve, reflecting the dependence of the amplitude of forced vibrations of the vibration installation on the value Δ of the gap of a given period of harmonic force.



(a) (b)

Fig. 4. Skeletal and amplitude curves of the vibration installation: (a) at $\Delta/a_0=1{,}36$, (b) at $\mu=1{,}5$

Fig. 4, a shows the first – O_1B, and Fig. 2, b – the second – ОД skeletal lines of the vibration installation. The first - ABC and the second amplitude curves of the АДЖ are also shown there. Point C of the intersection of the amplitude curves with the skeletal lines corresponds to the resonant oscillations of the vibration system. S_0, there are two ways to implement the resonance mode: by changing the frequency of the harmonic force ω at the same $\Delta = \text{const}$ time, or by changing Δ it with a constant frequency of the harmonic force ω. The second method is technically simpler, due to the change Δ than the frequency ω. The curve of dependence of the amplitude of forced oscillations of the vibration installation on Δ or the weight Q is the curve of adjustment of the parameters of the vibration installation.

To determine the energy balance of a vibration system, the expression for the harmonic force acting during one period is written as follows:

$$F(t) = -F_0 \sin\left[\omega(t - t_1) + \varphi\right] \text{ at } 0 \leqslant t \leqslant 2t_1, \tag{32}$$

$$F(t_*) = F_0 \sin\left[\omega(t_* - t_2) + \varphi\right] \text{ at } 0 \leqslant t_* \leqslant 2t_2. \tag{33}$$

Here, formula (32) determines the harmonic force for the stage of motion of the vibration unit in contact with the limiter, and formula (33) determines it in isolation from the limiter. The angle φ included in the formulas is less than the phase displacement angle between the harmonic force and the forced oscillations of the vibration installation on $\pi/2$. In contrast to the phase shift angle, which varies from 0 to π the angle φ will vary in the range from $-\pi/2$ to $\pi/2$. For resonant vibrations of the vibrator $\varphi = 0$.

The expression in order for the harmonic force to ensure the movement of the vibration unit in one period is written in the form:

$$A(F) = \int_0^{2t_1} F(t)dx + \int_0^{2t_2} F(t_*)dx. \tag{34}$$

As a result of the solution, we get:

$$A(F) = k_1(v) F_0 a \cos\varphi, \tag{35}$$

where

$$k_1(v) = \frac{k_0^2(v)}{4}\left[\frac{2v}{v^2-1}\left(v \sin\omega t_1 - \tan t_1 \cos\omega t_1\right) + 2v\left(v \sin\omega t_1 - \tan pt_1 \cos\omega t_1\right)\right] =$$

$$= \frac{k_0^2(v)}{2}\frac{v^3}{v^2-1} \times \left(v \sin\frac{pt_1}{v} - \tan pt_1 \cos\frac{pt_1}{v}\right). \tag{36}$$

The graph $k_1(v)$ of the function depending on pt_1 (Fig. 5) shows that the value $k_1(v)$ for $1 \leqslant v \leqslant 2{,}5\pi$. If the condition that $v = 1$, then $k_1 = \pi$, which corresponds to the harmonic oscillations of the vibration installation, is satisfied.

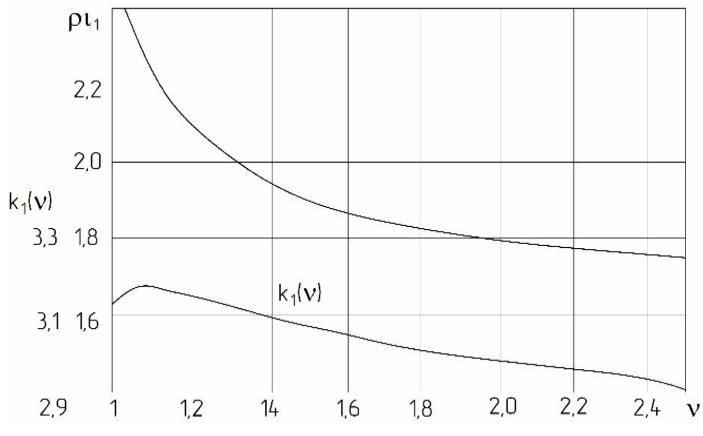


Fig. 5. Graphs of dependence $k_1(v)$ and magnitude pt_1 on v

Let us make an expression for the energy dissipated in the elastic limiter for one period of oscillation of the vibration unit:

$$W = \psi \frac{c_б a_к^2}{2} = \psi k_0^2(\nu) \frac{c_б a^2}{2}. \tag{37}$$

The condition for the energy balance of the vibrator is as follows:

$$A(F) = W.$$

Or, substituting the values of $A(F)$ and W we get equality:

$$k_1(\nu) F_0 \cos\varphi = 1/2\, \psi k_0^2(\nu) c_б a. \tag{38}$$

Based on equation (38), the following formula can be obtained for the amplitude of vibrations of the vibrator:

$$a = \frac{2k_1(\nu)}{\psi k_0^2(\nu)\mu^2} \cos\varphi a_\infty, \tag{39}$$

where $a_0 = F_0 / (m\omega^2)$ is the amplitude of oscillations of the vibration installation out of contact with the oscillation limiter.

In the case of using an unbalanced oscillation exciter, the forcing force:

$$F_0 = m r_0 \omega^2,$$

where mr_0 is the static angular momentum of the mass of the imbalances.

Then the expression for a_0 will take the form

$$a_\infty = \frac{m r_0}{M_B}, \tag{40}$$

where M_B is the oscillating mass of the vibration installation.

Let's also make a formula for the average power, inelastic resistance of the oscillation limiter:

$$P = \frac{W\omega}{2\pi} = \frac{\psi}{4\pi} c_б a_к^2 \omega = \frac{\psi}{4\pi} \mu^2 k_0^2(\nu) a^2 \omega^2 m. \tag{41}$$

Energy dissipated in the volume of the concrete mixture when it is compacted

$$W_{б.с} = \bar{E} dm, \tag{42}$$

where $\bar{E}$ is the specific energy (J/kg) spent on compaction of a concrete mixture with a mass of $m_{b.s}$.

Conclusions

1. The equations of motion of a two-seater vibrating unit have been compiled and solved, in which the discrete parameters of the machine and the parameters of the concrete mixture are distributed. Energy parameters and dynamic parameters have been investigated and determined.

2. The influence of energy dissipation on the motion of the vibration system under study has been revealed. Thus, for a two-seat system with a small dissipation, in the case of a force acting on one of the masses (for example, m_1) for a given amplitude of oscillations of this mass, the ratio of the amplitudes of oscillations in the resonance mode is determined only by the parameters of the second mass and the ratio between the masses and the coefficient b_2.

3. When resonating at a frequency close to the natural frequency of the mass m_1 on which the force F_{01} acts, the difference between the inertial force acting on the second mass m_2 and the elastic force is not zero. Therefore, the effect of dissipation is negligible. In the case of resonance at the frequency caused by the second mass m_2, the role of scattering forces increases.

4. The energy parameters of the vibrating percussion system have been investigated and determined. The equations of motion of a vibro-percussion installation as a model with discrete-continuous parameters are given and solved.

5. Amplitude and skeletal characteristics are given, the influence of parameters on their change is determined, and two ways of implementing the resonance mode are revealed: by changing the frequency of the harmonic force, and at the same time the possibilities of controlling the movement of the installation by changing the frequency and the value of compression of the limiter by a constant force are determined.

6. Formulas for determining the energy parameters of a vibrating percussion installation, which take into account the discrete parameters of the machine and the distributed parameters of the concrete mixture, have been obtained.

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ВИЗНАЧЕННЯ ЕНЕРГЕТИЧНИХ ПОКАЗНИКІВ ВІБРАЦІЙНИХ МАШИН ДЛЯ УЩІЛЬНЕННЯ ТА ФОРМУВАННЯ БЕТОННИХ ВИРОБІВ ЗА РІЗНОЮ СИЛОВОЮ ФОРМОЮ НАВАНТАЖЕННЯ

Анотація. В роботі досліджено й визначено параметри й енергетичні показники вібраційних машин з гармонійними та віброударними режимами руху для ущільнення бетонних сумішей. Рівняння руху вібраційної системи «робочий орган машини- ущільнююче середовище» складені на основі гібридної дискретно-континуальної моделі, яка адекватно описує реальний процес ущільнення суміші. Здійснені розрахунки за отриманими аналітичними залежностями для визначення амплітуд коливань та енергії на коливання двомісної системи дозволили здійснити оцінку їх зміни в різних режимах роботи. Так, перший резонанс був зафіксований на частоті 5гц, а другий на частоті 35гц. Для дії зовнішньої сили на першу масу резонансний режим визначається параметрами другої маси, в тому числі коефіцієнтом розсіяння енергії та співвідношенням між цими масами. Визначені умови впливу дисипації в режимі резонансу на частоті, близькій до часткової власної частоти тої маси , на яку діє зовнішня сила.. Досліджено та визначено енергетичні параметри віброударної установки. Приведені та вирішені рівняння руху віброударної установки, як моделі з дискретно-континуальними параметрами.

Приведено амплітудні й скелетні характеристики, визначено вплив параметрів на їх зміну і виявлені два способи реалізації резонансного режиму: шляхом зміни частоти гармонійної сили та при цьому визначені можливості керування рухом установки зміною частоти і величина стиснення обмежувача постійною силою. Отримані формули для визначення енергетичних показників віброударної установки, які враховують дискретні параметри машини та розподілені параметри бетонної суміші.

Ключові слова: дискретно-континуальна модель, двомісна вібраційна установка, віброударна установка, ущільнення, суміш, енергія, резонанс, параметри, амплітуда, частота коливань.

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DETERMINATION OF ENERGY PARAMETERS OF VIBRATING MACHINES FOR COMPACTION AND FORMATION OF CONCRETE PRODUCTS ACCORDING TO DIFFERENT POWER FORM OF LOAD

Abstract. The paper investigates and determines the parameters and energy indicators of vibrating machines with harmonic and vibroshock modes of motion for compaction of concrete mixtures. The equations of motion of the vibration system "working body of the machine - compaction medium" are made on the basis of a hybrid discrete-continuum model, which adequately describes the real process of compaction of the mixture. Calculations were made on the basis of the obtained analytical dependencies to determine the amplitudes of oscillations and the oscillation energy of a two-seater The systems made it possible to assess their changes in different modes of operation. Thus, the first resonance was recorded at a frequency of 5 Hz, and the second at a frequency of 35 Hz. For the action of an external force on the first mass, the resonance mode is determined by the parameters of the second mass, including the energy dissipation coefficient and the ratio between these masses. The conditions for the influence of dissipation in the resonance mode at a frequency close to the partial natural frequency of the mass on which the external force acts have been determined. The energy parameters of the vibrating percussion system have been investigated and determined. The equations of motion of a vibro-percussion installation as a model with discrete-continuous parameters are given and solved.

Amplitude and skeletal characteristics are given, the influence of parameters on their change is determined, and two ways of implementing the resonance mode are revealed: by changing the frequency of the harmonic force, and at the same time, the possibilities of controlling the movement of the installation by changing the frequency and the value of compression of the limiter by a constant force are determined. Formulas for determining the energy parameters of a vibro-percussion installation, which take into account the discrete parameters of the machine and the distributed parameters of the concrete mixture, have been obtained.

Keywords: discrete-continuous model, two-seat vibration unit, vibration shock unit, compaction, mixture, energy, resonance, parameters, amplitude, oscillation frequency.

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Назаренко І.І., Запривода А.В., Бондаренко А.З., Слюсар В.С. **Визначення енергетичних параметрів вібраційних машин для ущільнення та формування бетонних виробів при різних силових формах навантаження** // Опір матеріалів і теорія споруд: наук.-техн. збірн., К.: КНУБА, 2024. – Вип. 113. – С. 18-28.
У статті досліджено та визначено енергетичні параметри та показники вібраційних машин з гармонійними та ударно-вібраційними режимами руху для ущільнення бетонних сумішей. Для аналізу рівнянь руху вібраційної системи «робочий орган машини – середовище ущільнення» використано гібридну дискретно-континуальну модель, що дозволяє реалістично моделювати процес ущільнення. Розрахунки показали резонансні частоти на рівні 5 Гц і 35 Гц, причому параметри маси системи впливають на розсіювання енергії та режими резонансу.
Табл. 0. Рис. 5. Бібліогр. 8.

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Nazarenko I.I., Zapryvoda A.V., Bondarenko A.Z., Slyusar V.S. **Determination of energy parameters of vibrating machines for compaction and formation of concrete products according to different power form of load** // Strength of Materials and Theory of Structures: Scientific-and-technical collected articles. – K.: KNUBA. 2024. – Issue 113. – P. 18-28.
The paper investigates and determines the energy parameters and indicators of vibrating machines with harmonic and vibroshock modes of motion for compaction of concrete mixtures. A hybrid discrete-continuum model was used to analyze the motion equations of the vibration system "machine's working body - compaction medium," allowing for realistic compaction process simulations. Calculations reveal resonance frequencies at 5 Hz and 35 Hz, with energy dissipation and resonance modes influenced by system mass parameters.
Tab. 0. Fig. 5. Ref. 8.

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