

Oleksiy Lanets, Volodymyr Borovets, Olena Lanets, Yaroslav Shpak, Vasyl Lozynskyy
Lviv Polytechnic National University, Lviv, Ukraine

SYNTHESIS OF STRUCTURE AND RESEARCH OF OPERATION OF RESONANCE TWO-MASS VIBRATING TABLE WITH ELECTROMAGNETIC DRIVE

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Abstract. An industrial necessity of creation of 100-Hz vibratory tables is grounded. The prospect of researches of just resonance electromagnetic vibratory tables is conditioned. The existing vibrating tables on the basis of electromagnetic drive are considered. It is noted that the stages and problems which arise during the creating of high-frequency two-mass vibration table of middle sizes constructed according to classic chart are outlined in the article. The principle scheme of an electromagnetic vibratory table which is to be developed is described. The basic analytical dependences for determining the coefficient of inflexibility of the resonance resilient system and hauling effort of the electromagnetic vibroexciters are obtained. The construction of the upper plate of vibrating table is grounded, and its first eigenfrequency is determined in order to avoid the coincidence with its forced frequency. The construction of vibrating table is given: its spatial model is presented, and the general draft and the draft of the two-mass vibratory system are also given. The inertia parameters of the oscillating masses are determined. The parameters of stiffness and structural parameters of the resonance resilient system are determined. Checking of the resilient elements for durability is conducted. The parameters of stiffness and structural parameters of vibration isolators are determined. The results of calculation of vibroexciter, according to the hauling effort with the use of the specially developed program, are presented. The theoretical gain-frequency description of vibratory table and time dependences of motion of the oscillating masses are presented. The experimental gain-frequency description of vibratory table and time dependences of motion of the oscillating masses, which absolutely coincide with the theoretical, are presented.

1. Introduction

Vibrating machines must satisfy the followings technical requirements: high reliability, longevity, efficiency, accordance to the sanitary norms, the immediacy of achievement of the set work mode, and et cetera. These circumstances induce scientists and engineers to search for new ways in creation of high-efficiency equipment.

2. Statement of the problem

Nowadays, there is a need of high-frequency vibrating tables (frequency of the forced vibrations 100 Hz) which are designed for making of sidewalk tile, facing tile, flows, etc. Needed sizes of the tables surface are from 1200×600 mm, the mass of load is no more than 120 kg, the maximal value of amplitude of working organ vibrations, especially in vertical direction, is 0.2 mm (equivalent to the overweight of 8 g at the forced vibrations frequency of 100 Hz).

Development and making of vibrating equipment contains a series of technically difficult and scientifically capacious decisions. First of all, it is due to the fact that working frequency of the vibratory table must be of 100 Hz. In such case, taking in account the presence of mobile connections in the drive, it will be under the action of high dynamic loadings, consequently it will be subjected to the intense wear. Such circumstances somewhat limit the use of mobile connections vibroexciters such as inertial, eccentric ones (it's working frequency is mainly under 50 Hz), especially those of piston and plunger. The increase in frequency leads to the a sharp decrease in the reliability of the drive. However unbalanced vibrating machines which work at the frequency of 100 Hz do exist, but the resource of their drive is low.

Taking into account the aforesaid, and especially the vertical direction of vibrations, in this case, it is the most rational to use resonance electromagnetic vibrating tables, on which it is structurally simple to achieve the rectilinear vibrations with the frequency of 100 Hz. The work of vibrating tables on the basis of electromagnetic drive is based on the unique phenomenon in oscillating processes – resonance, which provides the minimum consumption of electric power, and their creation and research is a prospective engineering task caused by the necessities of production.

3. Review of the literary sources

Nowadays, some attempts to make the vibratory tables on the basis of the electromagnetic drive have been taken [1-4]. As to principle of their work, they are two-mass or multimass ones, with harmonic motion or vibroshock. Vibrating tables with the electromagnetic vibroexciters occur mainly as small sized and laboratory equipment.

On the department of mechanics and automation of mechanical engineering, it was developed and made a series of experimental prototypes of high-frequency (100 Hz) vibratory tables for the compression of concrete mixtures (Fig.1- Fig. 4) whose construction is based on the theory of synphasic vibrations [2].

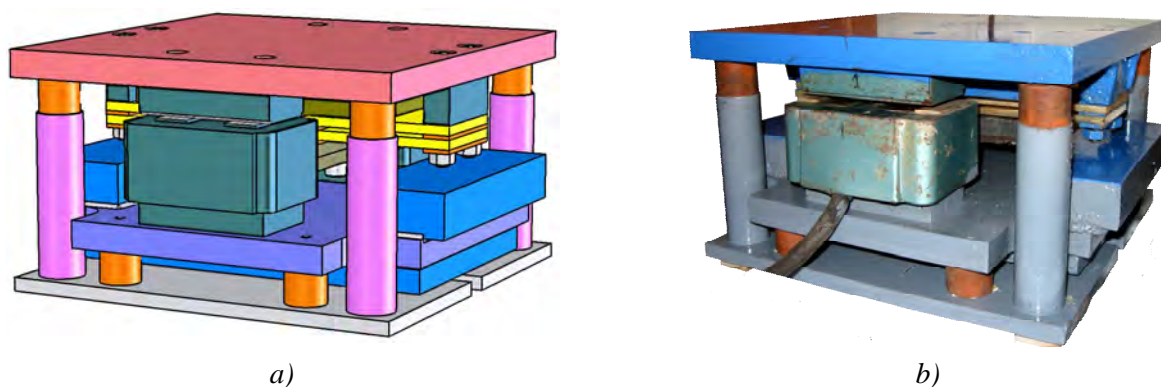


Fig. 1. Spatial model (a) and created on its basis without rectifier 100-Hz electromagnetic vibratory table (b) with the sizes of working body surface $300 \times 300 \text{ mm}$

(**Technical description of experimental vibrating table:** the working range of amplitudes of vibrations 0...0,3 mm; maximal overload which operates on a detail on the tables surface -12g; the power, kW – 0,35; power supply voltage 220 V from a network 50 Hz, overall sizes of vibrating table, mm: width – 308, length – 330, height – 174; mass is 67 kg)

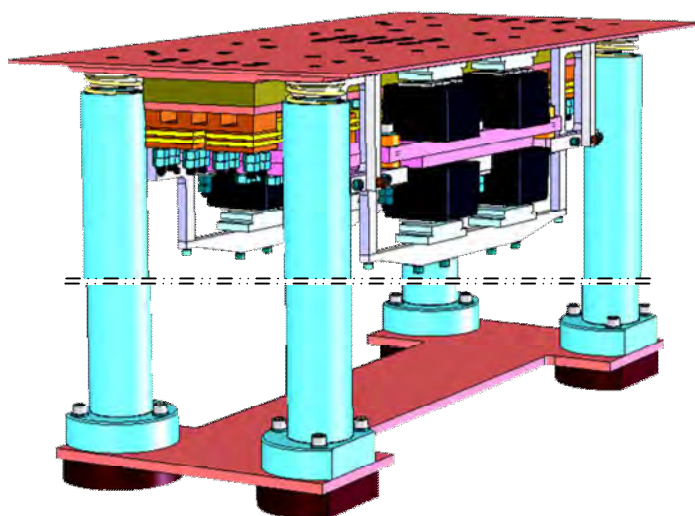
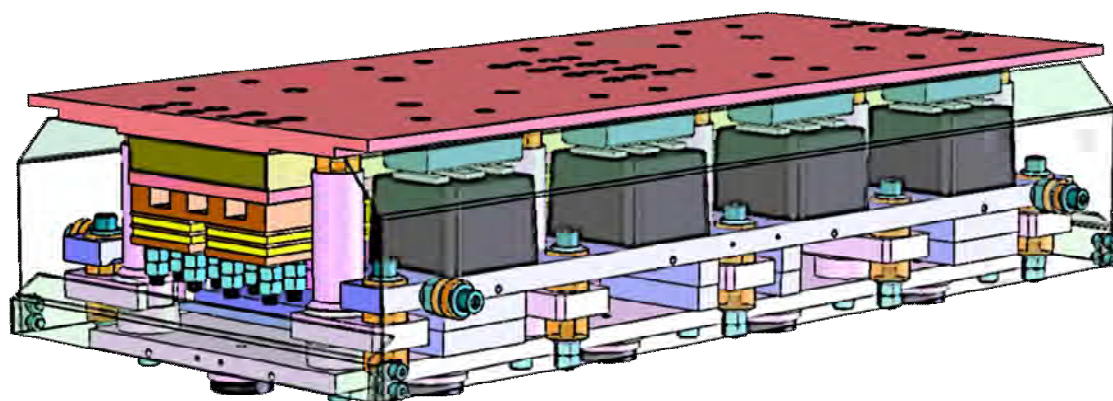


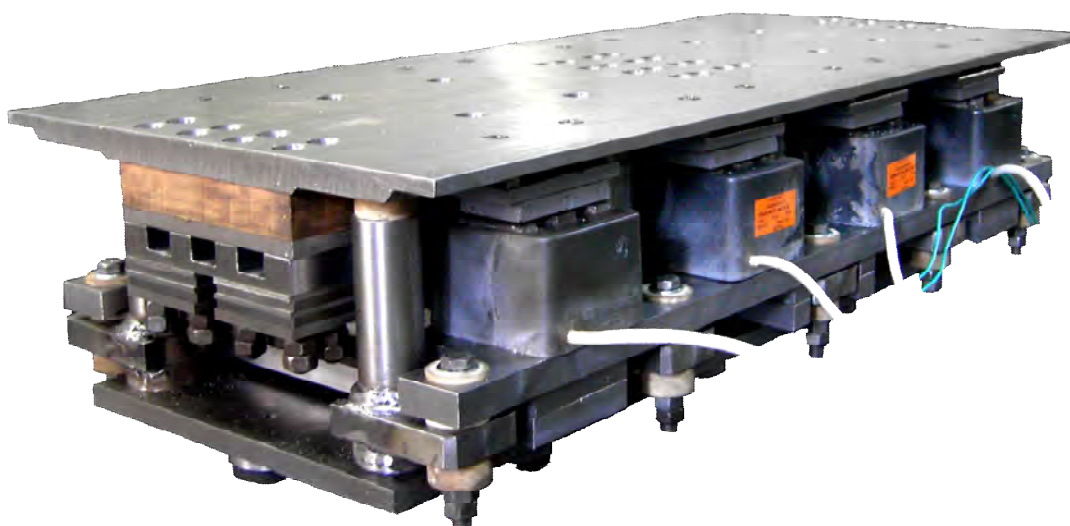
Fig. 2. Spatial model of 100-Hz electromagnetic vibratory table with rectifier. The sizes of working body surface $1000 \times 500 \text{ mm}$



a)



b)

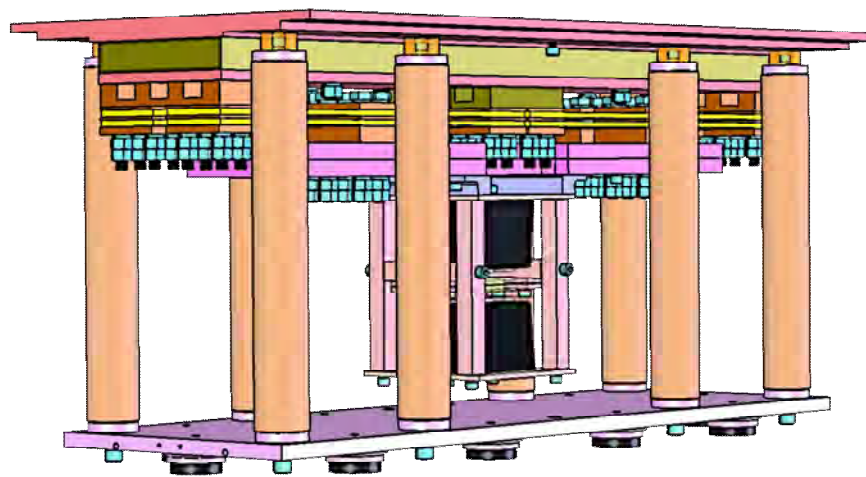


c)

Fig. 3. Spatial model (a) and created on it's basis without rectifier 100-Hz electromagnetic vibrating table.

The sizes of working body surface 900×400 mm with (b) and without (b) hoods

(**Technical description of vibrating table:** the working range of amplitudes of vibrations 0–0,3 mm; maximal overload which operates on a detail on the tables surface – 12 g; power, kW – 1,4; power supply voltage 220 V from the network 50 Hz; overall sizes of vibrating table, mm: width -400, length – 924, height – 222; mass is 246 kg)



a)



b)

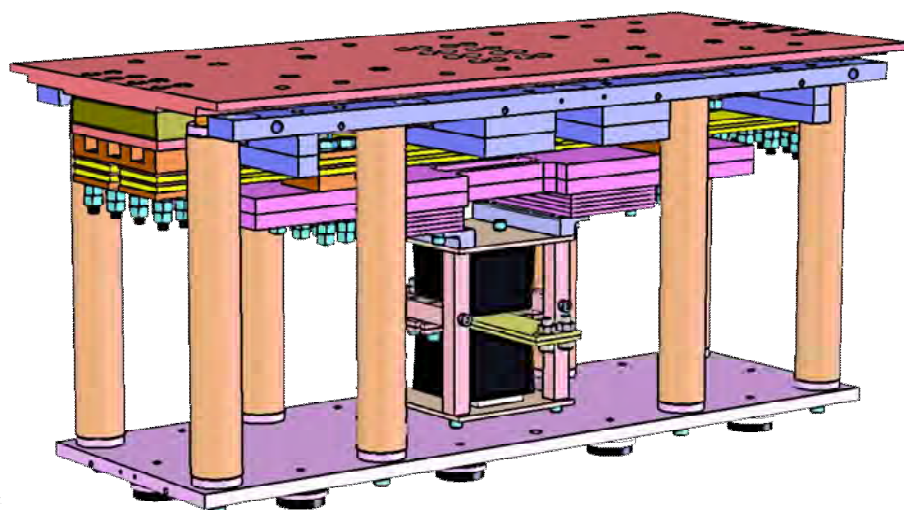
Fig. 4. Spatial model (a) and created on its basis high-efficiency without rectifier 100-Hz electromagnetic vibratory table (b). The sizes of working tools surface $900 \times 400 \text{ mm}$

(Technical description of vibrating table based on data of the experiment: the amplitude of vibrations 0,1 mm; overload which operates on a detail on the tables surface – 4g; power – 10 W; a power from the sound generator of 3Г -33 is 100-Hz harmonious signal; overall sizes of vibrating table, mm: width – 440, length – 900, height – 430; mass is 170 kg)

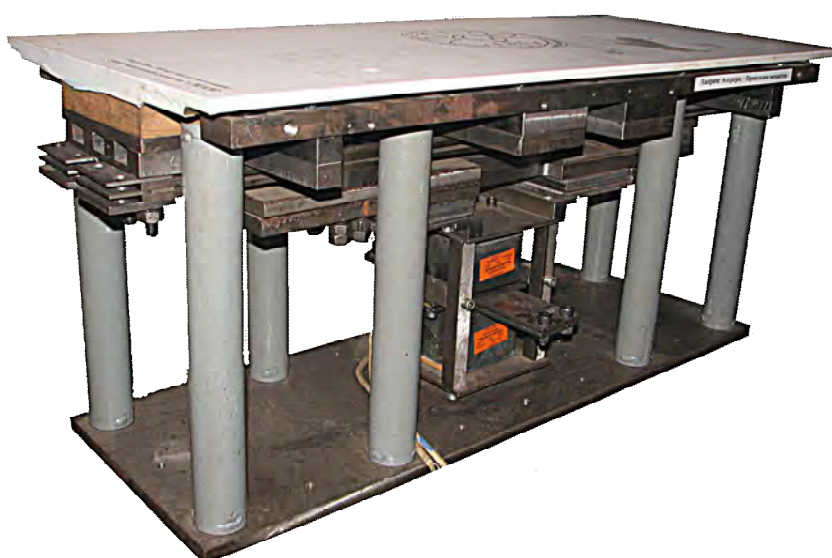
It was used two-way vibroexciter for high-efficiency vibrating table (Fig 4). The circumstance that the frequency of the forced vibrations of the system is 100 Hz, while the construction required 100-Hz of sine-wave signal, some complicated it. It was decided to transfer the oscillating system of vibrating table on frequency of the forced vibrations 50 Hz (Fig. 5). In such case the machine control system is not needed, because a power supply comes from the network 220 V through a diode bridge. Vibrating table showed a good result in the work.

4. Raising of the task

In this article the authors will expose the stages and will describe the problems which arise up during the creation of high-frequency two-mass vibrating table of medium sizes, designed according to classic scheme. From the structural point of view it will be resonance two-mass with harmonious vibrations. From the authors point of view that is interesting and perspective to test this scheme for structures with medium sizes.



a)



b)

Fig. 5. Spatial model (a) and 50-Hz vibratory table with an electromagnetic drive with rectifier created on its basis (b)

(Technical description of vibratory table: the working range of amplitudes of vibrations 0...0.2 mm; maximal overload which operates on a detail on the surface of the table – 8 g; power, kW is 0.6; power supply voltage 220 V from a network 50 Hz; overall sizes of vibratory table, mm: width – 400, length – 900, height – 430; mass is 214 kg)

5. Description of two-mass model of vibratory table

Two-mass vibratory table with an electromagnetic drive (Fig. 6), in which rectilinear vibrations are realized along an x -axis according to independent coordinates x_1 and x_2 , contains reactive mass m_2 and active mass m_1 , formed from the working body m_{po} and conventionally added part $m_{np} = k_{np} m_{3a6}$ of the the mass of load environment (here k_{np} is a coefficient which represents the particle of load environment m_{3a6} , that conventionally added to the working body). An anchor is rigidly fastened to the active mass, and the core with the coil of electromagnet fastened to reactive mass. Oscillating masses are connected between each other by the resilient system with stiffness c_{12} . The vibratory table is set on immobile basis on the vibration isolators with stiffness c_{i3} , which fastened to mass m_1 .

Vibrations forced in the system under the action of power disturbance with the amplitude of the force vector F . Variable voltage 220V from the power source 50 Hz ($\omega = 314 \text{ rad/s}$) is given directly on the terminals of coil, and that is why an electromagnet, which works only on attracting, will generate power impulses with the frequency of forced vibrations 100 Hz ($\Omega = 2\omega = 628 \text{ rad/s}$). Movement of vibrating masses will be carried out exceptionally along the axis x , because the vector of power disturbance passes through the center of the masses of system. As dampers the coefficients of viscid friction which are proportional to the speed are introduced in dynamic model. Coefficient μ_1 represents external viscid resistance of motion of the mass m_1 , caused by the influence of the mass of load m_{3a6} . Coefficient μ_{12} describes dispersion of energy in the resonance resilient system with inflexibility c_{12} ; μ_{i3} is viscid friction in the damping resilient elements with the inflexibility c_{i3} .

A priori laying, that the law of motion is harmonious, the system of differential equations for a linear two-mass structure has the known kind of the linear system:

$$\begin{cases} m_1 \ddot{x}_1 + c_{12}(x_1 - x_2) + \mu_1 \dot{x}_1 + \mu_{12}(\dot{x}_1 - \dot{x}_2) = -F \sin \Omega t; \\ m_2 \ddot{x}_2 + c_{12}(x_2 - x_1) + c_{i3}x_2 + \mu_{i3} \dot{x}_2 + \mu_{12}(\dot{x}_2 - \dot{x}_1) = F \sin \Omega t. \end{cases} \quad (1)$$

It will be exposed below why the law of change of indignation force is given in a sinewave kind.

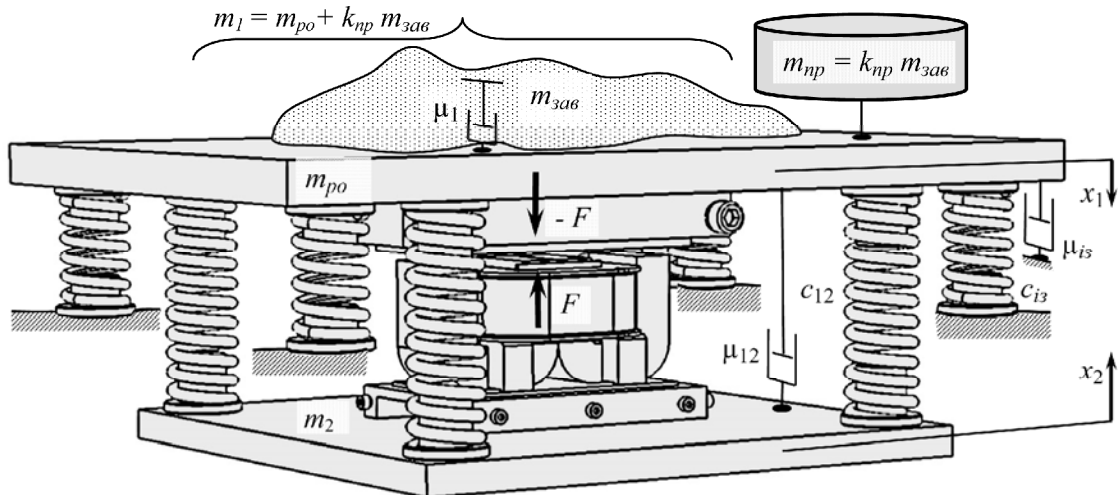


Fig. 6. Principle scheme of the electromagnetic vibratory table

We search the analytical expressions of motion of the masses according to two independent degrees of freedom using the general methods of solution of system of differential equations in the next form:

$$x_1 = X_1 e^{i\Omega t} \text{ and } x_2 = X_2 e^{i\Omega t},$$

where X_1, X_2 is amplitudes of linear forced vibrations according to independent co-ordinates x_1 та x_2 . It is possible to get dependences for establishment of magnitudes X_1 та X_2 putting this expressions in (1) and shortening in every part of the system of equations member $e^{i\Omega t}$. In matrix presentation according to amplitudes of vibrations of the masses it will look like:

$$\begin{bmatrix} \left(c_{12} + c_{i3} - m_1 \Omega^2 + \right) & -c_{12} - i\mu_{12} \Omega \\ \left(+i(\mu_{i3} + \mu_1 + \mu_{12}) \Omega \right) & \left(c_{12} - m_2 \Omega^2 + \right) \\ -c_{12} - i\mu_{12} \Omega & \left(+i\mu_{12} \Omega \right) \end{bmatrix}^{-1} \times \begin{bmatrix} -F \\ F \end{bmatrix} = \begin{bmatrix} X_1 \\ X_2 \end{bmatrix}. \quad (2)$$

If designate:

$$k_{11}(\Omega) = c_{12} + c_{i3} - m_1 \Omega^2 + i(\mu_{i3} + \mu_1 + \mu_{12}) \Omega; \quad k_{12}(\Omega) = k_{21}(\Omega) = -c_{12} - i\mu_{12} \Omega;$$

$$k_{22}(\Omega) = c_{12} - m_2 \Omega^2 + i\mu_{12} \Omega,$$

a solving of the system (1) as expressions for amplitudes of oscillations X_1 та X_2 of the masses m_1 та m_2 respective in stable modes will presented by followings dependences:

$$X_1(\Omega) = \frac{F \cdot (k_{12}(\Omega) + k_{22}(\Omega))}{\Delta}; \quad X_2(\omega) = \frac{-F \cdot (k_{11}(\Omega) + k_{21}(\Omega))}{\Delta}, \quad (3)$$

where

$$\Delta = k_{12}(\Omega)k_{21}(\Omega) - k_{11}(\Omega)k_{22}(\Omega), \quad (4)$$

is determinant of the matrix of coefficients of the unknowns in the equation system (2).

6. Synthesis of inertia-stiffness and force parameters of vibratory table

Forming characteristic equalization of the mechanical oscillating system, equating (4) to a zero, ignoring the coefficients of viscid resistance and inflexibility of vibration isolators, it is determined the value of inflexibility c_{12} through the unique natural frequency $\Omega_0 = \Omega/z$ of oscillations of the system. The expression turns into the known kind:

$$c_{12} = m_{36} (\Omega/z)^2, \quad (5)$$

where $m_{36} = m_1 m_2 / (m_1 + m_2)$ – erected mass; z – resonance adjusting of the system; Ω – angular frequency of the forced vibrations. We should note, that during the establishment of inertia and stiffness parameters of the system, it is ignored the coefficients of the viscid friction in order to avoid complex values. It is also ignored inflexibility of vibration isolators, as it does not have substantial influence on the redistribution of the parameters in the oscillating system. For that, it is necessary to satisfy the next condition: the frequency of natural oscillations of vibratory table, as conventionally solid on soft elastic elements must be at 5K8 times below than forced. However, an account of dissipative forces and inflexibility of vibration isolators is required when modeling the work of vibratory table.

Despite the fact, that the unaccounted dissipative forces and inflexibilities of vibration isolators when determining inertia-stiffness parameters lays an error in the model of the system, the results obtained with sufficient for engineering's calculations accuracy, and final analytical dependences are formed mathematically simply and definitely.

Power of electromagnetic drive N_{36} , necessary for indignation of n -mass oscillating system determined as follows [5]:

$$N_{36} = \frac{\sqrt{6}}{4} \frac{\Omega^3}{\eta} \sum_{i=1}^n \left(\frac{X_i^2 m_i}{\lambda_i} + X_1^2 m_{np} \right), \quad (6)$$

where X_i , λ_i – the amplitude of vibrations and the dynamic factor of mass m_i ; m_{np} – particle of boot environment which conventionally joins to the working body (in this case to mass m_1); X_1 – amplitude of the oscillations of mass m_1 ; η – coefficient of useful effect of drive (electromagnetic vibro-exciter). In our case values of dynamic factors of the first and second masses have the next form:

$$\lambda_1 = X_1(\Omega) / (m_1 \Omega^2 / F) = \left| m_1 \Omega^2 (k_{12}(\Omega) + k_{22}(\Omega)) / \Delta \right|; \quad \lambda_2 = \left| m_2 \Omega^2 (k_{11}(\Omega) + k_{21}(\Omega)) / \Delta \right|, \quad (7)$$

The feature of this construction is that the electromagnetic vibro-exciter are connected according to scheme without rectifier, that is, when exciting force affects on oscillating mass from one side and works only on attracting. It generates the presence of constant component from the exciting force, which from the first approaching is determined from the expression, which describes the law of change of one-arctic voltage of coils of electromagnetic vibro-exciter. Decomposition of this function in to a number of Fourier, on the condition that an electromagnet feeds from a network $\omega = 314 \text{ rad/s}$ ($\nu = 50 \text{ Hz}$), looks like:

$$p(t) = \left(\frac{2}{\pi} - \frac{4}{\pi} \left[\frac{\cos 2\omega t}{3} + \dots \right] \right) P_{ei}, \quad (8)$$

where P_{el} – total amplitude value of force impulse of electromagnetic vibro-exciter at indignation without rectifier. Actually it is necessary to perform the calculations of vibro-exciter (Fig. 7) after this amplitude value. A selection of a constant member from (8) will give the particle of constant component of attraction. In such case the constant component of effort (Fig. 7) will be determined as:

$$P_{nocm} = \frac{2}{\pi} P_{el}, \quad (9)$$

a variable component (Fig. 7) with working frequency of the forced vibrations $\Omega = 2\omega$ will be equal:

$$P_{zapM}(t) = \frac{4}{3\pi} P_{el} \cos 2\omega t. \quad (10)$$

From (10) the amplitude value of the exciting force of harmonious component on frequency $\Omega = 2\omega$, actually which drives the oscillating system to motion, will be:

$$P_{zapM} = F = \frac{4}{3\pi} P_{el}. \quad (11)$$

From the other side it is possible to set a needed value of F through the parameters of the system (amplitude of vibrations of working body X_1 , inertia parameters of the masses, stiffness of resilient elements), using the first expression from (3):

$$F = \frac{X_1(\Omega)\Delta}{(k_{12}(\Omega) + k_{22}(\Omega))}. \quad (12)$$

Equating expressions (11) and (12), the total amplitude value of impulse of indignation of electromagnetic vibro-exciter, which is expressed through the necessary amplitude X_1 of vibrations of the working body on frequency of the forced vibrations $\Omega = 2\omega$, is:

$$P_{el} = \frac{3\pi X_1(\Omega)\Delta}{4(k_{12}(\Omega) + k_{22}(\Omega))}. \quad (13)$$

If the force F , the expression (12) will be given through the dynamic factor of the active mass, using the expression (7):

$$F = \frac{X_1(\Omega)m_1\Omega^2}{\lambda_1(\Omega)}, \quad (14)$$

than the expression (13) can be written as:

$$P_{el} = \frac{3\pi X_1(\Omega)m_1\Omega^2}{4\lambda_1(\Omega)}. \quad (15)$$

Exactly after this parameter of exciting force, the value of which is determined in accordance with (15), an electromagnetic vibro-exciter will be calculated.

At first glance it is possible to attribute the presence of constant component P_{nocm} to the the significant disadvantage of vibro-exciter without rectifier, because in such case the hauling effort is lost, the static bending in resilient elements occurs, the possibility of the sticking of an anchor to core with a coil in electromagnetic vibro-exciter appears. However, it is some leveled for the high-frequency systems, as the oscillating masses have relatively low amplitudes of vibrations, and the air gap can be maximally minimized, that significantly increases the coefficient of useful effect of the drive. In addition, analyzing (11), it is clear that practically $4/2\pi \approx 2/3$ of the traction is perceived by the system. And it means that the traction which is need for vibro-exciter without rectifier compared to vibro-exciter with rectifier is only 1.57 times more, but not 2 times as at first glance. In other words, to set the oscillating system in motion, where 8 vibro-exciter without rectifier (8 electromagnets) are used, it is necessary to use 3 vibro-exciter with rectifier (vibro exciter with rectifier is 2 vibro-exciter without rectifier), that means that 6 electromagnets without rectifier, with power 85 % of vibro-exciter, or it is necessary 2 vibro-exciter with rectifier, that means 4 electromagnets with power 128 % of vibro-exciter without rectifier. The use of

vibro-exciters with rectifier structurally complicates and increases the cost of system and requires additionally control system to generate 100-Hz sine-wave signal.

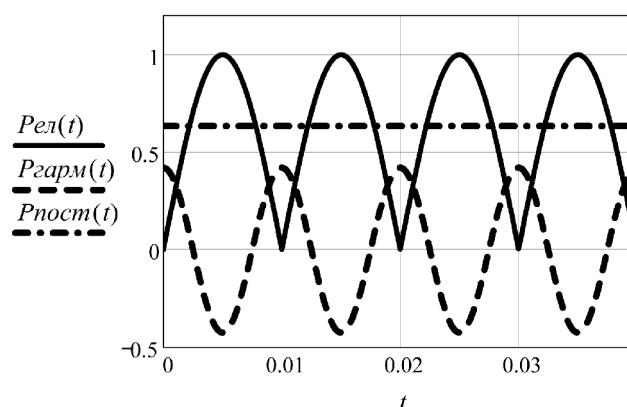


Fig. 7. Visualization of decomposition of function $p(t)$ after the first two members on condition that $P_{el} = 1 \text{ N}$

7. Synthesis of the working body of the vibrating table

The most important task during the usage of vibrating tables for the compression of concrete mixture, is the achievement of stable value of the amplitude of vibrations on the whole area of the working body. It is especially important when the working body works on a high-frequencies with the small values of the amplitude of vibrations. It is explained that under the action of exciting force, except of the useful vertical vibrations of the working body as solid, the additional vertical vibrations (parasitic) vibrations appear. Imposition of useful and parasite vibrations on the plane of working body will lead to unevenness of the value of amplitude in certain areas of working body which can entail appearance of defective product (insufficient and uneven closeness of wares, water is not enough forced out).

Even on the stage of planning there is a necessity to calculate the value of the first eigen-frequency_of transversal vibrations of working body. For providing of the stable and even compression of billets in the developed construction, it is necessary that the eigen-frequency should be at least in $3\mathbf{K}4$ times higher than the forced. The simplest method of these calculations is the use of the method of finite elements. Computer design of the mechanical oscillating system with distributed masses by the method of finite elements can be performed in the software product of Cosmos, which works in the shell of software product Solid Works.

So, for vibratory table on the picture 1 with sizes $300 \times 300 \times 20 \text{ mm}$ and the mass $m = 14 \text{ kg}$, the eigen-frequency of vibrations of the working body (of the square plate) will be $\nu = 724 \text{ Hz}$ (Fig. 8, a). In an order to attain the same value of eigen-frequency of vibrations for vibratory tables on the picture 2-picture 4 with the sizes of working body $900 \times 400 \text{ mm}$, using a monolithic steel plate, we need a plate 116 mm thick (Fig. 8, б), and its mass will be already $m = 327.8 \text{ kg}$. It means, if increasing the area of the working body in 4 times, so the weight increasing in 23 times.

So, increasing of the sizes of construction and adhering to those technical parameters of machine, as in it's experimental standard, can grow such concept as "complication" in several times. For an exit from this situation it is necessary to carry out the followings steps: facilitating the construction, and most likely losing in value of eigen-frequency of vibrations, going out on those structural parameters which will be acceptable from the technological and economic point of view.

The working body of vibrating table structurally can be made by many methods, main, that: the natural vibrations of the working body, which arise due to the coincidence of frequency of the forced and own vibrations, should be minimized, as for the body with the distributed mass. So it is necessary, that on the frequency of the forced vibrations 100 Hz all it's working surface should carry out the oscillating motion as absolutely solid, for that the working body must be stiff enough; at the same time a working body must be light, structurally simple and cheap in manufacture.

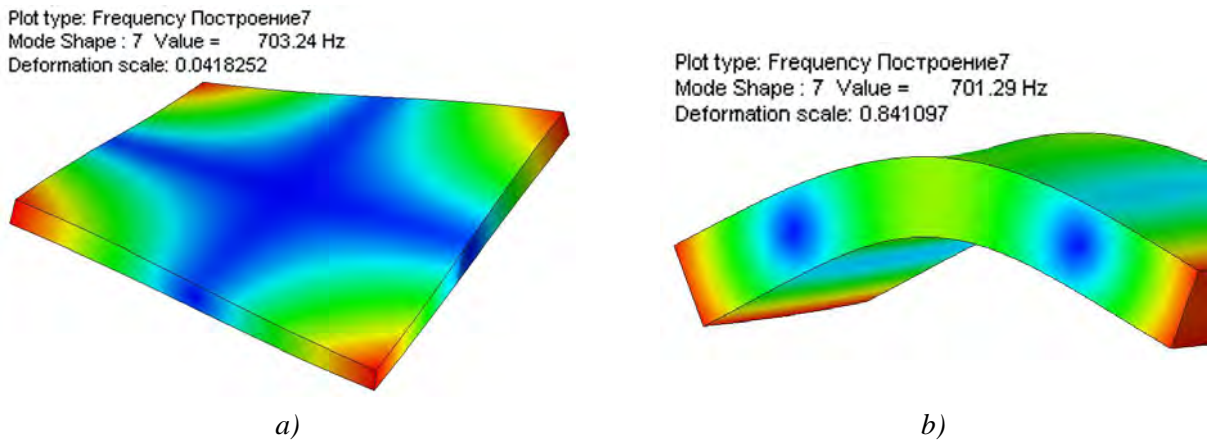


Fig. 8. The result of calculation of eigen-frequencies of a working body with sizes:
 a) - 300×300×20 mm; б) - 900×400×116 mm

The next example of the working body is taken in a quality of structural decision, that it is represented on the Fig. 9. As we can see, the upper steel plate is 25 mm thick, on edges it narrows, that does it lighter and at the same time stiff. Such thickness is conditioned by the necessity to hide the heads of screws M12 under an internal hexahedron in a plate. Narrowed lower plate provides the considerable reduction of the material, substantially not influencing on the stiffness of all construction. The lower steel plate, 8 mm thick, posted with upper due to the inserted between them channels №10. From below to the lower plate under the resilient elements the shoes are fastened.

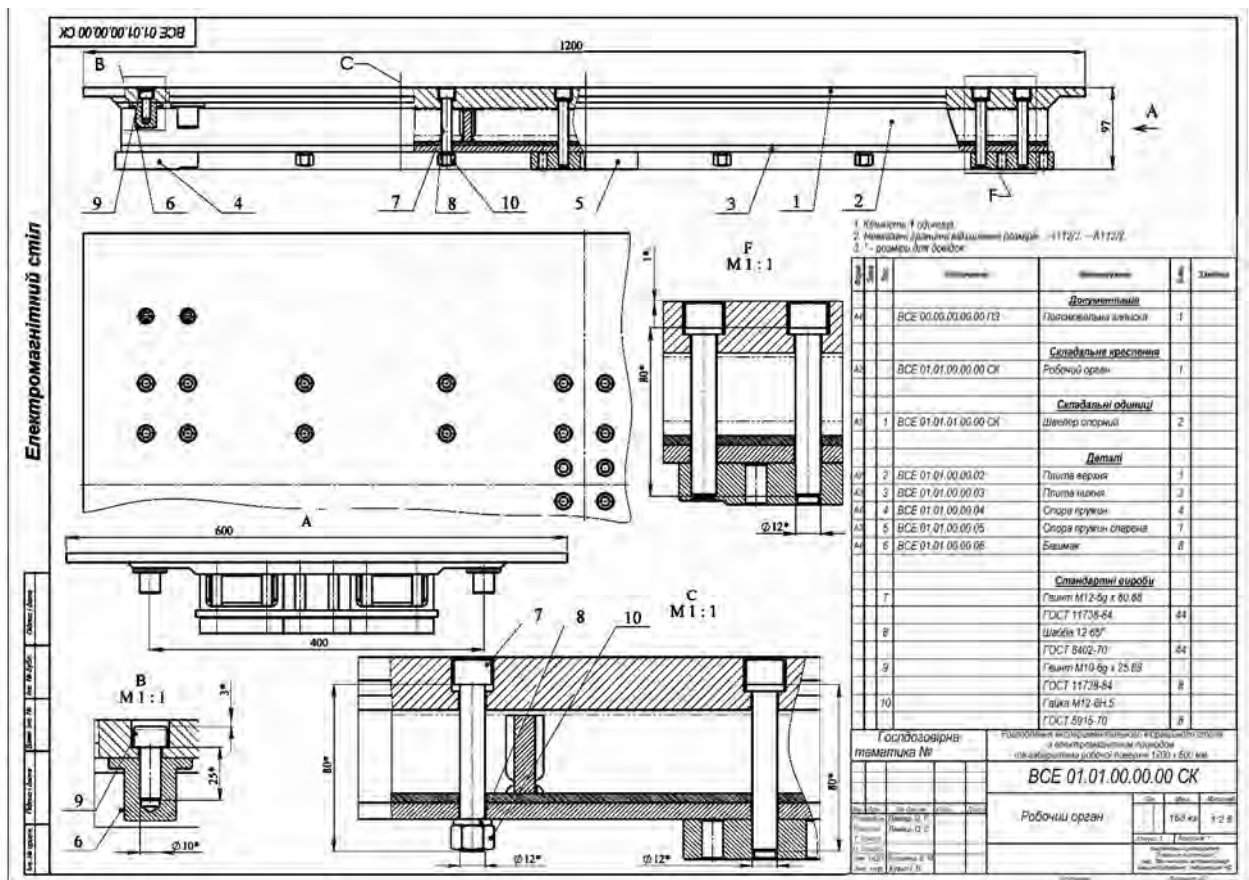


Fig. 9. Assembly drawing of the working body

Such structural decision allows to attain high stiffness of construction at its relatively small weight. It is possible to find the analogy of this structural decision in the form of rolled steel. So, for the example the girders, also have two steel shelves, posted after a height and connected between itself with the thin membrane. Such construction of the working body is relatively light ($m = 163 \text{ kg}$) and at the same time stiff (Fig. 10). As evidently, it's eigen-frequency of vibrations is $\nu = 300 \text{ Hz}$, that is an acceptable decision, since the eigen-frequency of vibrations of the working body, as of the body with the distributed mass, in 3 times higher than the forced frequency and a high probability of the absence of the parasite amplitudes of vibrations on the working body practically.

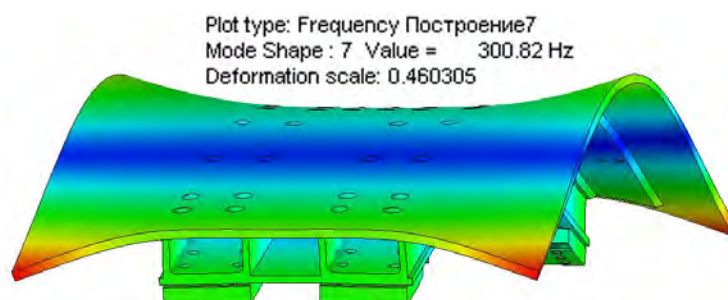


Fig. 10. The result of calculation of the working body with the sizes $1200 \times 600 \text{ mm}$ as of the body with the distributed mass, where it's first eigen-frequency of vibrations is defined

8. The development of solid-state model of vibratory table

Using above results of design, the solid-state model of vibratory table (picture 11) was created and a design documentation was developed in a software product Solid Works 2006 (picture 12 and picture 13).

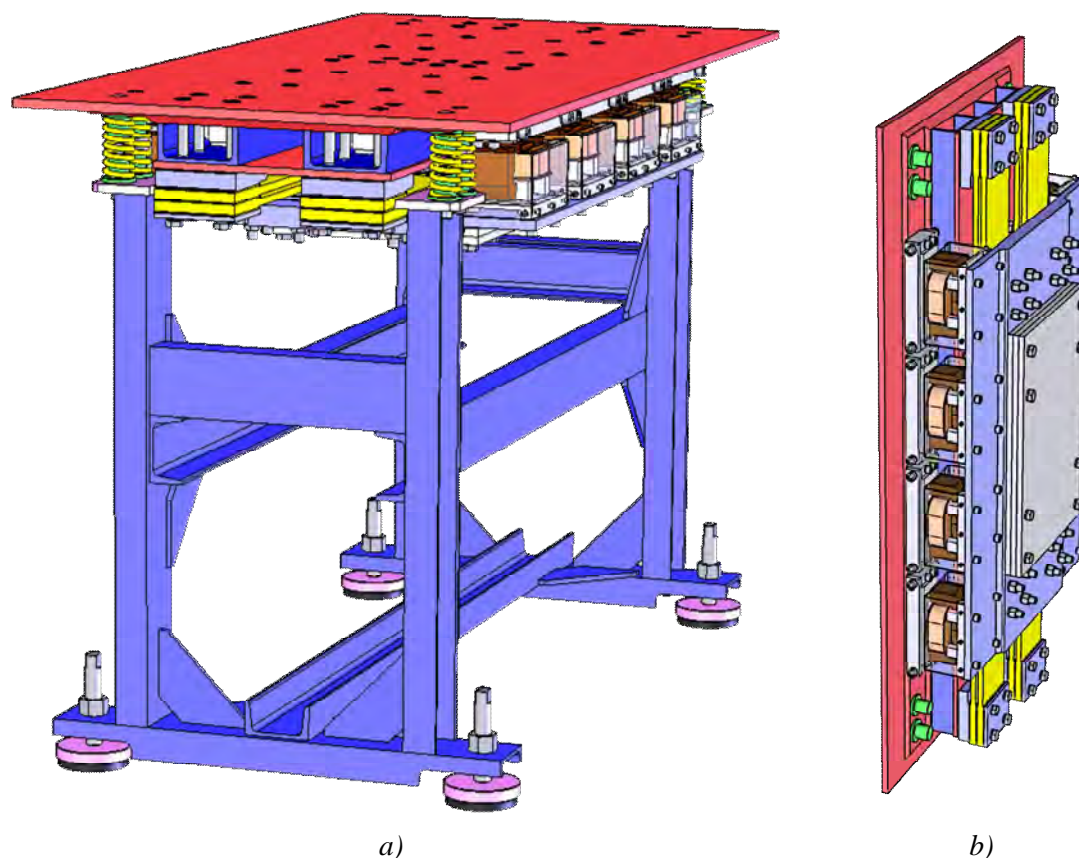


Fig. 11. Spatial models of: a) – vibratory table; b) – two mass oscillating system

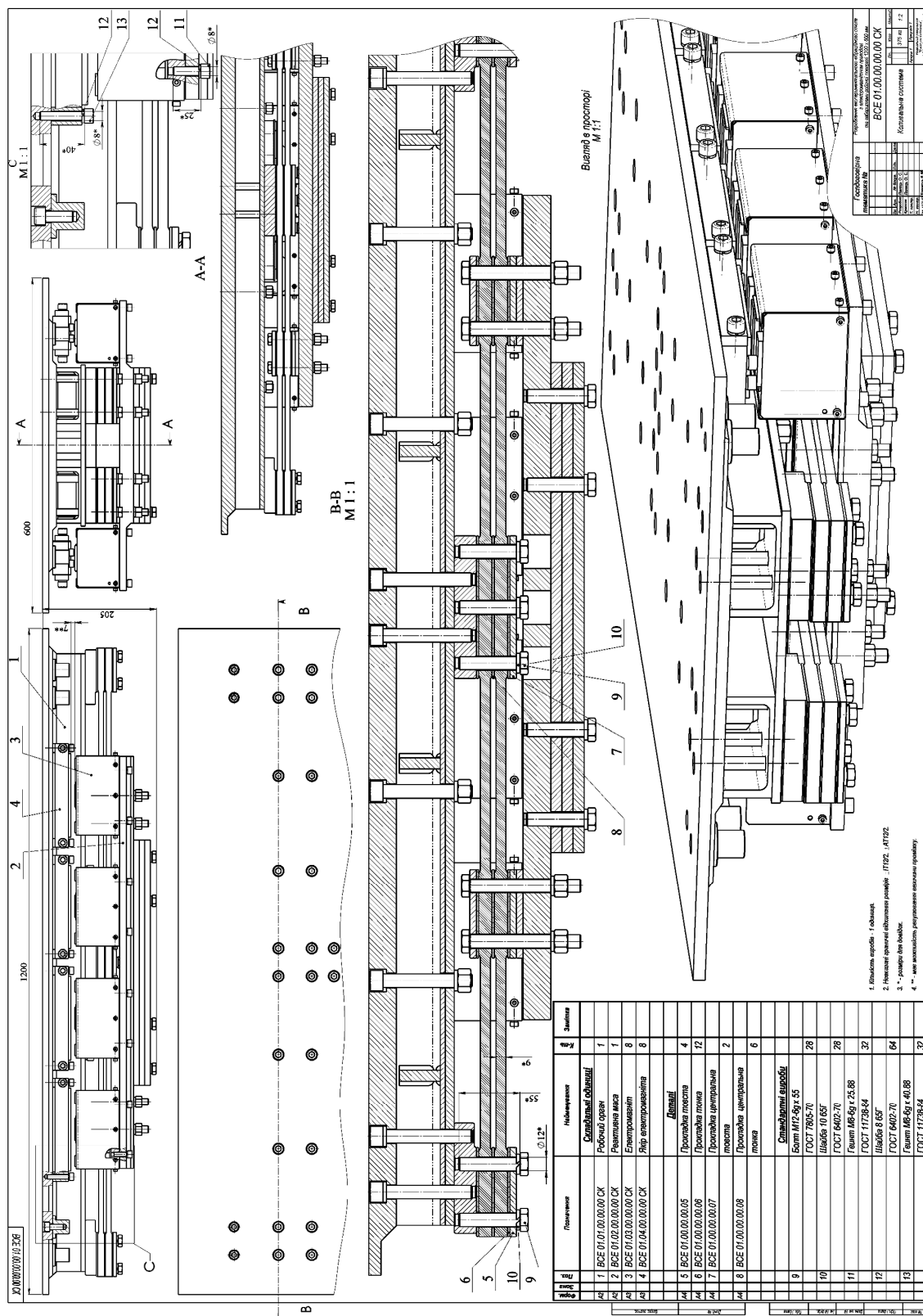


Fig. 13. Assembly drawing of the two mass oscillating system

9. The determination of the parameters of the mechanical oscillating system

Let's set the values of inertia parameters of the oscillating masses of the vibratory table. So, in our case, total mass of the two mass oscillating system is $M = 368.5 \text{ kg}$ (Fig. 14, a). Reactive mass without adjusting plates is $m_2 = 132.5 \text{ kg}$ (Fig. 14, b), and with the plates is $m_2 = 161.8 \text{ kg}$ (Fig. 14, c). Thus, the weight of the intermediate oscillating mass may vary near $\Delta m_2 = 29.3 \text{ kg}$. Such possibility of the change of the inertial parameter of reactive mass is related to the difficulty of predictions of the value of the ratio of crushing of elastic elements. Exactly with the help of this plates it is possible to bring the oscillating system to resonance, or to take away from it. Active mass is $m_1 = M - m_2 = 206.7 \text{ kg}$.

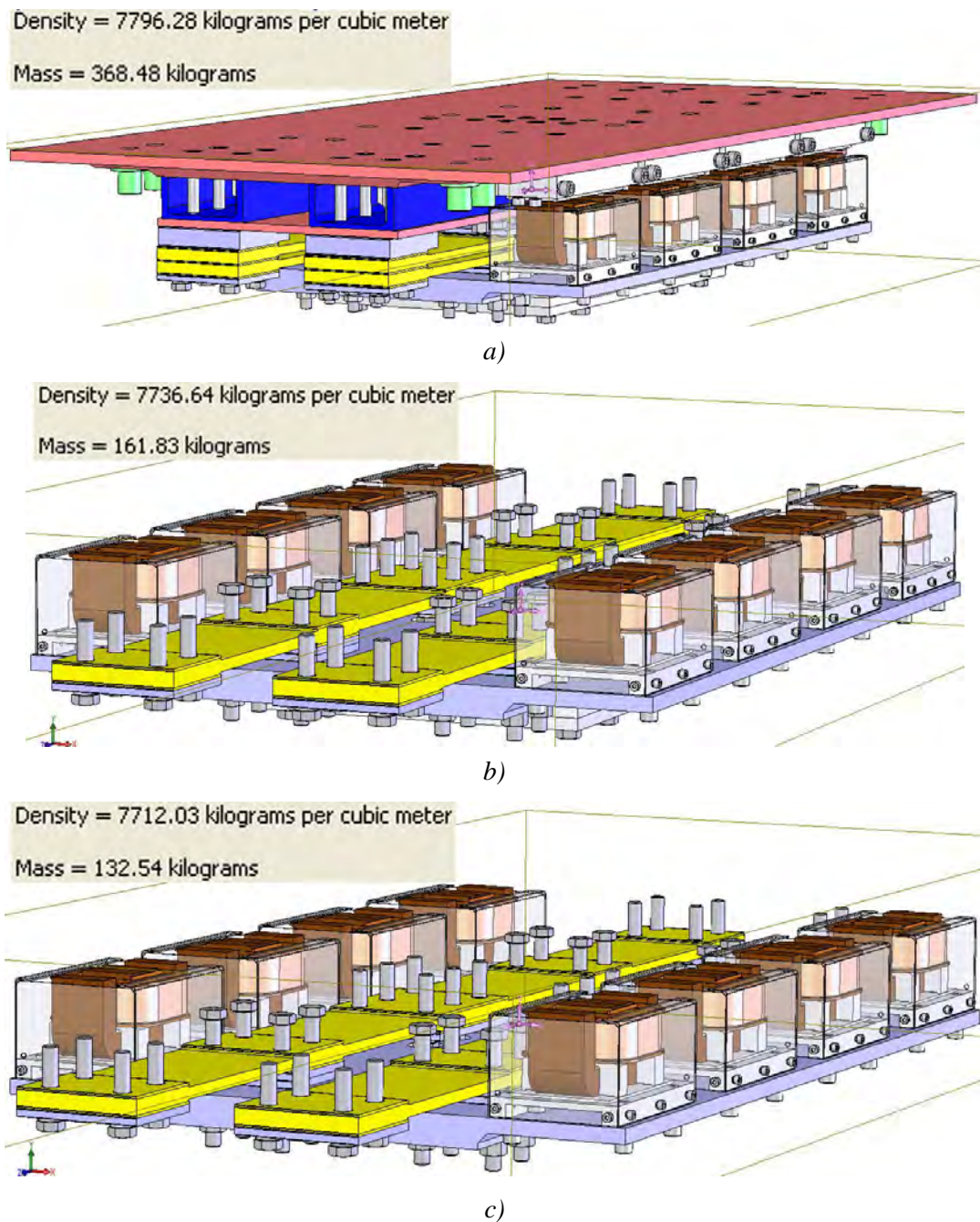


Fig. 14. Fragments of windows of software product of Solid Works 2006, where the mass is calculated: a) – two masses oscillating system of the vibratory table; b) and c) – reactive oscillating mass accordingly with adjusting plates and without them

9. The calculation of the resilient system of oscillation ground

Accepting a case, when $m_2 = 161.8 \text{ kg}$, and resonance adjusting of the system is $z = 0.97$, necessary stiffness of the resilient system will be (5):

$$c_{12} = \left(\frac{206.7 \cdot 161.8}{206.7 + 161.8} \right) \cdot \left(\frac{2 \cdot \pi \cdot 100}{0.97} \right)^2 = 3.808 \cdot 10^7 \text{ N/m},$$

In this case $m_{3g} = 90.75 \text{ kg}$. For a choice of the parameter of thickness b of a flat spring, at the known length l_c and width h of resilient element taking into account the amount i of resilient elements, it is necessary to use the next ultimate expression [2]:

$$b = l_c \sqrt[3]{\frac{c_1}{Ehik}}, \tag{16}$$

where E – the modulus of elasticity of the first kind; k – the coefficient of jamming, which takes into account imperfect tightening of screws and bolts, which have the pliability, in addition the jamming does not eliminate micro backlash for the resilient elements. All this reduces inflexibility of the resilient systems. Therefore in practice it is necessary to increase inflexibility of the resilient system relatively to the calculated value. Taking the ratio of crushing $k = 0.75$ and using the parameters from the Fig. 15, the thickness of resilient element in accordance to (16) will be:

$$b = 0.169 \cdot \sqrt[3]{\frac{3.808 \cdot 10^7}{2.1 \cdot 10^{11} \cdot 0.1 \cdot 16 \cdot 0.75}} = 0.009 \text{ m}.$$

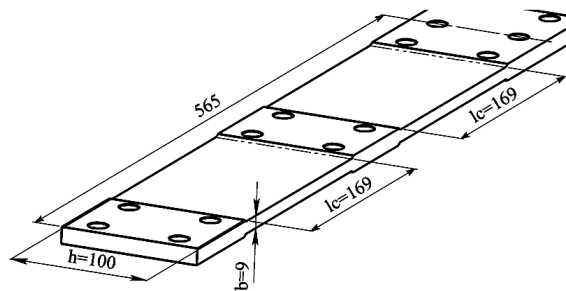


Fig. 15. Sizes of resilient element

10.1. Verification of stiffness of resilient element.

The design will carry out with the finite element method. 3D model of resilient element is broken on finite elements. Some areas are hardly jamming, the other load by the testing effort $R = 1000 \text{ N}$. The deflection of the element will be $\varpi = 1.937 \cdot 10^{-4} \text{ m}$ (Fig. 16), and therefore the stiffness of the resilient system is:

$$c_{12} = \frac{R}{\varpi} = \frac{1000 \cdot 8}{1.954 \cdot 10^{-4}} = 4.09 \cdot 10^7 \text{ N/m},$$

that is more than a calculating value. This is acceptable, because in a 3D model is not included the coefficient of jamming of resilient element.

10.2. Checking of resilient element for strength.

In the case when reactive mass matters $m_2 = 161.8 \text{ kg}$, and the active mass (working body) – $m_1 = 206.7 \text{ kg}$, correlation of amplitudes is 1.278 times. The relative amplitude of vibrations is determined as:

$$A_{bid} = A \cdot (1 + 1.278),$$

where A is amplitude of vibrations of the working body. Accept $A = X_1 = 0.2 \text{ mm}$ – equivalent to 8-th overloads, in such case it will be:

$$\xi = \frac{X_1 \Omega^2}{g} = \frac{0.0002 \cdot 628^2}{9.81} = 8.$$

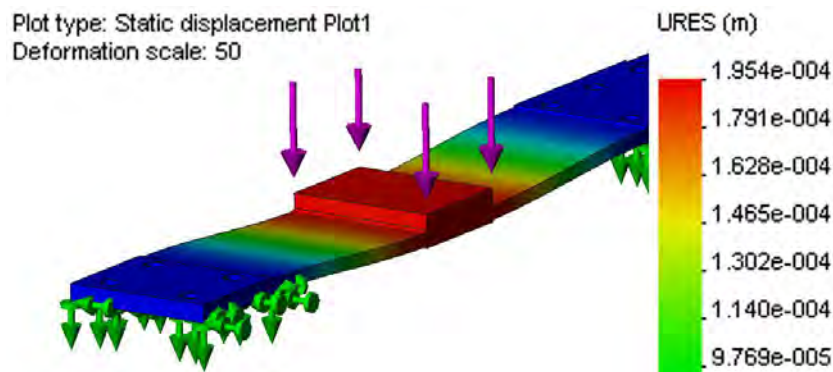


Fig. 16. The result of calculation by the finite elements method in the software product Solid Works of stiffness of the resilient system

On practice this parameter is enough to provide the effective compression of the concrete mixture. The calculation by the finite elements method, when in the solid model of the elastic rod rigidly pinched some areas, and the other areas get displacement on the value $A_{\text{gid}} = \varpi = 0.46 \text{ mm}$ without turning Fig. 17), indicates a significant margin of strength ($\sigma_{\text{max}} = 83 \text{ MPa}$). For steels 60C2, 60C2A, 60C2XΦA maximal allowable stress is $[\sigma_{-1}] = 170 \text{ MPa}$. According to known analytical dependence the maximal stress in resilient elements is:

$$\sigma_{\text{max}} = \frac{3EbA_{\text{gid}}}{l_c^2} = \frac{3 \cdot 2.1 \cdot 10^{11} \cdot 0.009 \cdot 0.00046}{0.169^2} = 91 \text{ MPa},$$

that agreed to the numerical calculation. Therefore this elastic system can withstand the load.

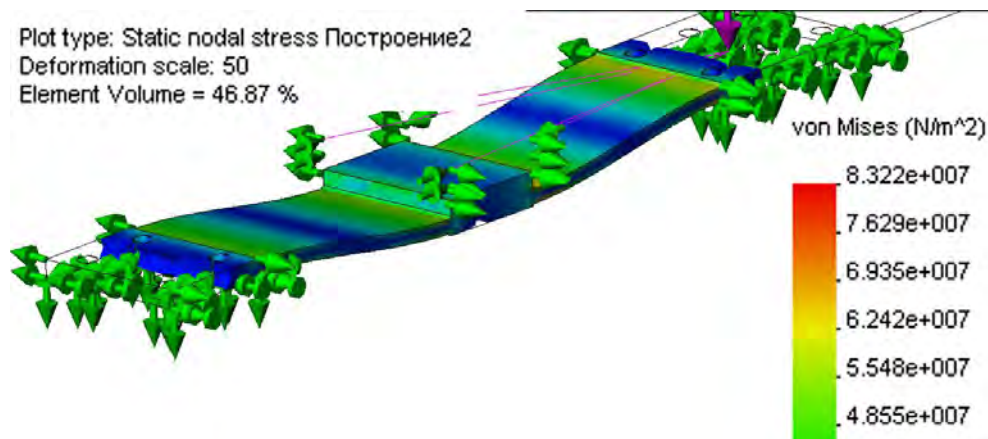


Fig. 17. The result of calculation by the finite elements method in the software product Solid Works of the resilient system on strength (a) (bending shown on the picture with a scale factor of 50)

10.3. Verification of eigen-frequency of oscillation of the system.

The conducting of frequency analysis will carry out with the use of the finite elements method in the software product Solid Works. For simplification of calculations the one-mass mechanical oscillating system will be examined (Fig. 18, a), it consists of the area of resilient element and given mass, which is especially on it ($m_{3g} = 90.75/8 = 11.34 \text{ kg}$). The virtual material with the density $\rho = 1 \text{ kg/m}^3$ is taken as a material of the resilient element, in order that it's mass did not influence on a calculation, as it is already taken into account in given mass. The material of load of given mass is also virtual with density

$\rho = 78500 \text{ kg/m}^3$ for reducing of it's sizes, which will allow to break up the system on finite elements more precisely. Taking into account that jamming is not ideal, actual eigen-frequencies will be below than calculating Fig. 18). In addition, we will be able using adjusting plates to approach the necessary eigen-frequency of vibrations $101\text{K}102 \text{ Hz}$.

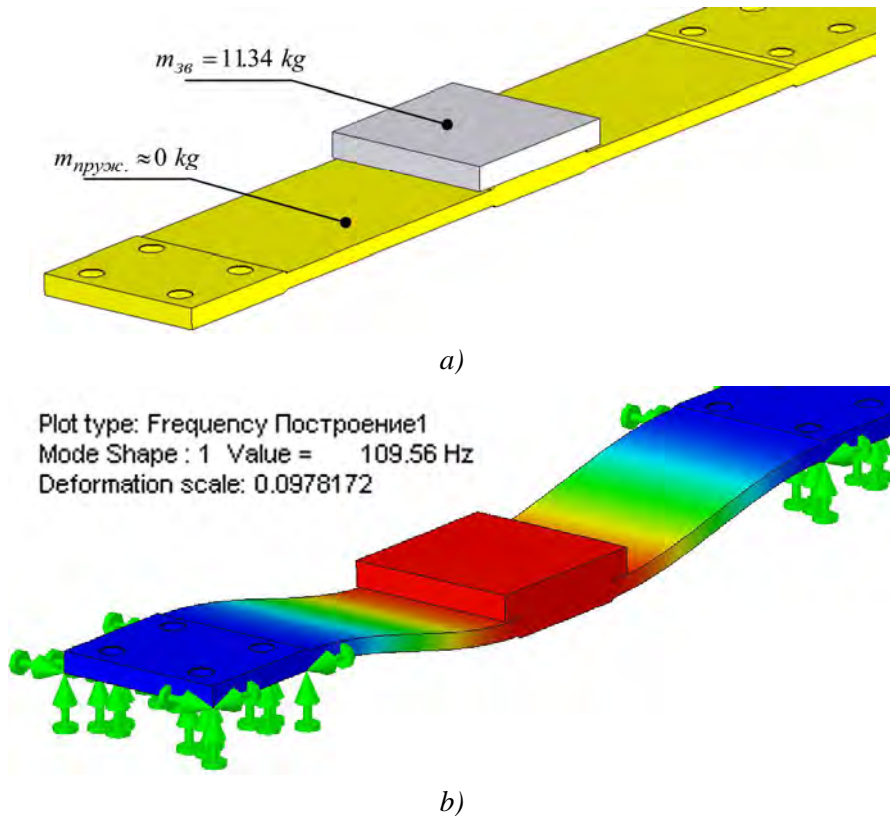


Fig. 18. The probed conditionally selected one-mass structure as an area of resilient element with the rigidly fastened given mass, which is on one area (a) and the result of calculation of eigen-frequency of vibrations of the conditionally selected one-mass structure (b)

11. The calculation of the parameters of vibration isolators

The inflexibility of vibration isolators set from a condition, that oscillating system of the vibratory table with the mass $M = 368.5 \text{ kg}$ and a load $m_{3ав.} = 120 \text{ kg}$ should have an eigen-frequency of vibrations on the vibration isolators 5...10 times below than the forced, so:

$$c_{i3} = (M + m_{3ав.}) \left[\frac{\Omega}{(5\text{K}20)} \right]^2 = (M + m_{3ав.}) (2\pi v_{i3})^2, \quad (17)$$

where v_{i3} – eigen-frequency of the oscillating system as a solid on vibration isolators. It is foreseen in the construction of the vibratory table (Fig. 12) $n = 8$ twisted springs, which execute vibro-isolating functions. There will be loading on one spring:

$$Q = \left(\frac{M + m_{3ав.}}{n} \right) g = \left(\frac{368.5 + 120}{8} \right) \cdot 9.81 = 600 \text{ N} .$$

Eigen-frequency of the oscillating system as a solid on vibration isolators (the one-mass oscillating system is examined), using (17), determined as:

$$v_{i3} = \frac{1}{2\pi} \sqrt{\frac{c_{i3}}{M + m_{3ав.}}} . \quad (18)$$

Let's put, that the eigen-frequency of the oscillating system is $\nu_{i3} = 5 \text{ Hz}$ (20 times less than the forced). In such case inflexibility of one vibration isolators (twisted spring) is in axial direction, concordantly (18) will be:

$$c_{i31} = \frac{(M + m_{3ae.}) (2\pi\nu_{i3})^2}{n} = \frac{(368.5 + 120)(2\pi 5)^2}{8} = 60266 \text{ N/m}.$$

Inflexibility of the twisted spring through its structural parameters is determined in obedience to the next dependence [6]:

$$c_{i31} = \frac{G \cdot d_0^4}{8 \cdot D^3 \cdot i_g}, \quad (19)$$

where G – is the modulus of elasticity of the second kind; d_0 – is a diameter of wire, D – is a middle diameter of winding of coils; i_g – is an amount of coils of wire. If to conduct the selection of spring by the adjustment of its length, for providing of the defined value of inflexibility c_{i31} , in the case when $d_0 = 6 \text{ mm}$, the needed amount of coils is set according to (19):

$$i_g = \frac{G \cdot d_0^4}{8 \cdot D^3 \cdot c_{i31}} = \frac{7.7 \cdot 10^{10} \cdot 0.006^4}{8 \cdot 0.032^3 \cdot 60266} = 6.3.$$

Let's accept, that $i_g = 6$ coils. Making the model of the twisted spring under load of 600 N , its sagging is set (Fig. 19).

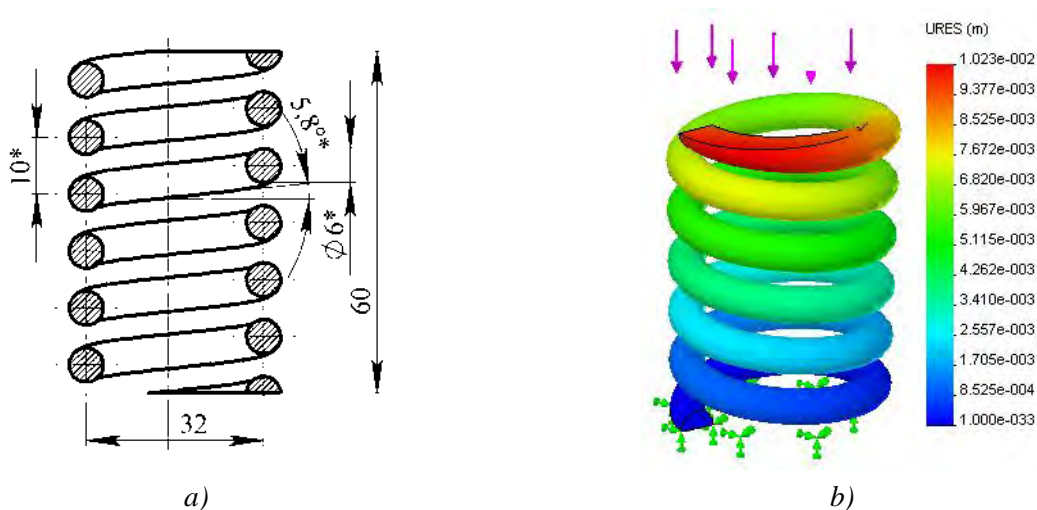


Fig. 19. Sizes of sagging of the twisted spring which executes vibration-isolating functions (a) and establishment of its size under the action of the calculating load

In obedience to the results of calculation on picture 19 sagging of the twisted spring is $\delta_{cm} = 0.00996 \text{ m}$, so the value of its coefficient of inflexibility in axial direction is:

$$c_{i31} = \frac{Q}{\delta_{cm}} = \frac{600}{0.001} = 60000 \text{ N/m}.$$

The eigen-frequency of the oscillating system of the vibratory table on eight vibration isolators according to (18) will be:

$$\nu = \frac{1}{2\pi} \sqrt{\frac{c_{i3}}{M + m_{3ae.}}} = \frac{1}{2\pi} \sqrt{\frac{60000 \cdot 8}{368.5 + 120}} = 4.99 \text{ Hz},$$

that is 20 times less than the forced and, surely, meets the requirements of vibro-isolation. So, with such indications, passing of the vibration on the foundation goes down at $20^2 = 400$ times.

The stress which arises up in a resilient element [6] will be:

$$\tau = \frac{8 \cdot k_{\delta} \cdot D \cdot Q}{\pi \cdot d_{\delta}^3} = \frac{8 \cdot 1.28 \cdot 0.032 \cdot 600}{\pi \cdot 0.006^3} = 290 \text{ MPa},$$

that is possibly for the static loading.

12. The calculation of the electromagnetic vibro-exciter

In the environment of QBasic the program for this purpose is compiled, which takes into account the bulge of the electromagnetic fields, this allowing to conduct a calculation more exactly. We'll carry out it from the followings conditions: total amplitude value of indignation of electromagnetic vibro-exciter under the terms, an overload is 11 g (the amplitude of vibrations of active mass 0.27 mm (relative 0.62 mm) on frequency 100 Hz) according to dependence (11): $P_{en} = 500 \text{ N}$; air interval $\delta = 0.0018 \text{ m}$. When calculating the electromagnetic vibro-exciter an overload 11 g is taken to ensure a supply of drive power, although it is foreseen that the vibratory table will work with an overload 8 g (nominal amplitude of vibrations of active mass is 0.2 mm). The initial data which was entered, and the results of calculation are resulted below:

A SIZE OF THE AMPLITUDE OF THE MAGNETIC INDUCTION VECTOR BM IN THE MAGNETIC CORE ? 0.97

A SIZE OF THE MAGNETIC INDUCTION VECTOR BI (0.6 ... 1) IN GAP ? 0.7

ACTUAL VALUE OF TENSION IS ON CLAMP. VIROZB. U= ? 220

FREQUENCY OF VIBRATIONS of N= ? 50

ENTER THE VALUE OF EFFORT F, WHICH DEVELOPS (DEVELOP) VIBROEXCITER (VIBROEXCITERS)

? 500

ENTER THE VALUE OF AMPLITUDE OF VIBRATIONS

? 0.00062

DO YOU WANT TO PUT YOUR OWN VALUE OF THE GAP SO ?

ENTER (Y, N)? y

SO= ? 0.0018

GAP SO= .0018

THE RESULTS OF THE PREVIOUS CALCULATION

DENOTATION OF THE MAGNETIC CORE Sh132*40

THE NUMBER OF TURNS OF THE COIL W= 763.811

THE DIAMETER OF THE WIRE WINDING D= 7.337951E-04

AN ELECTROMAGNET HAS ONE COIL

AN ELECTROMAGNET HAS NOT COUPLED MAGNETIC CORES

THE REFINED PART OF THE CALCULATION OF VIBROEXCITER HOW COIL WILL BE WINDED

1. MANUAL WINDING (ENTER 1)

2. ON THE MACHINE WITH PAPER 0.035 MM THROUGH 1 LAYER (ENTER 2)

3. MANUAL WINDING PAPER THROUGH 2 LAYERS (ENTER 3)

4. MANUAL WINDING PAPER THROUGH 1 LAYER (ENTER 4)

5. THE VALUE OF COEF. OF FILLING KU

? 4

303.7474

413.0006

506.7602

THE RESULTS OF THE REVISED CALCULATION

DENOTATION OF THE MAGNETIC CORE: Sh140*45

FACE A= of .04

FACE B= of .045

EFFORT FR= 506.7602

THE NUMBER OF TURNS $W = 609.8925$

CONDUCTOR DIAMETER $D = 7.884736E-04$

AN ELECTROMAGNET HAS ONE COIL

AN ELECTROMAGNET HAS NOT COUPLED MAGNETIC CORES

Thus, the chosen structural parameters of the W-shaped electromagnet are following: the diameter of the wire winding is 0.8 mm , the number of turns of the coil $\varpi = 610$, sizes of the core on Fig. 20).

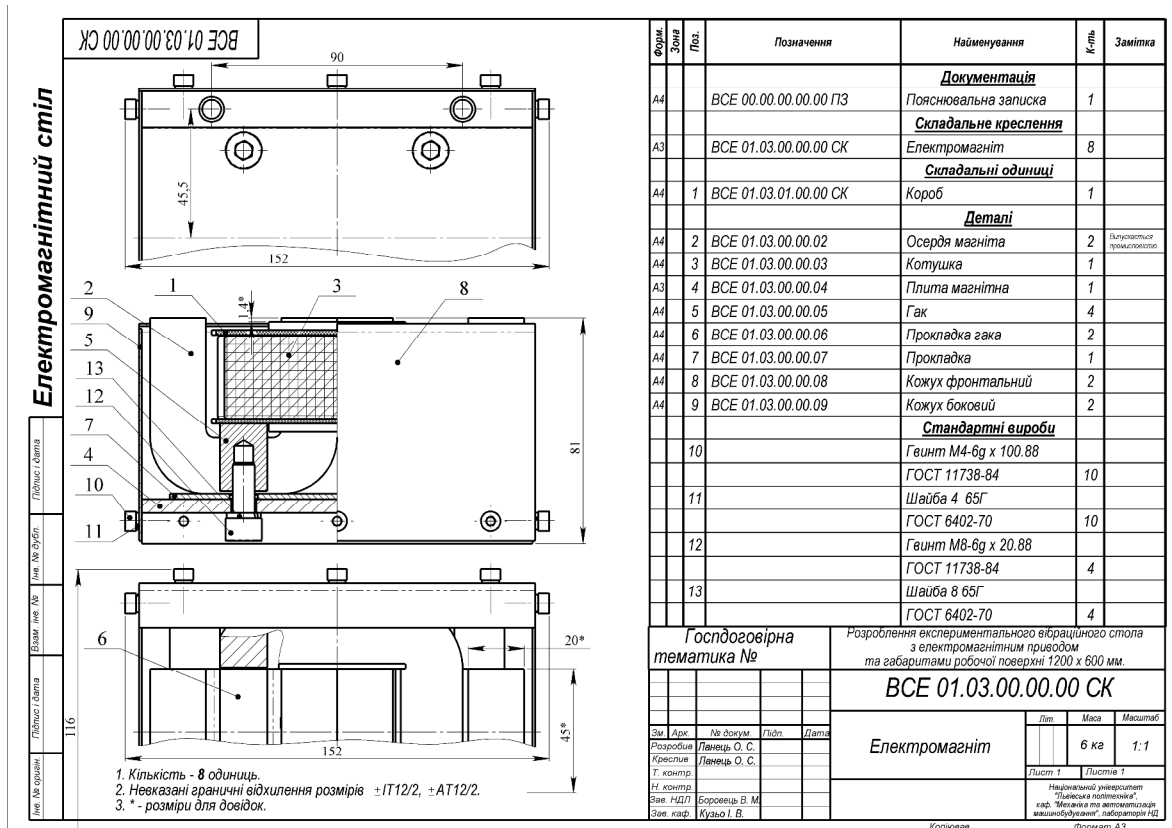


Fig. 20. Drawing of the core with a coil of an electromagnetic vibration exciter

The area of the cross-section of wire was accounted from the condition, that the density of current is $\rho = 3 \text{ A/mm}^2$. At the most unfavorable conditions (not taking into account $\cos(\varphi)$ between a current and voltage), one vibro-exciter can skip:

$$N_1 = UI = U \rho \frac{\pi d^2}{4} = \frac{220 \cdot 3 \cdot \pi \cdot 0.8^2}{4} = 330 \text{ W}.$$

So, the drive of the whole vibratory table is able to develop the power $N_{np} = 8 \cdot 330 = 2640 \text{ W}$. The question of overheating of the windings of the coil indirectly solved by limitation of the allowable value of current through a wire cross-section ($\rho = 3 \text{ A/mm}^2$). It is justified, as indeed during the work of the vibratory table, coils didn't overheat, and therefore a heat transfer is satisfactory.

13. The construction of frequency response

It is foreseen that when overload $8 g$ the necessary traction force of one electromagnet will be 370 N . Amplitude value of the total harmonic component of the exciting force of eight vibro-exciter according to (7) when overload $8 g$ is:

$$P_{\text{гарм}} = F = \frac{4}{3\pi} P_{\text{ел}} = \frac{4}{3\pi} (8 \cdot 370) = 1256 \text{ N}.$$

Putting the above parameters of the mechanical oscillating system in its mathematical model (1), it is possible to establish, that the offered construction will be capable (Fig. 21). The coefficient of viscosity μ_{12} accounted from a condition, that coefficient of the internal frequency-independent friction $\chi_{12} = 0.03 m^3 s$, and that is why $\mu_{12} = \chi_{12} c_{12} / \Omega = 1830 N \cdot s / m$. Other values of parameters of coefficients of viscosity are taken so: $\mu_1 = 2400 H \cdot s / m$, $\mu_{i3} = 230 N \cdot s / m$.

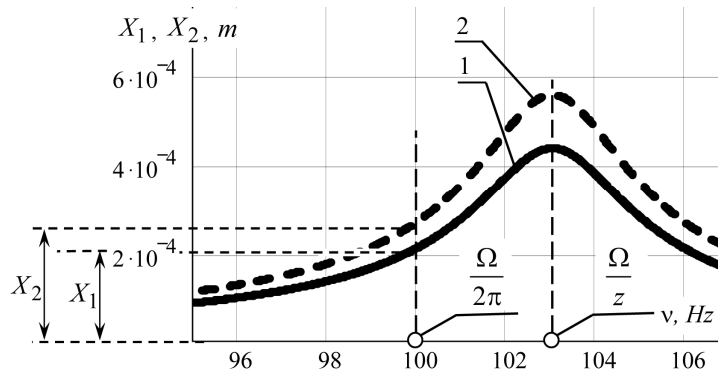


Fig. 21. Amplitude-frequency characteristic of the vibratory table, where: 1, 2 are amplitude values of oscillations accordingly of active and reactive masses

Using (7), it is set that on the frequency of the forced oscillations $\Omega = 628.3 \text{ rad/s}$ (100 Hz) value of the dynamic factor of mass m_1 in a near resonant area is practically identical to mass m_2 and is $\lambda_1 = \lambda_2 \approx 13$. Substituting the data in (6) on condition that the overload is $11 g$, and part of mass of the environment, which joined, is 100 kg (it is taken with a reserve), will get:

$$N_{3\phi} = \frac{\sqrt{6}}{4} \cdot \frac{628^3}{0.7} \left(\frac{0.000274^2 \cdot 206.8}{13} + \frac{0.00035^2 \cdot 161.8}{13} + 100 \cdot 0.000274^2 \right) = 2.2 \text{ kW} .$$

This result well conforms to the calculations resulted above, in fact the drive of the whole vibratory table is able to develop the power $N_{np} = 8 \cdot 330 = 2640 \text{ W}$. That is that in electromagnetic drive the power of vibratory table calculated with a margin.

Preliminary giving the system of differential equalizations (1) in a normal kind relatively higher derivative:

$$D(t, X) := \begin{bmatrix} X_1 \\ \frac{1}{m_1} \cdot [-\mu_{12} \cdot (X_1 - X_3) - c_{12} \cdot (X_0 - X_2) - \mu_1 \cdot X_1 - F(t)] \\ X_3 \\ \frac{1}{m_2} \cdot [F(t) - \mu_{12} \cdot (X_3 - X_1) - c_{12} \cdot (X_2 - X_0) - c_{iz} \cdot X_2 - \mu_{iz} \cdot X_3] \end{bmatrix}$$

$$S1 := \text{Rkadapt} \left[\begin{bmatrix} 0 \\ 0 \\ 0 \\ 0 \end{bmatrix}, 0, 4, 10000, D \right] \quad i := 0.. \text{last}(S1^{(0)})$$

and using parameters resulted above, in the software product of Matcad 13 with the use of the method of Runge-Kutti (Fig. 22) the system is solved. Thus $F(t) = |P_{el} \sin(\Omega t / 2)| \equiv F \sin(\Omega t)$. The amplitude of

vibrations on the working body as planned is $X_1 = 0.2 \text{ mm}$ (equivalent to 8 overloads). From the graphs we can see the static sagging of the resilient element $\delta_{nocm.}$ under the action of the constant component of the effort $P_{nocm} = 1884 \text{ N}$ (9). This sagging will be:

$$\delta_{nocm.} = \frac{P_{nocm}}{c_{12}} = \frac{1884}{3.808 \cdot 10^7} = 0.00005 \text{ m} = 0.05 \text{ mm} .$$

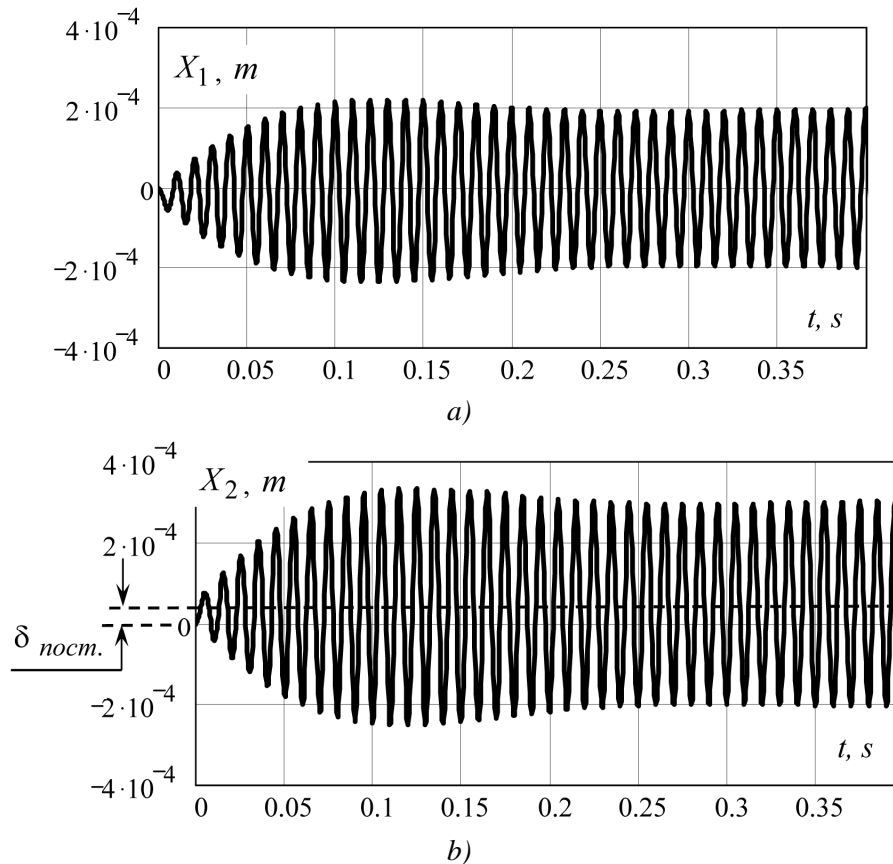


Fig. 22. Theoretical time dependency of motion of the masses m_1 (a) and m_2 (b) in the near resonant mode

14. Experimental verification

The experimental sample of the vibratory table was made (Fig. 23, a). The experiment was conducted with the use of laboratory equipment as follows. The mechanical characteristics were removed from an experimental sample by a two-channel USB-oscilloscope (analog-to-digital converter IRIS of the firm “Video Internet Technologies”), connected to the computer. The power was sourced from the autotransformer with the voltage 150 V . Connecting the sensor of vibro-accelerations (Fig. 23, d) to mass m_1 , the data were read with the use of software product for an USB-oscilloscope.

Let's set the amplitude of vibrations of the first mass which performs the functions of the working body. Taking into account that the sensor which is rigidly fastened to this mass tared ${}^1U_{ha1g} = 66 [mV/g]$ (Fig. 23, d), and amplitude value from the sensor is $U_1 = \frac{2.2 - 1.15}{2} = 0.525 [V]$ (Fig. 24, a), the conversion factor is ξ , and it is an index of overload, we count with the following conditions:

$$\begin{cases} 0.066 [V] \equiv g; \\ 0.525 [V] \equiv \xi g, \end{cases} \quad (20)$$

So, using (20), the overload on a working body will be:

$$\xi_1 = \frac{U_1}{U_{na} 1g} = \frac{0.525}{0.066} = 8.$$



Fig. 23. Experimental sample of the vibratory table (a); fragment of video, where the work of the vibratory table is represented (b); arrangement of equipment during an experiment (c) and sensor which is connected to mass m_1 (d) for the output of time dependences of motion of oscillating mass

As marked above, for the harmonious vibrations the overload ξ determined from the known dependence $\xi = \frac{X \omega^2}{g}$. Therefore, the amplitude of vibrations of the working body will be

$X_1 = \frac{8 \cdot 9.81}{(2 \cdot \pi \cdot 100)^2} = 2 \cdot 10^{-4} \text{ m}$, completely consistent with the results of mathematical modeling (Fig. 22).

The angular frequency of forced vibrations is indeed in the vicinity of 100 Hz (Fig. 24).

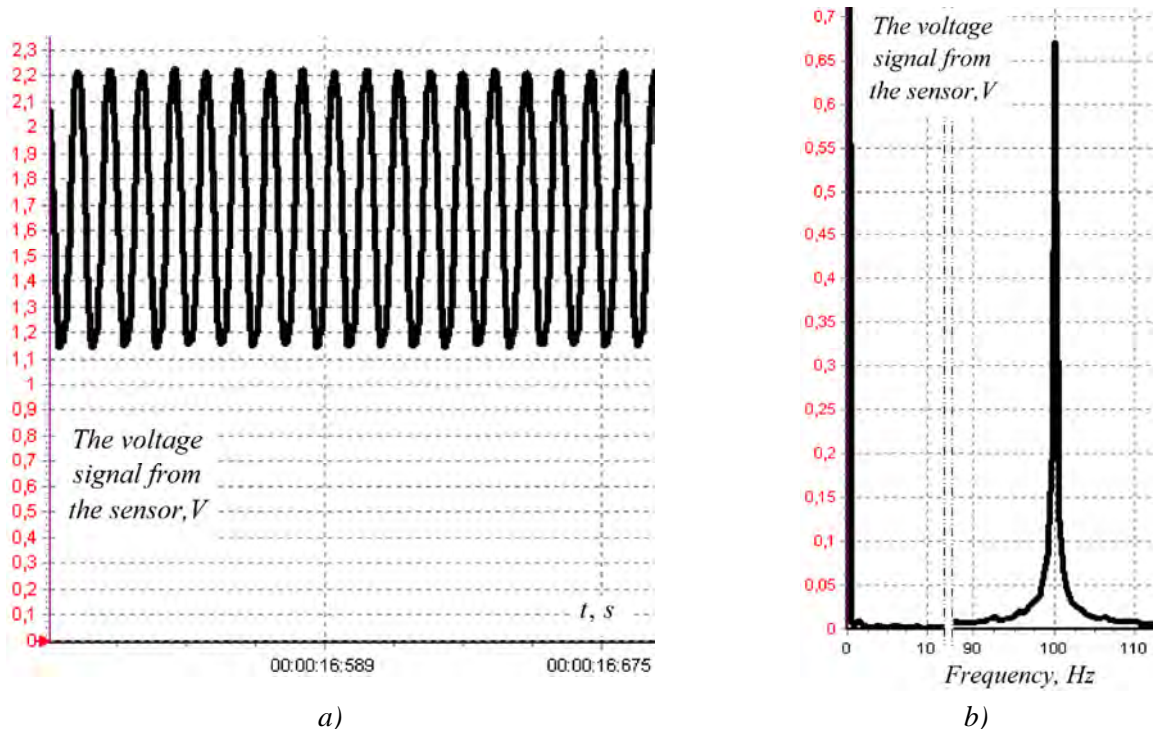


Fig. 24. Experimentally taken off: a - time dependence of motion of oscillating mass m_1 ; b - spectral description of motion of mass m_1 at steady-state oscillations of the system

The work of the vibratory table at a load was investigated. A working body was loaded with steel plates with the mass 120 kg, which were layed through a thick rubber pad. It imitated the real load well, as rubber has high dissipative properties and provides the higher particle of the accession of the mass. Taken characteristics at a supply voltage 150 V indicated that an overload had fallen down to 4 g (in 2 times). However, when raising the voltage to 220 V the vibratory table entered the working mode with an overload 8 g.

Conclusions

A resonance vibro-ground with an electromagnetic drive works at the forced vibrations frequency of 100 Hz ensuring the previously set technological parameters. The maximal power consumption is 2.2 kW; the amplitude of vibrations of working body of the mass $m_1 = 206.7 \text{ kg}$ is $X_1 = 0.2 \text{ mm}$ (overweight on the working body is 8 gravity accelerations).

The aforesaid example proved once again the reality of creation of medium sized vibro-equipment on the base of electromagnetic drive. Technical data are taken from such equipment confirm the efficiency of application of the suggested methods in designing the equipment of this type.

On the massive working body, under relatively small power loss high overloads are attained. This demonstrates significant advantages and prospects of the use of resonance oscillation machines with an electromagnetic drive.

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