

Розроблена нова схема збудження коливань робочих органів блоків вібраційної установки на основі зміни фазових кутів дебалансів між собою. Реалізація такої ідеї дозволяє за один оберт дебалансів здійснити ту кількість вібрордій на технологічне середовище, скільки віброблоків має установка. Таким чином реалізовано спектр частот, що значно підвищує ефективність технологічного процесу. Запропонована схема придатна для реалізації різних процесів із зменшенням енерговитрат у порівнянні з існуючими конструкціями вібраційних машин. Розроблена конструктивна схема вібраційної установки із чотирма вібраційними блоками. Вибрана математична модель на основі представлення параметрів машини дискретними, а оброблювальне середовище – континуальними. Моделювання робочого процесу вібраційної установки виконано на основі використання методу скінченних елементів. Скінченно-елементна модель складена шляхом апроксимації всіх несучих елементів, в тому числі і формують поверхню, двовимірними скінченними елементами.

Віброізоляційні опори та пружні елементи моделі прийняті тривимірними, так як процеси, що протікають в таких елементах конструкції, є більш складними з точки зору розсіювання енергії. Досліджений робочий процес енергоощадної вібраційної установки, що реалізує поліфазні коливання. Складені рівняння руху такої системи та визначені амплітуди та частоти коливань, що визначають цей рух. Здійснена оцінка розподілу амплітуд коливань по периметру рами, закріпленої на вібраційних блоках віброустановки. Визначено можливість ефективного використання поліфазного спектру коливань при виконанні процесів сортування та ущільнення матеріалів на основі реалізації зсувних та нормальних напружень. Запропонована схема енергоощадної вібраційної установки та визначені параметри відкривають реальну можливість для використання в різних галузях. Отримані результати використані при проектуванні енергозберігаючої конструкції віброустановки із врахуванням фазових кутів для ущільнення технологічних середовищ

Ключові слова: вібраційна установка, віброблок, дебаланс, модель, фазові кути, амплітуди, частоти і форми коливань

DETERMINATION OF THE WORKFLOW OF ENERGY-SAVING VIBRATION UNIT WITH POLYPHASE SPECTRUM OF VIBRATIONS

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Received date 14.08.2019

Accepted date 18.11.2019

Published date 24.02.2020

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1. Introduction

Modern economic development trends are formed on the use of machines and technologies that can ensure the minimization of energy costs with the implementation of high quality process performance. Vibration machines and

processes occupy a significant place in various sectors of the economy. At the same time, the existing vibration technology operates in a harmonious resonance mode, which is characterized by the completeness of the technological process and significant energy consumption. Therefore, the search for new solutions, both designs and research me-

thods of efficient and energy-saving vibration machines, is an urgent task.

With the development of computer modeling methods and the theory of vibration technology, a new direction has been outlined for the search for constructive solutions with a variable amplitude-frequency mode of vibrations and with non-linear characteristics, the use of resonance effects. The basis for achieving a number of these areas is the reliability of determining the adequacy of the calculation model to a real technological process. At the same time, the widespread adoption of such vibration technology is constrained by a number of existing reasons. So, the adequacy of the calculation model to the real process for a long time is based on simplified approaches to taking into account the mutual influence of working bodies and processing environments. This leads to negative results due to the difference between the calculated and real values of the parameters. This especially happens in situations where the numerical values of the vibrational masses of vibration machines and processing media are of the same order. The second reason is the forced use of resonance modes without purposefully taking into account the internal properties of the machine and the environment. Thus, the creation and use of equipment with operating modes that implement polyfrequency, polyphase vibrations is an urgent and important task, the solution of which can serve as the direction for creating a new class of technological vibration machines.

2. Literature review and problem statement

A wide range of applications and the effectiveness of vibration technologies and processes in various industries are noted in a number of works. This is a vibration technique for the manufacture of flat concrete slabs [1], which provides a simplified method for calculating parameters. This approach can't be used to calculate another class of machines. Empirical dependencies give satisfactory results only within the framework of using similar machine designs. The process of compaction of cement concrete mixtures is given in [2] based on a harmonious regime. The use of vibration technologies is also used for grinding materials [3]. In these works, the working processes of vibration technologies and processes in the implementation of standard modes without assessing the phase angle changes in machines are considered. However, an important point in the analysis of existing work is the movement assessment. From this point of view, these works deserve attention. An accurate assessment and analysis of these and other works will make it possible to more reasonably formulate methods for solving the problem. Thus, the work [4] is devoted to the study of the general problems of the motion of two-frequency resonant vibration devices with a pulsed electromagnetic drive. A rational method for the implementation of two-frequency resonant systems with multiple eigenfrequencies is considered. The effectiveness of the implementation of such operating modes is justified by the use of a pulsed electromagnetic drive with a vibration frequency of 50 Hz. However, the obtained results are given for the load conditions of the technological environment. In [5], studies are carried out using two types of feed bins as an example: with directional and independent vibrations. The problem is considered without taking into account shear stresses in the medium, it is modeled exclusively by a solid

body. The authors of [6] consider a dynamic system capable of accumulating internal energy without reference to a specific technological environment. Phenomena in complex nonlinear systems, as the authors note, are a promising direction and require additional research. In [7], the method is applied to nonlinear systems of active vibration control.

The application of such an integral method is that it requires a specialized solution to this control. One of the effective parameters that displays the dynamic process of motion is the method of measuring acceleration [8]. From the point of view of choosing assessment criteria, this is important, but it requires auxiliary equipment and consideration of the process of environmental impact. In [9], a passive device for controlling the process of influence by a vibration barrier is proposed. A vibration barrier is located in the soil and disconnected from the existing building. The principle of operation is based on the interaction of the structure-soil-structure between two vibrational structures and the soil. The structure, soil and device are considered linear and are modeled using an appropriate finite element approach. However, such a system with random action may not be adequate to the elastic properties of real soil. That is, the design scheme must take these circumstances into account. In [10], the vibrational characteristics of casing under the excitation force of an electric vibrator were studied using the main coordinate analysis method. The results show that the amplitude of vibration in the lower part of the column decreases with increasing length of the strapping column and the vibration frequency, and they are in inverse proportion; the vibration amplitude in the lower part of the casing string increases with increasing excitation forces and are in direct proportion. The functional dependence and explanations of the vibrator behavior in the contact zone with the well are not given in the work, since it is expected that in such systems a significant change in the parameters with the appearance of a wall effect. The conclusion that the vibration amplitude in the lower part of the casing string increases with increasing excitation forces is an obvious fact. In [11], numerical simulation of deep vibrational compaction of dry sand was performed. This compaction method compacts loose sands using shear deformation processes that provide horizontal vibrations of the vibration probe at the required soil depth. At the same time, it is necessary to note why damping is accepted according to the law of viscosity, if dry friction according to the Coulomb law is important in dry mixtures. There are no definitions of the effect of the phase shift between the force and the displacement at the point of application of force, since the phase angle determines the nature of the interaction of the working body and the medium. There is no justification why the effect of friction in the contact between the working body and the environment is insignificant. In [12], a study is conducted to determine the optimal mass ratio of the vibration valve for control vibration and the mass clamped by a rectangular plate. The problem of a dynamic vibration damper in discrete systems, known from the theory of vibrations, is applied. For distributed systems, such as a rectangular plate, it is necessary to automatically adjust for the possible occurrence of one of the natural vibrations of which there may be many. It is worth bringing a control system for the change in mass characteristics. This is because the vibration valve can effectively reduce the reaction across the entire plate solely by maintaining the mass ratio in the

insensitivity range of the plate vibrations. Noteworthy is the work [13] in which it is noted that insufficient rigidity of the unit of vibrators can lead to a significant change in the phase angle of synchronization and, as a result, the operation of the machine becomes uncontrolled. These results confirm the importance of the question of studying the effect of changes in the phase angle on the detection of synchronization and, as a result, on the operation of a vibration machine. The technique for modeling the influence of wave processes is given in [14] and can be applied to consider the influence of the medium for this problem. The modeling of machine parameters is useful [15], but it has been solved for the manufacture of other products. An important task in the study of the movement of vibration systems is to ensuring the reliability of systems, the possible occurrence of damage. This is solved by a number of methods, one of the most effective is given in [16], which can also serve to solve this problem with a real construction already created. The general technique for determining the parameters of machines [17] can be applied, but taking into account changes in the phase angles of rotation of the unbalances of the vibration blocks.

3. The aim and objectives of research

The aim of research is determination of the working process of an energy-saving vibration unit with a polyphase spectrum of vibrations to intensify the implementation of specific technological work processes.

To achieve this aim, the following objectives are identified:

- to justify and develop a design diagram of a vibration unit with a polyphase spectrum of vibrations;
- to research and analyze the main forms, amplitudes and frequencies of vibrations of the design of the vibration unit.

4. Development of a design scheme for a vibration unit with a polyphase spectrum of vibrations and a technique for studying its motion

Based on the analysis of previous studies, the idea arose to apply a new vibration excitation scheme with a change in the phase angles of unbalances between themselves on the vibration blocks of the vibration unit. The implementation of such an idea will allow for one revolution of imbalances to realize the number of vibration action per technological medium, how many vibration units the unit has. Thus, the implemented vibration spectrum would significantly increase the efficiency and reduce the energy costs of the technological process in comparison with existing designs of vibration machines. The implementation of just such a scheme opens up the possibility of obtaining shear stresses in the processing medium, as one of the factors for the acceleration of the regime and the reduction of energy.

The mathematical model of the design of the vibration unit (Fig. 1) is constructed according to the following assumptions. The block 1 of the unit is solids. The concrete mixture, which is in the form, is modeled by a system with distributed parameters. In the equations of motion of the general system «vibration unit – concrete mixture», the structural mass of the mold, vibration blocks and the mass of the concrete mixture are taken into account. The mass of the concrete mixture is taken into account by the wave

coefficient based on the method described in [18]. Gaskets 3 are modeled by elastic elements with a linear relationship between the resulting stress and strain. Unbalances 2 vibration blocks are located along the Y axis, are rotated along the X axis with certain angles ϕ_1 .

For this scheme, it is accepted: $\phi_1=0$; $\phi_2=60^\circ$; $\phi_3=120^\circ$; $\phi_4=180^\circ$. So, in one revolution of the shaft, the form with the concrete mixture will perceive four effects. To increase the effect of the action between the blocks and the form, elastic elements (rubber gaskets) are installed. Thus, without the use of special devices (additional shafts, gearboxes and motor devices), a polyphase mode is implemented, and in addition to waves generated along the X axis, a longitudinal wave along the Y axis also arises. Let's take the analytical description of the dynamics of such a system in the form:

$$m \cdot \ddot{y} + c(1 + f(z, t))y = F(z, t), \tag{1}$$

where m – reduced mass, taking into account the mass of the form, blocks and the reduced mass of the concrete mixture; c – elasticity of the rubber gaskets; $f(z, t)$ – function that determines the conditions of contact of the form with the gasket; $F(z, t)$ – perturbing force.

Given the accepted notation (Fig. 1): L – length of the site, δ_p – thickness of the rubber gasket. Given the operating conditions of the system:

$$f(z, t) = \begin{cases} \frac{L}{8} < z < \frac{L}{8} + b; nT < t < nT + t_k; \\ \frac{3L}{8} < z < \frac{3L}{8} + b; nT + \frac{\pi}{3\omega} < t < nT + \frac{\pi}{3\omega} + t_k; \\ \frac{5L}{8} < z < \frac{5L}{8} + b; nT + \frac{2\pi}{3\omega} < t < nT + \frac{2\pi}{3\omega} + t_k; \\ \frac{7L}{8} < z < \frac{7L}{8} + b; nT + \frac{\pi}{\omega} < t < nT + \frac{\pi}{\omega} + t_k. \end{cases} \tag{2}$$

where b – width of the contact zone of interaction; t_k – contact time; $n=0, 1, 2, \dots$

The perturbing force $F(x, t)$ will change according to a similar law to the function $f(x, t)$:

$$F(z, t) = \begin{cases} F_0 \sin(\omega \cdot t); \frac{L}{8} < z < \frac{L}{8} + b; \\ nT < t < nT + t_k; \\ F_0 \sin\left(\omega \cdot t + \frac{\pi}{3}\right); \frac{3L}{8} < z < \frac{3L}{8} + b; \\ nT + \frac{\pi}{3\omega} < t < nT + \frac{\pi}{3\omega} + t_k; \\ F_0 \sin\left(\omega \cdot t + \frac{2\pi}{3}\right); \frac{5L}{8} < z < \frac{5L}{8} + b; \\ nT + \frac{2\pi}{3\omega} < t < nT + \frac{2\pi}{3\omega} + t_k; \\ F_0 \sin(\omega \cdot t + \pi); \frac{7L}{8} < z < \frac{7L}{8} + b; \\ nT + \frac{\pi}{\omega} < t < nT + \frac{\pi}{\omega} + t_k. \end{cases} \tag{3}$$

The solution of equation (1) has the form:

$$Y(z, t) = \sum_{n=1}^{\infty} S_n(t) \cdot \sin\left(\frac{n \cdot \pi \cdot z}{L}\right). \tag{4}$$

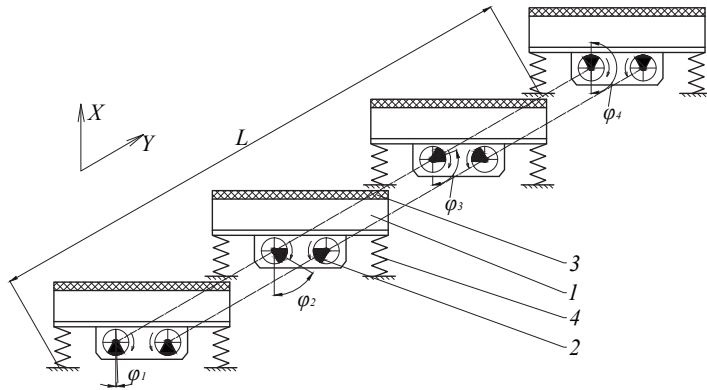


Fig. 1. The design scheme of the vibration platform with polyphase movement: 1 – vibration blocks; 2 – unbalance; 3 – rubber gasket; 4 – spring

Substituting solution (4) into the original equation (1), after the corresponding transformations, equations are obtained for determining the qualitative and quantitative picture and the working process. The simulation of the working process of the vibration unit is carried out on the basis of the finite element method using the MSC.NASTRAN calculation complex (MSC Software, Germany). The study of the model is carried out by the finite element method. The finite element model (Fig. 2) is composed by approximating all the supporting elements, including the shaping surfaces, by two-dimensional finite elements of the PLATE type, elastically deformed under the action of longitudinal force, bending moments in two planes and torque.

Vibration isolating supports and elastic elements of the model are adopted by three-dimensional SOLID-type FEs, since the processes occurring in such structural elements are more complicated from the point of view of energy dissipation.

The total number of finite elements turned out to be 19258, the number of nodes – 19912, the total number of desired variables – 20928.

It is accepted that at the extreme points of the support they are supported by the foundation and fixed, therefore, in the model, the extreme nodes of the supports are restrained in the X, Y, Z directions, and rotations along all three axes are also prohibited. It is believed that materials of the entire structure are deformed only in the elastic stage.

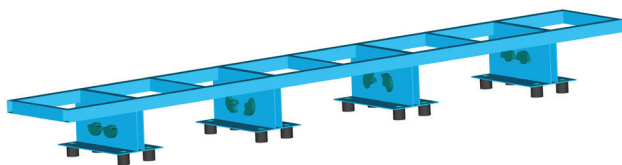


Fig. 2. General view of the 3D model of a block vibration unit

When checking the strength and determining the functioning of the system, which can be subjected to loads that differ in nature and which are applied in various combinations, it is advisable to first analyze the behavior of the structure under simple loads, and then complicate them with the addition of other perturbations.

To study the behavior of forming surfaces, a list and place of points on the surface of the form with a determination of the distribution of vibration amplitudes

depending on the location of the angle of the unbalance of the blocks is determined (Fig. 3).

To test the model and evaluate its functioning under the influence of static and dynamic loads, the following research steps are taken:

- performing modal analysis (Modes Analysis) and determining the main forms and their corresponding vibration frequencies;
- study of the structure movement under the action of a dynamic load based on dynamic analysis (Transient Analysis) at a given disturbance frequency.

In the dynamic analysis, the amplitudes of the structure vibrations in various sections (sections) are determined and the distribution of vibrations along the length of the structure is evaluated.

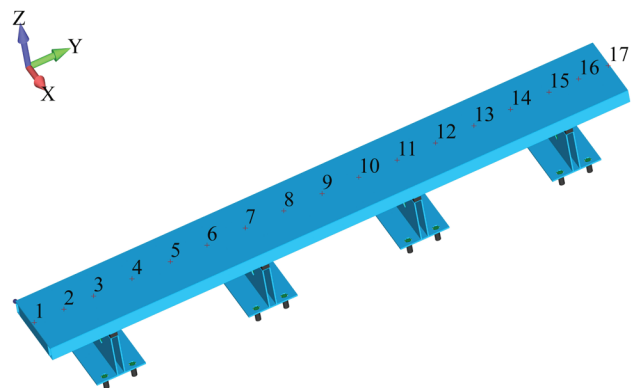


Fig. 3. The arrangement of measuring points on the surface of the form

5. Research and determination of the main forms, amplitudes and frequencies of vibrations of the design of the vibration unit

The performed studies allow to determine the forms of vibrations (Fig. 4).

So, at a vibration frequency $f=1.32$ Hz (Fig. 4, a), the vibration unit carries out vibrations in the second form in the transverse direction (X axis). This mode of vibration is due to the stiffness of the elastic supports 4 (Fig. 1). When the regime is implemented at a frequency $f=4.10$ Hz (Fig. 4, b), the vibrations are directed in the vertical direction. The implementation of vibrations in the mode at a frequency $f=16.60$ Hz (Mode 23) is a consequence of the influence of mold rigidity (Fig. 4, c). The vibrations are carried out due to the bending of the structure in the YZ plane. Such modes are insufficient for the concrete compaction process. Purely unidirectional vibrations with a low frequency do not cause stresses in the medium that do not exceed the continuity loss stress [15]. The operating mode in which the phase shift will be realized is possible with increasing to a frequency $f=27.40$ Hz (Fig. 4, d). This is due to the fact that with increasing frequency, the numerical values of the compressive stresses of the mixture layers and the stresses of the rupture of the layers increase significantly. Also, such a regime implements vibrations forming surfaces that are on elastic elements 3 (Fig. 1).

Thus, the operating mode of operation is adopted with a frequency of excitation of vibrations of 25 Hz. During the implementation of dynamic analysis, the obtained distribution

of the amplitudes of vibrations of the forming surfaces along the length of the structure for one period of vibrations (Fig. 5). The maximum value of the vibration amplitude in the framework of the studies is obtained on the length of the mold in the range 1.3–2.1 m in the range from $3/4 \pi$ to $7/4 \pi$.

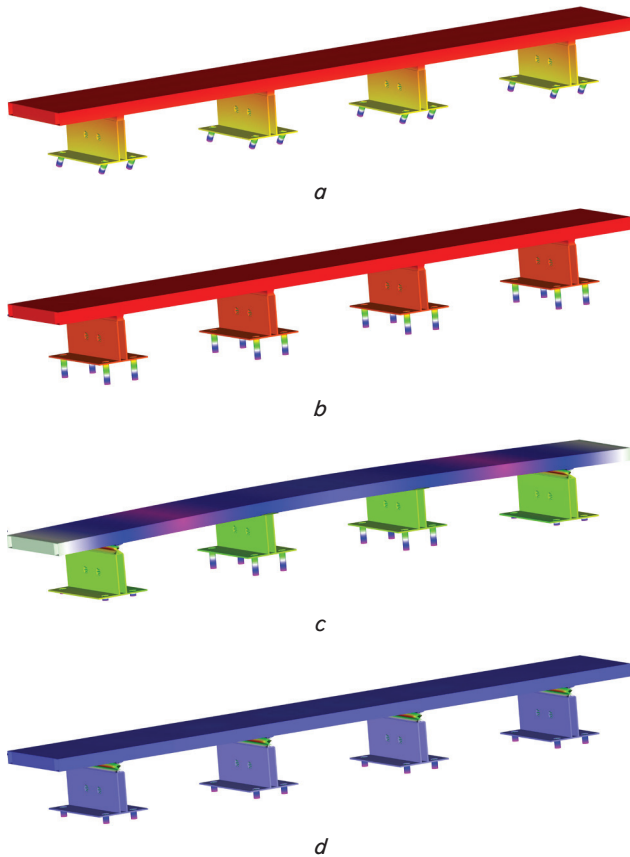


Fig. 4. Forms of structure vibration of the vibration unit:
 a – Mode 2, $f=1.32$ Hz; b – Mode 8, $f=4.10$ Hz;
 c – Mode 23 $f=16.60$ Hz; d – Mode 29 $f=27.40$ Hz

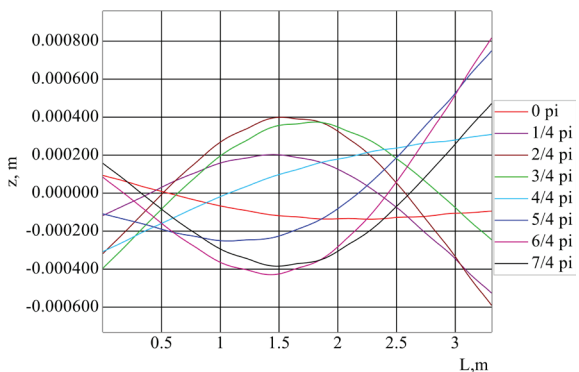


Fig. 5. Distribution of vibration amplitudes along the length of the structure for one vibration period

The surface motion at points 1, 2, 16, 17 in time (Fig. 6) indicates the presence of different values of the vibration amplitudes.

So, at point 16, the amplitude of the vibrations reaches a value of 0.6 mm, and at point 2, at the same time, it has a value of 0.2 mm, which is evidence of the presence of a wave process.

At other points (Fig. 7) the vibrograms of motion are identical in character to those considered above. The magnitude of the vibration amplitudes is smaller. At point 14,

the amplitude reaches 0.4 mm, and at point 3 in the same period of time, the amplitude of the vibrations is 0.1 mm. At points 14 and 15, there is practically no phase shift between the vibration amplitudes.

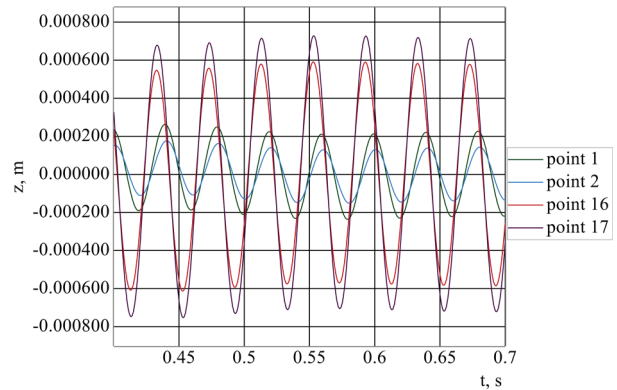


Fig. 6. Vibrograms of the movement of points 1, 2, 16, 17 in time

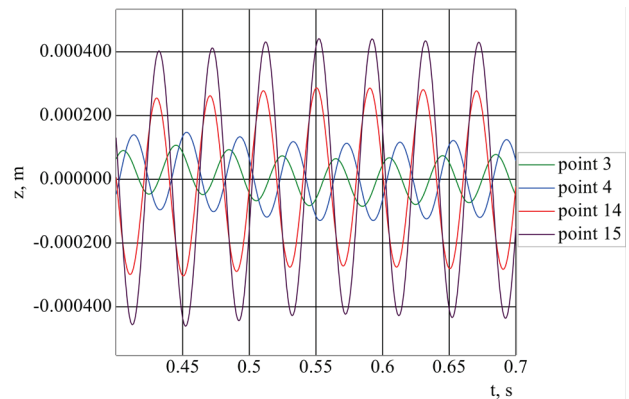


Fig. 7. Vibrograms of the movement of points 3, 4, 14, 15 in time

Vibrograms of motion at points 5, 12, 13 in time (Fig. 8) are similar in nature to the numerical values of amplitudes, as for points 1, 2, 3, 4 (Fig. 6, 7).

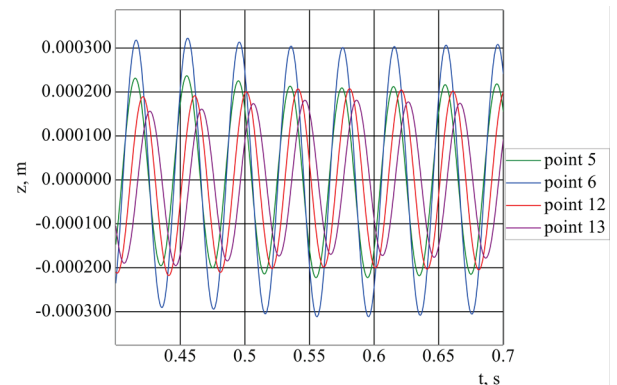


Fig. 8. Vibrograms of the movement of points 5, 6, 12, 13 in time

However, there are certain differences. So, it is recorded that at points 6, 12 and 13, the vibration amplitudes have numerical values close in magnitude in the range 0.18–0.23 mm. A change in the amplitudes of vibrations at points 7, 8, 9, 10, 11 in time (Fig. 9) is characterized by a certain deviation in numerical values and in phase.

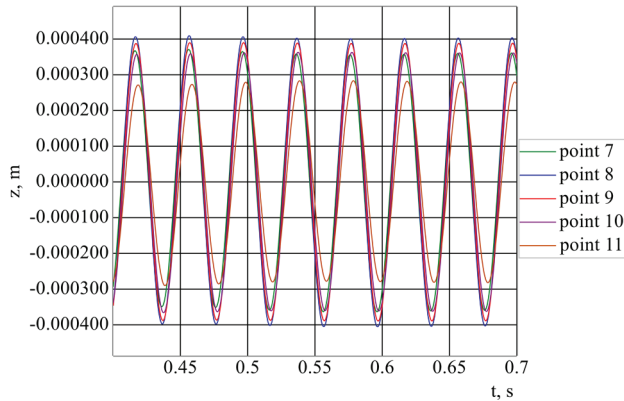


Fig. 9. Vibrograms of the movement of points 7, 8, 9, 10, 11 in time

At these points, the vibration amplitudes reach 0.27–0.40 mm, that is, the limit of change is the smallest in comparison with the movement of points 1, 2 and 16, 17 (Fig. 6).

6. Discussion of the research of the movement of an energy-saving vibration unit with a polyphase vibration spectrum

The research results indicate the presence of different in form and numerical values of the vibration amplitudes in the area of the vibration unit with a polyphase vibration spectrum. This is a fundamentally new result. It lies in the fact that a complex form of vibration, as an effective method of accelerated compaction of concrete mixtures, is realized by placing unbalances at a certain angle on each individual vibration unit. As a result, energy costs are reduced by 30 %, and the process of forming a concrete product is reduced by 20 %. The presence of various forms of the polyphase spectrum is confirmed by the forms of vibrations (Fig. 4, *b–d*) and the distribution of the amplitudes of the vibrations of the surface of the form along the length of the structure for one vibration period (Fig. 6–9, points 1–17). The use of such effects is determined by the overall dimensions of the product in plan and its height, it affects not only the phase reversal of the unbalance along the central axis of the vibration unit, but also the magnitude of the static moment of the unbalance.

This study has limitations in terms of considering a specific pattern of phase reversal of unbalances along the central axis of the vibration unit. Additional studies are needed with other schemes of phase reversal of unbalances along the central axis of the vibration unit. Such studies are planned as a continuation of the topic on the idea of optimizing the installation of phase angles on vibration blocks with a change in the frequency of vibrations over time. The proposed approach of phase reversal of unbalances along the central axis of the machine can be successfully applied for moving and sorting materials in mobile crushing and screening plants, which are widely used in various European countries, including Ukraine.

The disadvantages of this research include the magnitude of the selected values of the static moment of unbalance, which leads to a significant decrease in the amplitude of vibrations at some points (Fig. 7, points 3, 4). Further studies should optimize the parameters of the vibrations in their amplitude values within the limits provided for by the specific properties of the technological environment and the dimensions of the product.

7. Conclusions

1. The design scheme of an energy-saving vibration unit with a polyphase spectrum of vibrations is justified. The simulation of the working process of the vibration unit is carried out on the basis of the finite element method using the MSC. NASTRAN calculation complex (MSC. Software, Germany).
2. The main waveforms with numerical values of frequencies 1.32 Hz, 4.10 Hz, 15.60 Hz, 24.31 Hz are determined. The calculated value of the vibration frequency of the operating mode is 25 Hz. The most effective in the framework of the research is the waveform of 24.31 Hz. The implementation of this form of vibration provides an energy-saving mode of operation of the vibration unit.
3. A polyphase spectrum of the unit's vibrations is established and proposed at an excitation frequency of 24.31 Hz with which the vibration amplitudes of 0.27...0.6 mm are realized. The vibration mode of the calculation model is investigated, it is a confirmation of the energy-saving mode, it is a prerequisite for the calculation and creation of a new class of vibration unit.

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