EXPERIMENTAL STUDY OF RESONANCE VIBRATIONS OF THE VIBRATORY MACHINE EXCITED BY A BALL AUTO-BALANCER

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Розроблено дослідний зразок універсальної резонансної вібромашини широкого застосування. Основу вібромашини становить вібростол. Пружні опори дозволяють здійснювати платформі вібромашини, що має три ступеня вільності, три головних коливальних рухи, що відповідають трьом резонансним частотам. Віброзбудник має форму кульового автобалансира. Передбачається, що кулі в автобалансирі будуть застрявати на першій резонансній швидкості обертання валу. При цьому буде збуджуватися перша форма резонансних коливань.

Вібростол може використовуватися самостійно. Також на платформу можуть встановлюватися насадки з ситами для просіювання або сепарації сипучого матеріалу, ємність для галтовки, форми для цегли, плит тощо.

В результаті експериментальних досліджень встановлено, що належним вибором числа пластин в опорах, числа і маси куль можна забезпечити практично однакові динамічні характеристики вібромашини в конфігураціях вібростола і вібросепаратора. При цьому, при обертанні вала зі швидкостями, що перевищують першу резонансну частоту, платформа робить (практично чисті) вертикальні поступальні коливання. З ростом швидкості обертання валу збільшується амплітуда коливань платформи, а частота практично не змінюється. При перевищенні швидкості обертання валу на 15–20 % першої резонансної частоти, прискорення платформи стають достатніми для утворення киплячого шару на поверхні платформи. Зі збільшенням швидкості обертання валу зростання амплітуди коливань сповільнюється, що пояснюється як ковзанням куль по доріжці, так і нелінійністю опор при великих деформаціях.

Проведені дослідження підтверджують працездатність і універсальність розробленої вібромашини, є підґрунтям для подальшого удосконалення вібромашини

Ключові слова: резонансна вібромашина, віброзбудник, вібросито, вібросепаратор, вібростол, резонансні вібрації, ефект Зомерфельда

1. Introduction

Among such vibratory machines as screens, vibratory sieves, separators, the most energy-efficient are the resonance ones [1–12]. Such machines make it possible to excite significant vibrations of the platforms by vibration exciters with low mass and power. Therefore, there is a general issue related to designing new resonance vibratory machines.

To excite resonance oscillations, one can use passive auto-balancers of the ball-, roller-, pendulum-type [8–10]. The feasibility of this technique for single-mass machines was tested in papers [10–12].

It is a relevant task to use the results reported in [10–16] to design and test experimentally the dynamics of a prototype of the wide-use resonance vibratory machine, in which vibrations are excited by a ball auto-balancer.

2. Literature review and problem statement

Resonance machines are more energy efficient because they employ vibration exciters of lower mass to excite platform oscillations at a larger amplitude [1]. There are known techniques to excite the resonance modes of platform oscilla-

tions by the electromechanical [2], inertial [3] vibration exciters. These techniques require a vibratory machine control system with feedback, which makes the system much more complicated. As the load changes, the machine needs time to reconfigure. These shortcomings are not inherent in the vibratory machines with parametric vibration exciters [4]. In them, a vibration exciter is mounted onto a shaft and consists of the balls that are capable of rolling over the annular segments. The downside of the technique is that the parametric resonance occurs over a relatively narrow band of the rotor rotation speeds.

The simplest in design and most reliable in operation are the exciters of resonance vibrations whose work is based on the Sommerfeld effect [5]. This effect implies that a pendulum mounted on the shaft of an electric motor cannot accelerate and gets stuck at the resonance frequency of the vibratory machine's platform oscillations, which hosts the motor [6]. In order to prevent the electric motor from working under the mode of overload when the pendulum gets stuck, the electric motor was replaced by an air drive in [7]. The drawback of the new vibratory machines is the small efficiency of the air drive.

In accordance with the Sommerfeld effect, the balls [8], or pendulums [9], are stuck at the resonance frequency of rotor rotation in a ball- or pendulum-auto-balancer. Paper [10]

suggested exciting the direct resonance vibrations of the vibratory sieve box by a vibration exciter in the form of a ball auto-balancer. 3D simulation established that the ball jamming mode occurs under the small forces of viscous resistance acting in the supports of the box, on the balls when they move relative to the body of the auto-balancer. In a jamming mode, the balls come together, fail to catch up with the shaft onto which the auto-balancer is mounted, and get stuck at the resonance frequency of platform oscillations. The authors established that the characteristics of vibrations could be changed over wide ranges by altering the number and mass of the balls, the speed of shaft rotation, the rigidity of the supports, etc. It was found that the stuck mode of the balls was extremely resistant to a change in the shaft rotation speed, the mass on the platform. When different bodies are positioned on the platform, the balls almost instantly adjust to the new mass of the platform and continue to excite the resonance oscillations.

In work [11], the method's feasibility was confirmed experimentally at a specially built bench; paper [12] investigated it theoretically.

The results reported in [10–12] make it possible to design and manufacture a new resonance vibratory machine whose operation is based on the Sommerfeld effect.

Let us consider those studies whose results can be applied to test the new vibratory machine.

Paper [13] describes the results of the industrial test of a multi-vibratory inertial vibratory screen for ultra-fine screening. The tests were divided into dynamic and technological. In the course of dynamic tests, the law that governs the vibrations of the screen's box was investigated for vibratory motions, vibration speeds, vibration accelerations; the spectral analysis of vibrations was also performed. During technological tests, the authors determined the technological indicators of the vibratory screen operation. Such an approach is applicable for the case of a vibratory machine for particular use. In the case of a wide-use vibratory machine, the design of the machine can change significantly by installing a specific nozzle on a certain base. Therefore, the feasibility of such a machine should be investigated in separate stages. Dynamic testing is appropriate in the first phase.

The effect of the platform's own oscillation frequencies and the speed of the shaft rotation on the frequency of the pendulum, freely mounted on the shaft, getting stuck was experimentally investigated in [14]. It was established that depending on the speed of shaft rotation, the pendulum gets stuck at a certain resonance frequency of platform oscillations and excites the corresponding resonance oscillations at that. This approach may become a stage in the study of the dynamics of the resonance vibratory machine. The downside of the approach is that its implementation did not pre-identify all the own frequencies and the appropriate forms of platform oscillation.

The accuracy of balancing an axial fan's impeller by adjusting the mass and using ball auto-balancers was experimentally investigated in [15]. The authors described the procedure of balancing rotating parts in an assembly, determining the own frequencies and corresponding forms of fan casing oscillations, studying the fan dynamics. Those procedures and such equipment could complement the dynamic testing methods of resonance vibratory machines implemented in [13, 14].

Paper [16] gives formulae for calculating the total unbalanced mass created by multiple balls (rollers) in an auto-

balancer. The formulae are applicable for calculating the parameters for a vibration exciter in the form of a ball auto-balancer.

Thus, given the results reported in [10–12, 16], it is possible to design and manufacture a prototype of the resonance vibratory machine of wide use with a vibration exciter in the form of a ball auto-balancer. The experimental research methods outlined in works [13–15] could be applied for the dynamic testing of a new resonance vibratory machine.

3. The aim and objectives of the study

The aim of this study is to design and study the dynamics of a resonance single-mass vibratory machine of wide use.

To accomplish the aim, the following tasks have been set:
– to devise the concept of a vibratory machine, to build a model of a vibratory platform, to find theoretically the resonance frequencies and the appropriate forms of the platform oscillations;
– to calculate the parameters and create a prototype of the resonance single-mass vibratory machine;
– to perform experimentally the dynamic tests of the vibratory machine.

4. Procedure to study the platform oscillations and the equipment used

The results reported in papers [10–12] have been applied when designing a vibratory machine. The platform is elastically fixed and has several degrees of freedom. Therefore, the platform has several resonance frequencies and the corresponding forms of oscillations. According to the results from papers [10–12], the balls get stuck at the first (lowest) resonance speed. In this case, the first form of platform oscillation occurs. If the shaft rotates at speeds close to the second, third, and other resonance speeds, the platform may experience the noticeable corresponding components of the oscillations. Therefore, it is necessary to determine all resonance frequencies.

To theoretically determine the resonance frequencies and the appropriate forms of platform oscillations, a platform model on elastic-viscous supports is built and investigated. The elements from the theory of vibratory machines are employed, as well as the Lagrange equations of the second kind.

Experimentally, the resonance frequencies and the appropriate forms of platform oscillation are determined in line with the following methodology [15]. A controlled mechanical oscillation generator (Fig. 1) is used to initiate mechanical oscillations of the platform.
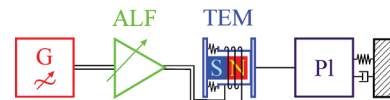

Fig. 1. Enlarged block-scheme of the controlled generator of mechanical oscillations [15]: G — signal generator; ALF — low-frequency amplifier; TEM — electromechanical transducer (vibrating speaker, PRC: 100 Watts, 4 Ohms, 0—20,000 Hz); Pl — platform

The generator's electromechanical converter is attached by magnets at a platform site to maximally create a certain form of resonance vibrations. Eight analog sensors-accelero-

meters GY-61 ADXL335 (Shenzhen HiLetgo Technology Co., Ltd., China) are placed on the platform in such a way as to most effectively capture the appropriate form of resonance oscillations. The signal generator creates sine signals, which are amplified by a low-frequency amplifier and sent to an electromechanical converter. By changing the vibration frequency by the generator, such frequencies are found at which the most intense oscillations of the platform of the appropriate form occur. The platform oscillation form is determined based on signals from the sensors-accelerometers.

The analog signals from eight sensors are processed by the modified device Oscilloscope Motor Tester MT Pro 4 (MLab, Ukraine) [15]. The device makes it possible to:

– digitize signals through 8 analog channels;

– operate under the modes of oscillograph, recorder, and spectrum analyzer.

The modernization involves installing 3.3 V stable power sources in the device, replacing BNC-type connectors with GX16-type connectors.

Experimentally, the platform motion is studied based on vibration speeds and vibration accelerations, measured by the analog sensors of vibration accelerations at the platform's control points.

The experiment also employs the device «Balcom-4» (Kinematica LLC, Russian Federation) for:

– dynamic balancing of the rotating parts in the assembly (shaft, pulley, vibration exciter housing);

– measuring the RMS of the total and reverse component of vibration velocity at 4 control points;

– spectral analysis of vibrations (by vibration velocity);

– the dynamic balancing of rotating parts in an assembly (shaft, pulley, a vibration exciter casing);

– measuring the RMS of the total and revolving component of the vibration speed at 4 control points;

– spectral analysis of vibrations (based on vibration speed).

The limits of the permissible absolute error in the RMS vibration speed measurement, at the base frequency (80 Hz) and over the working frequency range, mm/sec: $\pm(0.1+0.1 \cdot V_m)$, where V_m is the RMS of the measured vibration speed.

The CFM 210 (AS Privod, Ukraine) frequency converter is used to adjust the rotation frequency of the electric motor (shaft). It enables a continuous adjustment of the speed of shaft rotation with a sinusoidal current of frequency ranging from 0 Hz to 800 Hz.

Some additional information on the methodology of the current research is provided when presenting the study results.

5. Results of designing and studying the feasibility of a resonance vibratory machine

5. 1. The concept and model of a vibratory machine, resonance frequencies, and the forms of platform oscillations

The base of the vibratory machine is a vibratory table, which can be used independently. It is assumed that the vibratory table can host various attachments: a separator with sieves, a tumbling container, molds for bricks, slabs, etc.

Let us model and find the dynamic characteristics of the vibratory table. The base of the vibratory table is a mobile platform. The platform has mass m and dimensions $B \times H$ (Fig. 2, a). The platform is supported by four identical

supports in the form of flat bend springs. The distances between the supports are b, h. The total stiffness of the springs is k_z. The supports enable damping with a total ratio of b_z.
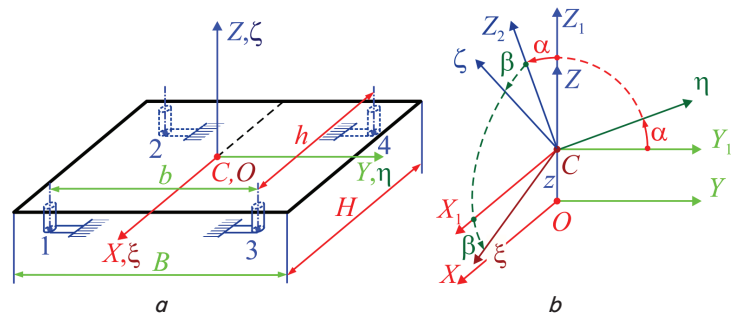

Fig. 2. A vibratory machine model: a — schematic of the vibratory machine; b — the platform motion kinematics

The motion of the platform will be described using two axis systems (Fig. 2). The fixed axes $OXYZ$ originate from the center of the masses of the stationary platform (Fig. 2, a). The movable axes $C\xi\eta\zeta$ coincide with the $OXYZ$ axes at a stationary platform and are rigidly connected to the platform. In the process of motion (Fig. 2, b), the $OXYZ$ axes move into the $C\xi\eta\zeta$ axes in the following way. At first, the $OXYZ$ axes move progressively along the Z axis to a distance determined by the coordinate z. As a result, the $OXYZ$ axes move into an intermediate position $CX_1Y_1Z_1$. Next, the $CX_1Y_1Z_1$ axes rotate at angle a around the X_1 axis. As a result, the $CX_1Y_1Z_1$ axes move into the intermediate $CX_1\eta Z_2$ axes. And finally, the $CX_1\eta Z_2$ axes rotate around the η axis at angle β. Thus, the platform has three degrees of freedom and its motion can be described by the following sequence of displacements:

$$OXYZ \xrightarrow{z} CX_1Y_1Z_1 \xrightarrow{\alpha} CX_1\eta Z_2 \xrightarrow{\beta} C\zeta\eta\zeta. \quad (1)$$

Let us believe that the generalized coordinates z, α, β and their derivatives are small values. Then the absolute deformations of the bend springs are equal to:

$$\Delta l_1 \approx z - b\alpha/2 - h\beta/2, \ \Delta l_2 \approx z - b\alpha/2 + h\beta/2,$$

$$\Delta l_3 \approx z + b\alpha/2 - h\beta/2, \ \Delta l_4 \approx z + b\alpha/2 + h\beta/2. \quad (2)$$

The potential spring energy is:

$$V = \frac{1}{2}\frac{k_z}{4}\left(\Delta l_1^2 + \Delta l_2^2 + \Delta l_3^2 + \Delta l_4^2\right) = \frac{k_z}{8}\left(4z^2 + b^2\alpha^2 + h^2\beta^2\right). \quad (3)$$

The kinetic energy of the platform's progressive motion is equal to:

$$T_{tr} = mv_C^2/2 = m\dot{z}^2/2,$$

where v_C is the module of speed of the center of the mass of the platform; a dot above the value denotes a time derivative.

The kinetic energy of the platform's rotational motion is equal to:

$$T_{rot} = J_\xi\omega_\xi^2/2 + J_\eta\omega_\eta^2/2 = J_\xi\dot{\alpha}^2/2 + J_\eta\dot{\beta}^2/2,$$

where, for an almost homogeneous rectangular plate, the axial moments of inertia are:

$$J_\xi = mB^2 \big/ 12, \quad J_\eta = mH^2 \big/ 12. \tag{4}$$

The platform's kinetic energy is:

$$T = T_{tr} + T_{rot} = m\dot{z}^2/2 + mB^2\dot\alpha^2/24 + mH^2\dot\beta^2/24. \tag{5}$$

The speeds of the ends of bend springs are:

$$v_1 = \Delta\dot{l}_1 \approx \dot{z} - b\dot\alpha/2 - h\dot\beta/2, \quad v_2 = \Delta\dot{l}_2 \approx \dot{z} - b\dot\alpha/2 + h\dot\beta/2,$$

$$v_3 = \Delta\dot{l}_3 \approx \dot{z} + b\dot\alpha/2 - h\dot\beta/2, \quad v_4 = \Delta\dot{l}_4 \approx \dot{z} + b\dot\alpha/2 + h\dot\beta/2. \tag{6}$$

The dissipative function is:

$$D = \frac{1}{2}\frac{b_z}{4}\left(v_1^2 + v_2^2 + v_3^2 + v_4^2\right) = \frac{b_z}{8}\left(4\dot{z}^2 + b^2\dot\alpha^2 + h^2\dot\beta^2\right). \tag{7}$$

The differential equations of the platform motion are derived from the Lagrange equations of the second kind:

$$L_z = \frac{1}{m}\left(\frac{d}{dt}\frac{\partial T}{\partial \dot{z}} - \frac{\partial V}{\partial z} - \frac{\partial D}{\partial \dot{z}}\right) = \ddot{z} + \frac{b_z}{m}\dot{z} + \frac{k_z}{m}z = 0;$$

$$L_\alpha = \frac{12}{mB^2}\left(\frac{d}{dt}\frac{\partial T}{\partial \dot\alpha} - \frac{\partial V}{\partial \alpha} - \frac{\partial D}{\partial \dot\alpha}\right) = \ddot\alpha + \frac{b_z}{m}\frac{3b^2}{B^2}\dot\alpha + \frac{k_z}{m}\frac{3b^2}{B^2}\alpha = 0;$$

$$L_\beta = \frac{12}{mH^2}\left(\frac{d}{dt}\frac{\partial T}{\partial \dot\beta} - \frac{\partial V}{\partial \beta} - \frac{\partial D}{\partial \dot\beta}\right) = \ddot\beta + \frac{b_z}{m}\frac{3b^2}{B^2}\dot\beta + \frac{k_z}{m}\frac{3b^2}{B^2}\beta = 0. \tag{8}$$

The differential equations of the platform motion are split into three independent equations. This means that the generalized coordinates are the main ones and the oscillations of these coordinates determine the resonance frequencies of the platform and the corresponding forms of the oscillations.

From (8), one finds the following resonance frequencies of the platform oscillations:

$$\omega_1 = \sqrt{\frac{k_z}{m}}, \quad \omega_2 = \frac{b}{B}\sqrt{\frac{3k_z}{m}}, \quad \omega_3 = \frac{h}{H}\sqrt{\frac{3k_z}{m}};$$

$$n_i = \omega_i/(2\pi), \quad /i = 1,2,3/. \tag{9}$$

The frequencies are numbered in ascending order: $\omega_1 < \omega_2 \le \omega_3$ $(n_1 < n_2 \le n_3)$. They correspond to the resonance forms of the oscillations shown in Fig. 3.
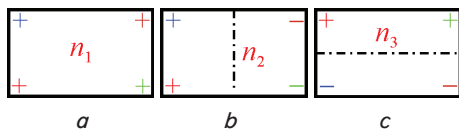

Fig. 3. Resonance forms of platform oscillations:
a — form 1, progressive along the vertical axis;
b — form 2, rotating around the smaller cross-axis of the platform; *c* — form 3, rotating around the larger cross-axis of the platform

A vibration exciter in the form of a ball auto-balancer is placed on the platform. The vibration exciter should excite the first form of resonance oscillations of the platform (the platform together with attachments and load) at the predefined amplitude. In this case, the platform without any additional kinematic restrictions should execute almost undisturbed sinusoidal vertical oscillations.

5. 2. A prototype vibratory machine
5. 2. 1. Description of the prototype vibratory machine

Fig. 4 shows the designed vibratory machine. The base of the vibratory machine is a vibratory table (Fig. 4, *a*). A separator (Fig. 4, *b*) was fabricated to serve as a nozzle.
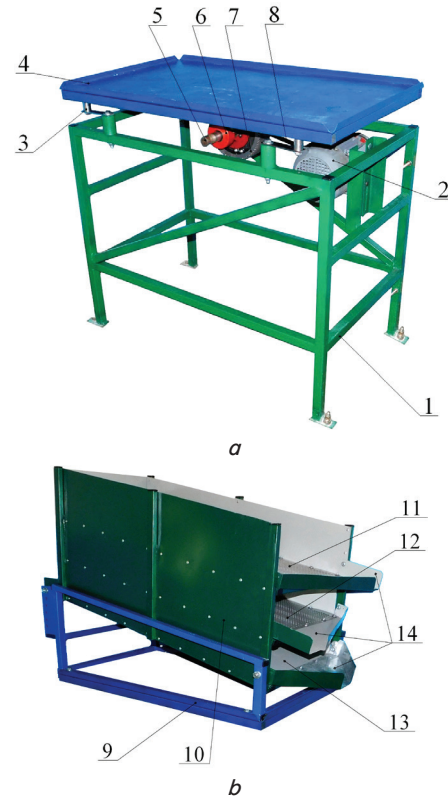

Fig. 4. Vibratory machine:
a — vibratory table (base); *b* — separator (attachment);
1 — bed, 2 — induction electric motor, 3 — elastic support,
4 — platform, 5 — shaft, 6 — support of the shaft,
7 — auto-balancer, 8 — belt transmission, 9 — frame,
10 — box, 11 — top sieve, 12 — lower sieve, 13 — pallet,
14 — fraction guides

The vibratory table consists of such elements (Fig. 4, *a*): bed 1, made from a profile pipe with a cross-section of 30 to 30 mm; induction electric motor 2; four identical elastic supports 3, made from a set of plates of springy steel; platform 4 mounted on elastic supports; shaft 5; shaft supports 6; vibration exciter 7; belt transmission 8.

The separator consists of such elements (Fig. 4, *b*): frame 9, made from a profile pipe with a cross-section of 30×30 mm; box 10, made of sheet steel; top 11 and bottom 12 sieves of different caliber; pallet 13, and guides 14, made of sheet steel.

The stiffness of the supports can be changed by changing the number of plates in the support from 1 to 8 (Fig. 5, *a*). For the dynamic balancing of rotating parts in an assembly (the shaft, vibratory machine case, lid, pulley), the body of the auto-balancer and the pulley have holes with thread (Fig. 5, *b*). In these holes, one can screw the bolts of different lengths (mass) as corrective loads.
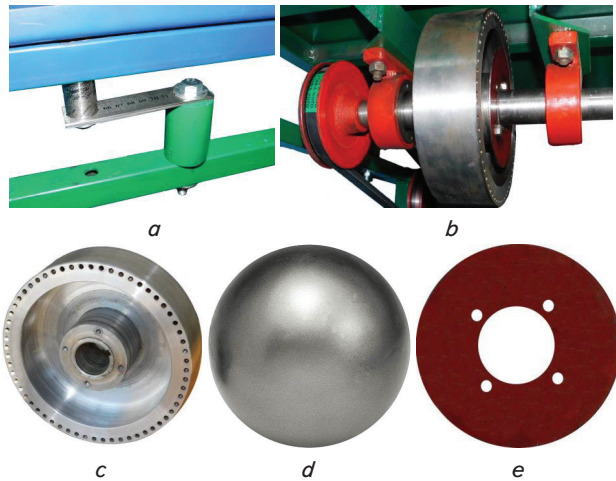
Fig. 5. Elements of the vibratory machine structure:
a — elastic support; b — rotating parts in an assembly (shaft,
vibration exciter case, cover, pulley); c — vibration
exciter case; d — ball; e — cover

The vibration exciter consists of the following parts: the case (Fig. 5, c); a ball or several balls (Fig. 5, d); a cover preventing the ball from falling out of the vibration exciter case (Fig. 5, e).

One or more balls can be placed in the vibration exciter's case. The diameters (mass) of the balls can be changed.

The platform has mass $m_p = 35$ kg and dimensions $B = 1000$ mm, $H = 600$ mm. The separator's mass is $m_s = 60$ kg. The distance between the supports is $b = 840$ mm, $h = 570$ mm.

Table 1 gives the main parameters of the vibratory machine. Calculation coefficients:

$$k_{min} = m_{min}(2\pi n_1)^2 = 35 \cdot (2\pi \cdot 10)^2 = 138.2 \cdot 10^3 \text{ N/m},$$

$$k_{max} = m_{max}(2\pi n_1)^2 = 95 \cdot (2\pi \cdot 10)^2 = 375 \cdot 10^3 \text{ N/m},$$

$$d_{max} = 2a_{max}/(2\pi n_1)^2 = 2 \cdot 20/(2\pi \cdot 10)^2 = 10.13 \cdot 10^{-3} \text{ m},$$

$$R_1 = 0.5(D_r - D_b) = 0.5(0.190 - 0.042862) = 0.0736 \text{ m},$$

$$R_2 = 0.5(D_r - D_{b2}) = 0.5(0.190 - 0.041275) = 0.0744 \text{ m}, \quad (10)$$

The unbalanced mass, created by n identical balls, is calculated from formula [16]:

$$S(n, r_b, m_b) = \frac{m_b(D_r - d_b)^2}{2d_b} \sin\left(n \cdot \arcsin\frac{d_b}{D_r - d_b}\right). \quad (11)$$

Find by using (11):

$$S(1, r_b, m_b) = 0.0238 \text{ kg} \cdot \text{m}, \quad S(3, r_b, m_b) = 0.0634 \text{ kg} \cdot \text{m}.$$

Note:

$$m_{max}/m_{min} = 95/35 = 2.714,$$

$$S(3, r_b, m_b)/S(1, r_b, m_b) = 0.0634/0.0238 = 2.661.$$

Therefore, the oscillations of the vibratory table are to be excited by a single ball, and the oscillations of the vibratory separator – by three balls.

For the vibratory table, the estimated resonance frequencies are:

$$n_1 = \frac{1}{2\pi}\sqrt{\frac{k_z}{m_{min}}} = \frac{1}{2 \cdot 3.142}\sqrt{\frac{140 \cdot 10^3}{35}} = 10 \text{ Hz},$$

$$n_2 = \sqrt{3}\frac{b}{B}n_1 = 1.732 \cdot \frac{820}{1000} \cdot 10 = 14.20 \text{ Hz},$$

$$n_3 = \sqrt{3}\frac{h}{H}n_1 = 1.732 \cdot \frac{574}{600} = 16.57 \text{ Hz}. \quad (12)$$

The total stiffness $k_{min} = 138.2 \cdot 10^3$ N/m is provided by three plates in each support.

For the vibratory separator, the calculated resonance frequencies are the same but at the total spring rigidity of $k_{max} = 373.33 \cdot 10^3$ N/m. Such stiffness is ensured by eight plates in each support.

5. 2. 2. The dynamic balancing of rotating parts in an assembly

After the manufacture of the vibratory machine, the dynamic balancing of the rotating parts in an assembly was carried out. Balancing is necessary to ensure that the imbalance of the shaft does not excite other forms of resonance oscillations of the platform (different from direct progressive motion).

Table 1

Basic parameters of the vibratory machine

No.	Parameter	Value	Note
1	Platform mass m_{min}, kg	35	Known
2	The largest mass of the loaded platform with a nozzle m_{max}, kg	95	Known
3	First resonance frequency n_1, Hz	10	Known
4	The greatest amplitude of vibration acceleration a_{max}, m/s²	20	Known
5	Weight of a steel ball m_b, kg	0.325	Known
6	Ball diameter d_b, mm	0.0215	Known/to be calculated
7	Diameter of a ball track D_r, m	0.19	Known
8	The largest span of vibration displacement d_{max}, mm	10	To be calculated
9	Total coefficient of support stiffness k_z, H/m	$140 \cdot 10^3 \div 375 \cdot 10^3$	To be calculated
10	Distance from the longitudinal axis of the rotor to the center of the ball mass R, m	0.0735	To be calculated

The balancing protocol is shown in Fig. 6.

Fig. 6. Protocol of the dynamic balancing
of rotating parts in an assembly

The first correction plane is located on the casing of the vibratory exciter (test mass of 3.56 grams). The second correction plane is located on the pulley (test mass of 3.85 grams). The vibration sensors were located on the supports of the shaft (Fig. 5, *b*). After balancing, the revolving components of the RMS vibration speed do not exceed 0.3 mm/s, which is less than the sensitivity of the balancing device.

The residual RMS vibration speed, measured by the «Balcom-4» instrument at control points 1–4 on the platform, located above the supports, do not exceed 6 mm/s. The relatively large residual non-revolving vibration speeds are caused by the belt transmission.

5. 3. Determining the dynamic characteristics of the vibratory machine
5. 3. 1. Determining the resonance frequencies and the appropriate forms of platform oscillations

Eight control points were located on the platform's working surface (Fig. 7). Eight single-axis GY-61 ADXL335 sensors were installed at these points.
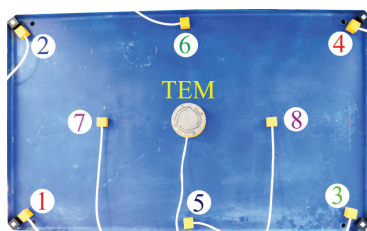

Fig. 7. Arrangement schematic of acceleration sensors 1—8
and TEM (electromechanical transducer)

An electromechanical converter was installed on the platform to maximally excite a certain form of resonance oscillations. The result of the experiments is the derived resonance frequencies:

$$n_1 = 10.04 \, \text{Hz}, \quad n_2 = 14.54 \, \text{Hz}, \quad n_3 = 16.67 \, \text{Hz}. \tag{13}$$

As one can see, the actual values of resonance frequencies are no more than 2.5 % different from the estimated. The discrepancies are explained by the fact that the platform in the calculations was considered a rectangular slab.

The experimentally-found oscillation forms are consistent with the forms derived theoretically. Fig. 8, *a* shows a fragment of the vibration accelerations oscillogram acquired from the MT Pro 4 oscillograph for the third resonance form of platform oscillations.
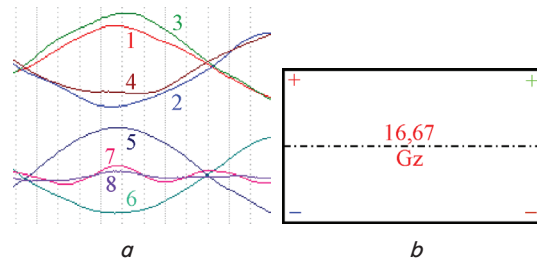

Fig. 8. The third resonance form of platform oscillations:
a — fragment of vibration accelerations oscillogram;
b — appropriate form of platform oscillations; 1—8 — signal
corresponding to sensor No. 1—8

The oscillogram (Fig. 8, *a*) shows that at a frequency of 16.67 Hz the platform executes a third form of resonance oscillations (Fig. 8, *b*) as the signals: 1, 3, 5 are in phase; 2, 4, 6 are in phase and are in the opposite phase to signals 1, 3, 5; 7.8 – almost do not change as they are on the line of the nodes.

5. 3. 2. Studying the oscillations of a vibratory machine in the configuration of a vibratory table

The platform weighs 35 kg, one support is formed by three plates. Oscillations are excited by one ball weighing 0.325 kg. The current frequency varies from 5 to 35 Hz with a 5 Hz increment. Fig. 9 shows the dependence of the amplitude of the variable component of vibration acceleration on the frequency of shaft rotation.
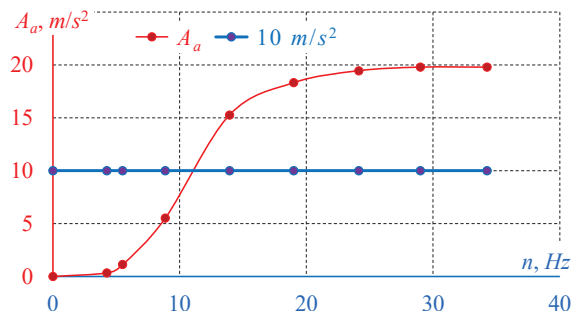

Fig. 9. Amplitude of the changing component of the platform
vibration acceleration

When the amplitude of the changing component of the vibration acceleration reaches 10 m/s^2, a boiling layer appears (Fig. 10). The boiling layer appears at the shaft rotation frequencies exceeding 12 Hz.

The lowest shaft rotation frequency, at which one observed a steady jam of the ball, was 4 Hz. At a lower frequency, the ball ceased to roll along the track, fell, and then hovered near the lowest position.

As one can see from Fig. 9, changing the shaft rotation speed can increase the amplitude of the platform oscillations. However, when the shaft rotation frequency exceeds 30 Hz, the amplitude of the oscillations almost stops growing, similar to that the platform's oscillation amplitude stops growing. Theoretically, the amplitude should continue to grow [12].

However, it turned out that this is prevented by the ball sliding along the track, which occurs when the shaft rotation frequency exceeds 30 rev/s.

Fig. 10. The appearance of a boiling layer on the platform

The vibration speed oscillogram, built for control points 1, 2 (Fig. 11, *a*), shows that the platform's oscillations are progressive, harmonic. The spectral diagram, at 10 Hz, exhibits a single peak (Fig. 11, *b*), corresponding to the vibrations caused by a ball jamming at the first resonance frequency.

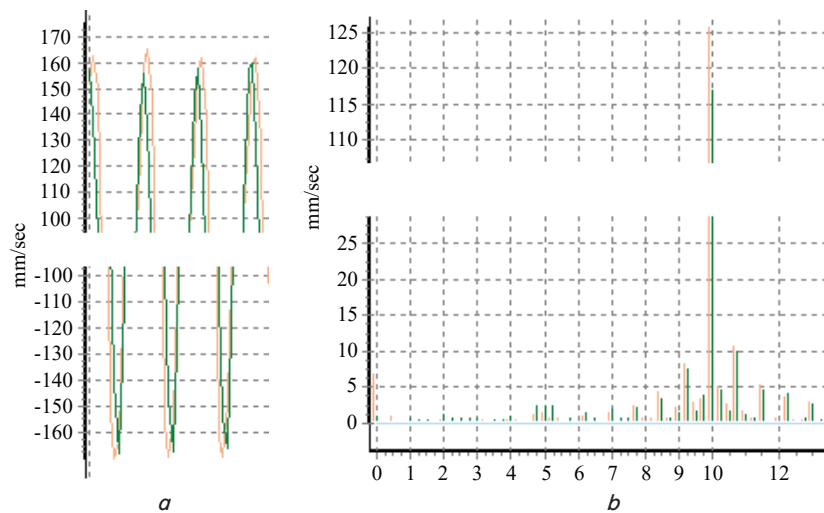A similar vibration speed oscillogram and a spectrum of frequencies are obtained at points 3, 4.

Fig. 11. Fragments (for the shaft rotation speed of 13.93 rev/s): *a* — vibration speed oscillograms at points 1, 2; *b* — frequency spectra at points 1, 2

5. 3. 3. Studying the oscillations of a vibratory machine in the configuration of a vibratory separator

The weight of the platform with a nozzle is 95 kg, one support is formed by eight plates. The oscillations are excited by three identical balls weighing 0.324 kg and a diameter of 42.862 mm. The current frequency varies from 5 to 35 Hz with a 5 Hz increment. Fig. 12 shows the dependence of the amplitude of the variable component of vibration acceleration on the shaft rotation frequency.
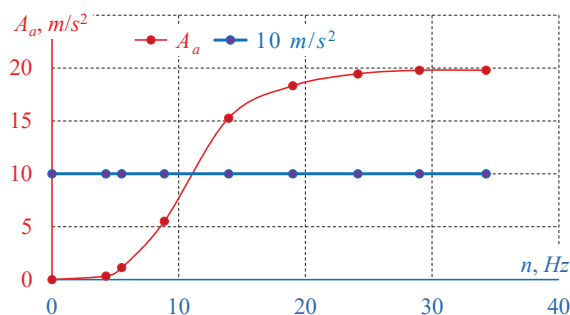

Fig. 12. Amplitude of the variable component of the platform vibration acceleration

The magnitude of the amplitude of platform oscillations almost coincides with the amplitude obtained for the vibratory table. However, the amplitude increases throughout the entire range of change in the speed of shaft rotation, which corresponds to the theory [12]. A boiling layer appears at the shaft rotation frequency exceeding the frequency of 11.5 Hz. The vibratory sieve becomes operational when a boiling layer appears.

6. Discussion of results of studying the rectilinear translational vibrations of the vibratory machine's platform, excited by a ball auto-balancer

Based on the results reported in [10–12], the concept of a new resonance vibratory machine has been suggested; its operation is based on the Sommerfeld effect. The base of the vibratory machine is a vibratory table, which can be used independently. In addition, the platform of the vibratory machines can host a variety of attachments in the form of a sieve, separator, molds for bricks, slabs, a tumbling capacity, etc. That provides for the versatility of the vibratory machine, its wide scope of application.

To implement the concept, a design was chosen in which a rectangular platform rests on four identical bend springs (Fig. 2). In this case, the platform has three degrees of freedom and can execute three main oscillatory motions (Fig. 3), corresponding to three resonance frequencies (9).

According to the concept, of all possible types of the kinematic motion of the platform, only those oscillations should be excited that correspond to the lowest resonance frequency of the platform oscillations. At the same time, almost undisturbed progressive vertical oscillations of the platform should be initiated without additional kinematic restraints, guides, etc. To this end, a ball auto-balancer is used.

A prototype vibratory machine (Fig. 4) implements the proposed concept. In order to change the vibration parameters:

– the bend springs are formed by plates whose number in a single support can be changed from 1 to 8 (or more);

– the auto-balancer's case can host one or more balls; balls of different mass can be used;

– the speed of shaft rotation can be controlled by a frequency converter.

Before operating the vibratory machine in a certain configuration, it needs to be pre-configured. When installing massive attachments on the platform, one needs to increase both the stiffness of the supports and the mass of the ball or the number of balls.

The configured vibratory machine is controlled by changing the speed of the shaft rotation. To increase the amplitude of platform oscillations, it is necessary to increase the shaft rotation frequency [12].

As a result of experimental studies, it was found that the actual resonance frequencies of the platform (13) are no more than 2.5 % different from the estimated (12). At the same time, the actual and calculated forms of resonance oscillations are the same.

The appropriate choice of the number of plates in the supports and the number of balls can ensure almost matching dynamic characteristics of the vibratory machine in the configurations of a vibratory table and a vibratory separator (Fig. 9, 12). In this case, when the shaft rotates at speeds exceeding the first resonance frequency, the platform makes almost undisturbed vertical progressive oscillations (Fig. 11). As the shaft rotation speed increases, the platform's oscillation amplitude increases while the frequency practically does not change (Fig. 9, 12). If the shaft rotation speed exceeds, by 15–20 %, the first resonance frequency, the accelerations of the platform are enough to form a boiling layer at the surface of the platform. As the shaft rotation speed increases, the platform's oscillations amplitude increases. When the shaft rotation speed exceeds 30 rev/s, the growth of the vibration amplitude slows down, which is due to both the sliding of the balls along the track and the non-linearity of the supports at large deformations.

Thus, the current work shows the feasibility of the proposed concept of a vibratory machine and the operability of the developed experimental prototype. However, it should be noted that the work investigated the dynamic characteristics of the machine itself rather than the execution of technological functions by the machine.

In the future, it is planned to study the productivity of the vibratory machine in various configurations, as well as improving its design.

7. Conclusions

1. The concept of a universal resonance vibratory machine of wide use has been proposed. The base of the vibratory machine is a vibratory table, which can be used independently. The platform can host attachments with sieves for sifting or separating a loose material, tumbling containers, molds for bricks, slabs, etc. The platform oscillations are excited by a ball auto-balancer. It is assumed that the balls in the auto-balancer would be stuck at the lowest (first) resonance frequency of the platform oscillations. At the same time, progressive vertical harmonic oscillations of the platform would be excited.

To implement the concept, a design was chosen in which a rectangular platform is installed on four identical bend springs. In this case, the platform has three degrees of freedom and can execute three main vibrational motions corresponding to three resonance frequencies.

2. A prototype vibratory machine implements the proposed concept. In order to change the vibration parameters:
– the bend springs are formed by plates whose number in a single support can be changed from 1 to 8;
– the auto-balancer's casing can host one or more balls; balls of different mass can be used;
– the speed of shaft rotation can be controlled by a frequency converter.

Before operating the vibratory machine in a certain configuration, it needs to be pre-configured. When installing attachments on the platform, the rigidity of the supports must be increased. To excite oscillations of a heavier (loaded) platform, one needs to increase the weight of the ball or the number of balls.

The configured vibratory machine is controlled by changing the speed of the shaft rotation. To increase the amplitude of platform oscillations, it is necessary to increase the shaft rotation frequency.

3. Experimental studies have established that the appropriate choice of the number of plates in the supports, the number and mass of the balls could ensure almost matching dynamic characteristics of the vibratory machine in the configurations of a vibratory table and a vibratory separator. At the same time, when the shaft rotates at speeds exceeding the first resonance frequency, the platform executes (almost undisturbed) vertical progressive oscillations. As the shaft rotation speed increases, the platform's oscillations amplitude increases while the frequency practically does not change. If the shaft rotation speed exceeds, by 15–20 %, the first resonance frequency, the accelerations of the platform are sufficient to form a boiling layer at the surface of the platform. As the shaft rotation speed increases, the platform's oscillations amplitude increases. When the shaft rotation speed exceeds 30 rev/s, the growth of the vibration amplitude slows down, which is due to both the sliding of the balls along the track and the non-linearity of the supports at large deformations.

This study has confirmed the feasibility of the proposed concept of a vibratory machine and the operability of the designed experimental sample.

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