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RESEARCH BY A 3D MODELLING OF THE SCREEN BOX FLAT TRANSLATORY VIBRATIONS EXCITED BY A BALL AUTO-BALANCER

Досліджено вібрації короба грохоту із плоским поступальним рухом короба, збуджені кульовим автобалансиром. 3D моделюванням динаміки грохоту отримано закон усталених вібрацій короба у числовому виді. За допомогою програмного пакета для статистичного аналізу Statistica вібрації ідентифіковано як двочастотні. Розбіжність між результатами 3D моделювання і знайденим законом двочастотних коливань не перевищує 1-го відсотка

Ключові слова: віброзбудник, двочастотні вібрації, 3D моделювання, дебаланс, резонансна вібромашина, автобалансир, грохот

Исследованы вибрации короба грохота с плоским поступательным движением короба, возбуждаемые шаровым автобалансиром. 3D моделированием динамики грохота получен закон установившихся вибраций короба в числовом виде. С помощью программного пакета для статистического анализа Statistica вибрации идентифицированы как двухчастотные. Расхождение между результатами 3D моделирования и найденным законом двухчастотных колебаний не превышает 1-го процента

Ключевые слова: вибровозбудитель, двухчастотные вибрации, 3D моделирование, дебаланс, резонансная вибромашина, автобалансир, грохот

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1. Introduction

Among vibration machines, the most potentially productive are mechanisms that combine in themselves advantages of resonant and dual-frequency vibration machines [1–4]. The resonant operating mode provides vibrations of big sizes of the box and weight at the minimum expenditure of energy and the minimum loads on the drive parts [1, 2]. An operating mode of dual frequency or more ensures increased productivity and performance of additional technology processes [3, 4].

It is suggested to excite dual-frequency vibrations of the screen box with various kinematics of motion by a passive auto-balancer [5]. In different fields of production, screens with a flat translational motion of the box are widely applied [2]. In this regard, it is essential to check the possibility of exciting dual-frequency vibrations by a passive auto-balancer at such kinematics of the box motion.

2. Literature review and problem statement

Among vibration machines, the most power-effective are resonant [1]. The resonant mode of vibrations provides a possibility of using a small drive to set in motion large boxes of screens at a minimum expenditure of energy [2].

A further increase of energy efficiency and productivity of resonant vibration machines is provided with using in them two [3] and more [4] frequency vibroexciters.

In study [5], it is suggested to excite resonant dual-frequency vibrations of the screen box by a method of using a passive auto-balancer. According to the method, the auto-balancer is mounted on a rigid shaft, and the shaft is installed in a rigid support on the screen box. When the shaft is rotated at superresonance velocities, there appear slow and fast harmonic oscillations of the box. Meanwhile, the slow vibrations are vibrations of the box at its own resonant frequency. At this frequency, corrective weights get stuck in the auto-balancer and, therefore, excite slow resonant vibrations. The rapid vibrations of the box take place at the frequency of the rotor rotation. They are excited by the unbalance on the auto-balancer body. The parameters of dual-frequency vibrations change over a wide range by changing the velocity of the rotor rotation, the unbalanced mass on the body of the auto-balancer, and the total mass of corrective weights. It is supposed that the method is applicable to screens with different kinematics of the box motion.

In work [6], proofs are provided on the operability of the method of a translational rectilinear motion of the screen box, and in work [7] – of a vibration rotation. The possibility of changing the characteristics of dual-frequency vibrations is confirmed for broad ranges. In the conducted studies, the dynamics of the screens are modeled by using the CAD system SolidWorks and its module Cosmos Motion.

Among vibration machines, screens with a flat translational motion of the box have become broadly applied [3]. Therefore, it is important to check the operability of the new method of exciting vibrations at such kinematics of the box motion.

It should be noted that the effects of corrective weights being jammed in auto-balancers at one of the resonant velocities of the rigid rotor rotation with isotropic support have been investigated in many studies [8–14]. In work [8], a generalized model of vibration mechanics is offered, and the equations of its motion are received. It is shown that the Sommerfeld effect, which is the phenomenon of stability of the upper position of a pendulum on a vibrating basis, the effect of the pendulum "jamming" at resonance frequencies of the system, and others are special cases of motion within the generalized dynamic model. It allows extending some results received for rotors with auto-balancers to screens with vibroexciters in the form of auto-balancers. Let us consider works [9–14] in more detail.

In [9], a study is undertaken to explore static balancing of a flat model rotor on an isotropic support with a double-pendulum auto-balancer. Equations of the motion of the auto-balanced rotor have been worked out. Their numerical integration has revealed the effect of the pendulums' jamming at the (only) resonant velocity of the rotor rotation.

In [10], two different correction planes are considered for a dynamic balancing of a rotor by two double-pendulum auto-balancers within a model of a long rigid rotor on two isotropic supports. The study reveals jamming of pendulums at one of two resonant velocities of the rotor rotation.

In work [11], the phenomenon of the pendulum jamming is studied experimentally. The experimental apparatus is a motor mounted on a metal plate that is mounted on a fixed base by means of four springs. The pendulum is mounted on an electric motor shaft with a possibility of free rotation. It is established that at certain ratios between the system parameters the electric motor rotor rotates at a working angular velocity, and the angular velocity of the pendulum rotation matches one of the fundamental frequencies of the plate vibrations.

Thus, in works [9–11], attention is mainly given to the conditions of an existing effect of pendulums' jamming at resonant velocities of the rotor rotation. The characteristics of vibrations arising meanwhile are not studied.

In work [12], a rotor on two axisymmetric elastic supports is considered being counterbalanced in one correction plane with a ball auto-balancer. The study develops differential equations on the rotor system motion. The numerical integration of the differential equations on the rotor motion has revealed an effect of the balls' jamming at the smallest resonant velocity of the rotor rotation. In work [13], this effect is investigated analytically. An approximate solution of the differential equations on the motion of a rotor system with an auto-balancer has been received. The first approximation (by the small parameter) already establishes that vibrations of the longitudinal axis of the rotor are of dual frequency. A smaller frequency is equal to the smallest resonance frequency of the rotor rotation, and a higher frequency corresponds to the frequency of the rotor rotation. Moreover, vibrations at a smaller frequency generate balls' jams, and at a higher frequency – rotor unbalance.

In work [14], it is considered how a flat model of the rotor on an isotropic support is counterbalanced with a two-ball auto-balancer. Numerical methods are used to determine the boundaries of the system parameters inside of which there occurs the effect of pendulums' jamming.

The results of studies [9–14] prove that passive auto-balancers (pendular, ball, or roller) in principle can excite vibrations that are close to a dual frequency of a screen box making a flat translational motion. However, possible dual-frequency vibrations of the box are not investigated, and their proximity to dual-frequency vibrations is not estimated either.

Therefore, the present study is focused on researching steady vibrations of a screen box with a flat translational motion of the box excited by a ball auto-balancer, as well as the law of vibrations and its parameters.

3. The purpose and objectives of the study

The purpose of the study is to test and define steady vibrations of a screen box with a flat translational motion and with vibrations excited by a ball auto-balancer.

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

– to develop (in a CAD system) a 3D model of a screen box with its flat translational motion and a ball auto-balancer for exciting vibrations;

– to use the 3D modeling of the dynamics to receive a signal in the form of vibro-displacements of the screen box;

– to study the motion trajectory of the box points to check the qualitative level for determining if the box vibrations are really of dual frequency;

– to test the assumption that the law of the box vibrations at a dual frequency can help determine the coefficients in the law with the use of methods of statistical analysis;

– to compare the law of the box motion that is found by the 3D modeling with the dual-frequency law that is identified by methods of statistical analysis.

4. Methods of researching the dual frequency of flat forward vibrations of a screen box excited by a ball auto-balancer

The 3D model of the vibration machine and the dual-frequency vibroexciter in the form of a ball auto-balancer is developed in the CAD system SolidWorks.

The modeling of the vibration machine dynamics is carried out with the use of the Cosmos Motion module.

The setup and the testing of the model are based on the main data from the theory of vibration machines [15].

The box makes a flat translational motion. Thus, it is possible to study the law of motion of the box by the motion of one its points. Further, the law of flat forward vibrations of the box is investigated by the motion of the center of mass of the box in a vertical plane. The motion trajectory of the center of mass of the box is estimated with the use of elements of the theory of flat transcendental curves [16].

We suggest defining the type of the excited vibrations of the box with the use of elements of the theory of vibrations. The coefficients in the law of dual-frequency vibrations are identified by the least-squares method, which is implemented in the software package Statistica for statistical analysis [17].

5. The results of researching flat translational vibrations of the screen box excited by a ball auto-balancer

5. 1. Description of the 3D model of the screen

For carrying out the research, the 3D screen model described in work [7] has been modernized. The 3D model consists of such main parts (Fig. 1): the box 1, a replaceable sieve 2, a pliable box support 3, a rigid support 4, shafts 5, the body of the auto-balancer 6, balls 7, and an unbalanced mass 8 on the auto-balancer body. Due to the modernization, it has been made possible for the box of a size of 200×300 mm to make a flat translational motion. The motion of the 3D screen model is described on the three mutual and perpendicular axes X, Y, and Z forming the right rectangular Cartesian coordinate system. The Z-axis is directed parallel to the longitudinal axis of the shaft at not deformed springs (Fig. 1). The X-axis is directed vertically up. The Y-axis is horizontal. When carrying out the experiments, gravity forces are taken into account.

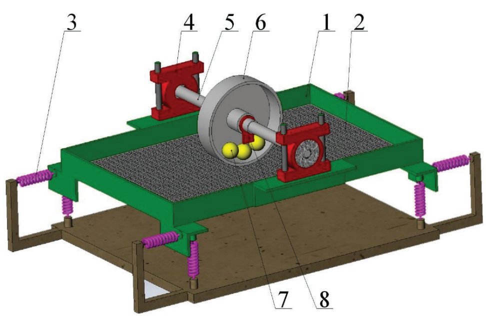

Fig. 1. A 3D model of a screen box with its flat translational motion

The adjustment and testing of the 3D model have revealed such key parameters influencing the stability of dual-frequency vibrations: the coefficient of the support rigidity k; the coefficient of the viscous resistance force of the support B_S; the unbalanced mass on the auto-balancer body M_D; the total weight of the balls M_{cw}; the box weight M; the shaft speed ω_r; the coefficient of the viscous resistance force against the balls' motion h_{cw}.

Such values of the parameters have been accepted by default: M_D=20 g, M_{cw}=80 g, M=2,000 g, and ω_r=3,000 rpm.

The coefficient of the support rigidity has been chosen so that the frequency of the center of mass of the box own vibrations on elastic shock-absorbers could be 11 s^{-1}, with the total coefficient of rigidity of all springs being k=4 N/mm (default value).

The adjustment and testing of the 3D model are held by elementary tests [7].

5. 2. Testing of the research methods

Vertical vibrations of the center of mass of the box have been investigated in the absence of the shaft rotation and with motionless balls in the auto-balancer. According to the linear theory of vibrations, their law must correspond to the law of free damped oscillations [15]:

$$X = \alpha \cdot e^{-ht} \cdot \cos(\omega_{res} t + \beta) + H. \quad (1)$$

Here ω_{res} is the frequency of the box's own vibrations; α and β are the parameters depending on the entry conditions; H is the constant shift on the X-axis.

The 3D modeling has resulted in a table of the dependence of the X coordinate of the center of mass of the box on time (because of its big size, this table is not provided in the article). In the table, time changes through equal steps. It is noteworthy that both the step of the time change and the "measurement" intervals of the coordinates can vary.

The obtained data have been processed through the software package Statistica for statistical analysis. The package, with the use of the method of the least squares, defines the unknown coefficients α, h, ω_{res}, β, and H by introducing the law of damped oscillations (1).

The parameters are identified at various time intervals corresponding to the 1st, 3rd, 6th, and 11th slow vibrations of the screen box. During one slow vibration of the box, the X coordinate of its center of mass was "measured" 74 times.

The results of the identification are shown is Table 1. In the table, the maximum discrepancy between the valid coordinates (found by the 3D modeling) and the coordinates found under the dual-frequency law (1) throughout the whole interval of the time change is estimated in %.

Table 1 shows that the process of determining the parameters' values is steady (robust), and the parameters practically do not change with changes in the time interval.

Fig. 2 contains graphs of the changes of the X coordinate of the center of mass of the box constructed by formula (1) and by the results of the 3D modeling.

Show that:

– for both short and long intervals of time, the law of vibrations is largely accurately described by the law of free damped oscillations (1);

– an increase in the time period does not virtually increase the discrepancy between the actual coordinates and the coordinates found under the law of damped oscillations (1).

Table 1

Results of identifying the coefficients in the law of free damped oscillations of the box

№	Number of vibrations (measurements)	α	ω_{res}	B	h	H	Discrepancy (%)
1	1 (74)	2.11795	68.06436	0.09496	6.32108	37.88034	1
2	3 (222)	2.11146	68.06179	0.09468	6.23060	37.88016	1
3	6 (444)	2.11109	68.05190	0.09398	6.22681	37.88049	1
4	11 (814)	2.11092	68.05156	0.09403	6.22534	37.88015	1

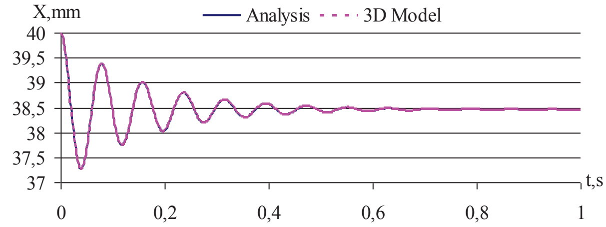

Fig. 2. Graphs of changes in the X coordinate of the center of mass of the box corresponding to free damped oscillations

5. 3. An assumption on the law of motion of the box and the identification of the parameters entailed by the law

An assumption on the law of motion of the box will be based on studying the trajectory of motion of the center of mass of the box.

The experiments of the first group have revealed separate components of the complex motion of the box. Therefore, the first experiment was made in the absence of the balls (Fig. 3, *a*) and the second – in the absence of an unbalance (Fig. 3, *b*). The 3D modeling has revealed that in both experiments the center of mass of the box moves in a circle of a corresponding diameter.

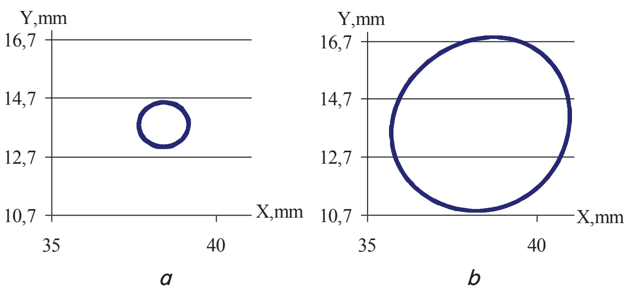

Fig. 3. The trajectory of the motion of the center of mass of the box in the vertical plane: *a* — in the absence of balls; *b* — in the absence of an unbalance

It allows assuming that the box motion is the sum of two motions:

– a slow circular motion at an angular velocity of corrective weights rotation ω_{res} and an amplitude of B;

– a fast circular motion at an angular velocity of the rotor rotation ω_r and an amplitude of A.

As the circular motions occur in one direction, the curve by which the center of mass of the box moves has to be an epitrochoid [16].

The experiments of the second group were used to prove that, in the presence of balls in the auto-balancer and an unbalance on the auto-balancer body, the component motions are integrated, and the box motion proceeds along the epitrochoid. The study considers the influence produced on the trajectory of motion by the center of mass of the box at an angular velocity of the rotor rotation.

It has been established that if the ω_r frequency exceeds the ω_{res} frequency an integral k number of times, the motion is periodic; in this case, the epitrochoid is closed, and it has (k–1) petals (Fig. 4, *a*).

If the ω_r frequency exceeds the ω_{res} frequency an irrational k number of times, the motion is quasi-periodic; in this case, the epitrochoid is opened, and it has (k–1) petals (Fig. 4, *b*).

At the identical ratio between the frequencies ω_r and ω_{res} (k=const) and a small unbalance, the epitrochoid is shortened (Fig. 5, *a*). With an increased unbalance, the epitrochoid becomes normal (Fig. 5, *b*), and then it gets extended (Fig. 5, *c*).

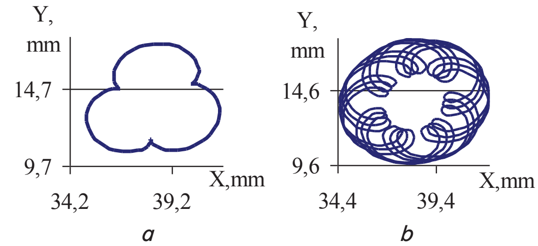

Fig. 4. The epitrochoids: a relational dependence between the frequency of the rotor rotation and the frequency of the balls' rotation: *a* — closed k=4; *b* — opened k=3+$\sqrt{2}$

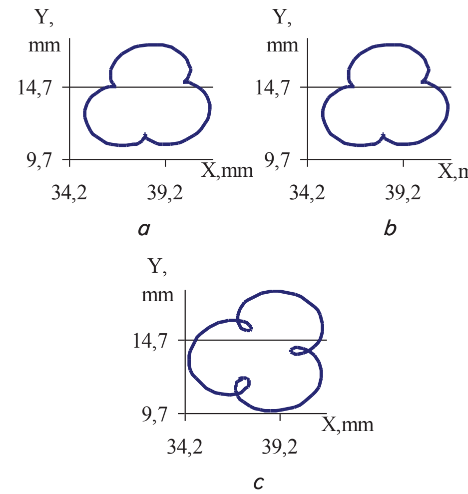

Fig. 5. The epitrochoids (k=4) when the unbalance is: *a* — small M_D=20 g; *b* — average M_D=50 g; *c* — big M_D=80 g

Thus, the conducted tests prove that the box makes a complex flat translational motion, which is the sum of a circular motion at the frequency of the rotor rotation and a circular motion at the frequency of the balls' rotation (with the fundamental frequency of the platform vibrations). It allows assuming that the Cartesian coordinates X and Y of the center of mass of the box change under the law of dual-frequency vibrations:

$$X = A\cdot\cos(\omega_r t+\delta)+B\cdot\cos(\omega_{rez} t+\gamma)+H,$$

$$Y = A\cdot\sin(\omega_r t+\delta)+B\cdot\sin(\omega_{rez} t+\gamma). \qquad (2)$$

Here, A is the amplitude of vibrations at the frequency of the rotor rotation; ω_r is the frequency of the rotor rotation; t is the time; δ is a phase; B is the amplitude of vibrations at the frequency of the balls' rotation; ω_{res} is the frequency of the balls' rotation; γ is a phase; H is a displacement on the X coordinate.

5. 4. Identification of the coefficients of dual-frequency vibrations

The 3D modeling has produced a table of dependence of the coordinates of the center of mass of the box on time (in the present article, it is not provided). In the table, time changes over equal steps like in the experiment described in section 5. 2. Then the obtained data were processed through the Statistica software package for statistical analysis. This software package, with the use of the least-squares method, has determined the unknown coefficients A, ω_r, δ, B, ω_{res}, γ, and H within the law of dual-frequency vibrations of the box (2).

The results of the identification are shown in Table 2. In this table, the maximum discrepancy between the valid coordinates (found by the 3D modeling) and the coordinates found under the dual-frequency law of vibrations (2) throughout the whole interval of time changes is estimated in %.

The data, obtained through the software package Statistica for statistical analysis, were used to construct graphs of changes in the X coordinate of the center of mass of the box at various intervals of time (Fig. 6). The constructed graphs have practically matched the graphs constructed by the CAD system Solidworks. The discrepancy does not depend on the size of an interval and does not exceed 1 %.

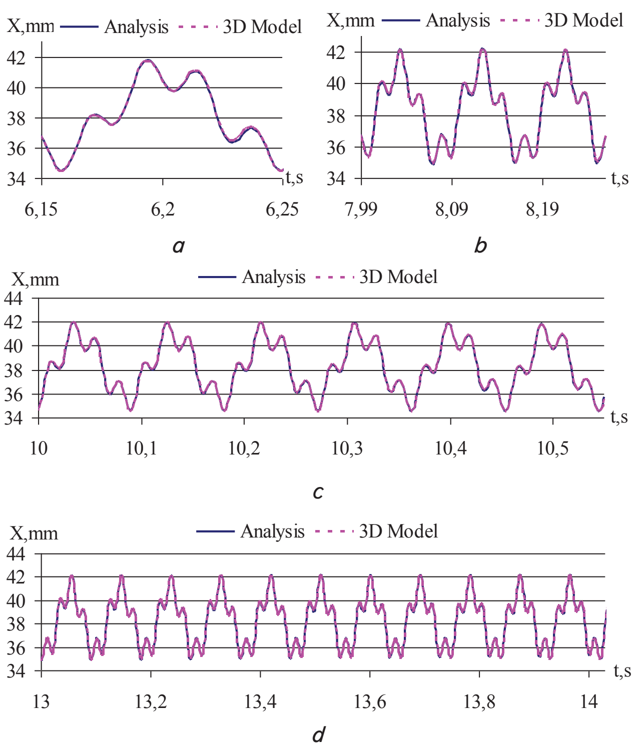

Fig. 6. Graphs of changes in the X coordinate of the center of mass of the box through time: *a* – 1; *b* – 3; c – 6; *d* – 11, slowing the box vibrations

Table 2

Results of identifying the parameters of dual-frequency vibrations

№	Number of slow vibrations (measurements)	A	ω_r	Δ	B	ω_{res}	γ	H	Discrepancy (%)
1	1 (74)	1.1761	276.4491	1.3224	2.6842	68.8971	6.2863	38.2881	1
2	3 (222)	1.1774	276.3858	2.4894	2.7346	68.8342	9.4845	38.2882	1
3	6 (444)	1.1782	276.4798	15.5549	2.6862	68.7929	13.0746	38.2882	1
4	11 (814)	1.1777	276.4670	6.3324	2.7211	68.8605	18.4193	38.2888	1

The graphs (Fig. 6) and Table 2 show that:

– process of calculating the values of the coefficients in the law of dual-frequency vibrations (2) is steady (robust) as the values of the coefficients practically do not depend on a time interval at which they are calculated;

– for both short and long intervals of time (during several slow vibrations of the box), the discrepancy between the "actual" law of vibrations for the box and the dual-frequencies does not exceed 1 %.

Thus, it is possible to claim, with a high degree of accuracy, that the ball auto-balancer excites practically ideal dual-frequency vibrations of the box.

A two-level experiment was further undertaken to determine the dependence of the amplitudes of fast and slow vibrations on the unbalanced mass across the auto-balancer body M_D and the total masses of the balls M_{cw}. The value of the masses changed in the range of 20 grams to 40 grams (Table 3).

Table 3

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

№	Mass		Amplitudes		Frequencies		$\frac{A}{M_D}$	$\frac{B}{M_{CW}}$
	M_D	M_{cw}	A	B	ω_r	ω_{res}		
1	40	40	0.6037	1.3497	276.4588	68.84183	0.015093	0.0337425
2	20	20	0.3031	0.6748	276.4621	68.81166	0.015155	0.0337437
3	20	40	0.3035	1.3473	276.4456	68.82109	0.015175	0.0336825
4	40	20	0.6094	0.6745	276.4480	68.81185	0.015235	0.0337264

The experiment has determined that the relation of the amplitudes to the corresponding masses does not change. It confirms that the values of the amplitudes are directly proportional to the unbalanced mass and the total mass of the balls.

6. Discussion of the research results on flat translational vibrations of a screen box excited by a ball auto-balancer

The conducted tests have proved that the ball auto-balancer excites almost ideal dual-frequency vibrations of the screen box that makes a flat translational motion. The discrepancy between the law of motion received by the 3D modeling and the law received by the methods of statistical analysis is less than 1 %.

Furthermore, the auto-balancer works as two independent inertial vibroexciters. The first vibroexciter is formed by the balls that are closely pressed to each other. The balls cannot catch up with the shaft; they gather together to rotate around the longitudinal axis of the shaft at the frequency of the box's own vibrations. It generates slow resonant circular vibrations of the screen box. The second vibroexciter is formed by an unbalanced mass on the auto-balancer body. It excites fast circular motions of the box at the shaft speed. The motion trajectories of the box points in the resulting motion are flat transcendental curves – epitrochoids.

The Cartesian coordinates of the box points fluctuate under dual-frequency laws. The first frequency corresponds to the velocity of the balls' rotation, and the second relate to the velocities of the shaft rotation. The amplitude of slow vibrations is directly proportional to the total mass of the balls, whereas the amplitude of fast vibrations relates to the unbalanced mass.

The approach that was applied in the present study to research vibrations excited by a passive auto-balancer can be used also in natural experiments. Moreover, the vibro-displacement (vibro-velocity or vibro-acceleration) of the box can be received with the use of relevant sensors.

It should be noted that the results have been received by a 3D modeling of the dynamics of a specific stand of a vibration machine. Therefore, the numerical results cannot be used for designing other vibration machines. However, the approach that is applied in this study can be used without basic changes for any other vibration machine with a flat translational motion of the box.

Further it is planned to carry out experiments at a natural stand to confirm the results obtained through the 3D modeling, as well as to create and analyze a mechanical-mathematical model of a screen with a dual-frequency vibroexciter of vibrations in the form of a passive auto-balancer.

7. Conclusions

1. The study has proved that an effective method of obtaining the law of motion on separate points of a screen box in the numerical form is to create a 3D model of the screen in a CAD system and to simulate its dynamics. The obtained data are considered as a signal the type of which is to be defined.

2. The research of the motion trajectory of the center of mass of the box has shown that the motion is the sum of two circular motions:

– a slow circular motion at an angular velocity of the balls' rotation;

– a fast circular motion at an angular velocity of the rotor rotation.

The motion trajectory of the center of mass of the box is an epitrochoid. The projections of the center of mass of the box onto the coordinate axes in the motion plane allegedly change under the law of dual-frequency vibrations.

3. Under the assumption that the projections of the center of mass of the box produce dual-frequency vibrations, the software package Statistica for statistical analysis was used to choose the coefficients under a relevant law. Eventually, it has been established that:

– the process of determining the values of the coefficients is steady (robust), and the coefficients practically do not change with changes in the time interval;

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

– the amplitude of fast vibrations is directly proportional to the unbalance on the auto-balancer body.

4. For both short and long intervals of time (during several slow vibrations of the box), the discrepancy between the law of vibrations of the center of mass of the box that was found through the 3D modeling and the law of dual-frequency of vibrations found by the method of statistical analysis does not exceed 1 %.

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Дослідження проведено у рамках дискретної моделі гнучкого двохопорного ротора, що балансується двома пасивними автобалансирами, розташованими біля опор. Отримано систему диференціальних рівнянь, що описують процес автобалансування. Встановлено, що основні рухи, за умови їх існування, є стійкими на зарезонансних швидкостях обертання ротора. Проведено оцінку перебігу перехідних процесів за коренями характеристичного рівняння

Ключові слова: гнучкий ротор, автобалансир, автобалансування, основні рухи, стійкість, перехідні процеси

Исследования проведены в рамках дискретной модели гибкого двухопорного ротора, балансируемого двумя пассивными автобалансирами, расположенными возле опор. Получена система дифференциальных уравнений, описывающая процесс автобалансировки. Установлено, что основные движения, при условии их существования, устойчивы на зарезонансных скоростях вращения ротора. Проведена оценка протекания переходных процессов по корням характеристического уравнения

Ключевые слова: гибкий ротор, автобаланcир, автобалансировка, основные движения, устойчивость, переходные процессы

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RESEARCH OF STABILITY AND TRANSITION PROCESSES OF THE FLEXIBLE DOUBLE-SUPPORT ROTOR WITH AUTO-BALANCERS NEAR SUPPORT

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1. Introduction

Rotors of many gas turbine engines, turbine units, centrifugal machines and so forth work at high velocities of rotation. Thereof they behave as flexible [1, 2]. The form and rotor unbalance of the flexible rotor depend on the current velocity of rotation, change from temperature, wear of the rotor and so forth. Therefore, the balance of such rotors during operation is worthwhile to be constantly corrected by passive auto-balancers [3–13]. In the latter corrective weights (pendulums, balls or rollers) under certain conditions come to positions in which they counterbalance the rotor. Then carry out steady state motions which it is accepted to call the main. There are also secondary motions on which auto-balancing does not come and auto-balancers increase vibrations of the rotor.

For balancing of a wide class of flexible rotors by passive auto-balancers in practice it is necessary to have a certain method which operability is theoretically proved. At justification of the method the conditions of existence and stability of the main motions (conditions of occur of auto-balancing) are defined and transition processes are estimated.

The most perspective method of balancing of flexible rotors on two pliable supports is the method in which two auto-balancers are located as close as possible to supports [3]. Below operability of this method is proved.