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Simulation and development of energy-efficient vibration machines operating in resonant modes

***Abstract**. Studied in the paper are the dynamic characteristic of a vibrating machine with two vibrating elements. The vibrating machine is simulated as a discrete-continuous system. Set up are the motion equations, and determined are the basic parameters of the vibration machine. Defined are the criteria for the first and the second stability ranges of the vibration machine as a resonance system. Determined are the optimal numerical values of the basic parameters of the vibration machine which are required for ensuring energy-efficient operation of the machine. Proposed is a new energy-efficient vibration machine designed for forming concrete foundation blocks.*

Streszczenie.** W artykule opisano dynamiczną charakterystykę wibrującej maszyny z dwoma elementami wibrującymi. Maszyna wibracyjna jest symulowana jako dyskretny system ciągły. Ustawione są równania ruchu i określone są podstawowe parametry maszyny wibracyjnej. Zdefiniowane są kryteria pierwszego i drugiego zakresu stabilności urządzenia wibracyjnego jako systemu rezonansowego. Określone są optymalne wartości liczbowe podstawowych parametrów maszyny wibracyjnej, które są wymagane do zapewnienia energooszczędnej pracy maszyny. Proponowana jest nowa energooszczędna maszyna wibracyjna przeznaczona do formowania betonowych bloków fundamentowych. **(Symulacja i rozwój energooszczędnych maszyn wibracyjnych działających w trybach rezonansowych).

Keywords: actuator, mathematical model, energy efficiency, resonance vibrations.
Słowa kluczowe: siłownik, model matematyczny, efektywność energetyczna, drgania rezonansowe.

Introduction

Energy saving is very important in developing new technologies and industrial equipment. The existing technologies and industrial equipment do not comply sufficiently with modern requirements concerning energy saving. One of the methods for solving the problems associated with energy saving is the development of vibration machines rated for operation in resonant vibration modes. Resonance vibration machines are widely used in the mining industry, the building-and-construction industry, and other industries. It is reasonably to use resonance vibration machines in processes where the vibration machine actuator acts directly on the material being processed. Such processes include processes for transporting, sorting, mixing, and compacting materials. Effective operation of resonance vibration machines is possible if the elasticity and dissipation properties of the vibration machine-medium system are accounted for, because such properties have a direct effect on the operational mode of the vibration machine. The further development of resonance vibration machines will be based on the combination of elastic and dissipative forces acting in the vibration machine-medium system. For this reason, it is necessary to use new methods for simulating, studying, and designing such systems.

Analysis of study results and publications

One of the basic problems concerning vibration machines consists in developing and validating their mathematical models. Paper [1] contains information on simulating vibrations in complex systems with indefinite nonlinearities, which are often spatially localized. Studied in this paper is the method, based on inverse optimization, for simulating dynamic nonlinearities in locally nonlinear systems.

Studied in the paper [2] are the factors that cause changes in the dynamic response of a system consisting of two plates connected by a single fastening element. Notwithstanding the fact that paper [2] does not relates to the studies discussed in paper [1], paper [2] is of interest due to its conclusions on the elasticity modulus. It is determined that the elasticity modulus of the system depends on the system dynamic response. There is no information about the system operating modes in which the said effect becomes apparent.

Paper [3] contains the results of analysis of nonlinear dynamic systems. It is considered that the use of the conventional linear Kelvin rheological model for the analysis is over-simplification. The authors of the paper propose another method based on consideration of elastic and dissipative forces. The authors use an ordinary differential equation of third order for the analysis of vibrations in a degenerate system. Analyzed are all-harmonic vibrations, and used for the analysis is the method for setting up the system equations on the basis of energy balance and power balance. There is no information about interaction between systems with different properties and about the possibility of nonlinear vibrations.

Paper [4] contains the results of studying dynamic decay in linear mechanical systems with two degrees of freedom when two excitation forces act simultaneously on the system. Studied in the paper are the characteristic features of dynamic interaction between the system elements under action of external excitation forces. Under certain conditions, it is possible to control the system elasticity parameters.

The characteristics of natural vibrations in nonlinear systems with light damping, which are presented in paper [5], were used for the experimental studies of such vibrations. The study method is based on the estimation of the instantaneous frequency and amplitude of the envelope of damped vibrations after a period of steady-state vibrations in the mechanical system. The high accuracy of the results is ensured due to the assumption that the expected dissipation forces in the system do not significantly affect the vibrations in the system. This assumption is valid if dissipation in the system medium does not depend on frequency.

Paper [6] contains the characteristics of a nonlinear oscillator with two degrees of freedom. The characteristics are used for describing the oscillator operation on exposure to an external excitation force. Proposed in the paper is the analytical method for considering natural and forced vibrations. The authors of the paper performed the analysis, on the basis of energy balance, for predicting resonance operating points where the characteristics for natural vibrations intersect with the characteristics for forced vibrations. The authors propose the method for calculating accuracy in predicting the resonance operating points. The resonance operating points so predicted are compared with the corresponding resonance operating points on characteristics obtained in numerical simulation.

Proposed in paper [7] is the method for analyzing a two-stage nonlinear system with linear and nonlinear elasticity characteristics. According to this method, the two nonlinear differential equations of the system with two vibrating elements are reduced to a single nonlinear differential equation. This method is effective. Used in the analysis is also a linear differential equation of second order based on a variation method.

As is stated in paper [8, 21], the motion equations set up for vibration machines apply to vibration machines with two vibrating elements, in which the vibration machine actuator and the medium with material being processed are simulated as solid media. Presenting concrete mixture as a solid medium is improper [9]. The studies discussed in paper [10] demonstrate that concrete mixture on exposure to vibrations has elastic properties, and therefore is characterized as a medium with distributed parameters [13,14, 20].

Purpose of the paper

The purpose of this paper consists in determining stable ranges of resonance vibrations when a vibration machine is in operation. According to this purpose, the following tasks should be performed:

1. To develop the physical and mathematical models of a resonance vibration machine-medium system and to solve the motion equations of the system.
2. To study the parameters that are required for ensuring stable resonant modes and to determine methods for stabilizing resonant modes in operation of the vibration machine.
3. To propose a high-efficient vibration machine of new design.

Methodology for developing the mathematical model of the resonance vibration machine-medium system and setting up the system motion equations

In this paper, it is assumed that concrete mixture has elastic properties, and such properties should be represented in the mathematical model of the concrete mixture. Therefore, the mathematical model of the vibration machine-concrete mixture should be presented as a discrete-continuous mathematical model. When analyzing such a model, some problems occur, which are specifically characteristic for nonlinear resonance systems. To solve these problems, paper [11] proposes the method for changing from the discrete-continuous mathematical model to a discrete mathematical model with consideration for the wave properties of concrete mixture. For forming a new calculation model, it is necessary to determine the ratio between the period of an excitation force and the time of propagation of waves in concrete mixture. With this provision (see Figure 1), it is possible to change from the combined discrete-continuous mathematical model to a discrete mathematical model [18, 19, 22].

According to the design model (see Figure 1b), for the upper vibrating element and the upper spring element with discrete parameters, the wave properties of concrete mixture are taken into consideration [12]. In this case, the combined discrete-continuous mathematical model for the vibration machine-concrete mixture system should be such as is shown in Figure 2.

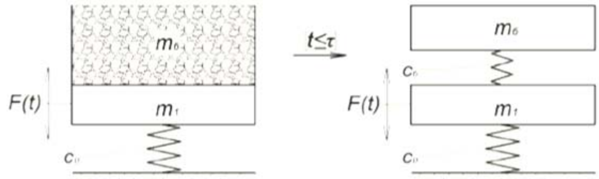


Fig. 1. Change from the combined discrete-continuous mathematical model (a) to a discrete mathematical model (b)

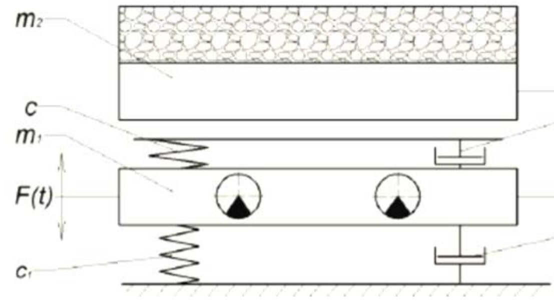


Fig. 2. Design model of the studied resonance vibration system: m_1 – reactive weight; m_2 – active weight, including the weight of the concrete mixture; c_0, c_1, c – elasticity coefficients for vibration-isolating, resonance, and vibration-limiting elements, correspondingly; b_0, b_1 – energy dissipation coefficients for the corresponding spring elements; x_1, x_2 – displacements of the vibrating elements m_1 and m_2; $F(t)$ – external excitation force

The shock-and-vibration machine with a vibration limiter (see Figure 3) is characterized by two vibration modes, that is, the mode with separation of the vibrating element from the vibration limiter and the mode without separation of the vibrating element from the vibration limiter. The condition for the mode without separation of the vibrating element from the vibration limiter is the following [15, 16, 17]:

$$x_0 \le \delta \quad (1)$$

where: x_0 is the amplitude of forced vibrations of the vibrating element m_2 in a steady state without separation of the vibrating element from the vibration limiter, δ is the deformation of the vibration limiter under gravity, $\delta = F_{cm}/c$, and c is the stiffness coefficient of the vibration limiter.

If this condition is not fulfilled, the vibrating element m_2 vibrates with separation from the spring element, with periodical impact action on the spring. In this case, superharmonic and subharmonic vibrations are possible.

According to the study technique, the following sequence of operations for performing the specified study tasks is accepted:

1. Setting up the motion equations for shock-and-vibration machine system in the mode of vibrations with separation of the vibrating element from the vibration limiter and in the mode of vibrations without separation of the vibrating element from the vibration limiter. In order to reduce the number of parameters in the equations, the equations should be set up in dimensionless form.
2. Determining and specifying the initial values of displacement, velocity, and time in the shock-and-vibration machine system equations.
3. Comparing the system equations at instants of time at which the mode of vibrations with separation of the vibrating element from the vibration limiter changes to the mode of vibrations without separation of the vibrating element from the vibration limiter, that is, when the velocities of the vibrating elements and spring elements are equal.
4. Determining the shock-and-vibration machine system parameter ranges in which the system operating mode is stable.

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

According to the study technique, the equation of the shock-and-vibration machine system with two vibrating elements in the operating vibration mode without contact between the vibrating elements is the following:

(2) $$m_{np}\ddot{x}^{(-)} + b_1\dot{x}^{(-)} + c_1 x^{(-)} = \alpha F \begin{Bmatrix} \sin \\ \cos \end{Bmatrix} (\omega t),$$

where: $m_{np} = \frac{m_1 m_2}{(m_1+m_2)}$ m is the reduced weight of the system, b_1 is the resistance coefficient, c_0 is the support elasticity coefficient, and α is the coefficient that accounts for the position of the point at which the excitation force F_0 is applied, and x is the reduced displacement of the vibrating elements with no contact between the elements.

The initial condition for vibration accelerations is the following:

(3) $$\ddot{x}^{(-)}\Big|_{t=0} = a_c; \ddot{x}^{(-)}\Big|_{t=t_1} = a_k.$$

The acceleration values are limited according to (3). Time is measured from the instant when the acceleration is maximum. With the specified initial condition, the solution of equation (2) for the displacement of the vibrating element is the following:

(4) $$x^{(-)} = e^{-\tilde{b}_1 t}\left[A_1 \cos(\tilde{\omega}_0 t) + A_2 \sin(\omega_0 t)\right] + \frac{\alpha F_0 \begin{Bmatrix} \sin \\ \cos \end{Bmatrix}(\omega t - \phi)}{\sqrt{(\omega_0^2 - \omega^2) + 4\tilde{b}_1^2\omega^2}}$$

where:

$$\tilde{b}_1 = \frac{b_1}{2m_{np}}; \tilde{\omega}_0 = \sqrt{\omega_0^2 - \tilde{b}_1^2}; \omega_0^2 = \frac{c_0}{m_{np}}; \phi = arctg\left\{\frac{2\tilde{b}_1\omega}{p_1^2 - \omega^2}\right\}.$$

The vibration velocity is determined as follows:

$$\dot{x}^{(-)}\Big|_{t=t_1} = -\tilde{b}_1 e^{-\tilde{b}_1 t_1}\left[A_1 \cos(\tilde{\omega}_0 t_1) + A_2 \sin(\tilde{\omega}_0 t_1)\right] +$$

$$+e^{-\tilde{b}t_1}\left[-\tilde{\omega}_0 A_1 \sin(\tilde{\omega}_0 t_1) + \tilde{\omega}_0 A_2 \cos(\tilde{\omega}_0 t_1)\right] + + \frac{2F_0\omega \begin{Bmatrix} \cos \\ -\sin \end{Bmatrix}(\omega t_1 - \phi)}{\sqrt{(\omega_0^2 - \omega^2)^2 + 4\tilde{b}_1^2\omega^2}}.$$

The coordinates A1and A2 are determined as follows:

$$A_1 = \frac{\Delta A_1}{\Delta}; A_2 = \frac{\Delta A_2}{\Delta}; \Delta = (\tilde{b}_1^2 + \tilde{\omega}_0^2)^2 \sin(\tilde{\omega}_0 t_1);$$

(5) $$\Delta A_1 = a_c(\tilde{b}_1^2 - \omega_0^2)\sin(\tilde{\omega}_0 t_1) - 2\tilde{b}_1\tilde{\omega}_0 \cos(\tilde{\omega}_0 t_1) + a_k 2\tilde{b}_1\tilde{\omega}_0 e^{\tilde{b}_1 t_1};$$

$$\Delta A_2 = a_k e^{\tilde{b}_1 t_1}(\tilde{b}_1^2 - \tilde{\omega}_0) - a_k\left\{(\tilde{b}_1^2 - \tilde{\omega}_0^2)\cos(\tilde{\omega}_0 t_1) + 2\tilde{b}_1\tilde{\omega}_0 \sin(\tilde{\omega}_0 t_1)\right\}$$

Let us to designate $\dot{x}^{(-)}\Big|_{t=t_1} = V.$

With contact between the vibrating elements, equation (2) can be presented as follows:

(6) $$m_{np}\ddot{x}^{(+)} + (b_1 + b)\dot{x}^{(+)} + (c_0 + c)x^{(+)} = \alpha F_0 \cos\left[\omega(t - t_1)\right],$$

where: $x^{(+)}$ is the displacement of the reduced system subject to forces in equation (5).

Let us to designate

$$\dot{x}^{(+)}\Big|_{t'=0} = V; t' = t - t_1, \; x^{(+)}\Big|_{t'=0} = 0.$$

The vibration velocity similarly to equation (5) is the following:

$$V = -\tilde{b}_1 e^{-\tilde{b}_1 t_1}\left[A_1 \cos(\tilde{\omega}_0 t_1) + A_2 \sin(\tilde{\omega}_0 t_1)\right] +$$

$$+e^{\tilde{b}_1 t_1}\left[-\tilde{\omega}_0 A_1 \sin(\tilde{\omega}_0 t_1) + \tilde{p}_1 A_2 \cos(\tilde{\omega}_0 t_1)\right] + \frac{\alpha F_0\omega\left[-\sin(\omega t_1 - \phi)\right]}{\sqrt{(\omega_0^2 - \omega^2)^2 + 4\tilde{b}_1^2\omega^2}}.$$

Let us to present the equation in dimensionless form. For this purpose, it is necessary to impose the following designations:

(7) $$m = m_{np};\; b = b_1 + b_0;\; F(t') = \alpha F_0 cos(\omega t');$$
$$\Omega = \frac{\alpha F_0}{mV};\; \tau = \Omega t' = \Omega(t - t_1);\; \varepsilon = \frac{mV^2}{\alpha F_0};\; z = \frac{x^{(+)}}{\varepsilon};$$
$$k = \frac{1}{\Omega}\sqrt{\frac{c_1 + c_2}{m}};\; \delta = \frac{b}{2\sqrt{(c_1 + c_2)m}};$$
$$f(t) = \frac{\alpha F_0 cos(\omega/\Omega \cdot \tau)}{m\varepsilon\Omega^2};\; f(0) = \frac{\alpha F_0}{m\varepsilon\Omega^2}.$$

Accordingly, the following equation is obtained:

(8) $$z'' + 2\delta k z' + k^2 z = f.$$

The solution of equation (8), with the specified initial conditions and designations of parameters, provides the possibility to determine the parameters that affect the stability of the resonance vibration system [13,14, 15].

1. The period τ_* for which the maximum deformation of the vibration limiter is achieved:

(9) $$\tau_* = \frac{\arccos\delta}{k\sqrt{1-\delta^2}} + \frac{f(0)}{k^2 V} + \frac{1}{k^2 V}\left[\frac{\delta}{V}f^2(0)\right].$$

It is evident that the actual period t_* for which the maximum deformation of the vibration limiter is achieved is determined depending on the period τ_* as follows:

$$t_* = \frac{\tau_*}{\Omega} + t_1.$$

2. The maximum deformation of the vibration limiter:

(10) $$a = \exp(-\delta k \tau_*)\left[c_1 \cos(k_1\tau_*) + c_2 \sin(k_1\tau_*)\right] + \frac{f(\tau_*)}{k^2}$$

where:

$$k_1 = k\sqrt{1-\delta^2}; c_1 = \frac{f(0)}{k^2}; c_2 = \frac{V}{k\sqrt{1-\delta^2}} + \frac{\delta f(0)}{k^2\sqrt{1-\delta^2}}; c_2 \gg c_1$$

The actual deformation is $x_{ma}^{(+)} = \varepsilon a$.

3. The period of impact action:

(11) $$\tau_{\gamma\partial} = \frac{\pi}{k\sqrt{1-\delta^2}} + \frac{1 + exp(\pi\delta/\sqrt{1-\delta^2})}{k^2 V} + \frac{1}{k^3}\left\{\frac{f(0)}{V^2}\delta\left[\frac{\pi\delta}{\sqrt{1-\delta^2}}\right]^2\right\}$$

$$t_{\gamma\partial} = \frac{\tau_{\gamma\partial}}{\Omega}$$

4. The velocity recovery coefficient:

(12) $$R = exp\left(-\frac{\pi\delta}{\sqrt{1-\delta^2}}\right) + \frac{1}{k}\frac{2f(0)\ln R_0}{V\sqrt{\pi^2 + \ln^2 R_0}} + \frac{1}{k^2}\frac{f^2(0)}{2v^2}\left(\frac{1}{R_0} - R_0\right),$$

where:

(13) $$R_0 = \exp\left(-\frac{\pi\delta}{\sqrt{1-\delta^2}}\right).$$

5. The velocity at the end of interaction between the vibrating elements:

(14) $$V_+ = -R_0 V + \frac{2\delta}{k^2}(1 + R_0)f(0) + \frac{1}{k^2}\cdot\frac{f^2(0)}{V}\cdot\frac{(1 - R_0)}{R_0}.$$

The actual velocity is $V_+ = \xi\Omega.$.

6. The energy absorbed by the vibration limiter due to impact action:

(15) $$W_{0n} = (1 - R^2)\frac{mV^2}{2}.$$

Equations (9) … (15) characterize the parameters of the operating modes of the shock-and-vibration machine. The calculations demonstrate that the period of impact action t_{imp} decreases with the increase of the elasticity coefficient c of the vibration limiter. If the coefficient c is constant, the period of impact action decreases with the increase of the weight m_2. The period according to the determined parameters is 0.01 … 0.018 c. For the frequency ω = 157 rad/s, the period is a fraction of 0.33 … 0.43 of the vibration period T. The vibration period is determined as T = t_{imp} + t_c, where t_c is the period of motion of the vibrating element without contact with the vibration limiter. If $V = x\omega$, the period of impact action is determined as follows:

$$t_{уд} = \frac{2}{\omega}\left(\pi - 1 - \frac{1}{R}\right).$$

(16) If $R \to 1$, $$t_{уд} = \frac{2}{\omega}(\pi - 2);$$

For $R \approx 0{,}5$, $$t_{уд} = \frac{2(\pi - 3)}{\omega}.$$

The value R is determined as $|a_{max}| \times t_{imp}$, where $|a_{max}|$ is the maximum impact acceleration. As the velocity is $V_+ = -RV$ after an impact and V before the impact, the acceleration is determined as follows:

$$|a_{max}| = \frac{(R+1)V}{t_{уд}} = \frac{\omega x (R+1)}{t_{уд}}. \tag{17}$$

Table 1 contains data on the velocity recovery coefficient and period of impact action for vibration frequency ω = 150 rad/s.

Table 1. Values of the velocity recovery coefficient (R) and period of impact action (t_{imp})

ω, s^{-1}	150	150	150	0,53	150	150
R	0,5	0,6	0,7	0,8	0,9	1,0
t_{imp} (s)	0,0019	0,0063	0,0095	0,012	0,014	0,015

The values of the basic parameters affecting the stability of vibrations demonstrate that for providing acceleration $a_{max} \approx 5g$ in impact action, the period of the impact action should be $t_{уд}$ = 0,003 s at $R \approx$ 0,3...0,4. The values of t_{imp} can be determined from equation (16).

The values of the basic parameters affecting the stability of vibrations in the vibration machine at ω = 150 rad/s and $x_0 = 7 \cdot 10^{-4}$ m are presented in Table 2.

Table 2. Values of the parameters affecting the stability of vibrations in the vibration machine at ω = 150 rad/s and $x_0 = 7 \cdot 10^{-4}$ m

$\|a_{max}\|$, relative to g	3	3	4	4	5	5
δ	0,36	0,28	0,36	0,28	0,36	0,28
$R=R_0(\delta)$	0,3	0,4	0,3	0,4	0,3	0,4
$t_{уд}$, c	0,046	0,0050	0,0035	0,0038	0,0028	0,0030
$\|a_{max}\|$, relative to g	6	6	8	8	10	10
δ	0,36	0,28	0,36	0,28	0,36	0,28
$R=R_0(\delta)$	0,3	0,4	0,3	0,4	0,3	0,4
t_{imp} (s)	0,0023	0,0025	0,0017	0,0019	0,0015	0,0014

The results of the analysis demonstrate the existence of several stability ranges that are determined according to the following equations:

$$\xi_1 = \sqrt{\frac{(m_1 + m_2)c_1}{m_1 m_2 \omega^2}},$$

$$\xi_2 = \sqrt{\frac{(m_1 + m_2)}{m_1 m_2}\frac{(c_1 + c_2)}{\omega^2}}, \tag{18}$$

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

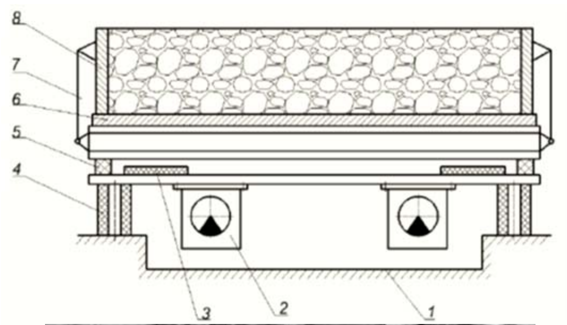
The parameters ξ_1 and ξ_2 in equations (18) take into account the weight of the vibration machine (see Figure 3) and the presence of two elastic elements with the elasticity coefficients c_1 and c. The parameter ξ_1 is the ratio of the system natural frequency, when the vibrating elements do not contact with spring element, to the squared forced vibration frequency ω_2. The parameter ξ_2 is the ratio of the system natural frequency, when the vibrating element is in contact with spring element, to the squared forced vibration frequency ω_2. The parameter f is the ratio of the reduced weight of the vibration machine to the excitation force. If the weight m of the vibration machine is specified, the parameters that should be determined are the elasticity coefficient c and the mass static moment of unbalance m0r0, since the natural vibration frequency ω depends, generally, on the vibration machine design. Additionally to the parameters ξ and f, the stability range depends on the period τ of contact between the vibrating elements. The first stability range in the resonance vibration mode is provided if the values of the parameters ξ and f are determined as follows [14, 15]:

$$0{,}8 \le \xi_2 \le 1{,}3$$

$$1{,}3 \le f \le 2$$

The value of the parameter ε may be 4 ≥ε≥ 2. These values represent the second stability range. With consideration for the first stability range, a new shock-and-vibration machine (see Figure 3) for forming hollow concrete foundation blocks was proposed.

a)



b)

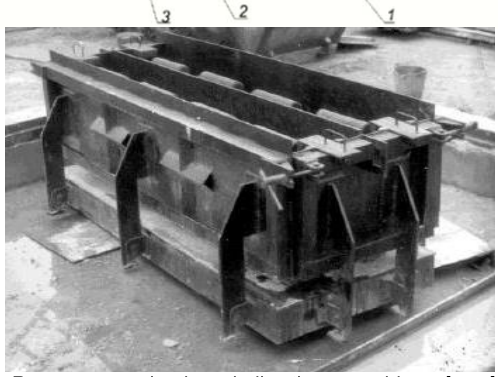


Fig. 3. Resonance shock-and-vibration machine for forming concrete foundation blocks: a) – Schematic diagram; b) – Construction design; 1 – Lower frame; 2 – Vibration exciter; 3 – Vibration limiter; 4 – Vibration-isolating support; 5 – Spring support; 6 – Pan; 7 – Equipment for closing and opening the side plates; 8 – Side plate

Conclusions

1. Developed is the mathematical model of a resonance vibration machine system. This model takes into account the elastic, inertial, and dissipative properties of

the actuator of the machine and material being processed.
2. Analyzed and determined are the parameters that affect the resonance operating mode of the vibration machine. It is determined that, for vibration frequency 100 … 160 rad/s, the resonance vibration mode is possible if the ratio of the period of impact action to the vibration period is 0.25 … 0.31. If the parameters $\xi \geq 1.7$ and $f = 2.5$, it is possible to set up the second stability range with minimum energy consumption in supporting vibrations.
3. Proposed is a new resonance shock-and-vibration machine designed for forming hollow concrete blocks.

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