This paper reports a study into the movement of vibratory machines for various technological purposes that determined their stable zones. These zones warrant that the predetermined parameters of energy saving and energy-efficient mode are maintained. The structural scheme of energy transmission within the elements of a vibratory machine has been built. It is common for any design of the vibratory machine and its operating modes. The machine estimation scheme has been constructed taking into consideration a technological load, which is a certain manufacturing environment or a material subject to processing based on the appropriate technology. Underlying the motion equations built is a substantiated discrete-continual model of the vibratory machine and processing environment. The estimation scheme takes into consideration possible structural solutions for a vibratory machine whose movement modes are harmonious or impact-vibrational. The adopted scheme is a resonance vibration-impact system. This study into the movement and establishing the zones of stability has been adapted to simpler and more complex systems by reducing a combined discrete-continual model to the discrete one. The result reveals a qualitative pattern of the vibratory machine movement ensuring the specified mode of its operation. It was found that at the predefined frequency of impacts and weight of a working body, the efficiency of the impact-vibratory machine is determined by the impact speed. The distribution of the basic parameters of such vibration systems has been estimated; stability cards for different zones have been built.

This very approach opens up new possibilities for designing highly efficient vibration equipment. A stable resonance mode makes it possible to significantly reduce the energy cost of the manufacturing process and warrant the rational parameters of vibratory machine operation specified by the technology. The results obtained were applied for the development of methods for calculating and constructing a new class of vibratory machines that implement appropriate energy-saving stable zones of the workflow

Keywords: vibratory machine, discrete and continual models, modes, parameters, stability zones

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DETERMINING THE REGIONS OF STABILITY IN THE MOTION REGIMES AND PARAMETERS OF VIBRATORY MACHINES FOR DIFFERENT TECHNOLOGICAL PURPOSES

I. Nazarenko
Doctor of Technical Sciences, Professor, Head of Department*

O. Dedov
Doctor of Technical Sciences, Associate Professor*
E-mail: dedovvcbk@ukr.net

I. Bernyk
PhD, Associate Professor, Head of Department
Departament of Microbiology and Processing Technology
Vinnytsia National Agrarian University
str. Sonyachna, 3, Vinnytsia, 21027

I. Rogovskii
PhD, Senior Researcher
Research Institute of Engineering and Technology***

A. Bondarenko
PhD (candidate of technical sciences), Head of Department
Department of mechanical engineering
Odessa State Academy of Civil Engineering and Architecture
Didrichson st., 4, Odessa, Ukraine, 65029

A. Zapryvoda
PhD, Associate Professor
Department of Architectural Constructions**

V. Slipetskyi
Deputy Director
Department of Procurement and Contract Policy
Corporation «DBK-ZHYTLOBUD»
Luhova str., 13, Kyiv, Ukraine, 04074

L. Titova
PhD, Associate Professor
Department of Technical Service and Engineering Management
named after M. P. Momotenko***

*Department of Machines and Equipment Technological Processes**
**Kyiv National University of Construction and Architecture
Povitroflotskyi ave., 31, Kyiv, Ukraine, 03037
***National University of Life and Environmental Sciences of Ukraine
Heroiv Oborony str., 15, Kyiv, Ukraine, 03041

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1. Introduction

Vibratory machines are widely used in many sectors of the national economy such as the construction, mining, food, chemical industries for executing various manufacturing processes: grinding, sorting, mixing, compaction. The effectiveness of their application is determined by the difference between modes and parameters, due to a particular technology.

At the same time, the universal factor is the vibration effect on the processed environment or material as a result of which the environment is set into motion changing the state of rest with the manifestation of the appropriate manufacturing process. Existing vibration equipment mostly operates under a harmonious over-the-resonance mode with significant energy consumption. Estimation parameters are determined when using empirical dependences that produce reliable results only under conditions predetermined by the specific experimental conditions. The energy of vibration processes in different systems is key in searching for solutions aimed at designing energy-saving vibratory systems, machines, robots, devices. The application of resonance energy-saving regimes is constrained by the lack of generally accepted estimation models. The search for and application of reliable actual process models should be based on revealing the patterns of changes, during the manufacturing process, in the working bodies of vibratory machines and processed environments that are different in their physical properties. One way to solve the issue is the idea of applying a hybrid model that takes into consideration both discrete and distributed parameters. We have overcome the mathematical complexities related to solving such problems by reducing discrete-continual systems to discrete ones. Such a discrete model in the analytical movement equations of the system «vibratory machine – processing environment» retains continual properties in the form of wave coefficients. These coefficients in their analytical form take into consideration both reactive (elastic-inertial) and active (dissipative) resistance forces that occur in the real oscillatory process in any vibration system. Such a model adequately reflects the actual pattern of the progress of a vibration process while analytical estimation parameters provide for a stable resonance mode of vibratory machine operation. This approach defines the relevance of this work since it is advisable to conduct a study aimed at determining the movement and establishing the stable zones by reducing a combined discrete-continual model to the discrete one. Shorter time for the course of any process is a key paradigm for indicators such as material capacity, efficiency, performance.

2. Literature review and problem statement

Constantly improving, technology encourages the research and development of the motion theory and the methods for designing vibration systems and machines. Work [1] devised a new scheme for exciting the oscillations of the working bodies of vibratory machine units based on changing the phase angles of unbalanced masses. The implementation of such an idea allowed for one spin of the unbalanced mass to execute the number of vibration effects on the manufacturing environment in proportion to the quantity of vibratory units that the installation is equipped with. Thus, the authors implemented the spectrum of frequencies that greatly increases the efficiency of the manufacturing process. At the same time, the cited work's limitation relates to considering a specific scheme of the phase arrangement of unbalanced masses along the central axis of the vibration installation. More research is needed for other schemes of the phase arrangement of unbalanced masses along the central axis of the vibration installation. Paper [2] reports the results of a study that made it possible to design vibratory sites and vibration installations executing multicomponent oscillations excited by vibration exciters with a vertical shaft. The reported average frequency of oscillations is $\omega=157\ s^{-1}$. Along with that, this frequency can be used only for loose mixtures, which narrows the scope of such oscillation frequencies. However, there remained unresolved issues about increasing the oscillation frequencies. In addition, the procedure for calculating parameters implies the use of empirical dependences, which produce satisfactory results only for a given structure [3]. The employment of the parametrical resonance [4] as a phenomenon of resonant strengthening of the system's movement is one of the areas towards reducing the energy intensity of a workflow. The proposed scheme of the implementation of the parametric resonance, when compared to the regular resonance, makes it possible to obtain a higher output power to accumulate the vibration energy. The disadvantage of the proposed solution is a complex design that reduces the reliability of its operation. Article [5] discusses the theory of oscillatory normal regimes and its applicability to general mechanical systems, including a resonance mode. Efforts to generalize modal analysis for nonlinear cases were considered. It was noted that the capability of mechanical systems to execute effective oscillatory movements can be significantly enhanced by adequately forming their inertia properties and those potential fields that act on them. This is very well understood for linear systems where the relationship connecting the physical parameters and the resultant free oscillations in mechanical structures is clearly explained by the well-established theory of linear modal oscillations. The cited work does not define the nature of dissipative forces and their impact on oscillations, which is a prerequisite for ensuring a stable resonance movement regime. This is noted by the authors of work [5]. They believe that further studies must account for the dissipation that is introduced into the system. A pattern of the qualitative and quantitative transformation of an ultrasonic device's force action on a processing environment is given in work [6]. The peculiarities of ultrasound device interaction with the technological environment, which have different rheological properties, have been investigated and identified. The authors established criteria that comprehensively evaluate the energy, force, acoustic, and time parameters of the process and determine the ratio of the wave resistance of the cavitating region to the initial resistance of the environment. The proposed method can be successfully used in other processes; however, it is necessary to conduct additional research into other materials to determine general patterns. The movement of resonance vibration devices was investigated in work [7]. However, when the effectiveness of the modes was determined, the authors did not specify which manufacturing environments fit the results obtained. The variable modes in the conditions of interaction between the vibroacoustic apparatus and processing environment [8] should be determined not only theoretically but also confirmed by using specific examples. Paper [9] uses a systematic way to select modal derivatives from a set of linear vibration modes based on the example of straight and curved beams, as well as panels. The effectiveness of such an approach requires evaluation for vibration systems, as a process of their interaction. Studies of nonlinear characteristics and application of the effects of combination modes are reported in work [10], which considers the issues related to an oscillatory effect on the nonlinear dynamic systems arising in various fields of science. It is noted that such problems may have significant applied and theoretical importance, in particular due to that the general properties of systems can significantly affect oscillatory actions. This approach must additionally clarify the transition from the initial control equations of movements to the equations describing only a slower component of the movements. Article [11] outlines the theory of oscillatory modes and its applicability to general mechanical systems by employing the generalization of modal analysis to a nonlinear case. The article does not define the nature of dissipative forces and their impact on oscillations. Work [12] uses a dynamic modeling approach for the nonlinear vibration analysis of a piping system with clamps. It is shown that the hysteresis loop used is an important way of expressing the characteristics of energy dispersion in the system. However, there remained unresolved issues related to the procedure for recording hysteresis loops and the parameters used when they were measured. Work [13] reports a study into the dynamics of oscillatory systems described by differential equations with the added hysteresis nonlinearities for the case when a hysteresis loop moves clockwise. Similar to the previous work, there are no data on measurement methods of the degree of influence exerted by hysteresis nonlinearities. The use of a hysteresis loop to assess dissipative properties was described by the authors of study [14]. However, the proposals are aimed at ensuring an increase in the vibration protective properties only in a narrow field of spring mass resonance. The methodology for converting a discrete elastic-plastic model into a continual model reflects the views by the authors of [15]. This very approach, reported in [15], deserves special attention. Therefore, it is advisable to conduct a study aimed at determining the movement and establishing the stability zones by reducing a combined discrete-continual model to the discrete one. Less energy for the progress of any process is also important for indicators such as material capacity, efficiency, performance.

3. The aim and objectives of the study

The aim of this work is to determine the zones of stability for the modes and parameters of a vibratory machine in order to reduce energy consumption and improve the efficiency of the manufacturing process.

To accomplish the aim, the following tasks have been set:

– to substantiate the procedure and devise an estimation scheme for the vibratory machine taking into consideration the influence exerted by the technological environment;

– to investigate and analyze changes in the basic parameters affecting the stability zones of the vibratory machine;

– to establish parameters for the vibratory machine stability zones.

4. Devising an estimation scheme of the vibratory machine and a procedure to study its movements taking into consideration the influence exerted by the technological environment

An estimation scheme of the vibratory machine, regardless of its purpose within a particular manufacturing process, is determined on the basis of the adopted structural scheme (Fig. 1).

This structural scheme is general; it underlies building an estimation scheme of the machine taking into consideration a manufacturing loading, which is a certain environment or material to be processed. Most materials undergo three stages of processing in the direct or reverse direction thereby manifesting their properties in the form of rheological models (Fig. 2). In the forward direction, these may include compacting processes [1, 16]; in the opposite direction – the processes of vibroacoustic treatment of liquid environments [6, 8].

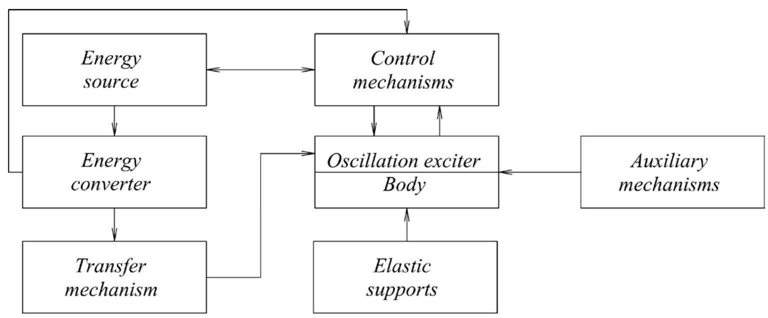


Fig. 1 Structural scheme of energy transfer in the elements of a vibratory machine

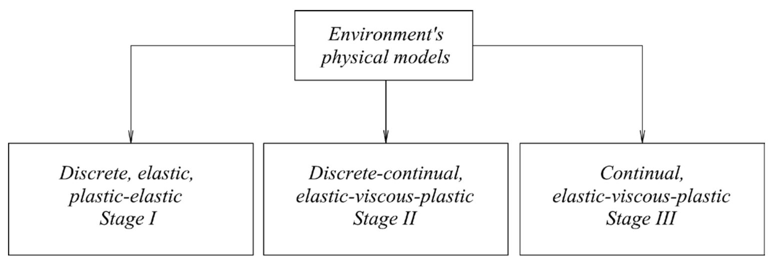


Fig. 2. Models of three stages of processing technological environments

The choice of a particular model is determined by the following considerations. A source model to use is a model with the distributed (continual) properties involving an assessment of the qualitative pattern of the time of wave propagation compared to the period of oscillations. If there is a wave process in an environment under the influence of a particular force, it would propagate at the following rate, c [8]:

$$c=\sqrt{E/\rho}, \quad (1)$$

where E is the elasticity module; ρ is the environment density.

The state of elastic perturbation would propagate over a distance of l during [1]:

$$\tau=l/c=l\sqrt{\rho/E},$$

Then the ratio of the time of oscillation propagation τ to the period of forcing T, which was used in work [16] for vibration systems, can be recorded as:

$$1>\tau/T<1. \quad (2)$$

Given (2), we obtain two conditions. If condition (2) is met in the following form:

$$\tau<T, \quad (3)$$

where T is the period of oscillations $T=2\pi/\omega$ (ω is the circular frequency of oscillations). It follows from (3) that the system can be modeled with discrete inertial or elastic parameters. This is due to that during the period of oscillations the time of oscillation propagation does not affect the change in the state of the system. At the same time, taking into consideration only the mass (inertial) or elastic forces is a complete idealization, which is why the discrete schemes for actual tasks must account for the elastic and inertial properties.

Under another condition (2) in the form:

$$\tau \geq T, \tag{4}$$

we obtain a pattern of the impact exerted by the time of oscillation propagation on a change in the state of the system. Under condition (4), the system in its movement should take into consideration wave phenomena and be modeled with the distributed parameters.

Thus, the vibration system «machine – environment», considering different features of their performance, will be represented, in terms of modeling, as the subsystems with the discrete (machine) and distributed (environment) parameters. This assumption is the following condition for choosing a mathematical model. Thus, one subsystem of the system is capable of accumulating the energy that transfers from one form to another (reactive resistance), while the second – energy scattering (active resistance) [1]. Such assumptions underlie the movement equations of a vibratory machine, taking into consideration the manufacturing loading. The final assumption is the choice of the law to change the elastic and dissipative characteristics of the original model, both the vibratory machine and environment. Nest, the research methodology implies the implementation of the following operations. The equations of the system movement are built under modes with and without detachment. Then, to reduce the number of parameters, the equations are reduced to a dimensionless form while determining the initial conditions for movement, speeds, and time. Next, the initial conditions are accepted; the stages in the movement of the system in contact and without contact are considered. Parameters that affect the stability of the system are determined; its stability card is constructed; the boundaries of changes in the movement parameters are set. An estimation scheme of the vibration system «machine – environment» is shown in Fig. 3.

In the estimation scheme (Fig. 3, *a*), m is the mass of the machine and the reduced mass of the manufacturing environment. The elasticity coefficients c_1 and c_2 determine the impact of elastic forces on the movement dynamics of the vibration system «machine – environment». The coefficients b_1 and b_2 determine the impact of dissipative forces in the system. The x_0 and F parameters are the distance between the head and the limiter and the amplitude of the forced harmonious force. The movement is carried out at an oscillation frequency of ω, and t is the current time. In the scheme of determining the resistance forces of an environment (Fig. 3, *b*), the following designations are adopted. The displacement is denoted via x; the external forcing – $F(t)$. The longitudinal displacement of the current cross-section of the technological environment column is indicated by $u(z, t)$. The reaction of the technological environment to oscillations in the cross-section $z=0$ is accepted as $Rl_{z=0}$, the reaction of the technological environment in the cross-section $z=h$ of the column is denoted via $Rl_{z=h}$. The mass m is determined from the following expression:

$$m = m_b + m_c, \tag{5}$$

where m_b is the weight of the vibratory machine executing oscillations; m_c is the mass of the technological environment derived from determining the reaction (Fig. 3, *b*) according to the procedure given in work [8]. We accept the wave equation of the displacement of the current cross-section of the environment's column in the following form [8]:

$$\frac{\partial^2 u}{\partial t^2} = c^2 (1 + i\gamma) \frac{\partial^2 u}{\partial z^2}. \tag{6}$$

In the given wave equation, $\partial^2 u/\partial t^2$ is the acceleration of an environment's layer. The parameter c is the speed of movement in the environment's layer; the γ coefficient is the resistance of the environment, which characterizes the quantitative amount of energy dispersion. The $\partial^2 u/\partial z^2$ term defines the second derivative from the movement of the environment's layer u along the coordinate of the environment's column, the size of z. An imaginary unit i is an imaginary unit that shows the rotation between the elastic and dissipative component at angle $\pi/2$. The difference between the analytical dependence of the environment's resistance from that reported in work [8] is that the resistance rate of the environment that characterizes the quantitative value of energy dispersal was determined on the basis of another dependence.

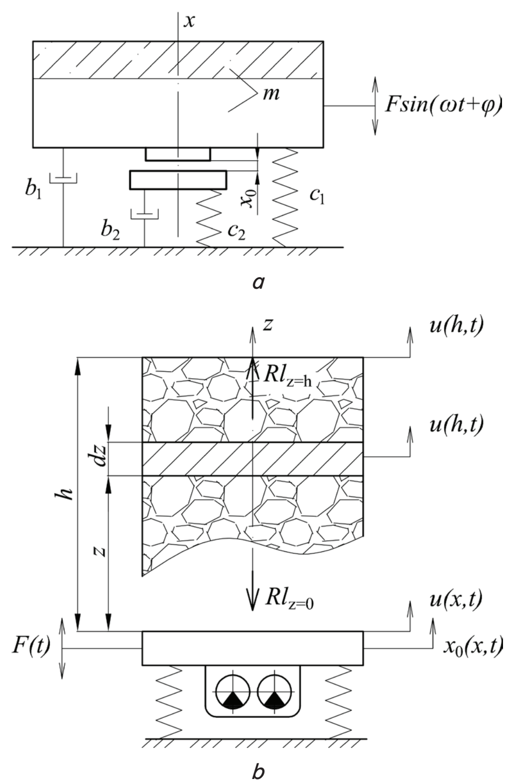


Fig. 3. Vibration system «machine – environment»: *a* – estimation scheme; *b* – scheme for determining the resistance forces of an environment

Following the approach of studies [1] and [8], and by omitting the intermediate operations, we obtain analytical expressions to determine the mass of the environment in the corresponding parts of the period:

– for part of the system's movement period in the absence of contact with the oscillation limiter, $0 \leq t \leq \tau_1$:

$$m'_{\delta} = \frac{2\rho \cdot c \cdot S \cdot \left[\sin\left(2\frac{\omega}{c}h\right) + \gamma/2 \cdot S \cdot h \cdot \left(\omega / c \cdot h \cdot \gamma\right)\right] \cdot \tau_1}{\left(\gamma/4 + 1\right) \cdot \left[\cos\left(2\frac{\omega}{c}h\right) + c \cdot h\left(\frac{\omega}{c}h \cdot \gamma\right)\right] \cdot (\tau_1 + \tau_2)}; \tag{7}$$

– for part of the period of the joint movement of a vibration installation and the limiter of oscillations, $\tau_1 \le t \le \tau_2$:

$$m'_\delta = \frac{2\rho \cdot c \cdot S \cdot \left[\sin\left(2\frac{\omega}{c}h\right) + \gamma/2 \cdot S \cdot h \cdot \left(\omega / c \cdot h \cdot \gamma\right)\right] \cdot \tau_2}{\left(\gamma/4 + 1\right) \cdot \left[\cos\left(2\frac{\omega}{c}h\right) + c \cdot h\left(\frac{\omega}{c}h \cdot \gamma\right)\right] \cdot \left(\tau_1 + \tau_2\right)}. \quad (8)$$

The resulting formulae (7), (8) can now be used for more complex structural schemes of vibratory machines. Their discrete form simplifies the mathematical system «vibratory machine – environment»; however, when performing numerical calculations, formulae (7), (8) considered the parameters of a wave process in an environment. An important result is that when the system moves beyond an impact zone ($0 \le t \le \tau_1$), the value of the weight of the mixture differs from the mass at the time of impact ($\tau_1 \le t \le \tau_2$); these values are the same only for the linear system ($\tau_1 = \tau_2 = T/2$).

5. The study and analysis of changes in the basic parameters affecting the stability zones of a vibratory machine

The movement equation for the vibration system «machine – environment» (Fig. 3, *a*) takes the following form:

Provided $x \le x_0$:

$$m\ddot{x} + b_1\dot{x} + c_1 x = F_a \cos(\omega t + \varphi). \quad (9)$$

Provided $x > x_0$:

$$m\ddot{x} + \left(b_1 + b_2\right)\dot{x} + c_1\left(x - x_0\right) + c_2 x = F_a \cos(\omega t + \varphi), \quad (10)$$

where φ is the initial phase of the forcing.

By dividing equations (9) and (10) by m, after the transform, we obtain:

$$\ddot{x} + h_1\dot{x} + \omega_1^2 x = \frac{F_a}{m_2}\cos(\omega t + \varphi) \text{ at } x \le x_0; \quad (11)$$

$$\ddot{x} + h_1(1+\varepsilon)\dot{x} + \omega_1^2(1+\gamma_1^2)x = \frac{F_a}{m_2}\cos(\omega t + \varphi) + \omega_1^2 x_0 \text{ at } x > x_0, \quad (12)$$

where $h_1 = \frac{b_1}{m}$; $h_2 = \frac{b_2}{m}$; $\omega_1^2 = \frac{c_1}{m}$; $\varepsilon = \frac{h_2}{h_1}$; $\gamma_1^2 = \frac{c_2}{c_1}$.

To proceed to the dimensionless coordinate system, we introduce the following designations:

$$x = \frac{F_a}{m\omega_1^2}\eta; \quad x_0 = \frac{F_a}{m\omega_1^2}\eta; \quad \tau = \omega t.$$

Then:

$$\dot{x} = \frac{F_a}{m\omega_1^2} \cdot \frac{d\eta}{d\tau} \cdot \frac{d\tau}{dt} = \frac{F_a\omega}{m\omega_1^2}\eta'; \quad \ddot{x} = \frac{F_a\omega^2}{m\omega_1^2}\eta''$$

(in the last expressions, and hereafter, the strokes indicate the differentiation for τ).

By substituting the resulting expressions in (11) and (12), we obtain the following equations:

$$\gamma^2\eta'' + \gamma\beta_1\eta' + \eta = \cos(\tau + \varphi) + \eta \text{ at } \eta \le \eta_0; \quad (13)$$

$$\gamma^2\eta'' + \gamma\beta_1(1+\varepsilon)\eta' + \left(1+\gamma_1^2\right)\eta = \cos(\tau + \varphi) + \eta \text{ at } \eta > \eta_0, \quad (14)$$

where

$$\gamma = \frac{\omega}{\omega_1}; \quad \beta_1 = \frac{h_1}{\omega_1}.$$

Equation (13) is integrated under initial conditions $\eta(0) = 0$; $\eta'(0) < \eta'_0$ until the moment τ_1, when $\eta(\tau) = 0$. The moment of collision between the head and limiter τ_1 is accepted as the impact onset. Equation (14) is integrated under initial conditions $\eta(0) = 0$; $\eta'(0) < \eta'_1$.

After the impact, the following condition is met:

$$\eta\left(\tau_y\right) = 0; \; \eta'\left(\tau_y\right)\eta_0.$$

We solved equations (13) and (14) using the Mathcad software in a wide range of parameters of the system «vibration installation – concrete mixture».

Impact speed, as the main parameter of the efficiency of an impact-vibratory machine, depends on the speed recovery coefficient on impact, the parameters that characterize the elastic and dissipative properties of the system. The maximum dimensionless impact speed is determined from the following expression:

$$\eta'_{\max} = -\frac{1}{(1-R)\left(1-\gamma^2\right)}. \quad (15)$$

Considering (15), after simple transforms, we obtain an expression for the maximum value of the phase angle:

$$\varphi_{\max} = \frac{3}{2}\pi. \quad (16)$$

The calculations employing the Mathcad software have confirmed that our study had revealed the important effect of phase angles on the achievement of a resonance regime at maximum speed. It was found that a maximum of the dimensionless speed is achieved at phase angle $\varphi_{\max} = 270°$. The $\varphi_{\max}$ phase angle value is reduced with an increase in time with the limiter, that is the time, which, in turn, grows with a decrease in the elasticity coefficient c_2. For a linear system, the maximum value of the phase angle is $\varphi_{\max} = 180°$. In a linear system, the time of an impact is conditionally accepted as the moment when the system passes the position of equilibrium.

Thus, a $\varphi_{\max}$ value changes from 180° to 270°. The dependences of angles φ_{1opt}, φ_{2opt}, and φ_{4opt} on γ_1 at different values of the elasticity coefficient are shown in Fig. 4, 5.

It follows from the diagrams in Fig. 4 that the angles φ_{1opt} and φ_{4opt}, in the transition from a nonlinear system with a completely rigid limiter to the linear one ($c_2 = 0$), monotonously change at $\pi/2$. Under the same conditions, the φ_{2opt} angle changes at 0.3 radians only (~17°).

The φ_{4opt} angle corresponds to the phase angle of the forcing at the time when the head is detached from the limiter; φ_{1opt} – at the time of resistance of the head to the limiter; φ_{2opt} – at the time of the maximum deformation of the limiter, that is, at the time when the speed exceeds the resonance.

Consider the way the system parameters affect the values of the phase angles φ_1 and φ_2. Fig. 6, 7 show the diagrams of impact speed dependences on frequency, as well as the phase-frequency characteristics for a series of values, over a wide range of changes in other parameters of the vibration system.

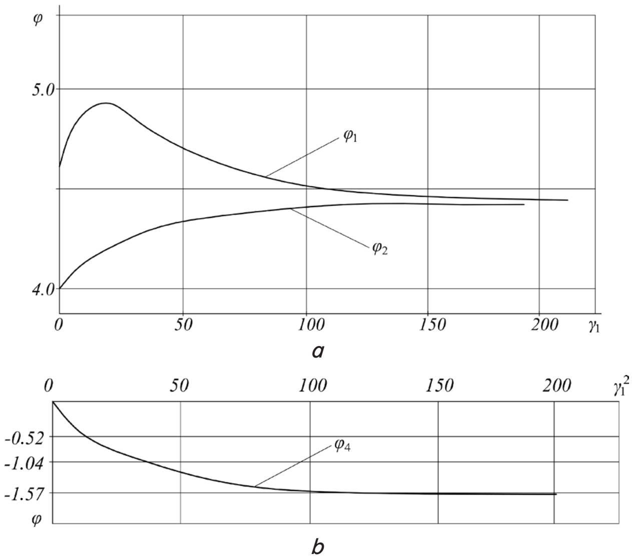


Fig. 4. Change in phase angles: *a* – φ_{1opt}, φ_{2opt}; *b* – φ_{4opt}

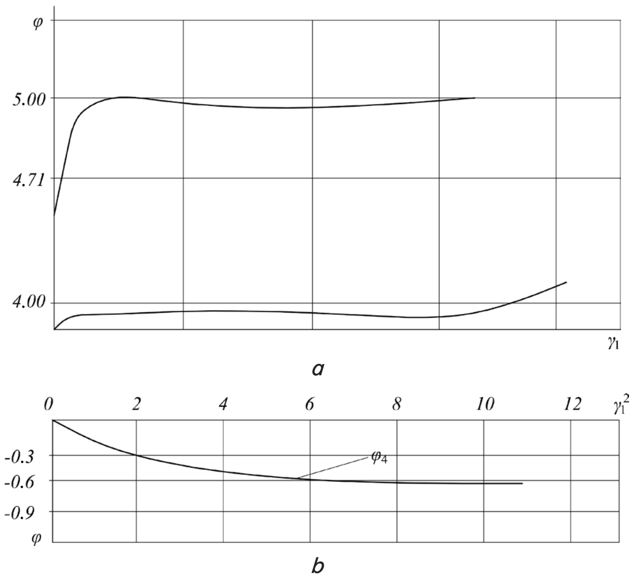


Fig. 5. Dependence of phase angles on the elasticity coefficient of the limiter: *a* – φ_{1opt}, φ_{2opt}; *b* – φ_{4opt}

Fig. 7 shows the following diagrams:

Fig. 6 shows the diagrams for three values of change in the reduced dissipative parameter: $\varepsilon = h_2/h_1$, $\beta_1 = h_1/\omega_1$, the ratio of frequencies $\gamma = \omega/\omega_1$ and elastic forces: $\gamma_1^2 = c_2/c_1$. Curves 1, 1*a,* and 1*b* are constructed for the following values: $\beta_1=0.4$; $\xi_0=0$; $\varepsilon=10$; $\gamma_i^2=10$; $\nu=1$. Curves 2, 2*a*, 2*b* – for the following values: $\beta_1=0.46$; $\xi_0=0$; $\varepsilon=10$; curves 3, 3*a*, 3*b* – for the following values: $\beta_1=0.46$; $\xi_0=0$; $\varepsilon=10$.

– curves 1, 1*a,* constructed for the following values: $\nu=1$; $\beta_1=0.4$; $\xi_0=0.075$; $\varepsilon=10$;

– curves 2, 2*a,* constructed for the following values: $\nu=1$; $\beta_1=0.4$; $\xi_0=0$; $\varepsilon=3$;

– curves 3, 3*a,* constructed for the following values: $\nu=1$; $\beta_1=0.3$; $\xi_0=0$; $\varepsilon=10$;

– curves 4 and 4*a*, constructed for the following values: $\nu=2$; $\beta_1=0.4$; $\xi_0=0$; $\varepsilon=10$.

These dependences show that over a wide range of changes in parameters (recalculated for the dimensional parameters of the displacement range x_0 from 0 to 6 mm, the mass doubled, the resistance b_z increased by 3.3 times), at a maximum impact speed, for each value γ_1^2, the constancy of the φ_{1opt} and φ_{2opt} phase angles is maintained at an accuracy of up to 0.1 rad. The optimum values for the phase angles φ_1, φ_2, at which the maximum impact speed is achieved, for machines $\gamma_1^2=10$ and 100, are equal to: $\varphi_{1opt}=1.3\pi$; $\varphi_{2opt}=1.5\pi$; and $\varphi_{1opt}=1.4\pi$; $\varphi_{2opt}=1.5\pi$, respectively.

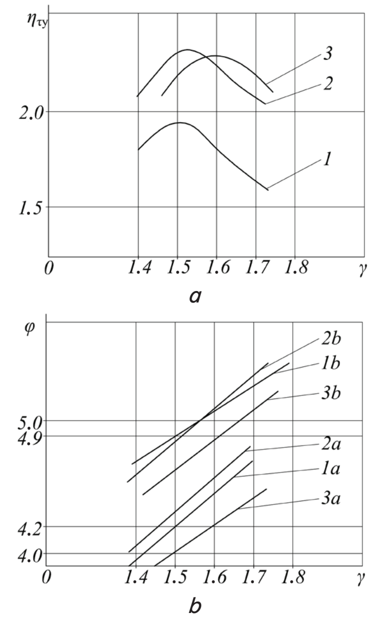


Fig. 6. Change in the basic parameters of a vibratory-impact machine: *a* – impact speed η on frequency γ; *b* – phase angles φ_1 and φ_2 on frequency γ

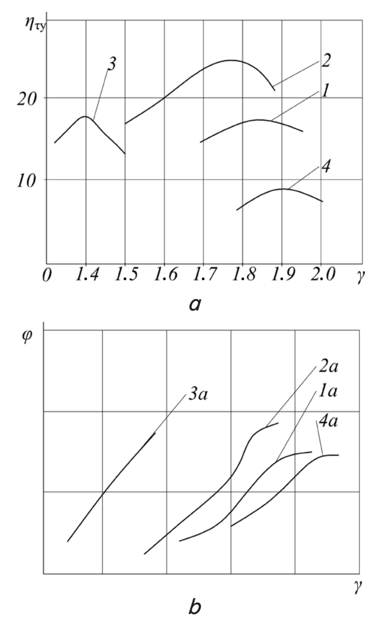


Fig. 7. Changes in the main parameters of a vibratory-impact machine: *a* – impact speed η on frequency γ; *b* – phase angles φ_1 and φ_2 on frequency γ

6. Establishing the parameters for a vibratory machine's stability zones

As established above, the vibratory-impact system with an oscillation limiter (Fig. 3, *a*) executes two movements: detachable and non-detached. To analyze and define stability parameters, we accept some simplifications that do not affect the nature of movement ($b_1=b_2=0$). Then:

– at $t<t_b$, the movement equation is:

$$m\ddot{x}=F(t);$$

– at $t<t_b$, the movement equation is:

$$m\ddot{x}+(c_0+c)x=F(t);$$

In case of the smallness $c_1\sim 0$ (on condition of the mass vibration insulation) m from the foundation, a half-scope at this stage of movement is:

$$x_0\le\delta; \qquad (17)$$

where x_0 is the amplitude of forced oscillations of mass m under a steady mode without breaking away from the limiter; δ is the static deformation of the limiter under the influence of weight force: $\delta=F_{cm}/c$; c is the stiffness of the limiter. When considering the non-detachable regime at $\Delta=x_0$, we obtain a system with one degree of freedom whose oscillation amplitude takes the following form: $x_0=F_0/(c-m\omega^2)$, or, taking into consideration that there is a resonance $c=m\omega_0^2$:

$$x_0=\frac{F_0}{m\omega^2}\left(\frac{1}{\frac{c}{m\omega^2}-1}\right).$$

Condition (17), considering $\delta=F_{cm}/c$, and $c=m\omega_0^2$ can then be recorded in the following form:

$$\frac{F_{cm}}{F_0}=\frac{\xi^2}{\xi^2-1}, \qquad (18)$$

where

$$\xi=\sqrt{\frac{c}{m\omega^2}}.$$

The ξ parameter determines the ratio of the system's eigenfrequency $m\omega_0^2=c/m$ when the mass is in contact with the spring to the frequency of forced oscillations ω^2. Consequently, at the predefined mass of a vibratory machine m, the parameters to be determined are the rigidity of elastic elements c and the external force F_0, since the frequency of forced oscillations ω is typically set by the appropriate technology. Thus, for a single-mass vibratory machine, there are the following criteria that define the zone of its stability:

$$\xi=\sqrt{\frac{c}{m\omega^2}}; \qquad (19)$$

$$\xi=\sqrt{\frac{c}{m\omega^2}};\quad f=\frac{F_{cm}}{F_0}. \qquad (20)$$

It should be taken into consideration, in this case, that in addition to finding the ξ and f parameters, one needs to know the time τ, the contact between the mass and spring. It is obvious that it makes up a certain proportion of the entire period of oscillations. The next step is to determine a change in the parameters' boundary reflecting the stable operation mode of impact-vibration systems. Since the impact-vibratory machines chosen in this study are the resonant ones, it is likely that $\xi\ge 1$. For example, at $\xi=1.3$, (19) would produce $f=1/8$. Based on the expression for the eigenfrequency of oscillations $\omega_0=\sqrt{c/m}$ under condition (17), one can determine the contact time between the mass m and the oscillation limiter:

$$t_k=\frac{2\pi}{\omega_0}=2\pi\sqrt{c/m}. \qquad (21)$$

Therefore, the ξ, f, and τ parameters are the criteria that define the mode of operation implemented under condition (17).

In case of violation of this condition, the movement of mass m would be executed with a separation from the spring under variable impacts against it. At the same time, the following modes are possible: one-impact, that is, over a single period of movement, the mass m executes one free flight and, consequently, one impact against the spring; super-harmonic, when over one period of change in the forcing frequency there are several impacts; sub-harmonic, when the number of impacts is n times less than the period of change in the forcing frequency.

In Fig. 8, *a*, frame 1 rests on supports 5, providing in this way the machine's vibration insulation. According to the scheme depicted in Fig. 8, *b*, the frame of working body 2 directly rests on the foundation by vibration-insulating supports 5, and the forcing force is applied to frame 6. In this case, similar to the scheme in Fig. 8, *a*, there is a need to establish auxiliary elastic links 4. The simplest scheme, in structural terms, is the one in Fig. 8, *c*; oscillation limiter 3 is installed between working body 1 and mold 7.

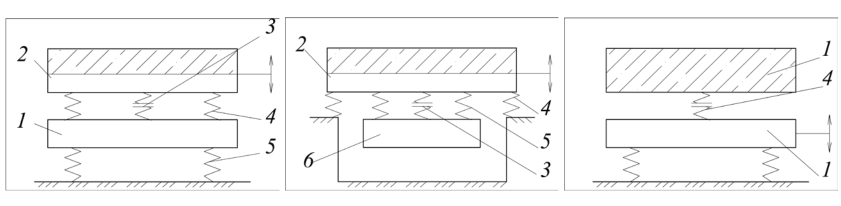
The ξ and f values can also be taken from a stability card.

Consider the equation of movement of such systems, find the ξ and f parameters, and determine their difference from expression (19), which was derived for a single-mass vibratory-impact machine. Provided that the forcing is applied to the mass m_1, we obtain:

$$m_1\ddot{x}_1+(c_0+c_1)x_1-c_1x_2=F_0\cos\omega t,$$

$$m_1x_1+(c_0+c_1)x_1-c_1x_2=F_0\cos\omega t. \qquad (22)$$

Equations (22) reflect the movement of the system in case of the contact violation.



a *b* *c*

Fig. 8. Schemes of two-mass vibration-impact machines:
a — the reactive mass rests on the foundation via supports; *b* — the weight of a working body rests via supports on the foundation; *c* — the external forcing is applied to the reactive mass

In a similar way, we obtain the equations for the joint movement of masses m_1 and m_2:

$$m_1\ddot{x}_1+(c_0+c_1+c)x_1-c_1x_2=F_0\cos\omega t,$$

$$m_1\ddot{x}_1+(c_1+c_2)x_2-(c_1+c)x_2=0, \qquad (23)$$

where C is the rigidity of oscillation limiters 3 (Fig. 8).

The derived system of equations (22), (23) can be somewhat simplified considering ($c_0 \ll c$) and the obvious connection $x=x_1+x_2$. By following these conditions and deducting the first equation from the second one, we obtain the following notation form for equations (22) and (23):

– at $x<0$

$$\ddot{x}+c_1\left(\frac{1}{m_1}+\frac{1}{m_2}\right)x=\frac{F_0}{m_1}\cos\omega t; \qquad (24)$$

– at $x>0$

$$\ddot{x}+(c_1+c)\left(\frac{1}{m_1}+\frac{1}{m_2}\right)x=\frac{F_0}{m_1}\cos\omega t. \qquad (25)$$

The number of variables in the resulting systems (24) and (25) can be reduced by applying the new dimensionless parameters (time τ and coordinate η):

$$\tau=\omega t;\ \ \eta=\eta_1-\eta_2;\ \ \alpha=\frac{m_1}{m_2};$$

$$\eta_1=\frac{m_2x_1\omega^2}{F_0};\ \ \eta_2=\frac{m_2x_2\omega^2}{F_0}. \qquad (26)$$

When substituting (26) in (24) and (25), we consider:

$$(\ddot{x}_i)_t=\omega^2(x_i)_t.$$

The new system of equations will be obtained following the appropriate transforms:

$$\alpha\eta_1-\eta_2=-\sigma\cos\tau;\ \ \ddot{\eta}_2+\xi_1\eta_2=-f-\alpha\cos\tau;$$

$$\ddot{\eta}_2+\ddot{\xi}_2\eta_2=-f-\sigma\cos\tau; \qquad (27)$$

here

$$\xi_1=\sqrt{\frac{(m_1+m_2)c_1}{m_1m_2\omega^2}},\ \ \xi_2=\sqrt{\frac{(m_1+m_2)}{m_1m_2}\frac{(c_1+c_2)}{\omega^2}}. \qquad (28)$$

$$f=\frac{m_2g}{F_0}\left(\frac{m_1+m_2}{m_2}\right). \qquad (29)$$

The σ sign in equations (27) accounts for the phase of a forcing. When comparing expressions (19) and (28), (20a) and (29), it is not difficult to notice that the difference between them is that the ξ_1 and ξ_2 parameters in dependences (28), while differing from (19), take into consideration the consolidated mass of the system (Fig. 8, *c*), as well as the presence of two rigidities c_1 and c_2. The experiments reported in [16] established that the rigidity c_1 represents a third of the support springs c_0; to determine the stiffness c and forcing F_0, the ξ_2 and f parameters for the implementation of the first stability zone (Fig. 9) are within the following limits:

$$0.8\le\xi_2\le1.3;\ \ 1.3\le f\le 2. \qquad (30)$$

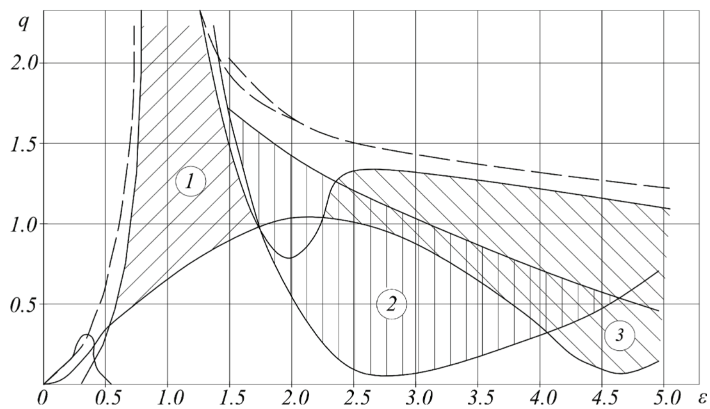


Fig. 9. Stability card of vibratory-impact machines

The ξ and f values can also be taken from a stability card.

The contact time τ also depends on the region of the stability zone. For the first zone, its averages can be taken as $\tau=(1/3)T$ [16], where T is the oscillation period of a vibratory machine.

Contact time, τ also depends on the area of the stability zone. For the first zone, its averages can be taken as [16], where T is the period of vibration oscillations.

When using the first stability zone, the amplification coefficient of oscillation amplitude k_A for the ratio of frequencies of forced oscillations ω to the natural oscillations ω_0, $\omega/\omega_0=0.87–0.91$, is $k_A=5–7$. Under this condition, energy costs for the manufacturing process decrease by 4–6 times. The discrepancy within these limits is due to the numerical values of the oscillation amplitude necessary to enable the technological process for a particular environment.

7. Discussion of results of studying the basic parameters of a vibratory machine and determining its stability zones

The results of our research show that at the predefined frequency of impacts and the weight of a working body, the effectiveness of an impact-vibratory machine is determined by the impact speed. This is a fundamentally new result, implying that a value of the phase angle decreases with an increase in time with the limiter, that is, the time, which, in turn, grows with a decrease in the elasticity coefficient. Our analysis of the analytical dependences and the above diagram (Fig. 7) has made it possible to establish the following. Over a wide range of change in the parameters at a maximum impact speed for each value, there is the permanence of the phase angles φ_{1opt} and φ_{2opt} at an accuracy of up to 0.1 rad. The optimal values of the phase angles φ_1, φ_2, at which the maximum impact speed is achieved, for impact-vibratory machines, at the numerical frequency ratio $\gamma_1^2=10$ and 100, are equal to $\varphi_{1opt}=1.3\pi$; $\varphi_{2opt}=1.5\pi$; and $\varphi_{1opt}=1.4\pi$; $\varphi_{2opt}=1.5\pi$, respectively. These results became the source information to determine a vibratory machine's stability zones. Thus, the parameter ξ determines the ratio of the system's eigenfrequency when the mass is in contact with the spring to the frequency of forced oscillations. The f parameter defines the weight ratio of the vibratory machine and external force. At the specified mass of a vibratory machine, we identified parameters to be determined. These include the rigidity of elastic elements and the external force, as the frequency of forced oscillations is typically set by the appropriate technology. By solving the equations of movement of two-mass vibration-impact machines, we derived expressions (28), (29), which differ from expression (19) for a single-mass vibratory-impact machine. When using the first stability zone, the amplification coefficient of oscillation amplitude k_A for the ratio of frequencies of forced oscillations to the natural oscillations ω_0, ω/ω_0=0.87–0.91, is k_A=5–7. Under this condition, energy costs for a manufacturing process are reduced by 4–6 times. The limitations of this study include the missing analysis of the second and third zones of stability of the modes and parameters of a vibratory machine. Studies to address this issue are planned as a continuation of this work's considerations about the idea of optimizing the parameters for other types of vibration equipment and other manufacturing processes. The proposed approach for designing an energy-efficient vibratory machine that would warrant the zone of stability of parameters in the resonance region could be used for other processes. Such processes include the moving, stirring, and sorting of materials.

8. Conclusions

1. We have substantiated the methodology and devised an estimation scheme for a vibratory machine taking into consideration the influence of the technological environment in the form of wave coefficients, which accounts for the discrete-continual model of a vibratory machine and the processing environment. The proposed estimation scheme adequately reflects the actual movement of the system and could serve a methodological procedure for investigating a similar class of vibratory machines.

2. The changes in the basic parameters affecting the stability zones of a vibratory machine have been investigated and analyzed. This is a new result, which reveals a qualitative pattern of the movement of a vibratory machine enabling the predefined mode of its operation. It was found that at the specified frequency of impacts and the weight of a working body, the efficiency of an impact-vibratory machine is determined by the impact speed. Over a wide range of changes in the parameters at a maximum impact speed for each value, there is the permanence of the phase angles φ_{1opt} and φ_{2opt} at an accuracy of up to 0.1 rad. The optimal values of the phase angles φ_1, φ_2, at which the maximum impact speed is achieved, for the impact-vibratory machines, at the numerical frequency ratio, are equal to $\varphi_{1opt}=1.3\pi$; $\varphi_{2opt}=1.5\pi$; and $\varphi_{1opt}=1.4\pi$; $\varphi_{2opt}=1.5\pi$.

3. We have established parameters for the vibratory machines' stability zones when using the first stability zone. The amplification coefficient of oscillation amplitude k_A, for the ratio of frequencies of forced oscillations ω to the natural oscillations ω_0, ω/ω_0=0.87–0.91, is k_A=5–7. Under this condition, energy costs for the manufacturing process decrease by 4–6 times.

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