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Implementation of dual-frequency resonant vibratory machines with pulsed electromagnetic drive

***Abstract**. The rational method of implementation of dual-frequency resonant systems with multiple eigenfrequencies of oscillations is considered. The efficiency of implementation of such operation modes is substantiated by the use of a pulsed electromagnetic drive with oscillations frequency of 50 Hz. The analysis of the vibrating system dynamics is carried out on the basis of numerical modelling of the system of nonlinear ordinary differential equations. The influence of inertia of auxiliary oscillating mass on the indexes of acceleration of the working device, namely on its maximum value and on the fundamental harmonics ratio, is investigated. The structure of a partial module, which is a means of modernization of single-frequency resonant systems, is proposed.*

Streszczenie.** Rozważana jest racjonalna metoda implementacji systemów rezonansowych o dwóch częstotliwościach z wieloma częstotliwościami drgań własnych. Efektywność realizacji takich trybów pracy jest uzasadniona przez zastosowanie impulsowego napędu elektromagnetycznego o częstotliwości drgań 50 Hz. Analiza dynamiki układu wibracyjnego przeprowadzana jest na podstawie numerycznego modelowania układu nieliniowych równań różniczkowych zwyczajnych. Zbadano wpływ bezwładności pomocniczej masy oscylacyjnej na wskaźniki przyspieszenia urządzenia roboczego, a mianowicie na jej wartość maksymalną i na współczynnik podstawowej harmonicznej. Zaproponowano strukturę modułu częściowego, który jest środkiem do modernizacji układów rezonansowych o jednej częstotliwości.* ***Implementacja systemów rezonansowych o dwóch częstotliwościach z wieloma częstotliwościami drgań własnych

Keywords: oscillations, vibration, resonance, electromagnetic drive, eigenfrequency.
Słowa kluczowe: drgania, rezonans, napęd elektromagnetyczny, częstotliwość drgań własnych.

Introduction

At present, the systems with regulated modes using the control and feedback facilities are becoming more and more widespread. The implementation of such systems is very relevant problem for effective operation of vibratory machines with resonant modes. This is due to the influence of technological factors on the dynamics of their oscillatory system, in particular on its amplitude-frequency characteristic [1–4]. In order to balance this effect out, the amplitude and frequency control is used. It allows to eliminate structural inaccuracies (errors), to provide stable characteristics due to the effect of a changing load. The publications [4, 5] are devoted to solving this problem.

In terms of introduction into practice, the resonant machines should be designed using an alternating current electromagnetic drive. This is due to the following factors:

- high reliability and durability due to the lack of elements and pairs (couples) of mechanical friction;
- simplicity of the machine starting up and stopping due to the absence of influence of the mechanical system on the drive dynamics, which is not typical for machines with an electromechanical drive using the effect of Sommerfeld [6].

Nevertheless, the vibratory resonant machines with single-frequency modes are of limited use [7]. The systems with two operation frequencies are much more effective [8-10]. In addition to the technological advantages of dual-frequency resonant systems, it is also necessary to take into account higher dynamic stability in comparison with single-frequency systems [11].

The dual-frequency systems based on the combined ball-inertial vibrator may also be promising at the present time [8]. The proposed design allows to use the resonance mode at the same frequency due to the use of the Sommerfeld effect. However, such a vibrator has the disadvantages of the mode regulation since the changing of the harmonic amplitude value at this frequency is directly proportional to the mass of balls that is constant.

Therefore, the modernization of existing resonant systems is an urgent problem. In order to do this, one must perfectly investigate the dynamic and structural features of the implementation of dual-frequency resonant systems, as well as perform analytical (mathematical) support in the form of computational techniques.

Premises of implementation of dual-frequency resonant systems

At first, the implementation of dual-frequency resonant systems with an electromagnetic drive should be considered based on the capabilities of the power unit. It is known that the steady-state fluctuations of the current in the electromagnetic circuit may be described by the following solution [12]:

(1) $$i(t)=\frac{U_0}{\sqrt{r^2+(\omega L)^2}}\sin\left(\omega t+\gamma-\frac{\pi}{2}\right)$$

where: U_0 – nominal value of the supply voltage, r – active resistance of the electromagnet coil, ω – cyclic frequency of oscillations, $L = k / 2\delta_0$ – inductance of the electromagnet coil, $k = \mu_0 S w^2$ – structural parameter of the electromagnet, $\mu_0 = 4\pi\cdot 10^{-7}$ [henry/m] – permeability of free space, S – surface area of the core poles, w – number of the coil turns, δ_0 – nominal value of the air gap between the armature and the core of the electromagnet, γ – phase difference (shift) between the voltage and the current.

In order to form the pulsed diagrams of electromagnets, the law of current fluctuations is as follows:

(2) $$\mathrm{i}(\mathrm{t})=\begin{cases}\mathrm{i}(\mathrm{t}), & \text{if } \mathrm{i}(\mathrm{t})>0,\\ 0, \ \mathrm{i}(\mathrm{t})\le 0.\end{cases}$$

The implementation of pulsed systems may be ensured due to the use of thyristor circuits [1]. Taking into account the wide spectrum of harmonics [13], which may be generated by the electromagnetic drive, the possibilities of their use for implementation of multi-frequency resonant systems are worthy of attention. The rectified (redressed) current supplied to the coils of electromagnets may be presented in the form of Fourier series with coefficients describing the amplitude values of the constant, fundamental and second multiple harmonics of the current:

$$i(t) = I_0\left(\frac{1}{\pi} - \frac{1}{2}\cos(\omega t) + \frac{2}{3\pi}\cos(2\omega t)\right) \tag{3}$$

where: $$I_0 = \frac{2U_0\delta_0}{\sqrt{(2r\delta_0)^2 + (\omega k)^2}}.$$

The law of changing of the tractive force of one-type electromagnets connected in parallel may be approximately determined by the Maxwell formula [12]:

$$f(t) = \frac{n\Phi(t)^2}{\mu_0 S} \tag{4}$$

where: $\Phi(t) = \dfrac{\mu_0 Swi(t)}{2\delta_0}$ – magnetic flux generated in the electromagnetic system due to the current passage (flow) through coil turns.

Performing some transformations, the previous expression may be presented as follows:

$$f(t) = \frac{nk}{4}\left(\frac{i(t)}{\delta_0}\right)^2 \tag{5}$$

Taking into account the time dependence of the current (3), the resultant tractive force of the electromagnetic drive will have the following frequency spectrum:

$$f(t) = \Xi\begin{pmatrix}\lambda_0 - \lambda_1\cos(\omega t) + \lambda_2\cos(2\omega t) - \ldots \rightarrow \\ \rightarrow \ldots \lambda_3\cos(3\omega t) + \lambda_4\cos(4\omega t)\end{pmatrix} \tag{6}$$

where: $\Xi = nk\left(\dfrac{I_0}{\delta_0}\right)^2$; λ_0=0,062; λ_1=0,106; λ_2=0,065; λ_3=0,02; λ_4=0,006 – harmonic coefficients of the pulsed tractive force of single-cycle electromagnets.

In order to carry out numerical analysis of the electromagnet tractive force, the following characteristics were adopted: U_0 = 220 V, ω = 314 rad/s, r = 10 Ω, □□ = 0.003 μ, S = 2.784 · 10^{-3}m^2, w = 800, n = 5, and the obtained amplitude values of the tractive force harmonics of the electromagnetic drive are presented in Table 1.

Table 1. Amplitude values of the electromagnets tractive force on the fundamental and multiple harmonics

Harmonic	0	ω	2ω	3ω	4ω
Calculated value of the harmonic	$\Xi\cdot\lambda_0$	$\Xi\cdot\lambda_1$	$\Xi\cdot\lambda_2$	$\Xi\cdot\lambda_3$	$\Xi\cdot\lambda_4$
Amplitude value of the force, N	269.8	461.3	282.9	117.5	26.1
The ration of the forces values on current and fundamental harmonics ω, %	58.6	100	61.3	25	5.3

The use of multiple harmonics of the pulsed tractive force is expedient for implementation of dual-frequency resonant vibrating systems with multiple eigenfrequencies of oscillations and with resonance relation of $z = \omega/\omega_{01} = 2\omega/\omega_{02}$ (Figure 1). In order to implement such systems, the following condition should be satisfied: z = (0.94 ... 0.96). The values of harmonics of fluctuations (for displacements or accelerations) will depend on the values of amplitudes and forms of self-oscillations. The latter ones are defined by inertial and stiffness parameters of the oscillating system.

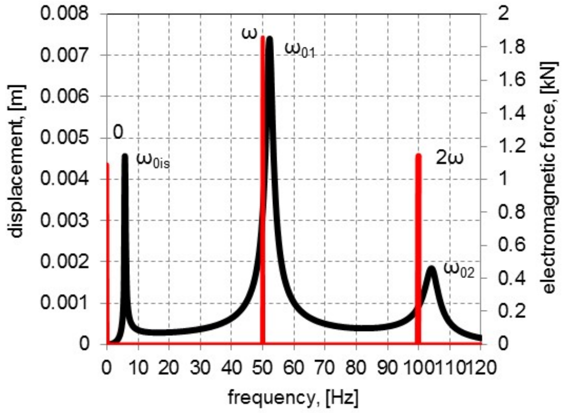


Fig.1. Amplitude-frequency characteristic of implementation of dual-frequency resonant operation mode with pulsed electromagnetic excitation

The formation of the machine structure starts with the determination of its technological part. One of the oscillating masses should be taken into consideration, the influence on which does not cause the change of stiffness of the elastic elements defining the working eigenfrequencies of oscillations and the change of the air gap between the armature and the core of electromagnets. The design diagram of the machine that complies with these requirements is presented in Figure 2. The influence of the technological factors causes the change of the stiffness coefficient of isolators because $\omega_{0is} << \omega_{01}, \omega_{02}$.

The upper mass m_1 is considered as the loading (working) mass. It is connected with the masses m_2 and m_3 by the flat springs with stiffness coefficients c_1 and c_2, correspondingly. The electromagnets are placed on the masses m_1 and m_2. The mass m_1 is mounted on the machine frame using the isolators.

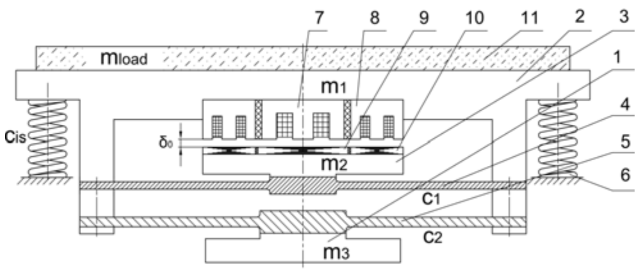


Fig.2. Design diagram of dual-frequency resonant vibrating table

The control method with pulsed thyristor circuit for connecting the electromagnets with the frequency of 50 Hz is proposed. The other electromagnets are connected reactively (directly with the power network) in order to implement the frequency of 100 Hz with the change of phase difference (Figure 3). Such circuit diagram allows rapid changing of operation modes, namely separately or simultaneously using the 50 Hz and 100 Hz modes, as well as influencing the amplitude values of certain harmonics. By means of changing the phase difference, one may influence the amplitude value of the fundamental harmonic.

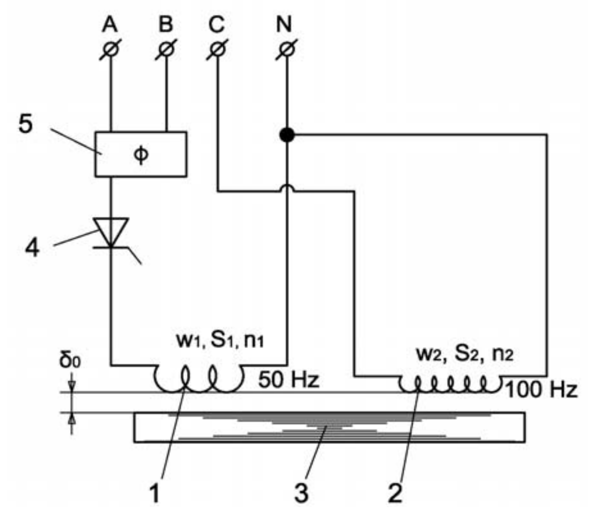


Fig.3. Circuit diagram of electromagnets connection for controlling the conditions of dual-frequency excitation

Dynamics and analysis of dual-frequency resonant machines

The system of nonlinear differential equations of electromechanical oscillations of dual-frequency vibrating systems with independent excitation (with pulsed 50 Hz and reactive 100 Hz circuits) may be presented as follows [14, 15,16]:

$$
(7)\quad \begin{cases}
\dfrac{k}{2\left(\delta_0-\left(y_1(t)-y_2(t)\right)\right)}\cdot \dot{i}_1(t)+\\
+\left[\begin{array}{l} r_1+\left(1-\Phi\left[i(t)\right]\right)\cdot r^{<->}+r^{<+>}+\\ +\dfrac{k\cdot\left(\dot{y}_1(t)-\dot{y}_2(t)\right)}{2\left(\delta_0-\left(y_1(t)-y_2(t)\right)\right)^2}\end{array}\right]\cdot i_1(t)=u_1(t);\\
\dfrac{k}{2\left(\delta_0-\left(y_1(t)-y_2(t)\right)\right)}\cdot \dot{i}_2(t)+\\
+\left[r_2+\dfrac{k\cdot\left(\dot{y}_1(t)-\dot{y}_2(t)\right)}{2\left(\delta_0-\left(y_1(t)-y_2(t)\right)\right)^2}\right]\cdot i_2(t)=u_2(t);\\
\left(m_1+a_{load}\cdot m_{load}\right)\ddot{y}_1(t)+c_1\cdot\left(y_1(t)-y_2(t)\right)+\\
+b_{load}\cdot m_{load}\cdot\omega\cdot\dot{y}_1(t)+b_1\cdot\left(\dot{y}_1(t)-\dot{y}_2(t)\right)+\\
+c_{is}\cdot y_2(t)+b_{is}\cdot\dot{y}_2(t)=f(t);\\
m_2\ddot{y}_2(t)-c_1\cdot\left(y_1(t)-y_2(t)\right)-b_1\cdot\left(\dot{y}_1(t)-\dot{y}_2(t)\right)+\\
c_2\cdot\left(y_2(t)-y_3(t)\right)+b_2\cdot\left(\dot{y}_2(t)-\dot{y}_3(t)\right)=-f(t);\\
m_3\ddot{y}_3(t)-c_2\cdot\left(y_2(t)-y_3(t)\right)-b_2\cdot\left(\dot{y}_2(t)-\dot{y}_3(t)\right)=0;\\
f(t)=\dfrac{n_1\cdot k}{4}\left[\dfrac{i_1(t)}{\delta_0-\left(y_1(t)-y_2(t)\right)}\right]^2+\\
+\dfrac{n_2\cdot k}{4}\left[\dfrac{i_2(t)}{\delta_0-\left(y_1(t)-y_2(t)\right)}\right]^2.
\end{cases}
$$

where: $u_1(t) = U_{01}\sin(\omega t+\varphi)$, $u_2(t) = U_{02}\sin(\omega t)$, – law of the supply voltage changing, φ – phase difference between the voltages on electromagnets, n_1 and n_2 – number of electromagnets connected in parallel in corresponding circuits, $r^{<->}$ and , $r^{<+>}$ – resistance of diode for negative and positive directions of current passage (flow), $\varphi[i(t)]$ – Heaviside function, which simulates the idealized current-voltage characteristic of the diode, $b_1 = \gamma c_1/\omega$, $b_2 = \gamma c_2/\omega$, $b_{is} = \gamma c_{is}/\omega$ – coefficients of viscous friction, $\gamma = 0.025$ – coefficient of internal non-elastic resistance (strength) of the springs material, a_{load} = -0.24 and b_{load} = 1.25 – coefficients, which take into account the influence of load on inertial and damping (dissipative) characteristics of the system.

The manipulated variables (parameters) are the values of nominal voltages U_{01}, U_{02} , φ (as well as the period of thyristor closing for the circuit with thyristor).

An important factor of rational design of machines is the modernization of basic existing models. Such a principle involves the use of existing single-frequency systems (50 Hz [12] or 100 Hz [15]) with the possibility of their structural and parametric transformation into dual-frequency ones. In order to modernize the vibrating table with the oscillations frequency of 100 Hz [15], the partial module (Fig. 4) with certain inertial and stiffness parameters should be introduced into the two-mass structure. The module is considered as single-mass oscillating system with the set of flat springs.

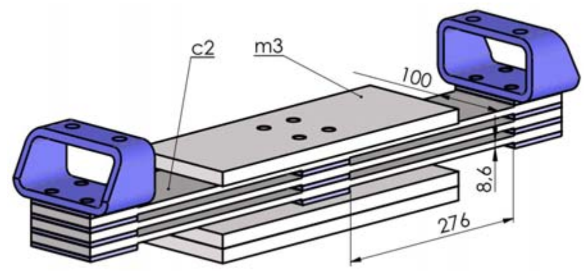


Fig.4. General view of the partial module

At the same time, it is known that stiffness coefficient c_1 may be determined by the same formulas as for two-mass resonant system. Depending on the certain basic structure (with 100 Hz or with 50 Hz operation mode), the calculation parameters may be defined by the following formulas [14]:

— for modernization of 100 Hz vibrating system

$$c_1=\frac{m_1 m_2}{m_1+m_2}\left(2\omega_{01}\right)^2,$$

$$(8)\quad c_2=\frac{m_2 m_3\left[\begin{array}{l}17m_2M-8m_1m_3-\\ 5m_2\sqrt{\dfrac{M\left[9m_2M-16m_1m_3\right]}{m_2}}\end{array}\right]}{2\left(m_1+m_2\right)\left(m_2+m_3\right)^2}\omega_{01}^2.$$

where: $\omega_{01} = \omega/z$ – frequency of free oscillations in the neighbourhood of the excitation frequency of ω = 314 rad/s with resonance relation (adjustment) z, $M = m_1 + m_2 + m_3$ – total mass of the system;

— if the modernization is carried out on the basis of machine with the working oscillations frequency of 50 Hz, so the formulas for calculation of stiffness coefficients are as follows

$$c_1=\frac{m_1 m_2}{m_1+m_2}\left(\omega_{01}\right)^2,\ (9)$$

$$c_2=\frac{m_2 m_3\left[\begin{array}{l}17m_2M-8m_1m_3+\\ +5m_2\sqrt{\dfrac{M\cdot\left[9m_2M-16m_1m_3\right]}{m_2}}\end{array}\right]}{8\left(m_1+m_2\right)\left(m_2+m_3\right)^2}\omega_{01}^2$$

In this way, the dual-frequency system with the modes spectrum of 50 Hz / 100 Hz and 100 Hz may be implemented. These modes may be ensured by means of switching the power circuit from pulsed one to reactive one (using the rectifier or directly).

Considering the value of the inertial parameter m_3 as unknown one, we may assume, that this parameter influences the kinematic characteristics and the harmonics ration of the working mass of the implemented machine. In addition, it is necessary ensure the calculation value of the stiffness coefficient c_2 and to carry out the verification of strength of flat springs being in a state of bending strain as a result of implementation of dual-frequency oscillations.

Basic results of dynamic analysis. Structural implementation and strength verification

In order to solve the system (7) with a help of known numerical methods, the following values of parameters were taken into account: $\omega = 314$ rad/s, z = 0.94, m_1 = 207 kg, m_2 = 161 kg, c_{is} = $2 \cdot 10^4$ N/m, c_1 = $4.046 \cdot 10^7$ N/m, $\varphi = 0$, n = 8, U_0 = $220\sqrt{2}$ V, r = 1 Ω, $\delta_0 = 0.0022$ μ, $S = 3.6 \cdot 10^{-3}$ m^2, w=640, $r^{<->}$ = 10^8 Ω, $r^{<+>}$ = 0.001 Ω. The mentioned parameters belong to real structure of vibrating table with the oscillations frequency of 100 Hz [14]. That's why, its structure may be used for further modernization without changing the existing parameters.

Taking into account the range of m_3 = (15 ... 115) kg, as well as the dependence of $c_2(m_3)$ in accordance with (8), the influence of this parameter on the working mass acceleration characteristics may be analysed. While changing the inertial parameters of the working mass, the maximal value of acceleration varies in the range of a_{1max} = (92.7 ... 70.52) m/s^2, and the harmonics ratio of acceleration is in the range of $A_{1[50Hz]} / A_{1[100Hz]}$ = 0.276 ... 1.116 (Fig.5).

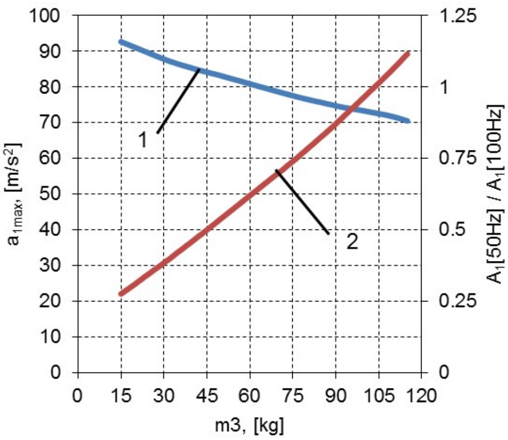


Fig.5. Influence of inertial parameters of the partial module on the characteristics of the working mass acceleration: 1 – on its maximal value, 2 – on the harmonics ratio

According to the value m_3 = 29kg, using formula (8), the value c_2 = 3.209 · 106 N/m has been calculated. These values ensure the acceleration harmonics ration of $A_{1[50Hz]}$ / $A_{1[100Hz]}$ = 0.379 when solving the model (7). Such a ration of harmonics is expedient for vibration consolidation of concrete mixes and for products moulding.

Time dependencies of the working mass acceleration and of the electromagnets tractive force are presented in Fig. 6. The implemented dual-frequency characteristic of the instantaneous acceleration and the pulsed characteristic of the tractive force may be observed.

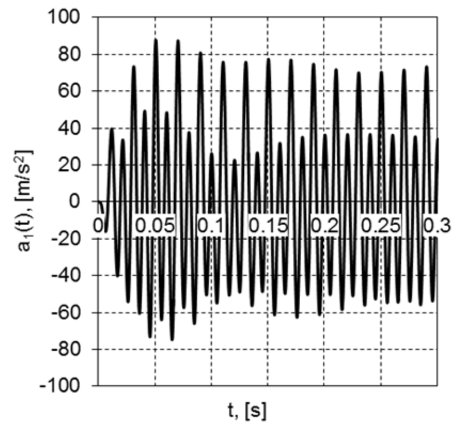
The basic requirements imposed on the structure of the partial module, in particular, to the set of flat springs, consist in the necessity to ensure the calculated value of eigenfrequency and strength of springs being in a state of bending strain. The required eigenfrequency of oscillations, which should be ensured by the springs, may be determined by the following formula:

$$f_{03} = \frac{\sqrt{c_2 / m_3}}{2\pi} = 52.95\,\text{Hz}\,.$$

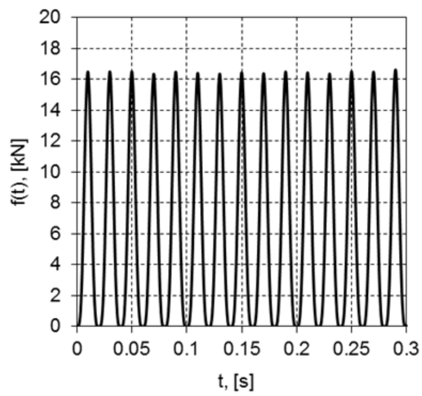
This value may be ensured by means of calculation of the thickness of one flat spring when its width and the working region length are already known. In order to determine the required thickness, the following formula may be used [14, 17, 18]:

$$h = l \cdot \sqrt[3]{\frac{c_2}{E \cdot b \cdot i \cdot k_f}}\,, \qquad (10)$$

where: l = 0.276 m – length of the spring working region, b = 0.1m – width of the spring, i = 6 – number of the working regions of all springs, E = 2.12 · 10^{11} Pa – elasticity modulus of the springs material (steel 60Si7), k_f = 0.85 – coefficient of fixity margin, which takes into account the imperfection of springs fixation.



a)



b)

Fig.6. Time dependencies of the working mass acceleration (a) and of the electromagnets tractive force (b)

Thus, taking into account the values of the parameters used, the calculated value of the thickness of one flat spring is h = 8.55 · 10^{-3} m. In addition, the calculation of oscillations eigenfrequencies of the partial module has been carried out using the SolidWorks Simulation software. The value of the first eigenfrequency f_{03} = 53.135 Hz (Fig. 7) has been obtained. This confirms the validity (reliability) of design calculations carried out, because the error of calculations equals 0,35%.

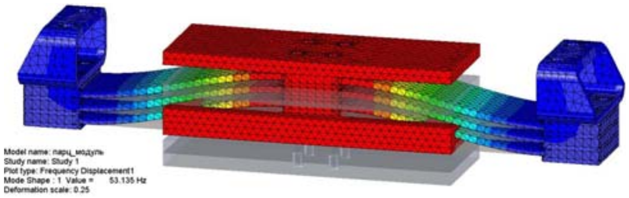
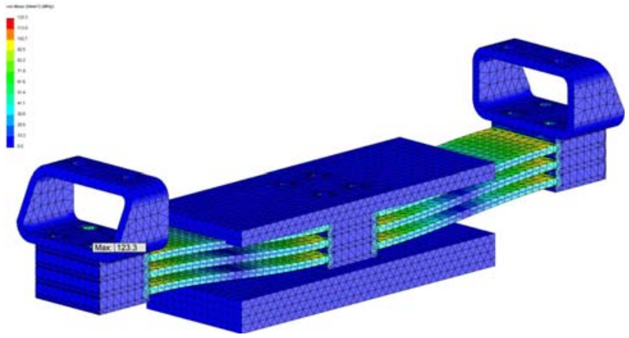
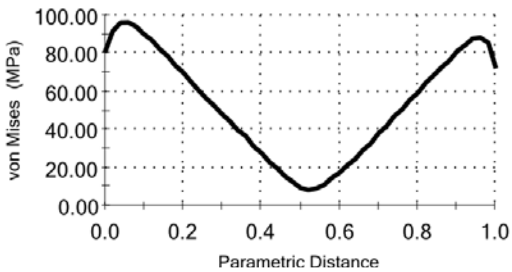


Fig.7. Determination of oscillations eigenfrequency of the partial module

Similarly, the strength verification of the set of flat springs may be carried out using analytical and numerical methods by means of CAE systems on the basis of finite element method (FEM). Based on the results of simulation, the maximal relative displacement of the oscillating masses m_2 and m_3 may be defined as x_{23} = max (x_2 - x_3, x_2 + x_3). After determination of the value $x_{23} = 1.8 \cdot 10^{-3}$ m, the result of calculation of the maximal equivalent stress in accordance with Mises criterion is 123.3 MPa (Fig. 8, a) in the holes for module fixation. The stress distribution along the working region of the flat spring is presented in parametric view in Fig. 8, b.



a)



b)

Fig.8. Stressed state of the partial module (a) and stress distribution along the working region of flat spring (b)

The maximal stress doesn't exceed 100 MPa. This means that the strength and the working capacity of springs are ensured. The reduction of equivalent stresses in springs may be ensured by increasing of their number in the set and of the working region length.

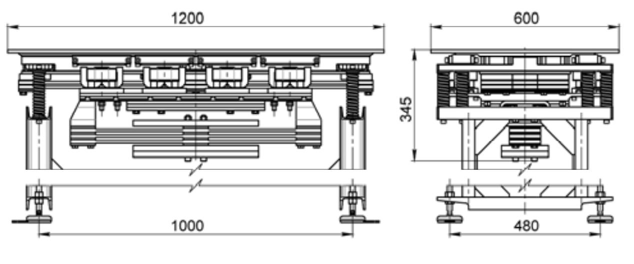


Fig.9. General view and overall dimensions of dual-frequency resonant vibrating table

The implemented vibratory machine (Fig. 9) has corresponding technical characteristics presented in Table 2. Despite the increased power consumption, the implemented structure may operate in two modes: in 100 Hz single-frequency mode and in 50 Hz/100 Hz dual-frequency one. This allows to change its operation efficiency according to the conditions of treatment and to the features of products being manufactured.

Table 2. Comparative technical characteristics of parameters of modernized vibrating table

Characteristic	Operation mode	
	single-frequency	dual-frequency
	100 Hz	50 Hz/100 Hz
Maximal overloading of the working device, $\Gamma = a_{1max}/g$	8.8 … 5.4*	9… 5.6*
Power, kW	3.4	4.8
Total mass, kg	397	
Overall dimensions, mm	1200x600x345	

* m_{load} = 120 kg

Conclusions

1. The efficiency of resonant systems with two working frequencies depends on the range of harmonics and on their amplitude values, which may be implemented by the working device. This values may be ensured by corresponding calculation of parameters of three-mass system with multiple eigenfrequencies. Using the pulsed excitation with 50 Hz frequency, the harmonics ratio depends on the inertial parameters of oscillating masses.
2. In order to control the mode parameters, it is expedient to use the circuit with independent excitation of harmonics, with the possibility of switching from 50 Hz circuit to 100 Hz one, with the phase difference between the circuits and with changing the nominal values of voltages.
3. The calculation formulas for determination of stiffness parameters of flat springs of the partial module, which is a means of structural and parametric modernization of real single-frequency (two-mass) systems, are presented. The design parameters of flat spring are calculated in order to ensure the prescribed elastic and inertial characteristics. The strength of designed springs is verified in the conditions of implementation of dual-frequency resonant mode with corresponding harmonics ration of acceleration.
4. The double-frequency system is implemented on the basis of real machine with initial working frequency of 100 Hz. Taking into account the synthesized operation modes, the design of machine is presented and its technical characteristics are analysed.

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