3D моделюванням досліджено процес збудження кульовим автобалансиром бігармонійних коливань, у котрих нижча частота співпадає з власною частотою коливань короба грохоту. Знайдені області зміни основних параметрів, всередині яких гарантовано настають двохчастотні вібрації, досліджений вплив основних параметрів зі знайдених областей на характеристики двохчастотних вібрацій

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

3D моделированием исследован процесс возбуждения шаровым автобалансиром бигармонических колебаний, у которых низшая частота совпадает с собственной частотой колебаний короба грохота. Найдены области изменения основных параметров, внутри которых гарантировано наступают двухчастотные вибрации, исследовано влияние основных параметров из найденных областей на характеристики двухчастотных вибраций

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

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RESEARCH INTO EXCITATION OF DUAL FREQUENCY VIBRATIONAL-ROTATIONAL VIBRATIONS OF SCREEN DUCT BY BALL-TYPE AUTO-BALANCER

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1. Introduction

Vibration machines found a wide application in the extracting industry [1], the agricultural production [2], the production of building materials, in the construction works [3], etc. The vibration machines which combine the advantages of the resonance machines and poly-frequency vibration machines are the most promising among them. On the one hand, the work in the resonance mode ensures the vibrations of the ducts of large sizes and mass with the minimum energy consumption. On the other hand, poly-frequency vibrations increase the productivity of the vibration machines [3–9].

In technical solutions [3–9], the frequencies of the excitable vibrations are rigidly connected with each other and their amplitudes cannot be changed in the process of the work of a vibration machine. This does not make it possible to change the characteristics of poly-frequency vibrations within wide limits. The resonance modes of the vibrations of the vibrator's operating unit can also be difficult to realize because of the low stability of this mode [3, 4]. Therefore, it is urgent to create the resonance poly-frequency vibration machines with the stable resonance mode and the possibility of changing within the wide limits of the vibrations characteristics.

2. Analysis of scientific literature and the problem statement

The resonance poly-frequency vibrating machines are the most promising among the vibration machines [4]. In such machines, as a rule, the basic technological process (separation, classification, sifting, etc.) occurs at the lowest frequency coinciding with the resonance frequency of the duct vibration, and the additional technological processes (sieve self-cleaning, a change in the mechanical properties of the workable material, etc.) occur at higher frequencies. In this case, a resonance mode ensures the vibrations of the duct of large sizes and mass with the minimum energy consumption and the minimum loads on the drive's components.

The duct vibrations at higher frequency can give the directly opposite properties to the workable material. For example, in the work [3], the vibrations of a duct with higher frequency contribute to the compaction of concrete, and in the work [4] they create "a boiling layer" with which they loosen the mineral raw material. So, the rapid vibrations depending on their frequency can improve or worsen the passage of different fractions of the workable material through the sieve [5], to increase or to reduce the productivity of the vibrating mills [6], etc. Therefore, in the poly-frequency vibration machines it is worthwhile having the capacity to change the characteristics of rapid

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vibrations in the working process of a vibrating machine for obtaining the required effect.

However, the vibration exciters used nowadays and the methods of the excitation of the poly-frequency vibrations do not make it possible to change the rapid frequencies of the duct vibrations and their amplitude in the process of vibrating machine operation [4]. This concerns the multimass vibration machines with several resonance frequencies [3], the vibration machines with the devices for the self-excitation of the repeated combinational parametric resonance [7], the vibration machines in which the super-harmonic vibrations are excited as the rapid vibrations due to the nonlinear transmission of a vibration drive [8] or the nonlinear characteristics of the elastic and compressible duct supports [9], etc.

For the purpose of overcoming the drawbacks of the poly-frequency vibration exciters described above, in the paper [10], the new method of excitation of the dual frequency vibrations in vibration machines with the use of the passive auto-balancers (AB) as the vibration exciter was proposed. The method uses a special state of motion of the corrective loads in the AB, which emerges with the relatively small resisting forces to their motion relative to the AB case. In this mode, the corrective loads are gathered together, cannot overtake the shaft, on which the AB is mounted and get stuck at the resonance frequency of the duct vibration, but the unbalanced case of the AB revolves synchronously with the shaft. In this case, the corrective loads excite slow resonance duct vibrations, and the unbalanced case excites the rapid duct vibrations.

The workability of the method was proved by a 3D simulation in the computer CAD SolidWorks [10] and was confirmed by a full-scale experiment [11] for the vibration machine with the vertical forward motion of the duct. In this case, since it was impossible to carry out the large number of experiments on the real screen, it was substituted with (miniature) screen stand with the variable kinematics of the duct motion. On the stand, it is possible to investigate the work of the dual frequency vibration exciter in the form of the ball or roller auto-balancer [11]. The stand allows: controlling relative motion of the corrective loads (spheres, rollers), changing the unbalanced mass of the shaft, its rotation frequency, changing the hardness of the elastic supports and the kinematics of the duct motion, measuring the duct vibratory accelerations and obtaining the spectrum of the vibration frequencies.

In the paper [12], the applicability of method for the modernization of the industrial screens was proved. For this, the 3D model of the light type screen GIL 42 with the single-frequency inertia vibration exciter and the two-frequency vibration exciter in the form of a ball auto-balancer was developed. With the help of the computer simulation, the process of the excitation of the vertical dual frequency duct vibrations was investigated. It was established that in the wide area of the change in the screen and auto-balancer parameters, the dual frequency vibrations are guaranteed to occur. The criterion of replacing the single-frequency vibration exciters to the dual frequency ones was developed.

The process of excitation of the dual frequency vibrations by the ball-type AB during the vibrational-rotational motion of the screen duct is studied in this work. Such screens have the simplest construction and as such ensure both high efficiency of screening (75–85 %) and durability, as well as the technological effectiveness of the construction [1].

3. The purpose and the tasks of the research

The aim of the work is the study of the process of excitation of two-frequency vibrations by the ball-type AB during vibrational-rotational screen duct motion.

To achieve the set goal, the following tasks were solved:

– to design a 3D model of the screen stand with vibrational-rotational duct motion and the ball-type AB for the excitation of the dual frequency vibrations;

– to identify main parameters of the screen and the AB that influence the duct frequency vibrations and to find the ranges of values of these parameters within which the dual frequency vibrations are guaranteed;

– to study the influence of a change in these parameters in the defined range on the characteristics of the dual frequency vibrations;

– to establish the mechanism of excitation of the AB of the dual frequency vibrations during the vibrational-rotational duct motion.

4. Methods of studying the process of the excitation by the ball-type AB of the dual frequency vibrational-rotational vibrations of the screen duct

The posed problems are solved by a computer 3D simulation.

For creating a 3D model of the vibrating machine and the dual frequency vibration exciter in the form of the ball-type AB, the computer program CAD SolidWorks is used.

The simulation of the dynamics of the vibrating machine is carried out with the use of the module Cosmos Motion.

The main information from the theory of the vibration machines is used during tuning and testing of the model.

5. Results of the study of the process of excitation by the ball-type AB of the dual frequency vibrational-rotational vibrations of the screen duct

The 3D model of the screen, described in the paper [10] was modernized. Due to modernization, the duct of 200×300 mm in size acquired the capacity to make the vibrational-rotational motions (Fig. 1). The 3D model consists of such main parts as: the duct 1, the changeable sieve 2, the supple supports 3, the hinged support 4, the supports 5, the shaft 6, the AB case 7, the spheres 8 and the unbalanced mass 9.

After adjusting and testing the 3D model, such main parameters influencing the stability of the dual frequency vibrations were defined: the rigidity coefficient of the supports k; the coefficient of the force of the viscous resistance of the supports B; the unbalanced mass on the case of the AB M_D; the summary mass of spheres M_{kg}; the mass of duct M; the frequency of the shaft rotation ω; the coefficient the force of the viscous resistance to the spheres motion h.

For conducting the experiments, such values of the parameters of the screen stand were accepted by default: the unbalanced mass on the AB case – M_D =13 g, the total mass of the spheres M_{kg} =28 g, the duct mass M=2000 g, the shaft rotation frequency ω=1500 r/min.

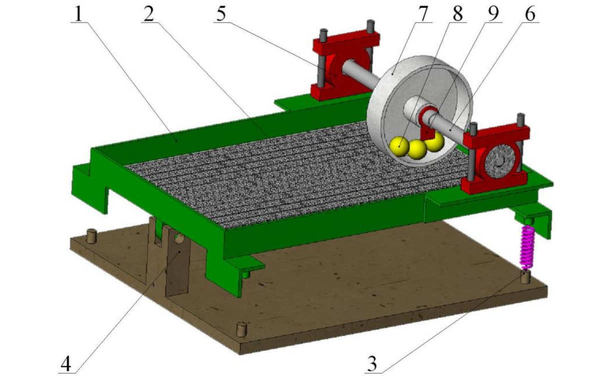


Fig. 1. 3D model of the screen stand with vibrational-rotational duct motion

The rigidity coefficient of the supports was chosen in such a way that the natural vibration frequency of the duct masses center (CM) on the elastic shock absorbers would account for 12 s^{-1}, in this case the summary rigidity coefficient of all springs accounted for k=5 N/mm (value by default).

At first, the coefficient of viscous resistance of supports B changed (Table 1). It was established that the dual frequency vibrations do not emerge with too low or too large forces of viscous resistance.

Table 1

Influence of the coefficient of the force of viscous resistance of the supports on the excitation of the dual frequency vibrations

B, H×s/mm	0,0001	0,0005	0,0007	0,0009	0,002	0,003	0,004
Damping time of the free duct vibrations, s	15	8	6	4	2	1	0,7
Dual frequency vibrations emerge	–	+	+	+	+	+	–

In further studies the value B=0,001 N×s/mm is accepted by default.

The following step was the determination of the range in the plane of M_{kg}×h, within which the dual frequency vibrations are guaranteed to occur. It was established that due to an increase in the total mass of the spheres M_{kg}, the range of a change in the coefficient h of the force of the viscous resistance to the spheres motion increases, within which the dual frequency vibrations (Fig. 2, *a*) emerge. In the graph, the horizontal axis reflects the value of M_{kg}, the vertical one – h. The range within which the dual frequency vibrations are guaranteed to emerge is shown by shading. In further studies, the value of h=0,00007 N×s/mm is accepted by default.

Similarly, the range where the dual frequency vibrations are guaranteed to emerge was found in the plane of the parameters k×B. In Fig. 2, *b* the horizontal axis reflects the value of the rigidity coefficient of the supports k, the vertical axis – the value of the coefficient of force of viscous resistance of the supports B.

As can be seen from the graphs in Fig. 2, the ranges of the dual frequency vibrations are relatively large, which makes it possible to change the characteristics of the vibrations by a change in the parameters from these ranges.

Let us explore the influence of the main parameters on the dual frequency vibrations.

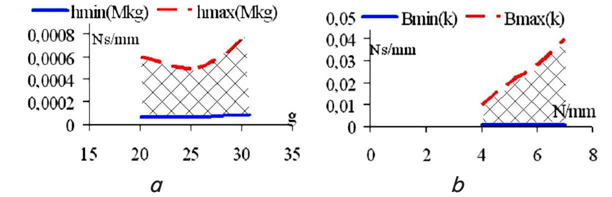


Fig. 2. Ranges of values of the parameters, within which the dual frequency vibrations emerge: *a* – M_{kg}×h; *b* – k×B

Since the screen duct makes the vibrational-rotational motion, the module of the vibration acceleration will increase from the minimum $|\bar{a}_O|=0$ to the maximum $|\bar{a}_O|=0$ value with distancing from the rotation center (Fig. 3). Therefore, the vibratory acceleration of the duct masses center $\bar{a}_C$ ($|\bar{a}_C|=|\bar{a}_N|/2$) was used to construct the diagrams of vibratory accelerations (Fig. 3).

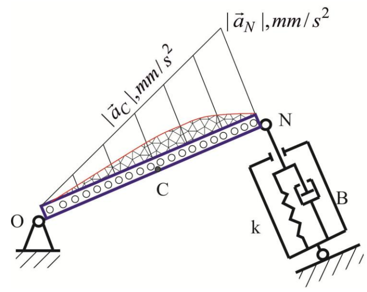


Fig. 3. Scheme of a change in the module of vibratory accelerations of the points of the screen duct depending on the distance from the rotation center

In all the diagrams of vibratory accelerations, the horizontal axis reflects the value of time, the vertical axis – the value of the projection of the vector of the vibratory acceleration of the duct masses center onto the vertical axis. Let us note that the use of this projection is justified during the vertical-progressive duct motion. With the vibrational-rotational duct motion, this projection also characterizes the vibrations, moreover, the more precise it is, the less the duct slope angle and the vibration amplitude are.

For exploring the influence of the duct mass on the characteristics of vibrations, the value of the duct mass changed in the range of 2000–3000 g (Fig. 4, *a–c*).

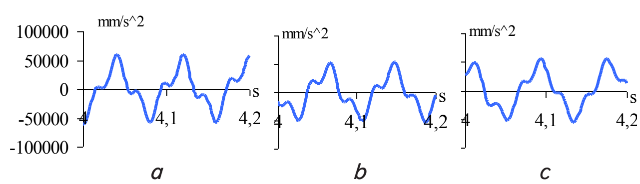
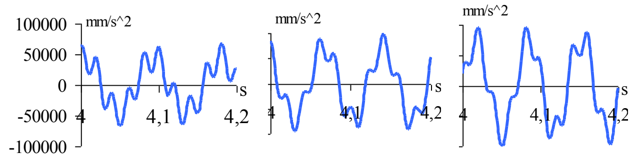


Fig. 4. Influence of the duct mass on the diagram of vibratory accelerations of its masses center: *a* – M=2000 g; *b* – M=2500 g; *c* – M=3000 g

It was established, that the resonance frequency of the duct vibrations decreases with an increase in the duct mass. In this case, the spheres automatically adjust to a change in the duct mass.

For studying the influence of the total mass of the spheres M_{kg} on the characteristics of the vibrations, the value of this mass changed in the range of 25–32 g (Fig. 5, *a–c*).

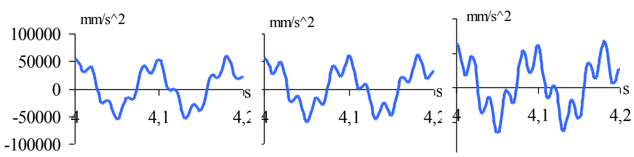


a *b* *c*

Fig. 5. Influence of the total mass of spheres on the diagram of vibratory accelerations of the duct masses center: *a* – M_{kg}=25 g; *b* – M_{kg}=28 g; *c* – M_{kg}=32 g

It was established that an increase in the total mass of the spheres in direct proportion increases the amplitude of slow vibrations of the duct masses center. This increases the vibration energy, directed toward fulfilling the main technical process (separation, classification, sifting, etc) in direct proportion.

For studying the influence of the unbalanced mass M_D on the characteristics of the vibrations, the value of the unbalanced mass was changed in the range of 13–23 g (Fig. 6, *a*–*c*).

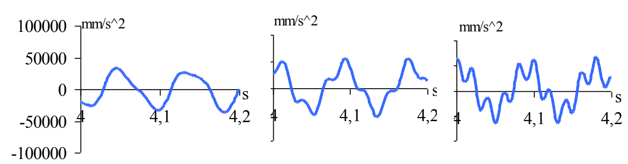


a *b* *c*

Fig. 6. Influence of the unbalanced mass on the diagram of vibratory accelerations of the duct masses center: *a* – M_D=13 g; *b* – M_D=18 g; *c* – M_D=23 g

It was established that an increase in the unbalanced mass on the AB case increases the amplitude of rapid vibrations of the duct masses center in direct proportion. This increases vibration energy, directed toward the duct self-cleaning and the change, through the vibrations, of the mechanical properties of the workable material in direct proportion.

For studying the influence of the frequency of the shaft rotation ω on the characteristics of the vibrations, the value of the frequency of the shaft rotation was changed in the range of 1500–3000 r/min (Fig. 7, *a*–*c*).



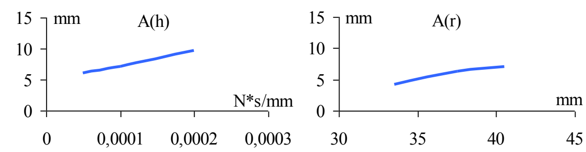
a *b* *c*

Fig. 7. Influence of the frequency of the shaft rotation on the diagram of vibratory accelerations of the duct masses center: *a* – ω=1500 r/min; *b* – ω=2300 r/min; *c* – ω=3000 r/min

It is established that an increase in the rotation frequency of the rotor increases the speed of rapid vibrations of the duct masses center in direct proportion. This, by the quadratic law, increases vibration energy, directed toward the duct self-cleaning and the change, through the vibrations, of the mechanical properties of the workable material.

The further studies were directed toward determining the parameters influencing the value of the amplitude of slow vibrations of the duct masses center. It was established that an increase in the coefficient h of the force of viscous resistance to the motion of the spheres leads to certain increase in the amplitude of slow vibrations of the duct masses center (Fig. 8, *a*). In the graph, the horizontal axis reflects value h, the vertical axis – the value of the amplitude of the motion of the duct masses center.

An increase r in the radius of the spheres rotation increases in direct proportion the amplitude of slow vibrations of the duct masses center (Fig. 8, *b*, the horizontal axis reflects value r, the vertical axis – the value of the amplitude of the motion of the duct masses center).



a *b*

Fig. 8. Dependence of the amplitude of slow fluctuations of the duct masses center on: *a* – the coefficient of the force of viscous resistance to spheres motion; *b* – the radius of the spheres rotation

At the same time, an increase in the coefficient of the force of viscous resistance of supports B leads to the decrease in the value of the amplitude of slow vibrations of the duct masses center (Fig. 9).

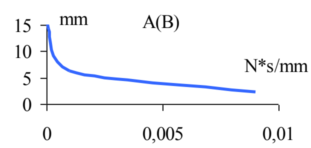


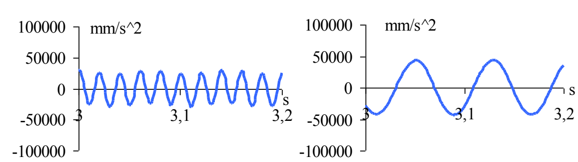
Fig. 9. Dependence of the amplitude of slow vibrations of the duct masses center on the coefficient of force of viscous resistance of the supports

In the graph, the horizontal axis reflects the value B, the vertical axis – the value of the amplitude of slow vibrations of the duct masses center.

Let us explore the mechanism of excitation of the dual frequency vibrations by the auto-balancer.

Since the dual frequency vibrations have two constituents, which emerge from the unbalance on the AB case and from the spheres respectively, it was decided to explore each constituent separately. For this, the simulation of the screen work in two modes was carried out:

– without the spheres (M_{kg}=0, Fig. 10, *a*);
– without the unbalance on the AB case (M_D=0, Fig. 10, *b*).



a *b*

Fig. 10. Diagram of vibratory accelerations of the duct masses center with the single-frequency vibrations from: *a* – unbalance on the AB case; *b* – the spheres

The obtained data were then processed in the system of the computer algebra Mathcad and the summary diagram of vibratory accelerations was built. The comparison of the built diagram with the diagram, obtained by simulation in CAD Solidworks (with the presence of both the spheres and the unbalance on the AB case) (Fig. 11) shows that they are almost identical (the largest divergences do not exceed 3 %).

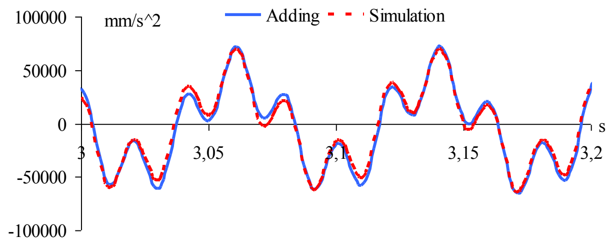


Fig. 11. Diagrams of vibratory accelerations of the duct masses, obtained by the addition of the single-frequency graphs (with the time shift by 0,01 s for rapid vibrations) and the 3D simulation

In Fig. 11, the horizontal axis reflects the time, the vertical axis – the projection of vibratory acceleration of the duct masses center onto the vertical axis.

The obtained results of the 3D simulation make it possible to formulate the following assumptions relative to the mechanism of the occurrence of the dual frequency vibrations:

– AB functions as two separate independent vibration exciters;

– the first vibration exciter forms the spheres and excites slow duct vibrations with the self-resonant frequency ω_{res};

– the second vibration exciter forms the unbalanced mass on the AB case and it excites the rapid duct vibrations with the frequency of the shaft rotation ω.

In Fig. 12, the mechanical model of the vibrator corresponding to the assumptions is built. In the diagram, the angle β determines phase displacement in the work of the vibration exciters, and the parameter δ – the proximity of the frequency of slow duct vibrations to the resonance frequency of its vibrations.

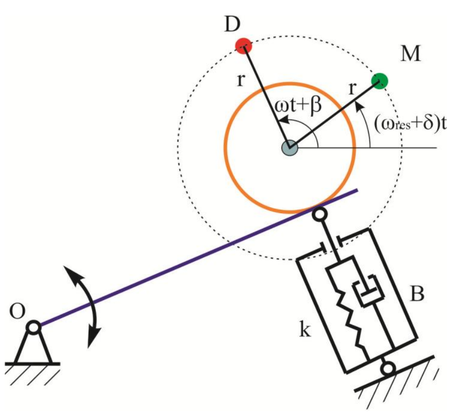


Fig. 12. The mechanical model of the vibrator

This model works only in the range of the system parameters, within which the dual frequency vibrations are guaranteed to occur. It makes it possible to find the law of the duct vibrations and to determine by it the values of the parameters, which ensure the desired dual frequency vibration parameters.

Let us note that in this model with the small resisting forces, the parametric resonances are possible [5, 6]. This mode of vibrations is undesirable for the considered vibration exciters. Its absence is ensured by the presence of the forces of viscous resistance.

6. Discussion of the results of the study of the process of excitation of the dual frequency vibrational-rotational vibrations of the screen duct by the ball-type AB

The performed 3D simulation confirmed the possibility of the excitation of the dual frequency vibrational-rotational vibrations of the screen duct by the ball-type AB. In this case, the relatively wide range of the change in the parameters of the vibrator and the AB, within which such vibrations are guaranteed to be excited. It is shown that it is possible to change the characteristics of the dual frequency vibrations in broad bands with a change in these parameters.

It should be noted that the results were obtained by the 3D simulation of the dynamics of a specific stand of the vibrator. Therefore, the numerical results cannot be used for designing other vibrators. However, the approach used in this work can be applied without the fundamental changes to any other vibration machine with the vibrational-rotational duct motion.

Further plans include: conducting experiments on a full-scale stand for confirming the results obtained by the 3D simulation; constructing and analyzing a mechanic mathematical model of the screen with the dual frequency vibration exciter in the form of the passive AB.

7. Conclusions

1. The 3D model of the screen stand with the vibrational-rotational duct motion and the ball-type AB for the excitation of dual frequency vibrations was designed. Geometric and mass-inertia parameters of the model completely correspond to a full-scale stand.

2. The main parameters of the screen and the AB, which influence the two-frequency vibrations were identified: the rigidity coefficient of the supports k; the coefficient of the force of viscous resistance of the supports B; the unbalanced mass on the AB case M_D; the summary mass of the spheres M_{kg}; the duct mass M; the frequency of the shaft rotation ω; the coefficient of the force of viscous resistance to the spheres motion h.

It was established that for the guaranteed excitation of the dual-frequency vibrations: with the summary mass of the spheres M_{kg}=28 g, the force coefficient h of viscous resistance to the spheres motion can change in the range of 0,00005–0,0006 N×s/mm; with the value of the rigidity coefficient of the springs k=5 N/mm, the coefficient of the force of viscous resistance of the supports B can be found in the range of 0,00001–0,02 N×s/mm.

3. It was established, that an increase in the rigidity coefficient of the supports k leads to an increase in the natural vibration frequency of the duct masses center. An increase in the coefficient of the force of viscous resistance of the supports B decreases the amplitude of slow vibrations of the duct masses center. An increase in the summary duct mass in the range of 2000–3000 g leads to the decrease in the resonance frequency of the vibrations of the duct masses center; however, in this case spheres automatically adjust to a change in the summary duct mass. An increase in the summary spheres mass in the range of 25–32 g in direct proportion increases the amplitude of slow fluctuations of the duct masses center.

This increases the vibration energy, directed toward the basic technical process (separation, classification, sifting, etc.) in direct proportion. An increase in the unbalanced mass on the AB case in the range of 13–23 g in direct proportion increases the amplitude of rapid vibrations of the duct masses center. An increase in the rotation frequency of the rotor in the range of 1500–3000 r/min in direct proportion increases the amplitude of the rapid vibration speeds of the duct. This increases the vibration energy, directed toward the duct self-cleaning and the change of mechanical properties of the workable material through the vibrations in direct proportion to the square of the rotation frequency of the rotor.

A change in the defined parameters allows: increasing by 3 times the amplitude of slow vibrations and increasing by more than 3,5 times the amplitude of rapid vibrations, changing by more than 15 times the relation between the amplitudes of the rapid and slow vibrations and changing by more than 3 times the relation between excitable frequencies.

4. The simulation showed that the AB works as two separate vibration exciters. In the first vibration exciter, the spheres rotate practically evenly with the resonance frequency of the duct vibrations, moreover, independent of its loads, the spheres automatically adjust to the frequency, by which they excite the slow resonance duct vibrations (12 Hz) with a large amplitude. In the second vibration exciter, the mass on the AB case excites the rapid duct vibrations with (any) existing non-resonant rotation frequency of the rotor.

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