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EXPERIMENTAL STUDY INTO ROTATIONAL-OSCILLATORY VIBRATIONS OF A VIBRATION MACHINE PLATFORM EXCITED BY THE BALL AUTO-BALANCER

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Експериментально досліджені поворотно-коливальні вібрації платформи вібромашини, збуджені кульовим автобалансиром.

Закон зміни віброприскорень на платформі вивчався з використанням датчиків акселерометрів, плати аналогово-цифрового перетворювача з USB інтерфейсом і персонального комп'ютера. Амплітуда швидких і повільних вібропереміщень платформи вивчалася з використанням лазерного променя.

Встановлено, що резонансна частота (частота власних коливань) платформи становить: 62,006 рад/с для маси платформи 2000 г; 58,644 рад/с – для 2180 г; 55,755 рад/с – для 2360 г. Похибка визначення частот не перевищує 0,2 %.

Кульовий автобалансир збуджує практично ідеальні двочастотні вібрації платформи вібромашини. Повільна частота відповідає швидкості обертання центру куль навколо поздовжньої осі вала, а швидка – швидкості обертання валу з прикріпленим до нього дебалансом. Двочастотний режим виникає в широкому діапазоні зміни параметрів і основні його характеристики можна змінювати зміною маси куль і дебаланса, кутової швидкості обертання валу.

Експериментально встановлено, що кулі застряють на частоті, меншою приблизно на 1 % резонансної частоти коливань платформи.

У припущенні, що платформа здійснює двочастотні коливання, в програмному пакеті для статистичного аналізу Statistica були підібрані коефіцієнти в відповідному законі. При цьому було встановлено, що:

– процес визначення величин коефіцієнтів стійкий (робастний), коефіцієнти практично не змінюються від зміни інтервалу часу вимірювання закону руху платформи;

– амплітуда прискорень від повільних коливань прямопропорційна сумарній масі куль і квадрату частоти застрявання куль;

– амплітуда швидких коливань прямопропорційна дебалансу на корпусі автобалансира і квадрату кутової швидкості обертання валу.

Розбіжність між законом руху, отриманим експериментально і законом, отриманим методами статистичного аналізу, менша 3 %.

Отримані результати роблять актуальними як аналітичні дослідження динаміки, розглянутої вібромашини, так і створення дослідного зразка вібромашини

Ключові слова: інерційний вібрюзбудник, двочастотні вібрації, резонансна вібромашина, автобалансир, інерційна вібромашина

1. Introduction

Among such vibratory machines as screeners, vibratory tables, vibratory conveyers, vibratory mills etc., very promising ones are the multi-frequency-resonance machines.

Multi-frequency vibratory machines demonstrate better performance [1], resonance machines are the most energy efficient [2], and the multi-frequency-resonance ones combine the advantages of both the first and second [3].

The excitation of two-frequency resonant vibrations is proposed to enable using the passive auto-balancers – ball, roller, pendulum [4]. The method is based on the Sommerfeld effect [5].

Designs of the new vibratory machines have been proposed and the operational performance of several such

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machines has been studied theoretically, employing 3D modeling, and conducting the field and computational experiments [5–8].

It is a relevant task to experimentally investigate the workability of the specified excitation method for dual-frequency resonance vibrations for a single-mass vibratory machine with the rotational-oscillatory motion of platform.

2. Literature review and problem statement

Paper [4] suggested using, as a dual-frequency vibration exciter, a passive auto-balancer of the ball-, roller-, or pendulum- type. Under certain conditions, correcting loads get together, cannot catch up with the shaft onto which the auto-balancer is mounted, and get stuck at the resonance frequency of platform oscillations. This is how the first, inertial vibration, exciter forms, stimulating slow resonant oscillations of the platform. The unbalanced mass is installed on the casing of an auto-balancer. This is how the second, inertial above-resonance vibration, exciter forms, exciting fast (above-resonance) oscillations of the platform. Dual-frequency vibration parameters are changed by altering the speed of shaft rotation, the unbalanced mass on the auto-balancer's casing, the total mass of loads.

The workability of the new method was investigated for a single-mass vibratory machine with a rectilinear translational motion of platform, applying 3D modeling [4], by conducting a field experiment [6], analytically [7], and [8] in the course of computational experiments [8].

Of relevance is the initiation of dual-frequency vibrations in single-mass vibratory machines with the rotational-oscillatory motion of platform. Such machines have a simple structure and a single resonance frequency, facilitating the task of their design.

It should be noted that the modes at which correcting loads get stuck were investigated:

[9] – for the rotor, mounted on two isotropic supports, which held a dual-pendulum auto-balancer;

[10] – within a flat model of the rotor on isotropic supports, which held a multi-pendulum (multi-ball) auto-balancer;

[11] – within a flat model of the rotor on isotropic supports, which held a multi-pendulum (multi-ball) auto-balancer;

[12] – for a flexible rotor with two dual-pendulum auto-balancers;

[13] – for a flat rotor with a two-ball auto-balancer, installed in the middle of a flexible weightless shaft;

[14] – within a flat model of the balanced rotor on isotropic supports, which held a two-ball auto-balancer;

[15] – for the rotor, mounted on two isotropic supports, which held two dual-pendulum auto-balancers;

[16] – within a flat model of the rotor on isotropic supports, holding a two-ball auto-balancer;

[17] – within a model in which the disc rotor is mounted onto a flexible shaft and the shaft holds a two-ball auto-balancer;

[18] – within a flat model of the rotor on anisotropic supports, holding a two-ball auto-balancer;

[19] – within a flat model of the rotor with a two-ball auto-balancer installed on isotropic supports attached to an elastic-viscous base that executes the rectilinear reciprocating motion.

In the cases specified, the rotor is mounted on axisymmetric (isotropic) supports. In all cases, one observed the modes at which loads get stuck at frequencies close to resonant.

However, the examined vibratory machine has a strong asymmetry of supports. It is therefore important to test whether in this case the passive auto-balancer would operate as two independent inertial vibration exciters and whether dual-frequency vibrations would be excited in this case.

Analytically, the stability of regimes for getting stuck was examined by Lyapunov at a small value of ratio of mass of the correcting loads to the rotor's mass. On the one hand, such studies are rather labor-intensive. On the other hand, the stability conditions obtained using this technique demonstrate (strongly) asymptotic properties [8]. Therefore, the conditions become inapplicable even at a slight increase in the small parameter. In contrast, the law of motion of the system, which corresponds to the mode when loads get stuck, shows no asymptotic properties and is almost uniformly suitable for all values of the system parameters.

Given the above, it is relevant initially to experimentally explore the law of rotational-oscillatory vibrations of a vibratory machine platform, excited by the ball auto-balancer.

3. The aim and objectives of the study

The aim of this work is to study experimentally the rotational-oscillatory vibrations of a vibratory machine platform, excited by the ball auto-balancer. Achieving the aim would make it possible to draw a conclusion about the operational performance of the examined vibratory machine and to address correctly the further analytical study into its dynamics.

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

– to determine the resonance frequencies of the platform with and without additional loads, to verify research methods;

– to establish the type of vibrations of the platform and to select coefficients for the assumed law on a change in the vibration accelerations of the vibratory machine platform employing the methods of statistical analysis;

– to investigate the operation of auto-balancer as a dual-frequency vibration exciter;

– to study the effect of parameters of the vibratory machine and auto-balancer on the characteristics of dual-frequency vibrations.

4. Methods to study the vibrations of a vibratory machine platform excited by the ball auto-balancer

The rotational-oscillatory motion of a vibratory machine platform is studied based on the motion of a platform frame. The frame is considered to be a solid body. Motion of the frame is studied based on vibration accelerations. This relates to that the vibration displacements (vibration velocity) of slow oscillations of the platform are significantly larger than the vibration displacements (vibration velocity) of rapid oscillations. Only the vibration accelerations are comparable.

In order to measure vibration accelerations, the acceleration sensor MMA6231Q 2AX 4 is used. The range of measured vibration accelerations is ±4g. The analog signal from the sensor is digitized by the analog-to-digital converter (ADC) board ADXL202EB-232A with a USB interface made by Analog Devices, Inc. (USA). The sensor is connected to the board, which is connected to a personal computer.

The measuring system operates under 3 modes:

1. Under the oscilloscope mode, a computer screen displays a chart of several oscillations of the signal acquired by the board.

2. Under the mode of spectral analyzer, a computer screen displays a chart of the oscillation spectrum.

3. Under the recorder mode, the signal acquired from the sensor is picked up and digitized.

The type of the excited vibrations of the platform is to be defined applying the provisions from the theory of oscillations [20].

The range of frequencies of the excited vibrations is determined under the mode of spectral analyzer.

The law of change in the vibration acceleration, acquired and digitized under the recorder mode is interpreted as a signal. The data obtained are subsequently stored in the form of a table.

An acceleration sensor allows acquiring an analog signal from the frame. The ADC board digitizes the signal N times per second and records it in the form of a table with two columns. The first column is time, the second column is the magnitude of a signal in volts.

Based on the conducted experiments, an assumption is made on that the platform oscillations are the dual-frequency ones. Coefficients in the law of change on the vibration accelerations at the two-frequency oscillations are identified by using the least squares method, implemented in the software package for statistical analysis Statistica [21]. To this end, the data from a table of vibration accelerations are processed by Statistica. Such an approach improves the accuracy of determining such vibration characteristics as the rapid and slow oscillation frequency. That greatly simplifies the process of determining the amplitudes of oscillations in corresponding vibration accelerations.

Coefficients in the law of change in the vibration accelerations are derived by at least three times. This makes it possible to both find the mean values for the coefficients and to evaluate the accuracy of determining these magnitudes.

We compare the laws of change in the vibration accelerations, obtained experimentally and when using the methods of statistical analysis.

We propose to examine the proportionality of change in the amplitude of the platform applying a laser pointer.

Additional details of the course of experiments are described in the statement of the results of experiments.

5. Results of studying the rotational-oscillatory vibrations of a vibratory machine platform excited by the ball auto-balancer

5. 1. Description of the laboratory bench for a vibratory machine

To investigate the vibrations excited by the ball auto-balancer, we modernized the bench constructed earlier [6] (Fig. 1). In the experiments, we chose the design of supports that enable the rotational-oscillatory motions of the platform (Fig. 1, *b*).

The bench enables:

– changing the number of balls in an auto-balancer;

– changing the mass of the unbalanced mass by using several identical unbalanced masses (Fig. 1, *c*);

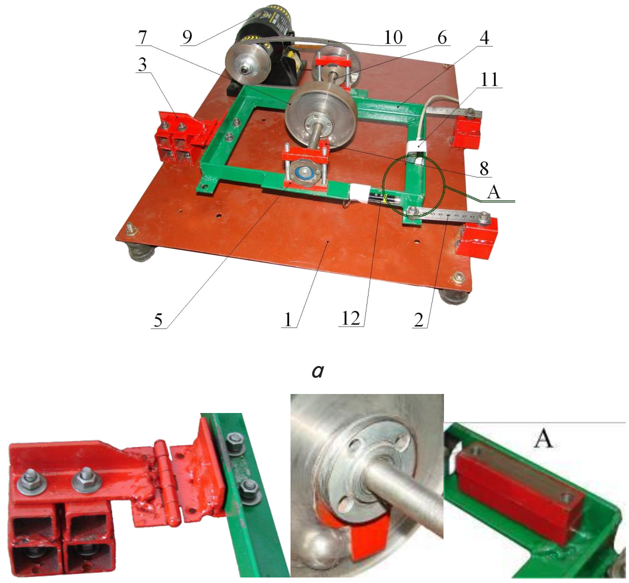
– changing the mass of the platform applying additional loads (Fig. 1, *d*);

– changing the frequency of shaft rotation, the rigidity of spring supports;

– measuring the vibration accelerations at the platform;

– observing the motion of balls relative to the casing of an auto-balancer under stroboscopic light;

– estimating the amplitudes of rotational-oscillatory vibrations of the platform based on the motion of the laser beam on the screen.



a

b *c* *d*

Fig. 1. Bench for a vibratory machine: *a* – general layout (1 – bed, 2 – elastic support of the platform, 3 – hinge support of the platform, 4 – platform, 5 – shaft support; 6 – shaft; 7 – auto-balancer; 8 – unbalanced mass; 9 – asynchronous motor; 10 – belt transmission; 11 – vibration sensor, 12 – laser pointer); *b* – hinge support of the platform; *c* – unbalanced mass and balls; *d* – additional loads, A – place to install additional loads

5. 2. Characteristics of the bench for a vibratory machine and testing the methods of research

Experiments were conducted for two balls in the auto-balancer. The mass of one ball is 21 gm.

The existence of a hinge support enables only one motion for the platform, which is the rotational-oscillatory motion.

In the absence of shaft rotation and at fixed balls in the auto-balancer, we investigated small rotational-oscillatory motions of the platform based on its vibration accelerations.

In accordance with the linear theory of oscillations, the law of change in the vibration accelerations must conform (formally) to the law of free damped oscillations [20]:

$$U = \alpha e^{-ht}\cos(\omega_0 t + \beta) + U_0, \qquad (1)$$

where ω_0 is the frequency the platform's natural oscillations; α, β are the amplitude and phase of oscillation; U_0 is a constant offset.

We turned the ADC board on under the oscilloscope mode, the platform deviated from the equilibrium position and was then abruptly released. Next, the oscilloscope readings were recorded during platform's oscillations.

The result of the conducted experiments is the derived dependence of the platform's vibration accelerations on time. In the dependence, time varies at an equal step. Note that both a step in the change of time and the interval of "measuring" the vibration accelerations may vary within broad ranges.

Next, the data acquired were transferred to the software package for statistical analysis Statistica. The software package employs the least squares method to determine unknown coefficients α, h, ω_0, β, U_0, included in the law of damped oscillations (1).

Parameter identification was carried out at different time intervals, corresponding to n=1, 3, 6, 9 slow oscillations of the platform of the vibration machine. During a single slow oscillation of the platform, its vibration accelerations are measured approximately 115 times. We computed mean values for parameters; the results of identification are given in Table 1. Similar results were obtained for the platform with additional masses of 180 gm and 360 gm (Table 1 shows the mean values of parameters). It also gives the percentage of the maximum difference η between the vibration accelerations that were established experimentally and based on law (1) (at all points of vibration acceleration measurement).

Table 1

Results of identification of coefficients in the law of change in the vibration accelerations (1)

No.	M, gm	n	α	ω_0, rad/s	β	h	U_0, V	η (%)
1	2,000	1	8.3825	61.9572	–61.0113	1.3975	1.3952	1
2		3	5.6559	61.4434	–59.2161	1.2510	1.3965	1
3		6	5.7121	62.1635	–55.4631	1.2572	1.3948	1
4		9	5.3431	62.2685	–55.8437	1.2337	1.3948	1
5		Mean	6.2734	61.9581	–57.8836	1.2849	1.3953	1
6	2,180	Mean	44.3388	58.7406	–69.7573	1.3960	1.3944	1
7	2,360	Mean	293.5435	55.7752	–41.7880	1.8388	1.3938	1

Fig. 2 shows diagrams of the platform vibration accelerations, constructed based on formula (1) and on the results of a full-scale experiment.

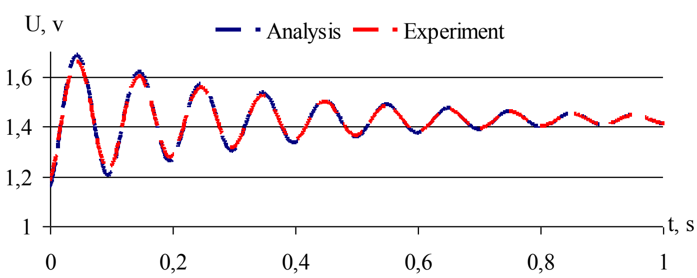


Fig. 2. Diagrams of platform vibration accelerations, corresponding to the free damping oscillations of the platform

Fig. 2 and Table 1 show that:

– the process of determining the magnitudes of parameters is stable (robust); the parameters almost do not change from the change in the number of measurements of vibration accelerations;

– the law of change in the platform vibration accelerations is described with a great accuracy by the law of free damped oscillations (1);

– the increasing interval of time almost does not increase the discrepancy between the actual vibration accelerations and those established in line with the law of damped oscillations (1);

– an increase in the mass of the platform reduces the frequency of natural platform oscillations.

The frequency of natural platform oscillations depends on the moment rigidity C of the elastic supports of the platform and on the momentum of inertia I of the platform relative to the axis around which the platform oscillates

$$\omega_{res_i} = \sqrt{C / I_i}, \quad / i = 1,2,3 /. \tag{2}$$

In its turn

$$I_2 = I_1 + \mu H^2, \quad I_3 = I_2 + \mu H^2 = I_1 + 2\mu H^2, \tag{3}$$

where μ=180 gm is the mass of a single additional load, H=0.2 m is the distance from the center of mass of the addi tional load to the rotation axis of the platform.

We derive from (2)

$$C = \omega_{res_i}^2 \cdot I_i. \tag{4}$$

Rigidity C of the elastic supports of the platform does not change. Since there is an overabundance of experimental data, we have introduced, in order to minimize, the following functional

$$\begin{aligned} F(C, I_1) = {} & [\omega_{res_1}^2 I_1 - C]^2 + \\ & + [\omega_{res_2}^2 (I_1 + \mu H^2) - C]^2 + \\ & + [\omega_{res_3}^2 (I_1 + 2\mu H^2) - C]^2. \end{aligned} \tag{5}$$

Based on the results of experiments, we determined values C=234.6789 N·m/rad and I_1=0.0610 kg·m^2 that minimize functional (5). The corresponding resonance frequencies, moments of platform inertia, as well as calculation errors, are given in Table 2.

Table 2

Natural (resonance) frequency of platform oscillations

No.	M, gm	ω_{rez}, rad/s – experiment	ω_{rez}, rad/s – calculation	Error, %	I, kg·m^2
1	2,000	61.9581	62.0063	0.078	0.0610
2	2,180	58.7406	58.6439	0.165	0.0682
3	2,360	55.7264	55.7752	0.088	0.0754

Table 2 shows that the error in determining the natural (resonance) frequencies of platform oscillations does not exceed 0. 2 %.

When adding loads, the mass of the platform grows by 18 %, and its axial inertia moment – by 24 %.

5. 3. Identification of coefficients in the dual-frequency law of change in the platform vibration accelerations

Experiments were conducted for two balls in the auto-balancer. The mass of one ball is 21 gm.

We determined the frequencies and intensity of oscillation components in the range up to 100 Hz (the mode of spectral analyzer). We observed at the spectral analyzer the presence of two components: at the frequency of 47 Hz – caused by the unbalanced mass; at the frequency of 9 Hz – caused by the balls (Fig. 3).

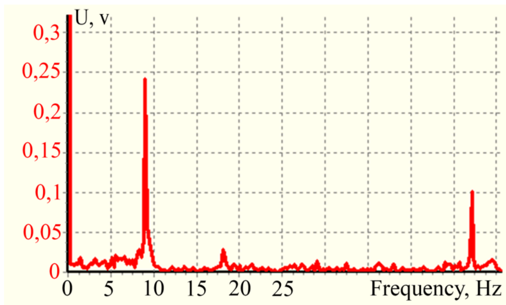


Fig. 3. Fragment of the software window under the mode of "Oscillation spectrum analysis"

The platform executes a complicated rotational-oscillatory motion caused by the sum of two circular motions – the unbalanced mass with the rotor rotation frequency and the balls with the natural oscillation frequency of the platform. This suggests that the platform vibration accelerations (formally) change in line with the law of dual-frequency oscillations [20]:

$$U = U_A \cos(\omega t + \delta) + U_B \cos(\omega_{rez} t + \gamma) + U_0, \qquad (6)$$

where U_A is the oscillation amplitude with the rotor rotation frequency; ω is the rotor rotation frequency; t is time; δ is the phase for rotor; U_B is the oscillation amplitude with a frequency of ball rotation; ωres is the ball rotation frequency; γ is the phase for balls; U_0 is a constant offset.

We derived experimentally a dependence of vibration accelerations on time. The data were transferred to the software package for statistical analysis Statistica. The software package employed the method of least squares to determine unknown coefficients U_A, ω, δ, U_B, ωres, γ, U_0, included in the law of dual-frequency oscillations (6).

The results of identification are given in Table 3. It also gives the percentage of maximum difference η between the vibration accelerations derived experimentally and the vibration accelerations established based on the dual-frequency law of oscillations (6).

Table 3

Results of identification of parameters of dual-frequency vibrations (for M=2,000 gm)

No.	n	UA, B	UB, V	ω, rad/s	ω_{res}, rad/s	δ, rad	γ, rad	U_0, V	η (%)
1	1	0.184	0.465	307.744	61.358	2.641	0.298	1.418	3
2	2	0.187	0.452	307.750	61.351	2.632	0.277	1.415	3
3	3	0.179	0.467	307.763	61.372	2.628	0.284	1.421	3
Mean value		0.183	0.461	307.752	61.360	2.634	0.286	1.418	3

Fig. 4 shows diagrams of change in the platform vibration accelerations, derived experimentally and by using the methods of statistical analysis.

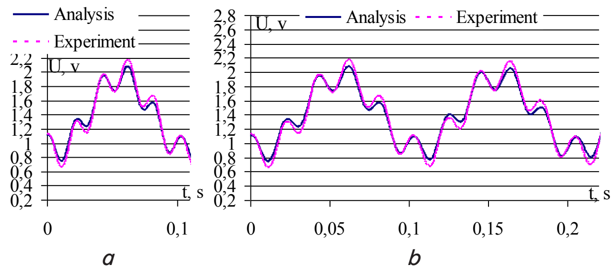


Fig. 4. Diagrams of platform vibration accelerations over the time of: *a* – a single; *b* – two slow oscillations of the platform

Diagrams (Fig. 4) and Table 3 show that:

– the process of computing the magnitudes of coefficients in the dual-frequency law of changes in the vibration accelerations (6) is stable (robust) as the magnitudes of coefficients almost do not depend on the time interval in which they are calculated;

– at both small and large time intervals (over the time of several slow oscillations of the platform), the discrepancy between the actual vibration accelerations and the vibration accelerations established based on the dual-frequency law (6) does not exceed 3 %;

– the frequency of slow platform oscillations equals 61.360 rad/s or 9.766 Hz, which is less by approximately 1 % than the natural oscillations of the platform.

Similar results were obtained for the platform with a single and the two additional loads.

Thus, despite the strong asymmetry in supports, the ball auto-balancer excites almost perfect dual-frequency platform oscillations. In this case, the balls get stuck at a frequency slightly less than the resonant one.

5. 4. Influence of the bench parameters on the characteristics of dual-frequency vibrations

The characteristics of dual-frequency vibrations are affected by such basic parameters as the mass of the unbalanced mass on the casing of the auto-balancer M_D, the total mass of balls M_{cw}, the mass of platform M, the shaft rotation frequency ω_r.

5. 4. 1. Impact of the combined mass of balls on the characteristics of vibrations

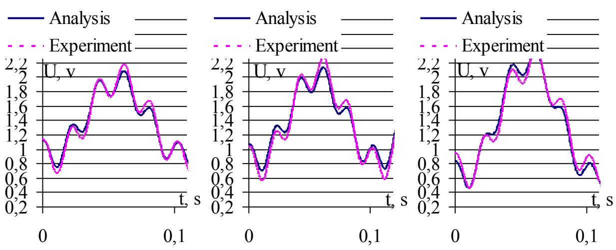
The experiment was conducted for 2, 3, and 4 balls. The mass of a single ball is 21 gm. We established under the oscilloscope mode, based on oscillograms (Fig. 5), that an in increase in the mass of balls directly proportionally increases the amplitude of slow oscillations of the vibratory machine platform.

It should be noted that the amplitude of slow oscillations is directly proportional to the balancing capacity of auto-balancer. However, even 4 balls occupy a small part of the treadmill. In this case, the balancing capacity of the auto-balancer is approximately derived from formula

$$S = nm(R - r), \quad (7)$$

where n is the number of balls, m is the mass of a single ball, R is the radius of the treadmill, r is the radius of the ball.

A similar experiment on determining the proportionality of change in the amplitude of the platform was conducted using a laser pointer (Fig. 6).



a *b* *c*

Fig. 5. Diagrams of vibration accelerations of platforms at different total mass of balls: *a* – M_{cw}=42 gm; *b* – M_{cw}=63 gm; *c* – M_{cw}=84 gm

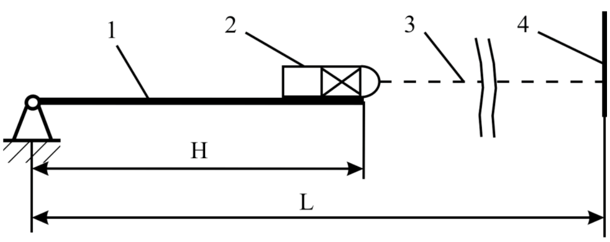
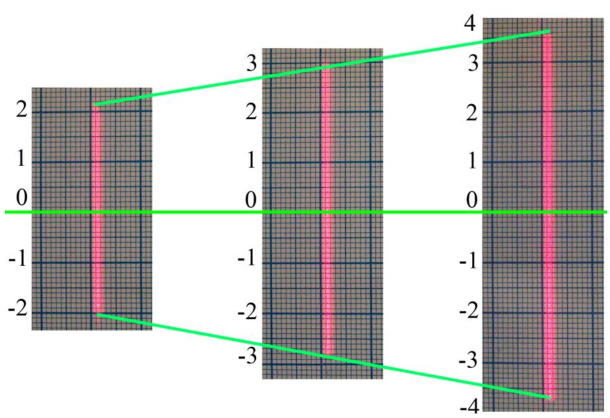


Fig. 6. Installation diagram of a laser pointer: 1 – platform, 2 – laser pointer, 3 – laser beam, 4 – graph paper

When conducting the experiment, the unbalanced mass was removed from the casing; we changed only the number of balls in the auto-balancer. During operation of the platform, the laser beam was projected onto a graph paper; we measured the length of the beam's projection *K* (Fig. 7). Positioning the graph paper at a distance *L* from the axis around which the platform oscillates makes it possible to increase the projection of a laser beam by *L*/*H* times.



a *b* *c*

Fig. 7. Testing the proportionality of change in the amplitude of platform's oscillations at a different total mass of balls: *a* – M_{cw}=42 gm; *b* – M_{cw}=63 gm; *c* – M_{cw}=84 gm

It was confirmed that the amplitude of slow oscillations of a vibratory machine platform is directly proportional to the summary mass of balls. In addition, we computed the respective swings of oscillations S at the far side of the platform (Table 4).

Table 4

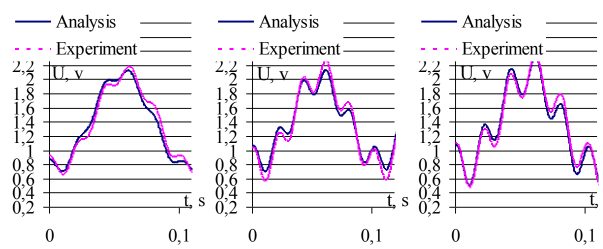
Computation of swings of slow oscillations at the far side of the platform

No.	H, mm	L, mm	K, mm	S, mm
1	200	3,200	41	2.625
2			58	3.750
3			72	4.688

5. 4. 2. Influence of mass of the unbalanced mass at the auto-balancer's casing on vibration characteristics

The unbalanced mass at the casing of the auto-balancer is rigidly fixed pendulums. The experiment was conducted for three values of mass of the unbalanced mass M_D: 10, 15, and 20 gm. The platform's mass is M=2,000 gm.

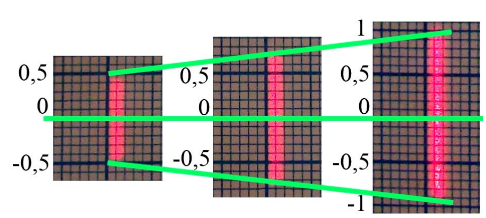
It was established under the oscilloscope mode, based on oscillograms (Fig. 8), that an increase in the mass of the unbalanced mass at the casing of the auto-balancer increases directly proportionally the amplitude of rapid oscillations of the platform.



a *b* *c*

Fig. 8. Diagrams of vibration accelerations of the platforms at a different total mass of the unbalanced mass: *a* – M_D=10 gm; *b* – M_D=15 gm; *c* – M_D=20 gm

Similar to the previous experiment, we conducted study using a laser pointer. When conducting the experiment, the balls were removed from the auto-balancer, we changed only the total mass of the unbalanced mass at the auto-balancer's casing. During vibrations of the platform, the laser beam was projected onto a graph paper; we measured the length of the beam's projection (Fig. 9).



a *b* *c*

Fig. 9. Testing the proportionality of change in the amplitude of platform at a different mass of the unbalanced mass M_D: *a* – 10 gm; *b* – 15 gm; *c* – 20 gm

It was confirmed that the amplitude of rapid oscillations of a vibratory machine platform is directly proportional to the mass of the unbalanced mass. We computed the respective swings of oscillations S at the far side of the platform (Table 5).

Table 5

Computing the swings of rapid oscillations at the far side of the platform

No.	H, mm	L, mm	K, mm	S, mm
1	200	3,200	9	0.625
2			14	0.875
3			18	1.25

5. 4. 3. Simultaneous influence of the total mass of balls and mass of the unbalanced mass on vibration characteristics

In order to obtain a more accurate quantification of the influence of mass of the balls and the mass of the unbalanced mass on dual-frequency vibrations, we conducted a two-level experiment. The result is given in Table 6.

It was established in the course of the experiment that the ratio of amplitudes of rapid and slow oscillations to respective masses almost does not change. This confirms that the values for the rapid and slow oscillation amplitudes are directly proportional to, respectively, the mass of the unbalanced mass or the total mass of the balls.

Table 6

Dependence of amplitudes of rapid and slow oscillations of platform (M=2,000 gm) on the total mass of the balls and mass of the unbalanced mass

No.	Mass		Amplitudes		Frequencies		$\frac{U_A}{M_D}$	$\frac{U_B}{M_{CW}}$
	MD, gm	M_{cw}, gm	U_A, v	U_B, v	ω_r, rad/s	ω_{res}, rad/s		
1	20	63	0.2841	0.6656	307.7412	61.6621	0.0142	0.0106
2	20	42	0.2855	0.4654	307.7501	61.6553	0.0143	0.0111
3	15	42	0.2184	0.4658	307.7498	61.7017	0.0146	0.0111
4	15	63	0.2175	0.6671	307.7511	61.6632	0.0145	0.0106

5. 4. 4. Influence of mass of the platform on vibration characteristics

The influence of mass of the platform on vibration characteristics was estimated based on the vibration acceleration diagrams, constructed under the mode of oscilloscope. It was established that:

– an increase in the mass of the platform reduces the frequency of slow oscillations of the platform;

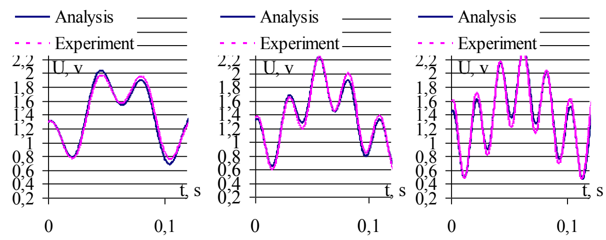
– the amplitude of accelerations due to the slow oscillations of the platform is proportional to the square of the frequency at which balls get stuck.

The same experiment implied studying the stability of dual-frequency vibrations against a change in the mass of the platform. To this end, during bench operation, we threw additional loads on the platform in the form of plasticine (total mass is up to 0.5 kg). It was established that a dual-frequency vibration mode is resistant to a change in the mass of the platform. Increasing the mass of the platform reduces the frequency at which loads get stuck. In this case, the balls automatically adjust to the changed mass of the platform. The frequency at which balls get stuck is approximately 1 % less than the resonance frequency of platform oscillations.

5. 4. 5. Influence of shaft rotation frequency on vibration characteristics

A change in the frequency of shaft rotation was brought about by installing pulleys of different diameter. The experiment was conducted for three values of shaft rotation frequency: 1,500, 2,250, 3,000 rpm.

The influence of shaft rotation frequency on vibration characteristics was estimated based on the diagrams of vibration accelerations under the oscilloscope mode (Fig. 10).



a *b* *c*

Fig. 10. Diagrams of the platform's vibration accelerations at a different shaft rotation frequency: a – ω_r=1,500 rpm; b – ω_r=2,250 rpm; c – ω_r=3,000 rpm

It was established that the amplitude of accelerations due to the rapid oscillations of the platform is proportional to the square of shaft speed.

6. Discussion of results of studying the rotational-oscillatory vibrations of a vibratory machine platform, excited by the ball auto-balancer

The research conducted has shown that the ball auto-balancer excites almost perfect dual-frequency rotational-oscillatory vibrations of a vibratory machine platform. Slow frequency corresponds to the rotational speed of the center of the balls around the longitudinal axis of the shaft, while the rapid one – to the shaft rotation speed. Balls get stuck at a rotation speed that is lower by about 1 % than the resonance frequency of platform oscillations.

The procedure for determining the law of change in a vibration acceleration using the method of statistical analysis has proven effective. The process of computing the magnitudes of coefficients for the dual-frequency law of change in the vibration accelerations is stable (robust) since the magnitudes of coefficients almost do not depend on the time interval in which they are computed. The discrepancy between laws obtained experimentally and by employing methods of statistical analysis is less than 3 %.

Rotational-oscillatory vibrations of the platform could be also effectively studied based on the motion of a laser beam.

It is possible to change, in wide ranges, characteristics of vibrations by altering the mass of the balls, mass of the unbalanced mass, mass of the platform and shaft speed. In this case, the amplitude of slow oscillations of the platform is directly proportional to the total mass of balls, while that of the rapid ones – to the mass of the unbalanced mass.

Increasing the mass of the platform increases its axial moment of inertia and reduces the resonance frequency of oscillations.

The amplitude of accelerations due to the slow oscillations of the platform is proportional to the square of the frequency at which balls get stuck. The amplitude of accelerations due to the platform's rapid oscillations is proportional to the square of shaft speed.

It should be noted that the results were obtained for a particular vibratory machine bench. Therefore, the results cannot be applied in full to other vibratory machines with a vibration exciter in the form of a passive auto-balancer. However, the approach taken in this paper could be applied, without fundamental changes, to other vibratory machines, including those with a different kinematics of platform motion.

In the future we plan to construct and analyze a mechanical-mathematical model of a vibratory machine with a rotational-oscillatory motion of the platform and a vibration exciter in the form of a passive auto-balancer.

7. Conclusions

1. An effective method for studying the platform's vibrations is the processing of digitized vibration accelerations of separate points of the platform using the methods of statistical analysis.

Resonance frequencies of platform oscillations: in the absence of additional masses, 62.006 rad/s or 9.869 Hz; with an additional mass of 180 gm, 58.644 rad/s or 9.333 Hz; with an additional mass of 360 gm, 55.775 rad/s or 8.877 Hz. An error in determining the resonance frequencies does not exceed 0.2 %. Increasing the mass of the platform reduces the resonance frequency of platform oscillations.

2. Assuming the platform executes the dual-frequency oscillations, we employed the software package for statistical analysis Statistica to select coefficients for the law of change in the platform vibration accelerations. It was found that:

– the process of determining the magnitudes of coefficients is steady (robust); coefficients almost do not change from altering the time interval;

– at both small and large time intervals (over a period of several slow oscillations of the platform), the discrepancy between the laws of change in the platform vibration accelerations that were established experimentally and by employing a method of statistical analysis does not exceed 3 %.

Thus, despite the strong asymmetry of supports, the ball auto-balancer excites almost perfect dual-frequency vibrations of the platform.

3. Auto-balancer operates as two separate vibration exciters. In the first one, the balls almost uniformly revolve around the longitudinal axis of a rotor at a frequency close to the resonance frequency of platform oscillations. In so doing, the balls excite slow resonant oscillations of the platform at a larger amplitude. In the second one, a mass at the auto-balancer's casing excites rapid oscillations of the platform at a current above-resonance frequency of the rotor rotation (48.98 Hz).

A change in the mass of the platform alters the resonance frequency. However, the balls automatically adjust to fit the current resonance frequency. In this case, the balls' rotation frequency (at which they get stuck) is slightly lower (up to 1 %) than the resonance frequency of platform oscillations.

4. Basic parameters of the system affect vibration characteristics in the following way:

– the amplitude of slow oscillations is directly proportional to the total mass of balls;

– the amplitude of rapid oscillations is directly proportional to the mass of the unbalanced mass at the auto-balancer's casing;

– the amplitude of accelerations due to the slow oscillations of the platform is proportional to the square of the frequency at which balls get stuck;

– the amplitude of accelerations due to the rapid oscillations of the platform is proportional to the square of the shaft rotation frequency;

– increasing the mass of the platform decreases the frequency of slow oscillations of the platform.

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