Ivan I. NAZARENKO[1], Mykola M. RUCHYNSKYI[1], Anatoliy T. SVIDERSKYI[1], Iryna M. KOBYLANSKA[2], Damian HARASIM[3], Aliya KALIZHANOVA[4,5], Ainur KOZBAKOVA[4,6]
Kyiv National University of Construction and Architecture(1), Vinnytsia National Technical University(2), Lublin University of Technology(3), Institute Information and Computational Technologies CS MES RK(4), Al-Farabi Kazakh National University(5), Almaty University of Power Engineering and Telecommunications (6)

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Development of energy-efficient vibration machines for the buiding-and-contruction industry

***Abstract**. This paper contains the results of the analysis of motions of a vibration machine actuator and the response of medium with material being processed to the actuator motions. The actuator and the medium in combination are simulated as a discrete-continuous system. Discussed in this paper are the basic parameters of such a system, as well as the modes of resonance, under-resonance, and over-resonance vibrations of the vibration machine actuator. In the paper, some variants of the vibration machine design, which are characterized by enhanced energy efficiency in generating vibrations, are proposed.*

Streszczenie.** Artykuł zawiera wyniki analizy ruchów siłownika maszyny wibracyjnej i reakcji ośrodka z materiałem przetwarzającym ruchy elementu. Połączenie siłownika oraz medium symulowane jest jako dyskretny system ciągły. W artykule omówiono podstawowe parametry układu a także stany rezonansowe, pod-rezonansowe oraz nad-rezonansowe drgań siłownika maszyny. Zaproponowano także szczególne warianty konstrukcji maszyny, które charakteryzują się zwiększoną efektywnością energetyczną w generowaniu drgań. **(Opracowanie energooszczędnych maszyn wibracyjnych dla branży budowlanej oraz konstrukcyjnej).

Keywords: vibration machine, energy efficiency, resonance vibrations.
Słowa kluczowe: maszyny wibrujące, efektywność energetyczna, wibracje rezonansowe.

Introduction

Vibration machines are used in processes for crushing, sorting, mixing, and compaction of construction materials, as well as in other processes that are characteristic for the building-and-construction industry. Vibration machines are designed, for the most part, for operation in the mode of over-resonance vibrations of the vibration machine actuator. In the design of such vibration machines, predominantly inertial forces are considered. Notwithstanding the fact that vibration machines are rated for steady-state operation, that is, for operation in conditions without transient processes, the energy efficiency of such vibration machines is low. Consequently, such vibration machines are characterized by that their reliability is reduced, materials consumption is high, and running time required for sufficient processing of materials is increased. Additionally, to the operation of vibration machines in the mode of normal harmonic vibrations of the vibration machine actuator, it is possible to operate vibration machines in modes of resonance harmonic vibrations with the frequency of the fundamental harmonic component and frequencies of other harmonic components, that is, to operate vibration machines in several harmonic vibration modes. The characteristic feature of such multimode vibration machines is the wide amplitude-frequency spectrum of the vibration machine actuator vibrations. For this reason, the use of multimode vibration machines is considered as perspective.

Analysis of study results and publications

Vibration machines are used in the building-and-construction industry in various processes associated with processing of construction materials [1]. Measurements of quantities related with vibration could be performed e.g. by using acoustic methods [2].

Discussed in paper [3] are systems with multimode shock-and-vibration machines, which are represented as dynamic systems with nonlinear operating characteristics. In such systems, vibrations are combined with shocks, and such shock-and-vibration machines operate in the mode of nonlinear vibrations of the machine actuator. In the paper, the optimal relationship between the period of an excitation force and the period of forced vibrations in the system with a shock-and-vibration machine is determined.

The operation of the vibration machines that are discussed in paper [4] are based on the nonlinear elasticity characteristics of medium with material being processed and change of the characteristics with time. As a result, the dynamic properties of the vibration machine are improved, and the vibration machine can operate in modes with higher harmonic components of the actuator vibrations. In such modes, energy consumption in operation of the vibration machine is significantly reduced.

Paper [5] proposes, in order to amplify the amplitude of the actuator vibrations, the operation of the vibration machine in the mode of parametric resonance vibrations instead of the operation in the mode of normal resonance vibrations. As the authors of this paper note, parametric resonance vibrations are characterized by a minimum threshold value of the vibration amplitude. In this case, the design of the vibration machine is complicated due to the necessity to install special springs, which are required for providing the vibration machine operation in the specified mode. The authors also note that, for increasing the efficiency of a process for exciting resonance vibrations, it is reasonably to provide the vibration machine operation in the mode of parametric resonance self-excited vibrations. To achieve the high energy efficiency of the multimode vibration machine, it is necessary to provide the availability of corresponding means for monitoring the parameters of the vibration machine actuator vibrations.

Discussed in paper [6] is the method for determining the time-dependant response of the medium with material being processed to the actuator vibrations. This method is based on the analysis of a hysteresis characteristic in connection with the actuator vibrations under the action of an excitation force. The method is efficient for analyzing the dependence of material deformation on stress in a medium with dissipative structure.

Paper [7] contains the results of analysis of the effect of finite constraints on the natural-vibration frequencies of a beam structure installed on an earth foundation. It is shown that, if the Winkler stiffness of the beam structure increases, the vibration frequency increases independent of the vibration mode. In the paper, there is no information on the mathematical model of the vibration machine-medium system, which would take into account the characteristics describing variations of dissipative forces. Other possible frequencies, which are distinctive from the basic vibration frequency, also are not considered.

Paper [8] proposes an experimental vibration machine designed for analyzing the vibration machine actuator vibrations at various loads and for determining interaction between the external excitation force and the amplitude of vibrations of the beam of the beam structure. The paper does not contain information about the foundation on which the beam structure is installed, and on the conditions required for generating the actuator vibrations with frequencies that are distinctive from the basic vibration frequency.

Discussed in paper [9] is the design problem that is characteristic for vibration machines with a specified structure. This design problem is stated as a problem with limitation of the operating characteristics of the vibration machine. The proposed methods for solving the problem apply only to linear steady-state vibrations.

In paper [10], it is shown that special measurement devices are required for analyzing modes and parameters of vibrations when performing static and dynamic tests of vibration machines.

Paper [11] contains the results of analysis of harmonic vibration modes. Discussed in papers [12, 13] is the mathematical model of a vibration machine-concrete mixture system with the discrete parameters of the concrete mixture as a medium. This mathematical model has specific limitations. As is shown in paper [14,15,16], if there is a specific relationship between the vibration period and the period required for the vibration wave to propagate within the medium, the mathematical model with discrete parameters is unreliable. In paper [17,18,19], it is noted that, for taking into account the viscosity and plasticity of a medium, it is reasonably to use a mathematical model in which the medium is characterized by distributed parameters. For analyzing the vibration machine actuator, it is reasonably to use a mathematical model in which the actuator is characterized by discrete parameters. So, the mathematical model of a vibration machine-medium should be combined, that is, should be presented as a discrete-continuous mathematical model [20,21,22].

The purpose of this paper consists in performing analysis of the vibration machine-medium system, which is required for determining stable operating modes of the system when designing new energy-efficient vibration machines for the building-and-construction industry. According to this purpose, the following tasks should be performed:

1. To develop the mathematical model of the system with consideration for the combination of vibrations of the vibration machine actuator and the response of the medium with material being processed to the actuator vibrations.
2. To determine the ranges of the system parameters in which operation of the vibration machine is stable in modes of harmonic vibrations and nonlinear vibrations.
3. To propose design variants for an energy efficient vibration machine.

Mathematical model of the system

In the mathematical model of a vibration machine-medium system, the vibration machine is characterized by discrete parameters, and the medium with material being processed is characterized by distributed parameters. According to such a discrete-continuous mathematical model, the forces acting on the medium with distributed parameters via contact between the vibration machine actuator and the medium are considered as discrete forces [16]. The forces acting form the medium on the actuator are considered as medium reaction forces. The medium reaction forces are divided into elastic-inertial forces and dissipative forces. The elastic-inertial forces from the vibration machine actuator react against the elastic-inertial forces from the medium. The equation that describes interaction between the forces is presented as an equation with discrete parameters. Solving this equation in order to determine vibration amplitude and power is possible by using standard methods. The vibration amplitude and power so determined are considered as discrete parameters. The equation coefficients are determined according to the distributed parameters of the medium [17,23]. The modes of operation of the vibration machine-medium comply with the following equation:

(1) $$nT_{\omega} = mT$$

where n is the number of periods of action of the external excitation force F, T_{ω} is the period of action of the excitation force, m is the number of periods of natural vibrations, T is the period of natural vibrations.

If $n = m$, harmonic vibrations with a basic resonance frequency are generated. If $m = 1$, under-resonance vibrations with a period $(1/n)T$ are generated. If $n = 1$ and $m \geq 2$, over-resonance vibrations are generated. Resonance vibrations with a fractional period $(m/n)T$ are also possible.

In designing vibration machines, the following assumptions are accepted:

1. The stabilization and control of a vibration machine-medium system is provided via excitation forces from the vibration machine and reaction forces from the medium, which act in combination.
2. The energy efficiency of a vibration machine with an elastic restoring force with a piecewise linear characteristic depends on vibration acceleration.
3. In determining the dynamic characteristics of the system it is necessary to take into account the natural vibrations of the system and the effect of higher harmonic components.

Analysis of the stability of a vibration machine-medium system in modes of harmonic vibrations

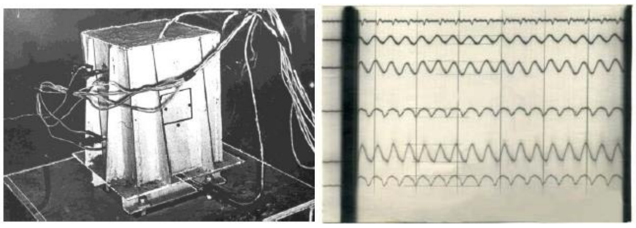
When simulating a medium with distributed parameters, there is the necessity to determine the wave properties of the medium and the areas of propagation of vibrations in the medium. Consideration for the dissipative structure of the medium is essential for mathematical simulation. Neglect of the dissipative structure, when determining the elastic-inertial component of the response of the medium to the vibration machine actuator vibrations, can cause erroneous results of the simulation. It is necessary to consider that, according to the vibration theory, the attached weight mc' for a medium without considering dissipation is determined as follows [24,25,26].

(2) $$m'_c = \left[\frac{\mathrm{tg}\left(\frac{\omega c}{h}\right)}{\frac{\omega c}{h}} \right] m_c$$

where m_c is the total weight of the medium, ω is the vibration frequency, h is the thickness of the medium layer in which the vibration wave propagates, c is the velocity of the vibration wave propagation.

If $\omega h/c = \pi/2+n\pi$, where n = 0, 1, 2, value $m_c' = \omega$, and such a value does not represent the facts.

For the analysis of interaction between the vibration machine actuator and the medium, studies were performed with the use of an experimental vibration machine (Fig. 1).



a) b)

Fig.1. Analysis of interaction between the vibration machine actuator and the medium with concrete mixture being processed:
a) Experimental vibration machine with installed sensors;
b) Vibration records for the area of contact between the actuator surface and concrete mixture (from the top downward, the following parameters are indicated: vibration phase, actuator vibration amplitude, concrete mixture amplitude, pressure on the concrete mixture)

The analysis results (see Figure 2) demonstrate that there is no separation of the concrete mixture layer from the vibration machine actuator surface.

a)

b)

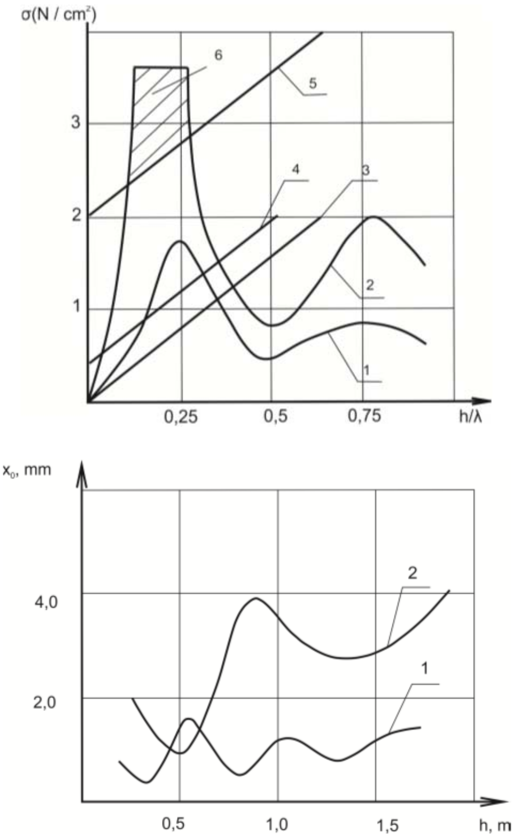


Fig. 2. Results of the analysis of interaction between the vibration machine actuator and the medium with concrete mixture being processed:
a) Variation of the dynamic pressure amplitude (σ) depending on the ratio of the concrete mixture thickness *(h)* to the vibration wave length (λ) (h/λ);
b) Limiting values of the vibration amplitude depending on the concrete mixture thickness at vibration frequencies of 25 Hz and 50 Hz;
1 - limiting values at a frequency of 25 Hz; 2 - limiting values at a frequency of 50 Hz; 3, 4, 5 - variation of pressure ρgh, $\sigma_c + \rho go$, and $\sigma_V + \rho gh$ correspondingly; 6 - area of separation of the concrete mixture layer from the actuator surface in the direction of vibration wave propagation.

For the analysis, the design model of the medium shown in Figure 3 was used. The system equations were obtained with consideration for the response of the medium to the actuator vibrations [14].

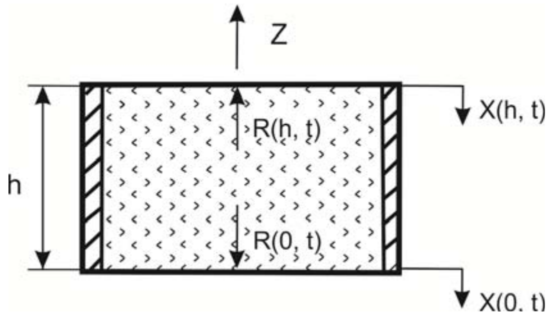


Fig.3. Design model of the medium

To determine the response of the medium, the following equation was used:

$$\frac{\partial^2 u(z,t)}{\partial^2 z^2} = \frac{\rho^*(z,t)}{E^*(z,t)} \cdot \frac{\partial^2 u(z,t)}{\partial t^2} \quad (3)$$

where $u(z,t)$ is displacement along Z axis depending on time t, $\rho^*(z, t)$ is the density of the concrete mixture, $E^*(z, t)$ is the Young's modulus.

The solution of equation (1) is presented as a Fourier series depending on the excitation force:

$$F(t) = \sum_{n=-\infty}^{+\infty} F_n e^{in\omega t} \quad (4)$$

where $\omega = 2\pi/T$, n = ±1, ±2. …, $F_n = \frac{1}{T} \int_{-r/2}^{r/2} F(\tau) e^{in\omega t}$.

The response of the medium to the actuator vibrations, with consideration for the boundary conditions (see Figure 3), is determined as follows:

$$R(0,t) = \sum_{n=-\infty}^{+\infty} Xn\omega n\omega_o n^2 \sqrt{a_n^2 + d_n^2 e} \quad (5)$$

m_c is the weight of the concrete mixture; a_n, d_n are phase factors.

The medium response contains two components, which are distinctive by aп and dп coefficients. These components characterize the effect of elastic-inertial (reactive) forces and dissipative (active) forces from the medium on vibrations in the complete vibration machine-medium system. The effect of these reactive and active forces was determined from calculations (see Figures 4-8) [28].

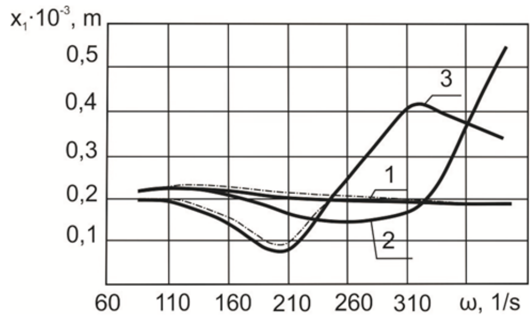


Fig. 4. Characteristics of the amplitude of the vibration machine actuator vibrations depending on the vibration frequency

If the frequency of the vibration machine actuator vibrations is ω= 310 1/s (see Figure. 4), and the thickness of the concrete mixture layer is h = 0.3 m, the actuator vibration amplitude is increased by a factor of two. The presence of resonance vibrations is confirmed by the characteristics in Figures 6 and 7. The experimental results presented in these figures are designated by dashed lines. These experimental results were used in designing the vibration machine for forming reinforced concrete plates.

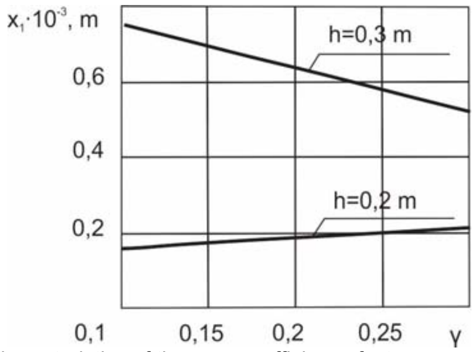


Fig. 5. Characteristics of the energy efficiency factor

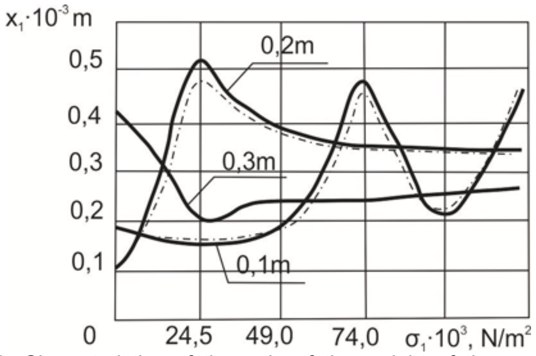


Fig. 6. Characteristics of the ratio of the weight of the concrete mixture to the weight of the vibration machine actuator

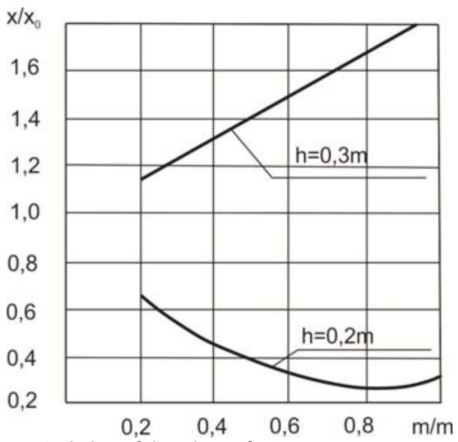


Fig. 7. Characteristics of the phase factor

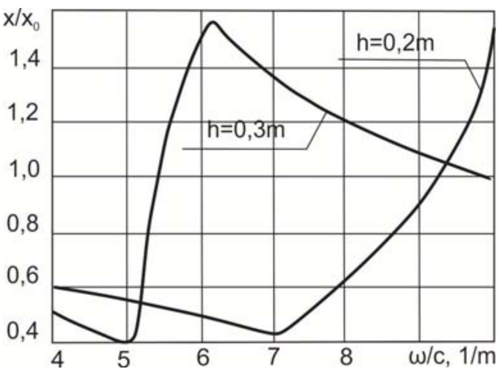


Fig. 8. Characteristics of the static pressure produced by the loading weight

Nonlinear vibration mode

The analysis of the nonlinear vibration mode is based on the equation that describes vibrations in the vibration machine-medium system under the action of an excitation force with a piecewise linear characteristic [14]. The equation is presented as follows:

$$(6)\quad m\dot{x} + b\dot{x} + \left\{ \begin{array}{ll} c_1 x, & |x| \le \Delta, \\ c_2 x + (c_1 - c_2)\Delta sign x, & |x| \ge \Delta, \end{array} \right. = F_0 \sin(\omega t + \phi)$$

where m is the reduced weight of the vibration machine including the actuator weight ma and the medium weight mm, x is the reduced displacement of the vibration machine actuator, b is the equivalent coefficient of the medium resistance, c1 and c2 are the coefficients of elasticity of the actuator and the medium, Δ is the value of displacement x which causes change of the restoring force, F_0, ω, and φ are the amplitude of the restoring force, vibration frequency, and phase angle of the excitation force.

The values of the equation (3) parameters are the following:

$$(7)\qquad m = m_a = m_m\ ,$$

where ma is the weight of the vibration machine actuator, mm is the weight of the medium,

$$(8)\quad m_m = \frac{\frac{SE}{\omega^2}\left\{\sum_{n=1}^{\infty}\sum_{n'}^{\infty} nn'\sqrt{a_n^2 + d_n^2}\, NN'\cos(\phi_n - \phi_{n'})\right\}^{\frac{1}{2}}}{\left\{\sum_{n=1}^{\infty}\sum_{n'}^{\infty} n^2 (n')^2 \sqrt{a_n^2 + d_n^2}\cos(\phi_n - \phi_{n'})\right\}^{\frac{1}{2}}}\ ,$$

where:

$$d_n = \frac{\omega n}{c_w \sqrt[4]{1+\gamma^2}} \sin\left[\frac{1}{2} arc\,\mathrm{tg}(-\gamma)\right];\quad \phi_n = n\omega t - arc\,\mathrm{tg}\left(\frac{d_n}{a_n}\right);$$

$$a_n = \frac{\omega n}{c_w \sqrt[4]{1+\gamma^2}} \cos\left[\frac{1}{2} arc\,\mathrm{tg}(-\gamma)\right];$$

$$N = \frac{\alpha_{11} sh(2\alpha_{1n} h) - \beta_{11}\sin(2\beta_{1n} h)}{ch(2\alpha h) + \cos(2\beta_{1n} h)}\ .$$

S is the area of contact between the actuator surface and the medium, E is the dynamic elasticity modulus, c_w is the velocity of propagation of elastic vibration wave in the medium in the direction of the applied excitation force, γ is the coefficient of energy scattering in the medium, h is the thickness of the medium layer, n is the number of a harmonic component.

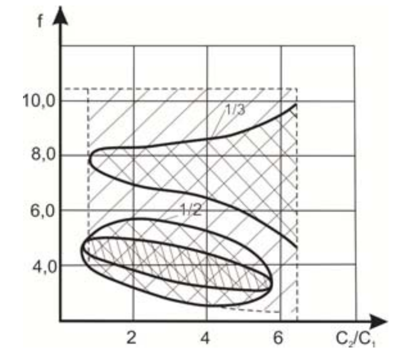
The values of parameters F_0, ω, b, and φ are determined by standard methods [14]. The solution of equation (9) is presented, for convenience, in dimensionless form:

$$\dot{y} + 2\delta\dot{y}\left\{ \begin{array}{ll} y & |y| \le 1 \\ \frac{c_2}{c_1} + \left(1 - \frac{c_2}{c_1}\right) signy ,\ , & |y| \ge 1 \end{array} \right. = f\sin(\nu\tau + \phi),$$

$$\phi_n = n\omega t - arc\,\mathrm{tg}\left(\frac{d_n}{a_n}\right);\quad N = \frac{\alpha_{11} sh(2\alpha_{1n} h) - \beta_{11}\sin(2\beta_{1n} h)}{ch(2\alpha h) + \cos(2\beta_{1n} h)}$$

In computer calculations performed for solving equation (5), the values of parameters $c1$, $c2$, f, and v were varied at values δ=0.02 and γ=0.2, which correspond to the level of energy loss in the vibration machine and medium [5]. The fulfillment of condition $nT_\omega = mT$ depends not only on the ratio of amplitudes of harmonic components but also on the amplitude of the external excitation force and the nonlinearity factor c2/c1 of the elasticity characteristic (see Figure 9a).

a)



b)

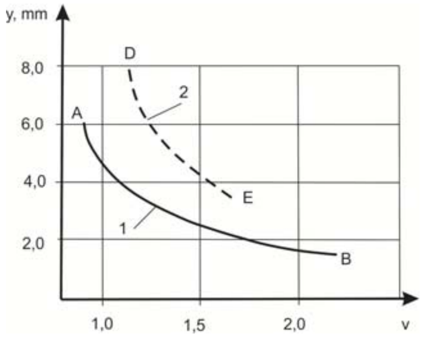


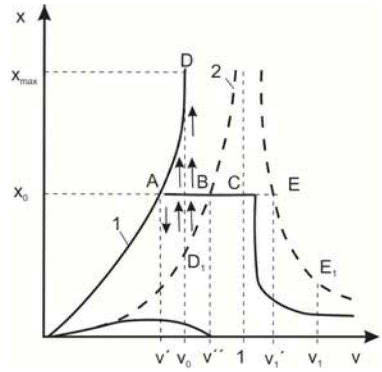
Fig. 9. Subharmonic vibrations in the vibration machine-medium system: a) area of subharmonic vibrations in coordinates *f* and c_2/c; b) Amplitude-frequency characteristics $y = f(v)$; 1 - for the basic vibration frequency; 2 - for the frequency with a freq. ratio of 1:2

The results of analysis of the amplitude-frequency characteristics of the system demonstrate that, with the specified system parameter values, the optimal relationship between the elasticity coefficients in a range of $3 \le c2/c1 \le 5$ is provided, and the coefficient of dynamic amplification of the vibration amplitude at under-resonance frequency is increased by a factor of 2 … 3. It is found that, for every vibration excitation frequency, there are limiting values of the excitation force, outside of which the stability of subharmonic vibrations does not ensured. The range of subharmonic vibrations can be extended by smooth variation of the vibration excitation frequency or excitation force amplitude f. Such extension of the subharmonic vibration range by variation of the excitation frequency or excitation force amplitude is evident in the change from amplitude frequency characteristic 2 to amplitude-frequency characteristic 1 in Fig. 9b.

When vibration excitation frequency increases, the ratio ω/ω0 reduces, and the operating point of the system moves to the left to point D of the characteristic in Fig. 10a. This point corresponds to the position of the vibration machine actuator in which the actuator touches the vibration limiter. When vibration excitation frequency further increases, the actuator vibration amplitude is restricted by the limiter, and the subharmonic vibration mode changes to a nonlinear vibration mode. In the frequency subrange that corresponds to section D-C in Figure10a, the vibration amplitude is limited to a value of x0, and the vibration acceleration increases. When vibration excitation frequency further increases, the system changes to a stable shock vibration mode, which is characterized by the position of the system operating point within section B-A in Figure 10a. The above mentioned characteristics of the system provide the possibility to set up required operation modes of the system by using program control facilities.

The amplitude-frequency characteristics of the system in the superharmonic vibration mode, which are shown in Figure 10b, illustrate the change of superharmonic vibration amplitude depending on the amplitude threshold value □. When the amplitude threshold value increases, the superharmonic vibration amplitude increases at all resonance frequencies, and modes of combination resonance vibrations with a frequency ratio of 3:2, which are detected at $\Delta > 2.5$, occur.

a)



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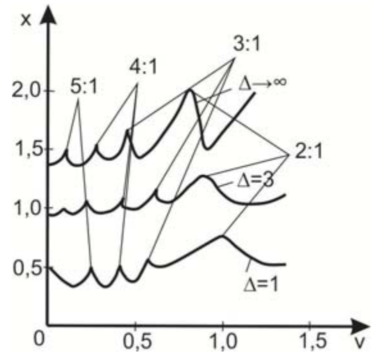

Fig. 10. Amplitude-frequency characteristics of the system in the superharmonic vibration mode:
a) Amplitude-frequency characteristics for the basic resonance vibration frequency in the system with a piecewise linear elasticity characteristic (1) and a nonlinear elasticity characteristic (2);
b) Superharmonic vibrations at various threshold amplitude values; A, B, C, D, and E - points of possible transition of the system from one operating mode to another operating mode*)*;

The results of analysis of the system amplitude-frequency characteristics demonstrate that increase of the threshold amplitude value causes insignificant change of the fundamental harmonic component amplitude and more significant change of higher harmonic component amplitudes of vibrations. For example, at the resonance frequency that corresponds to a frequency ratio of 3:2, amplification of amplitudes of harmonic components with different numbers is provided. At the resonance frequency that corresponds to a frequency ratio of 2:1, amplification of amplitudes of harmonic components with different signs is provided. At the resonance frequency that corresponds to a frequency ratio of 3:1, amplification of amplitudes of even harmonic components and attenuation of amplitudes of odd harmonic components is provided.

Design variants for an energy-efficient vibration machine

The results of studies performed were used in designing vibration machines which are characterized by the maximum concentration of energy, which is transferred from the vibration machine actuator to the medium with material being processed, due to the energy of higher harmonic components. The energy efficiency of the vibration machine is achieved due to the combination of impacts and vibrations (see Figure 11) by using additional vibration limiters with the corresponding stiffness and providing the optimal relationship between the impact action period and the vibration period.

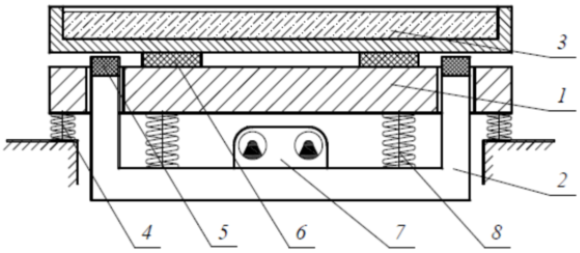

Fig. 11. Vibration machine:
1 – frame, 2 – impact device, 3 – form with material to be compacted, 4 – vibration-isolating supports, 5 – bumper for the impact device, 6 – additional bumper, 7 – vibration generator

The synchronous excitation of multiphase free vibrations in the vibration machine-medium system is provided by unbalance elements, which are rated for different vibration phases (see Figure 12).

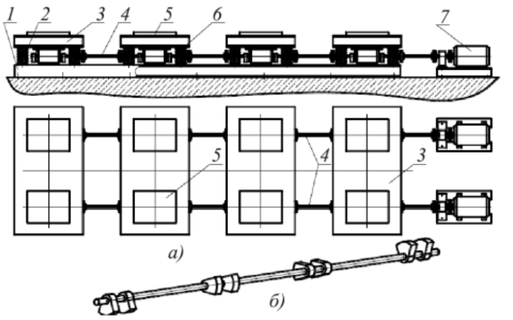

Fig. 12. Vibration machine for generating multiphase vibrations a) Appearance; b) Arrangement of unbalance elements on the shaft; 1 - frame, 2 - spring elements; 3 - section of the vibration machine; 4 - drive shaft; 5 - bumper of the impact device; 6 - unbalance element; 7 - driving motor

Conclusions

1. Mathematical model of a vibration machine-medium system, which consists of the mathematical model of the vibration machine actuator with discrete parameters and the mathematical model of the medium with distributed parameters is developed. The equations that describe motions in the system are based on the mathematical model with discrete parameters with consideration for the wave properties of the medium.
2. Determined are conditions for providing the stability of the system in operation in a harmonic vibration mode and a nonlinear vibration mode due to the combination of internal motions in the system.
3. Proposed are design variants for an energy-efficient multimode multiphase vibration machine, which provide the possibility to generate vibrations at frequencies of higher harmonic components, multiphase vibrations, and multifrequency vibrations.

***Authors**: D. Techn. Sc., Prof., Ivan I. Nazarenko, Kyiv National University of Construction and Architecture, 31 Povitroflotsky av. Kiyv, 03037, Ukraine, e-mail: i_nazar@i.ua; PhD. Techn. Sc., Ass. Prof., Mykola M. Ruchynskyi, of the Machinery and Equipment of Technological Processes, Kyiv National university of construction and Architecture, 31 Povitroflotsky av., Kyiv, 03037, Ukraine; PhD. Techn. Sc., Ass. Prof., Anatoliy T. Sviderskyi, Deparment of the Machinery and Equipment of Technological Processes, Kyiv National university of construction and Architecture, 31 Povitroflotsky av., Kyiv, 03037, Ukraine; Assistant Prof. Iryna M. Kobylyanska, Vinnytsia National Technical University, e-mail:akobilanskiy@gmail.com; MSc Damian Harasim, Lublin University of Technology, Nadbystrzycka Str. 38D, 20-618 Lublin, Poland.*

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