На створеному стенді грохота досліджені усталені вібрації короба, збуджені кульовим автобалансиром. Оцифрован закон зміни вібропришвидшень короба грохоту на усталеному русі. За допомогою програмного пакета для статистичного аналізу Statistica вібрації ідентифіковані як двочастотні. Розбіжність між законами зміни вібропришвидшень короба грохоту, знайденими експериментально і методами статистичного аналізу, не перевищує 1-го відсотка

Ключові слова: інерційний віброзбудник, двочастотні вібрації, резонансна вібромашина, автобалансир, інерційний грохот

На созданном стенде грохота исследованы установившиеся вибрации короба, возбужденные шаровым автобалансиром. Оцифрован закон изменения виброускорений короба на установившемся движении. С помощью программного пакета для статистического анализа Statistica вибрации идентифицированы как двухчастотные. Расхождение между законами изменения виброускорений короба, найденными экспериментально и методами статистического анализа, не превышает 1-го процента

Ключевые слова: инерционный вибровозбудитель, двухчастотные вибрации, резонансная вибромашина, автобалансир, инерционный грохот

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EXPERIMENTAL RESEARCH OF RECTILINEAR TRANSLATIONAL VIBRATIONS OF A SCREEN BOX EXCITED BY A BALL BALANCER

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1. Introduction

Among vibrating machines such as screens, vibrating sieves, separators, the machines with two-frequency vibration exciters are promising [1–9]. In such machines at the box (sieve, etc.) vibrations with the lower frequency, the main technical process such as separation, screening, etc. is carried out. The vibrations with the higher frequency provide self-cleaning of the box [2] and changes in mechanical properties of the processed material to increase the intensity of the main technological process [3].

However, the resonant vibrating machines are the most power-effective [10–12]. Therefore, it is actual to create the machines that combine the advantages of the two-frequency and resonant machines.

For exciting two-frequency resonant vibrations, it is offered to use passive auto-balancers of ball, roller or pendular type [14–18]. To date, the designs of new vibrating machines were proposed and their performance was investigated by 3D modeling. However, there is no experimental confirmation of the operability of the new method of the two-frequency vibrations exciting.

2. Literature review and problem statement

On the one hand, the research conducted in [4–8] proves the advantages of machines with two-frequency vibration exciters. Thus, two-frequency vibrations provide more efficient separation of the size of mineral raw materials [4], dehydration of coal [5] or fractionation of sand [6]. The efficiency of two-frequency vibrations was confirmed for single-mass [7] and two-mass [8] inertial vibrating machines. The general prospects for the development of machines with a polyharmonic law of platform vibrations are presented in [9], where the technical relevance of the task being solved is proved.

On the other hand, resonant machines are more energy-efficient, since their smaller-mass vibration exciters excite platform vibrations with a larger amplitude [10]. The resonant modes of the platform vibrations are widely used both for vibrating mills [11], and screens [12].

From the foregoing it follows that the combination of the two-frequency and resonant machines will improve the technological processes of processing raw materials and reduce energy costs [13].

In [14], it is proposed to excite the two-frequency resonance vibrations of the screen box with a vibration exciter in the form of a passive auto-balancer. For this purpose, the specific mode of motion of the corrective weights in the auto-balancer is used. It occurs at small forces of resistance to the motions of the corrective weights relative to the auto-balancer body. In this mode, the corrective weights are assembled together, cannot catch up with the shaft on which the auto-balancer is mounted and get stuck at the resonance frequency of the box vibrations. This excites the slow resonant vibrations of the box. The mass is mounted on the auto-balancer body (unbalanced mass). The body and unbalanced mass rotate synchronously with the shaft. This excites the fast vibrations of the box. The parameters of the two-frequency vibrations vary within wide limits by changing the rotor speed, the unbalanced mass on the auto-balancer body, the total mass of the correcting weights. It is assumed that the vibration exciter in the form of a passive auto-balancer is applicable for the screens with different kinematics of the box motion.

In [15–17], a new method of excitation of vibrations was investigated in the CAD SolidWorks using the Cosmos Motion module. The workability of the method was proved for the screens whose box motion is rectilinear translational [15], swinging [16] and plane translational [17]. The possibility of changing the characteristics of the two-frequency vibrations in wide ranges is confirmed.

In [18], the criterion of replacement of the single-frequency vibration exciter by two-frequency one is formulated. According to the criterion, a modernized machine must perform the main technological process in the same way as the basic machine. To do this, the slow box vibrations of the modernized machine must correspond to the box vibrations of the basic machine, both in amplitude and frequency. The mechanical-mathematical models of the basic and modernized machines were created. The parameters of the modernized machine, which provide this, were determined. The modernization of the GIL 42 screen made it possible to reduce the total mass of the rotating parts of the vibration exciter 3.4 times.

From the foregoing it follows that to further justify the operability of the new-type vibrating machines, it is important to confirm experimentally the results obtained by 3D modeling. It is actual for a particular screen to determine the law of vibrations of its box and to reveal the influence of the screen parameters on the vibration characteristics.

Similar problems were solved in [19–22] by indirect and direct methods.

In [19], indirect methods were applied. The occurrence of the biharmonic vibrations mode was determined by increasing the screen productivity.

In direct methods, individual vibration parameters are measured or the law of the box motion as a whole is investigated [20–22].

In [20], the registration of vibration accelerations at the measurement points was carried out by means of the universal registrar and analyzer of vibrations «VibroDon-3» made in Ukraine. The device is designed for the operational recording and analysis of the vibration field signals in the laboratory and industrial conditions. The research findings revealed the presence of the screen biharmonic operating mode.

In [21], the complex effect of the rotational speed of the vibration exciter shaft, the angle of direction and the vibration amplitude on the two-frequency vibration parameters was investigated. During the experiments, one of the parameters varied while others were fixed, and then the experiments were repeated with the other values of the fixed parameters.

In [22], the influence of the oscillation frequency and amplitude variation on the two-frequency vibration characteristics was investigated.

In [20–22], the type of the excited vibrations and their characteristics were investigated. Below, the emphasis is placed on the box vibration law and the influence of the vibration machine main parameters on this law.

3. The purpose and problems of the research

The purpose of the work is to research the steady vibrations of the screen box, excited by the ball balancer.

To achieve this purpose, it is necessary to solve the following problems:

– to create the screen stand;

– to define experimentally the screen stand characteristics, to test the methods for research of the screen box vibrations and to define the type of the box motion;

– to pick up the coefficients in the expected law of variation of the screen box vibration accelerations with the use of the statistical analysis methods;

– to compare the law of variation of the box vibration accelerations, found experimentally with the two-frequency law, identified by the statistical analysis methods.

4. Methods for the research of the screen box vibrations excited by the ball balancer

When developing the screen stand, the basic information from the vibrating machines theory [23] is used.

The box motion is studied by the motion of its frame. The frame is considered as a rigid body. The motion is studied by vibration accelerations. This is because the vibration displacements (vibration velocities) of the slow box vibrations are much greater than the vibration displacements (vibration velocities) of the rapid vibrations. Only vibration accelerations are comparable.

Vibration accelerations are measured by two identical sensors-accelerometers MMA6231Q 2AX 1,5. The sensor measuring range is 1.5 g. To digitize the signal, the analog-to-digital conversion board ADXL202EB-232A with the USB interface of the Motorola company is used. Sensors are connected to the board, and the board is connected to a personal computer.

The board works in 3 modes:

1. In the «Oscillograph» mode, two graphs of several oscillations of the signal captured by the board are displayed on the computer screen. The signals can be compared in magnitude and phase. One can determine their amplitudes and frequencies.

2. In the «Spectral analyzer» mode, the graph of the oscillation spectrum is displayed on the computer screen.

3. In the «Recorder» mode, the signals taken from two sensors are captured and digitized.

The translational linear motion of the box is checked in two ways:

In the first way, the laser level is used. It is consistently mounted on the short and long sides of the screen box. The absence of angular box vibrations is verified by the motion of the horizontal line formed by the laser ray. For this, the line is projected onto a special screen with horizontal lines.

In the second way, two identical vibration acceleration sensors are used. They are installed on the opposite long (short) sides of the box, facing each other. The board is used in the «Oscillograph» mode. Vibration acceleration graphs taken from two sensors are compared. At the translational motion of the box, the sensor readings are almost identical. At purely angular vibrations of the box, the readings of the sensors are in antiphase.

The type of the excited box vibrations is proposed to define with the use of the vibration theory elements [24].

The frequency spectrum of the excited vibrations is determined in the «Spectral analyzer» mode.

The law of vibration acceleration variation, digitized and recorded in the «Recorder» mode is interpreted as a signal. The obtained data is stored in the form of a table.

Based on the experiments, it is assumed that the vibrations of the box are two-frequency. The coefficients in the law of the vibration accelerations variation at two-frequency vibrations are identified with the use of the least-squares method implemented in the software package for statistical analysis Statistica [25]. The laws of the vibration accelerations variation, obtained experimentally and by statistical analysis methods are compared. Additional details of the experiments are described in the presentation of the experiments results.

5. The results of the research of translational rectilinear vibrations of the screen box, excited by a ball balancer

5. 1. Screen stand description

To research the vibrations excited by the ball balancer, the stand was created (Fig. 1). Depending on the support design, the box can perform such motions: translational rectilinear, swing vibrational, plane-parallel, translational in the vertical plane, translational in the horizontal plane.

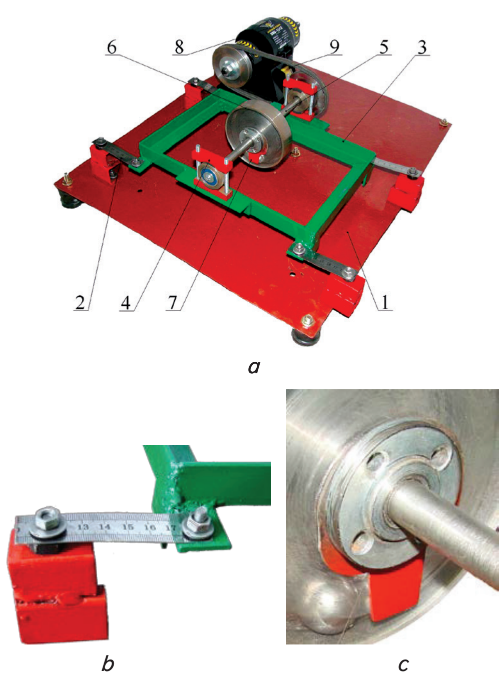

Fig. 1. The screen stand:
a — general view (1 — bedplate; 2 — elastic box support; 3 — box; 4 — shaft support; 5 — shaft; 6 — auto-balancer; 7 — unbalanced mass; 8 — asynchronous motor; 9 — belt gear); *b* — elastic support; *c* — unbalanced mass and balls

In the experiments that have been carried out, the support structures that allow the box to perform vertical rectilinear translational motion have been chosen (Fig. 1, *b*).

The stand allows:

– to change the number of the balls in the auto-balancer;

– to change the box mass by means of additional weights;

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

– to change the shaft rotation frequency, the rigidity of the elastic supports;

– to measure the box vibration accelerations;

– to observe the relative motions of the balls;

– to observe the box motion by means of the laser ray.

Because the box theoretically has several resonance frequencies and several corresponding forms of vibrations, it is necessary to define what form of vibrations the box makes.

5. 2. Checking of the screen box vibration straightness

Two ways of checking the screen box vibration straightness are offered.

The first way. Consistently the laser level is mounted on the short and long sides of the box (Fig. 2, *a*). It is studied how the laser ray in the form of a straight line moves after the box motion became steady. For this purpose, the ray is projected on the vertical scale consisting of parallel lines.

At first, the experiment has been made with balls, but in the absence of unbalanced mass. It is defined that irrespective of the installation site of the laser level, the line of the laser ray is parallel to the lines of the scale (Fig. 2, *b*). This demonstrates that the balls get stuck at the lowest resonance frequency, thus exciting the first form of the box vibrations.

Then, the experiment has been made in the absence of balls, but with the unbalanced mass (Fig. 2, *c*). The similar findings are obtained. However, this testifies to the symmetry of the supports and is caused by the arrangement of the vibration exciter in the box center.

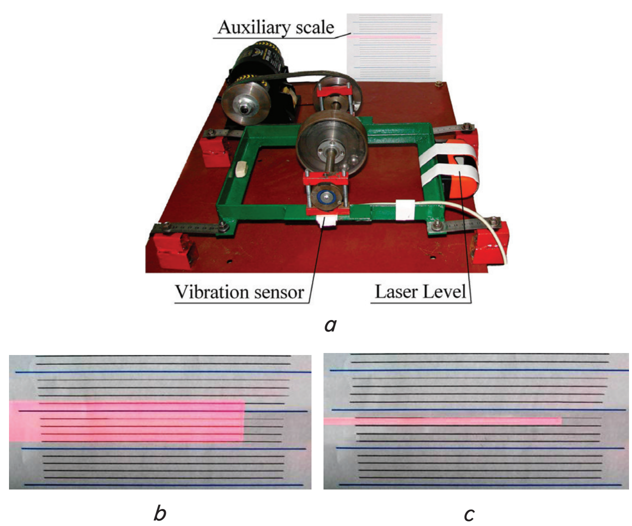

Fig. 2. Checking of the screen box vibration straightness: *a* — installation of the laser level on the box; *b* — laser ray motion in the presence of balls only; *c* — laser beam motion in the presence of unbalanced mass only

The second way. Two identical vibration acceleration sensors are installed on the long box sides, opposite to each other. Sensors accelerometers measure vibration accelerations on the box in the vertical direction. The instantaneous values of vibration accelerations in tension (volts) are defined. The sensor indications practically matched (Fig. 3).

As a result of checking the screen box vibration straightness in two ways, it is defined that the box makes almost pure rectilinear progressive vibrations, there are no angular components.

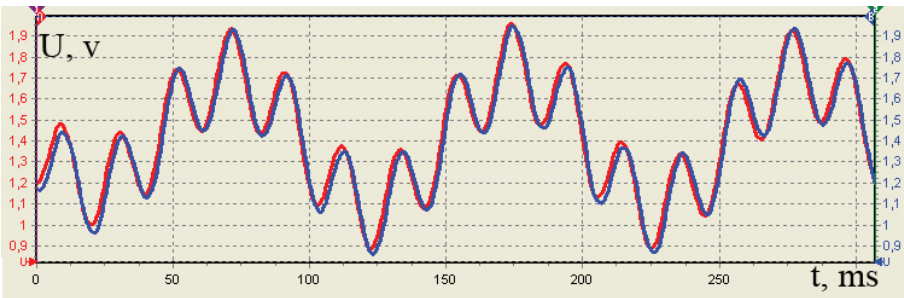

Fig. 3. The program window fragment in the «Oscillograph» mode

Frequencies and intensity of the vibration components in the range up to 100 Hz (the «Spectral analyzer» mode) were defined. On the spectral analyzer screen, there are two components: at a frequency of 48 Hz – from the unbalance mass; at a frequency of 9 Hz – from the balls (Fig. 4).

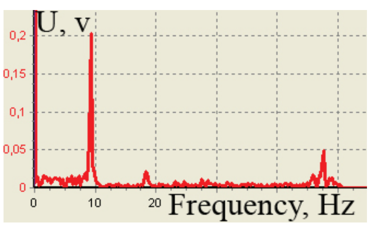

Fig. 4. The program window fragment in the «Spectral analyzer» mode

The characteristics of two-frequency vibrations are influenced by such main parameters: the mass on the auto-balancer body M_D, the total mass of the balls M_{cw}, the box mass M, the shaft speed ω_r.

In all experiments, M=2.000 g, ω_r=2.880 RPM.

5. 3. The screen stand characteristics and research technique testing

The elastic supports allow the box to make three principal oscillating motions:

– rectilinear translational motion in the vertical direction;

– angular oscillatory motions about the box central axis which is parallel to its short side;

– angular oscillatory motions about the box central axis which is parallel to its long side.

The principal oscillating motions, respectively, occur if:

– to raise a little the auto-balancer body and then to release it abruptly;

– to raise a little one short side of the box, to lower a little another one and then to release both abruptly;

– to raise a little one long side of the box, to lower a little another one and then to release both abruptly.

Box vibrations are investigated by vibration accelerations.

In the absence of the shaft rotation and fixed balls in the auto-balancer, box vertical vibrations have been investigated. In accordance with the linear theory of oscillations, the law of the vibration acceleration variation must correspond to the law of free damping oscillations [24]:

$$U_X = \alpha \cdot e^{-ht} \cdot \cos(\omega_{res} t + \beta) + U_0, \qquad (1)$$

where ω_{res} is the frequency of the box own vibrations; α, β are the parameters depending on the initial conditions; U_0 is the constant shift.

As a result of the experiments, the time dependence of the box vibration accelerations (is not provided because of a large size) has been obtained. In this dependence, the time varies with the equal step. Let us notice that can be any both the time variation step, and the interval of vibration acceleration measurements, can be arbitrary.

Then, the obtained data have been transferred to the software package for statistical analysis Statistica. The package, with the use of the least-squares method, defines the unknown coefficients α, h, ω_{res}, β, U_H included in the law of damped oscillations (1).

The parameters were identified at various time intervals corresponding to 1, 3, 6, and 9 slow oscillations of the screen box. During one slow oscillation of the box, its vibration accelerations were measured 115 times. The results of the identification are shown in Table 1. In the table, the maximum discrepancy η between the vibration accelerations found experimentally and under the law (1) is calculated in % (for all measuring points of the vibration accelerations).

Table 1

The results of identifying the coefficients in the law of the vibration acceleration variation (1)

№	n(n·115)	α	ω_{res}	β	h	U_0	η (%)
1	1 (115)	0.126	60.246	927.109	2.867	1.426	1
2	3 (345)	0.125	60.286	927.342	2.878	1.421	1
3	6 (690)	0.126	60.240	927.548	2.891	1.423	1
4	9 (1,035)	0.126	60.259	927.261	2.854	1.427	1
Average		0.126	60.258	927.315	2.873	1.424	1

The graphs of the box vibration accelerations constructed according to the formula (1) and the results of the natural experiment are presented in Fig. 5.

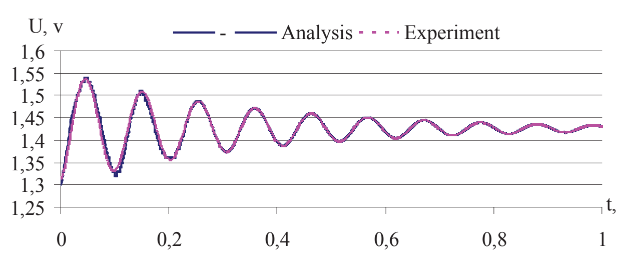

Fig. 5. The graphs of the box vibration accelerations, corresponding to free damped oscillations of the box

From Fig. 5 and Table 1 it can be seen that:

– the process of determining the values of the parameters is steady (robust), the parameters practically do not vary with the number of vibration acceleration measurements;

– the law of vibration acceleration variation corresponding to the first free vibration mode of the box is highly accurately described by the law of free damped oscillations (1);

– with the time period increasing, the discrepancy between the actual vibration accelerations and vibration accelerations found under the law of damped oscillations (1) practically does not increase;

– the first resonance frequency of the box vibrations is equal to 60.258 rad/s or 9.590 Hz.

5. 4. Identification of the coefficients in the two-frequency law of the box vibration acceleration variation

The box makes a complex translational rectilinear motion caused by the sum of the motion of the unbalanced mass with the rotor speed and the circular motion of the balls with the box vibration natural frequency. This allows assuming that vibration accelerations of the box vary under the law of two-frequency vibrations [24]:

$$U_X = U_A \cdot \cos(\omega_r t + \delta) + U_B \cdot \cos(\omega_{rez} t + \gamma) + U_0, \qquad (2)$$

where U_A is the amplitude of vibrations at the rotor speed; ω_r is the rotor speed; t is the time; δ is a phase for the rotor; U_B is the amplitude of vibrations at the balls rotating speed; ω_{res} is the balls rotating speed; γ is a phase for the balls; U_0 is a constant shift.

In the experiment, the time dependence of the box vibration accelerations (here is not provided) has been obtained. Then, the obtained data were transferred to the statistical software package Statistica. The software package, with the use of the least-squares method, has determined the unknown coefficients U_A, ω_r, δ, U_B, ω_{res}, γ, and U_H included in the law of two-frequency vibrations of the box (2).

The results of the identification are listed in Table 2. In this table, the maximum discrepancy η between the actual vibration accelerations (found experimentally) and the vibration accelerations found under the two-frequency law of vibrations (2) throughout the time variation interval is estimated in %. The amount of slow vibrations (measurements) is n (115n).

Table 2

The results of the identification of the two-frequency vibration parameters

№	n	U_A	ω_r	δ	U_B	ω_{res}	g	U_H	η (%)
1	1 (115)	0.0631	306.543	3.385	0.182	59.183	1.197	1.416	1
2	2 (230)	0.0642	306.679	3.334	0.188	59.752	1.199	1.418	1
3	3 (345)	0.0624	306.263	3.320	0.188	59.755	1.194	1.417	1
Average		0.0632	306.495	3.346	0.186	59.563	1.197	1.417	1

In Fig. 6, the diagrams of the box vibration accelerations, obtained experimentally and with the statistical analysis methods are constructed. The discrepancy between the two laws does not depend on the time interval size and does not exceed 1 %.

From the diagrams (Fig. 6) and Table 2 it can be seen that:

– the process of calculation of the coefficients values in the two-frequency law of vibration acceleration variations (2) is steady (robust) as the coefficients values practically do not depend on the time interval, on which they are calculated;

– on both small and large time intervals (during several slow box oscillations), the discrepancy between the actual vibration accelerations and the vibration accelerations found under the two-frequency law (2) does not exceed 1 %;

– the box slow vibration frequency is equal to 59.563 rad/s or 9.480 Hz, which practically matches (with an accuracy of 1.15 %) with the first resonance frequency of the box vibrations.

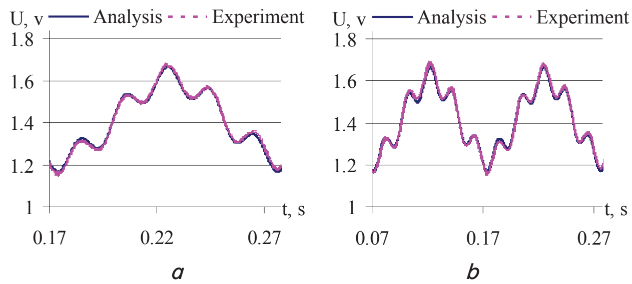

Fig. 6. Diagrams of the box vibration accelerations during the time of: *a* — one; *b* — two slow box oscillations

Thus, it can be stated with a high degree of accuracy that the ball balancer excites actually ideal two-frequency vibrations of the box.

5. 5. Influence of the auto-balancer parameters on the two-frequency vibration characteristics

The influence of the total mass of the balls on the vibration characteristics. The experiment was carried out for 2, 3 and 4 balls. The mass of one ball is 21 g. In the oscillography mode using the oscillograms (Fig. 7), it is defined that an increase in the balls mass directly proportionally increases the slow vibration amplitude of the screen box.

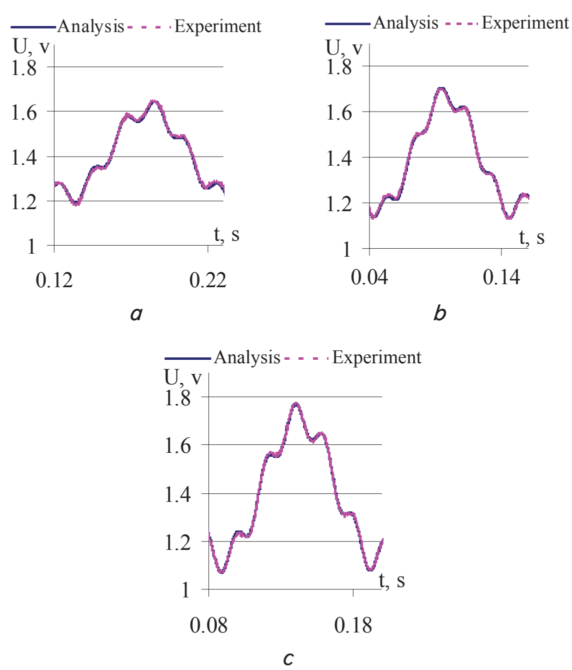

Fig. 7. Diagrams of the box vibration accelerations at different total masses of the balls: *a* — M_{cw}=42 g; *b* — M_{cw}=63 g; *c* — M_{cw}=84 g

It should be noted that the slow vibration amplitude is directly proportional to the auto-balancer balancing capacity. However, even 4 balls fill a small part of the racetrack. At the same time, the auto-balancer balancing capacity approximately is determined by the formula

$$S=nm(R-r), \quad (3)$$

where n is the number of balls, m is the mass of one ball, R is the racetrack radius, r is the ball radius.

Influence of the unbalanced mass on the vibration characteristics. As the unbalanced mass, rigidly fixed pendulums are used. The experiment was made for three values of the unbalanced mass: 10, 15 and 20 g.

In the oscillography mode according to the oscillograms (similar to Fig. 7 and not provided in the paper), it is defined that the increase in the unbalanced mass directly proportionally increases the box fast vibration amplitude.

For a more accurate quantitative evaluation of the influence of the mass of the balls and unbalanced mass on two-frequency vibrations, a two-level experiment has been made. Its findings are given in Table 3.

Table 3

Dependence of the fast and slow vibration amplitudes on the unbalanced mass and the total mass of the balls

№	Mass, g		Amplitudes, v		Frequencies, rad/s		$\frac{A}{M_D}$	$\frac{B}{M_{cw}}$
	M_D	M_{cw}	A	B	ω_r	ω_{res}		
1	20	63	0.0661	0.2518	307.4657	60.3999	0.0033	0.0041
2	20	42	0.0624	0.1878	306.2629	60.7545	0.0031	0.0044
3	15	42	0.0422	0.1898	306.4301	61.1148	0.0028	0.0045
4	15	63	0.0436	0.2504	306.0757	61.8666	0.0029	0.0041

As a result of the experiment, it is defined that the ratio of the amplitudes of the fast and slow vibrations to the corresponding masses practically does not change. This confirms that the amplitudes values of the fast and slow vibrations are directly proportional to, respectively, the unbalanced mass or the total mass of the balls.

6. Discussion of the results of the research of translational rectilinear vibrations of the screen box excited by a ball balancer

Elastic supports allow the box to make three principal oscillating motions corresponding to three resonant speeds of the rotor. Of all the possible types of the box kinematic motions, only those vibrations are excited which correspond to the lowest resonance frequency of the box vibrations.

As a result of investigations, it was defined that the box performs almost pure translational rectilinear vibrations, there are no angular components. In this regard, there is no need to impose additional kinematic constraint on the box motions.

The research has allowed defining that the ball balancer excites almost ideal two-frequency vibrations of the screen box. The slow frequency corresponds to the rotational speed of the balls centers around the longitudinal shaft axis, and the fast frequency corresponds to the shaft rotational speed.

The methodology of definition of the vibration acceleration variation law with the use of the statistical analysis method is effective. The process of the coefficients values calculation in the two-frequency law of vibration accelerations variation is steady (robust) as the coefficients values practically do not depend on the time interval, on which they are calculated. The discrepancy between the laws obtained experimentally and by the statistical analysis methods is less than 1 %.

The findings of 3D modeling are confirmed [15–17]. With the broad ranges, it is possible to change the vibration characteristics by changing the mass of the balls and unbalanced mass. At the same time, the slow vibration amplitude is directly proportional to the total mass of the balls, and the fast – the unbalanced mass.

It should be noted that the findings have been obtained for the specific screen stand. Therefore, the findings cannot be fully applied to other screens with vibration exciters in the form of passive auto-balancers. However, the approach used in this paper can be applied without basic changes to other screens, including with a different box motion kinematics.

Further, the creation and analysis of a mechanical-mathematical model of a screen with two-frequency vibration exciter in the form of passive auto-balancers are planned.

7. Conclusions

1. An effective method for obtaining the laws of variation of the vibration accelerations of the screen box points in a numerical form is carrying out full-scale experiments. The experiments have been carried out on the created screen stand and can be repeated on other screens, in which a ball balancer excites two-frequency vibrations.

2. The box of the created screen stand has three resonant frequencies and three corresponding forms of vibrations. At the same time, the corrective weights get stuck at the lowest resonance frequency, thus exciting the first form of the box vibrations. As a result, the box performs pure translational rectilinear vertical vibrations and there are no angular components.

The auto-balancer works as two separate vibration exciters. In the first one, the balls almost uniformly rotate with the resonance frequency of the box vibrations. In this case, the balls automatically adjust to this frequency, regardless of the box loading. Thus, the balls excite the slow resonant vibrations of the box (9.5 Hz) with a large amplitude. In the second vibration exciter, the mass on the auto-balancer body excites the fast box vibrations with the current above resonance frequency of the rotor (48.8 Hz).

3. Under the assumption that the box performs two-frequency vibrations, in the software package for statistical analysis Statistica, the coefficients in the law of variation of the vibration acceleration of the box have been picked up. At the same time, it has been defined that:

– the process of determining the coefficients values is stable (robust), the coefficients practically do not vary with the time interval;

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

– the fast vibration amplitude is directly proportional to the mass on the auto-balancer body.

4. At both large and small time intervals (during the time of several slow box oscillations), the discrepancy between the laws of variation of the box vibration acceleration, which have been found experimentally and by statistical analysis method, does not exceed 1 %.

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