Досліджений рух вібраційної установки, що реалізує складні просторові коливання. Розроблена конструктивна схема вібраційної установки. Здійснена оцінка розподілу амплітуд коливань по периметру рами, виявлені раціональні значення для ущільнення бетонної суміші. Визначено можливість реалізації багаторежимного спектру коливань вібраційної установки. Отримані результати використані при проектуванні та виготовленні раціональної конструкції рами з урахуванням складного розподілу напружень та деформацій

Ключові слова: вібраційна установка, віброзбуджувач, просторові коливання, напружено-деформований стан, частоти і форми коливань, скінченно-елементна модель

Исследовано движение вибрационной машины, которая реализует сложные пространственные колебания. Разработана конструктивная схема вибрационной установки. Осуществлена оценка распределения амплитуд колебаний по периметру рами, установлены рациональные значения для уплотнения бетонной смеси. Определена возможность реализации многорежимного спектра колебаний вибрационной установки. Полученные результаты использованы при проектировании и изготовлении рациональной конструкции рамы с учетом напряженно-деформированного состояния металлоконструкций

Ключевые слова: вибрационное оборудование, вибровозбудитель, пространственные колебания, напряженно-деформированное состояние, частоты и формы колебаний, конечно-элементная модель

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INVESTIGATION OF VIBRATION MACHINE MOVEMENT WITH A MULTIMODE OSCILLATION SPECTRUM

I. Nazarenko
Doctor of Technical Sciences, Professor, Head of Department*
E-mail: i_nazar@i.ua

V. Gaidaichuk
Doctor of Technical Sciences, Professor, Head of Department
Department of theoretical mechanics**
E-mail: viktor_gaydaychuk@bigmir.net

O. Dedov
PhD, Associate Professor*
E-mail: dedovvcbk@ukr.net

O. Diachenko
Postgraduate student*
E-mail: sanyadrg@gmail.com

*Department of machinery and equipment of technological processes**
**Kyiv National University of Construction and Architecture
Povitroflotskyi ave., 31, Kyiv, Ukraine, 03037

1. Introduction

Trends in the development of the construction industry are aimed at reducing the cost of energy resources with high quality of technological processes. In particular, the process of compacting mortars is one of the most responsible, since the future strength of the structure and the durability depend on it. In the process of wide application of the technology of frame-monolithic construction, the problem of forming large-sized products arose. Modern vibration technology of the construction industry does not fully meet the specified challenges of the industry. This is due to several reasons:

– the lack of specialized vibration machines for the formation of similar products;

– significant discrepancy between the existing physical and mathematical models describing the movement of the vibration machine and the concrete mix compaction;

– the use of ineffective regimes and parameters that do not take into account the actual processes taking place in the machine and in the sealing mixture;

– the lack of directional use of the vibration power due to the contribution of not only the main a and higher harmonics into the workflow.

The solution to this problem lies in the search for increasing the efficiency of the working process of the concrete mix compaction. Identification of new phenomena in the operation of compacting machines and their recording in the modeling of work processes. Perfection of models, which adequately correspond to real conditions of movement of the vibration machine.

2. Literature review and problem statement

An investigation of the dynamics of machines for compacting concrete mixtures has been devoted to a number of papers [1–3]. In [1], the design scheme of the machine is considered a system with discrete parameters, and the environment is modeled by a system with distributed parameters. The approach itself deserves attention. At the same time, the distribution of the amplitudes and frequencies of oscillations along the surface of either the frame or the shape is investigated. In [2], the cassette vibration machines used for formation of large products are considered. The movement of a concrete mix is studied without evaluating the effect of the shape-forming cassettes. In [3], oscillations of a vibration platform of a frame structure in a horizontal plane are investigated. However, a simplified design model of the machine is used in the form of a discrete one, and the concrete mix is taken into account by the empirical coefficient. Effects of wave processes aren't considered. Such method is reliable only within the limits of the performed studies and identical in design and parameters of the vibration machines. Thus, in the considered studies, there are no studies of the amplitude-frequency spectra of oscillations of vibration machines and simplified calculation models are applied. Application of continual models is more effective. It makes possible to take into account the propagation of waves, both in the construction of the frame of the vibration machine, and in the mixture compaction. This approach is the basis for determining the real distribution of the amplitudes and frequencies of oscillations and applying multi-mode effects. In [4], dynamic processes are considered on the basis of the energy balance. This approach is advisable to apply for models with a small number of unknowns in the study of nonlinear dynamic processes. Researches in [5] deserve attention. The authors apply the methodology of transformation of a discrete elastic-plastic model to a continual model, the equation of which is solved analytically as well as numerically. However, it is worth noting that this approach has been limited and the accuracy of the calculation results is low. An important aspect in the overall modeling system is elastic rubber elements. Studies of resonant phenomena in such elastic materials are discussed in [6] and [7]. The use of simple and rapid calculations based on energy methods [6] can be applied to rubber frame supports. The results of the same study [7] can be used to select elastic supports for a predetermined mode of oscillation. In addition to the static load, which are given in [8], it is necessary to take into account dynamic [9], in particular, to verify the obtained model. For these studies in the metal elements of the frame vibration control is also an important issue of synchronization of oscillation sources. Consideration of the mutual placement of the exciter and the dissipative properties indicated in [10] is an important aspect of modeling. It is necessary to refine the laws of variation of dissipative properties. An important factor in choosing a model is the acceptance of assumptions regarding the possible variation of model parameters in time under dynamic load. In [11], the model of plate movement under wave excitations is considered and a calculation method is proposed. In [12, 13] particular cases of rational constructions of planar elements are given, and optimization methods can be applied in the study of small and simple products. The stress-strain state of structures in a nonlinear analysis is rather complicated and requires a lot of resources, as indicated in [13]. Therefore, for complex computational models, the author recommends the use of finite element modeling. This is especially true for determining the qualitative and quantitative values of the loads [14]. In this case, structural differences in the form of holes, cutouts in the construction are possible, and also affect the model [15].

3. The aim and objectives of research

The aim of research is determination of the multimode vibration spectrum of the shape-forming surfaces of a vibration plant for intensifying the working process of compacting concrete mixes.

To achieve this aim, the following tasks are identified:

– substantiation and development of the design scheme of a vibration machine with a multimode oscillation spectrum;

– research and definition of the basic forms, amplitudes and frequencies of oscillations of a frame construction of a vibration machine;

– determination of the amplitude-frequency oscillation spectrum for creation of a new highly efficient vibration machine.

4. Technique for investigation of vibration machine movement with a multimode oscillation spectrum

The mathematical model of the construction of a vibration system is built on the following assumptions. The machine frame is a shaping surface and modeled by distributed parameters. Metal structures perceive only elastic deformations. The concrete mixture, which is on the shape-forming surface, is modeled by a system with distributed parameters. In the equations of movement of the general system «vibration machine – concrete mixture», the constructive mass and mass of the concrete mixture are taken into account. The mass of the concrete mixture is taken into account by the wave coefficient on the basis of the method given in [1]. Modeling of the working process of the vibration machine is performed on the basis of the finite element method using the MSC.NASTRAN computation complex (MSC.Software, Germany).

When creating a computer model of the investigated system, principles are applied that will ensure the simplicity and adequacy of the model, as well as the possibility of further research – the solution of other types of problems.

The study is carried out in three stages.

At the first stage, a static analysis of the stress-strain state of the structure under the action of all external forces (Static Analysis) is carried out in the nonlinear theory. At the second stage, based on the method of modes analysis, the basic shapes and frequencies of oscillations are determined. The third stage of the research is performed using dynamic analysis in the implementation of one of the forms of vibration (Transient Analysis), which is defined in the model analysis. Dynamic analysis determines the oscillation amplitude of the structure in different regions (sections) and estimates the change in the amplitude-frequency spectrum of the vibrational action.

5. Results of studies of the of vibration machine movement with a multimode oscillation spectrum

5. 1. Substantiation and development of the design scheme of a vibration machine with a multimode oscillation spectrum

The vibration machine is a frame structure that simultaneously functions as a mold for a concrete mix and consists of a welded box-shaped frame that is mounted on rubber elastic supports on a concrete foundation. The vibration machine is equipped with four, not symmetrical, installed pneumatic centrifugal exciters of high-frequency oscillations. On the frame are fixed two fixed beads and one movable board. To study the vibration machine, a geometric 3D model is created, on the basis of which a calculated finite-element model is developed (Fig. 1).

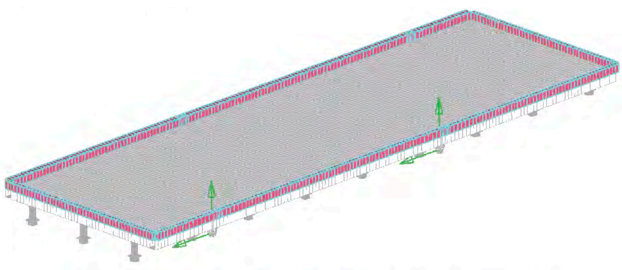

Fig. 1. Calculation of 3D model of the vibration machine for the formation and compaction of concrete mixtures

The finite element model is compiled by approximating all load-bearing elements of the frame by beam end elements, elastically deformed under the action of longitudinal force, bending moments in two planes and torque. The structures are modeled by finite elements of BEEM type: main beams – twin channel No. 20 (Fig. 2, *a*) secondary beams – channel No. 14 (Fig. 2, *b*); non-detachable beads – cross-section in Fig. 2, *c*; removable board – paired channel No. 16 (Fig. 2, *d*).

Shape-forming surfaces are modeled by plane elements of a given thickness of PLATE type, and elastic rubber supports with BEEM type elements. Under the condition of fixing the model, jamming is assumed from the movements and rotations of the elastic supports, on which the metal frame of the machine frame is based.

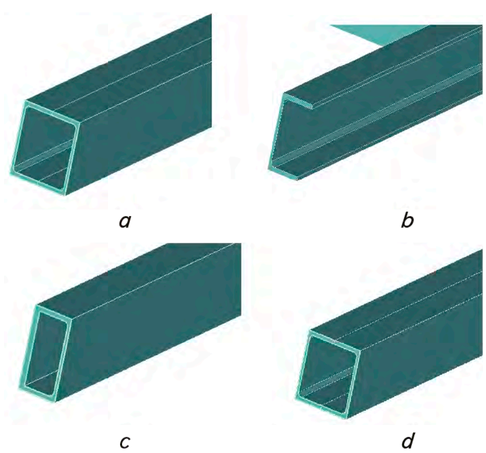

Fig. 2. Volumetric elements of the frame of the vibration machine for the formation and consolidation of concrete mixtures: *a* – the main beams of the frame; *b* – secondary beams of the frame; *c* – non-detachable sides; *d* – removable board

5. 2. Investigation and determination of the basic shapes, amplitudes and frequencies of vibrations in the frame structure of the vibration system

For the research and display of results, the settlement complex MSC.NASTRAN (MSC.Software, Germany) is used. The calculation of the frequencies and modes of oscillations is carried out in two stages. At the first stage, the static pre-stressing of the structure by all active forces is performed in the nonlinear theory, then the modal analysis (Modes Analys) is performed in the second stage, the result of which is the calculation of the frequencies and, respectively, the vibration modes.

The calculations are carried out with the aim of determining simple and more complex modes of oscillations. The choice consists in the possibility of implementing modes of operation with higher levels of energy transfer to the process medium. At the first stage, the investigations are carried out with the load of the structure only under the action of the force of gravity of the elements. Analysis of the calculation results shows that for installation of the first frequency ($f1=18.79$ Hz), the elements transfer transversely along the small horizontal axis of the system (Fig. 3), which is explained by the low stiffness of the supports in this direction.

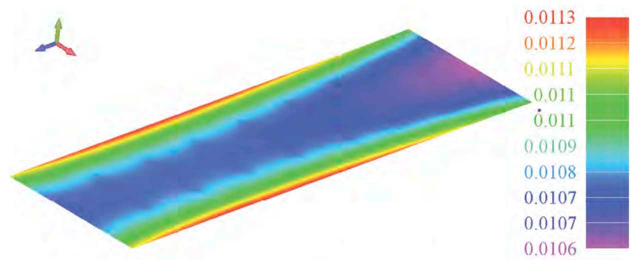

Fig. 3. The first form of oscillations of shape-forming surfaces (oscillation frequency 18.79 Hz)

In oscillations of the second frequency ($f1=18.89$ Hz), the waveform is similar to the first, but the movement is in the longitudinal direction (Fig. 4).

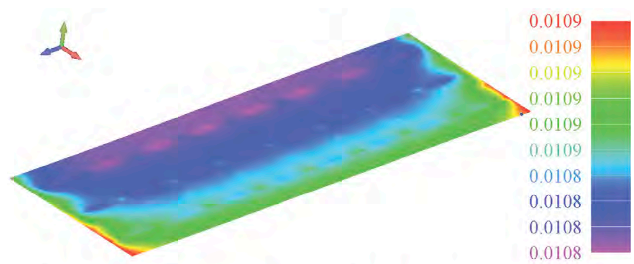

Fig. 4. The second form of oscillation of shape-forming surfaces (oscillation frequency 18.89 Hz)

In the oscillations of the third frequency ($f1=19.71$ Hz), the rotational movements of the system relative to its center prevail (Fig. 5).

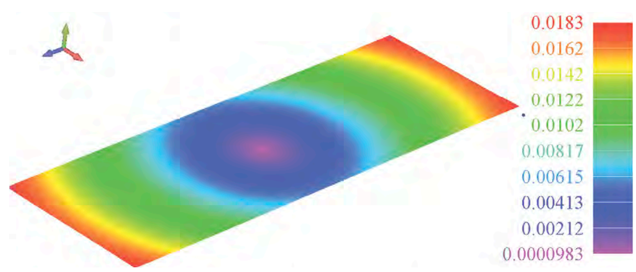

Fig. 5. The third form of oscillations of shape-forming surfaces (oscillation frequency 19.71 Hz)

Oscillations at higher frequencies are mainly due to vertical movements of the plate structure, connected to horizontal and twisting oscillations. For example, at a frequency of oscillations $f1=43.77$ Hz (Fig. 6), the forming surfaces of the plate perform predominantly vertical oscillations. The operating mode of the compaction equipment according to the following form of oscillation is certainly effective, but it is worth paying attention to the presence of two zones located at 1/4 of both edges of the surface. In these zones there are no oscillations (the amplitude of oscillations in these zones is equal to 0) and there is no process of compacting the concrete mixture.

Such forms of oscillations of the working element do not satisfy the conditions for effective compaction of the concrete mixture. Therefore, to realize the compaction, it is advisable to use complex, multifrequency oscillations, according to which zero values of the vibration amplitudes are absent.

Further investigations are carried out in the direction of searching for regimes of the corresponding shape of oscillations of the working member, along which zero amplitudes are absent. The mode of realization of internal resonant phenomena of directly forming plates is determined.

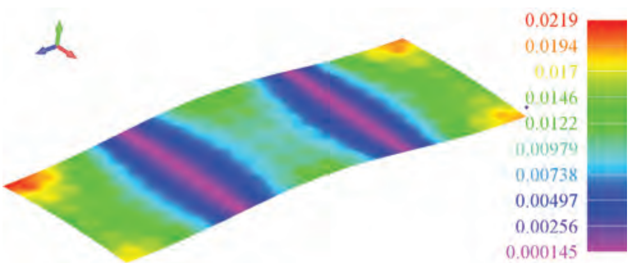

Fig. 6. Type of oscillations of shape-forming surfaces (oscillation frequency 43.77 Hz)

At the next stage of the research, the vibration mode of the shape-forming frames is realized in the frequency range of 170–190 Hz, realized using pneumatic oscillators. A number of forms of oscillations are obtained by processing the results of investigations. As an example, Fig. 7 shows the typical form of oscillations. Analysis shows a significant difference from the above. The second conclusion is that with this form of oscillation, for example, at a frequency $f1=182.50$ Hz (Fig. 7) there is a complex waveform. The realization of this form of oscillations provides resonance to the shape-forming plates, and the vibrations of the bearing structures of the vibration plant are insignificant.

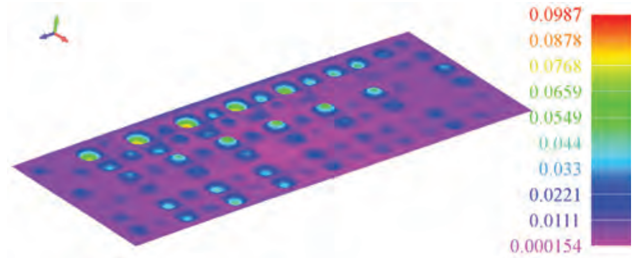

Fig. 7. Type of oscillations of shape-forming surfaces (oscillation frequency 182.50 Hz)

Comparing the obtained forms of oscillations, it is possible to note the following. The implementation of operating modes at high frequencies will give a new effect when compacting concrete mixes. This result is also obtained due to the resonance of the structural elements that have direct contact with the processed medium.

Due to this compaction mode, it will be possible to transfer the maximum energy from the working element to the process medium. A decrease in the energy capacity of the compaction process is also ensured due to the contribution of higher harmonics. In this case, the vibration amplitude of the bearing structures of the vibration set decreases.

5. 3. Determination of the amplitude-frequency spectrum of oscillations for the creation of a new highly efficient vibration machine

The study of the created system in the simulation of the dynamic load is realized using nonlinear time analysis. Dynamic load is four forces that vary in time with a frequency of 182.5 Hz in sinusoidal law. Forces are applied to longitudinal sides in pairs with displacement in the longitudinal direction (Fig. 1).

To determine the vibration amplitude of metal structures, shape-forming surfaces (plates), typical cross-sections in the transverse and longitudinal directions are identified (Fig. 8). The distribution of vibration amplitudes over the surface of the plate is analyzed from the values of the displacement of the nodes of the finite element grid in these sections.

Thus, the dependence of the vertical displacements in the transverse direction of the cross sections 1–1, 2–2, 3–3, 4–4, 5–5, 6–6 of the plate width is shown in Fig. 9, 10.

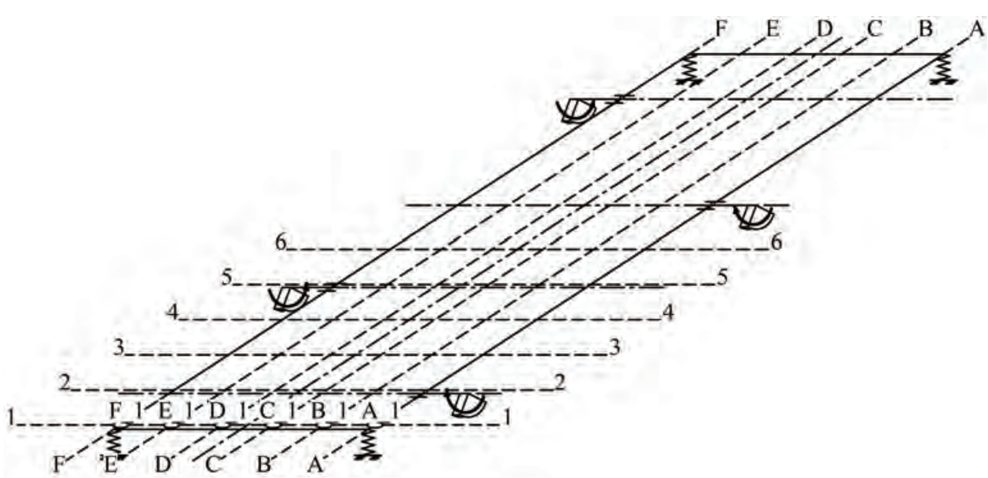

Fig. 8. The placement scheme of typical sections of nodes of a finite element grid

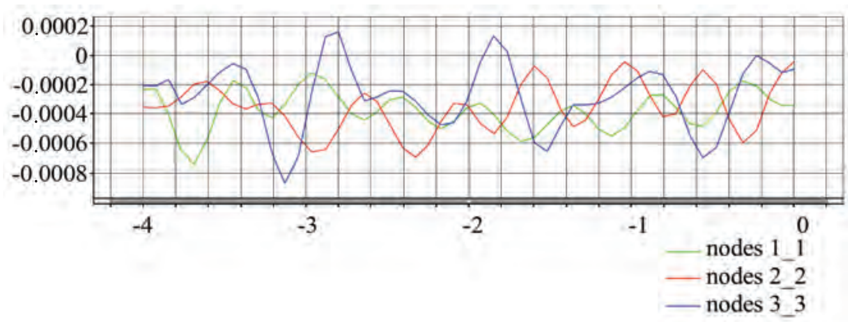

Fig. 9. Plate movement vertical for nodes in sections 1–1, 2–2, 3–3, (Time 0.3832)

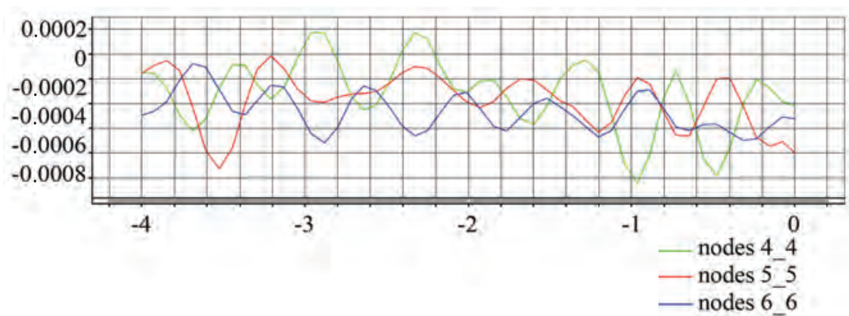

Fig. 10. Plate movement vertical for nodes in sections 4–4, 5–5, 6–6, (Time 0.3832)

In the first case (Fig. 9), the diameter 1–1 moves in opposite phase with a cross section of 2–2 and 3–3. This indicates the presence of a wave with a corresponding propagation length.

The oscillograms of the oscillations of the plate differ somewhat in shape from the oscillations. In phase, vertical displacements in the transverse direction of sections 1–1, 2–2, 3–3, are somewhat different from the transverse direction of sections 4–4, 5–5, 6–6.

The distribution of vertical nodal displacements in the longitudinal direction (the plate length – 12 m) of sections A–A, B–B, C–C, D–D, E–E, F–F (Fig. 11, 12) differs significantly from those considered above, both in magnitude and in nature of the change.

The longitudinal displacements of nodes along the length of the plate of sections A–A, B–B, C–C, D–D, E–E, F–F (Fig. 13, 14) change their numerical values monotonically without pronounced wave phenomena.

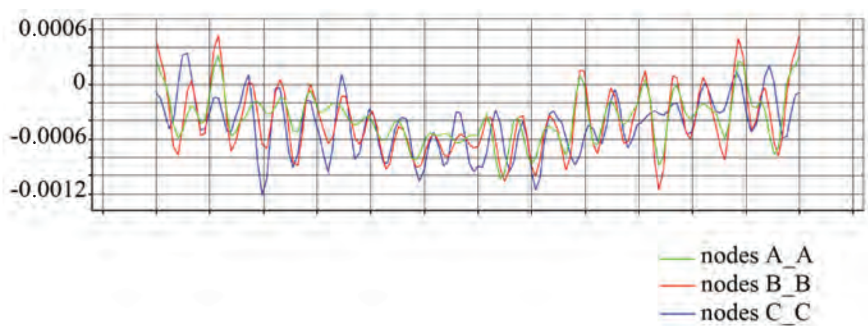

Fig. 11. Plate movement vertical for nodes in sections A–A, B–B, C–C, (Time 0.3832)

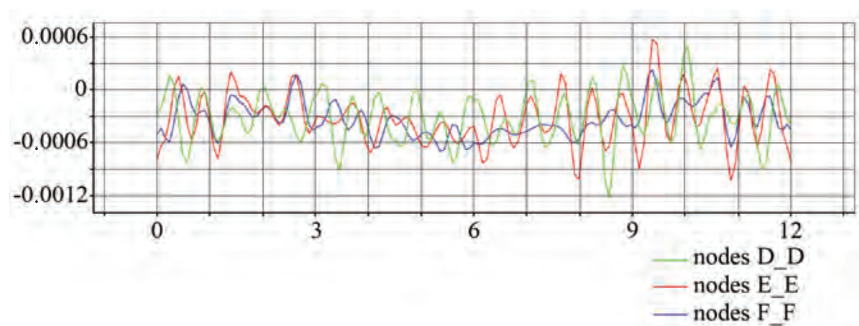

Fig. 12. Plate movement vertical for nodes in sections D–D, E–E, F–F (Time 0.3832)

The displacement in the nodes along the length of the plate in the sections A–A, B–B, C–C, D–D, E–E, F–F (Fig. 15, 16) has two peculiarities of their variation. The first is minor changes in numerical values between themselves. The second feature is the presence of characteristic peak bursts in the E–E section.

The evaluation and analysis of plate movements at six points of section 1–1 deserves special attention (Fig. 17–22). The need for reduction is due to the fact that the displacement of six points does not significantly change in the range 0–0.18 s in the magnitude of displacements in the transverse and longitudinal directions due to the force action of the vibrators (Fig. 8). Further in time, 0.45 to 0.78 s, transverse movements experience a certain influence of the complex shape of the vibrators. The change in vertical displacements, as can be seen from the graphs in the time bands 0.15–0.31 s and 0.51–0.66 s (Fig. 17, point A), manifests itself in the presence of a subresonant mode of oscillation. Another example is the presence of a super-resonance regime in the interval 0.40–0.72 s (Fig. 18, point B).

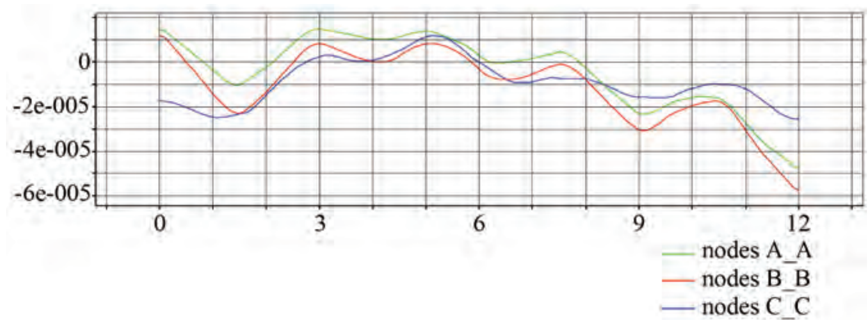

Fig. 13. Plate movement longitudinal for nodes in sections A—A, B—B, C—C, (Time 0.3832)

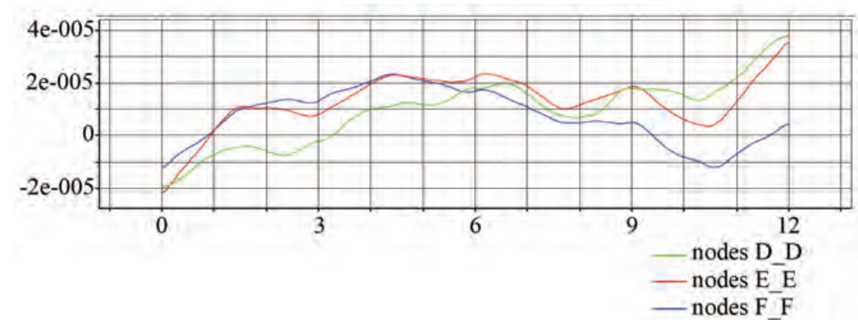

Fig. 14. Plate movement longitudinal for nodes in sections D—D, E—E, F—F, (Time 0.3832)

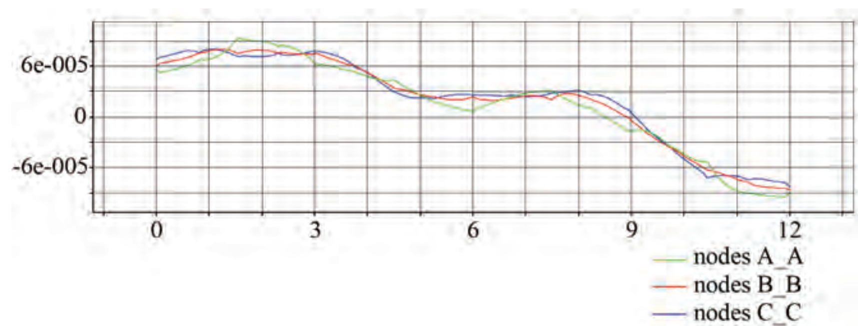

Fig. 15. Plate movement transverse for nodes in sections A—A, B—B, C—C, (Time 0.3832)

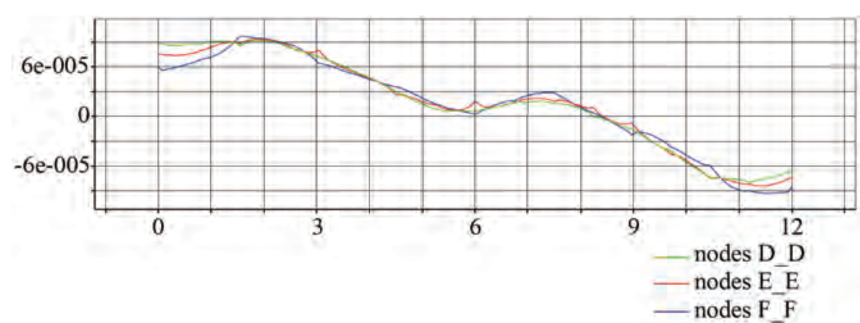

Fig. 16. Plate movement transverse for nodes in sections D—D, E—E, F—F (Time 0.3832)

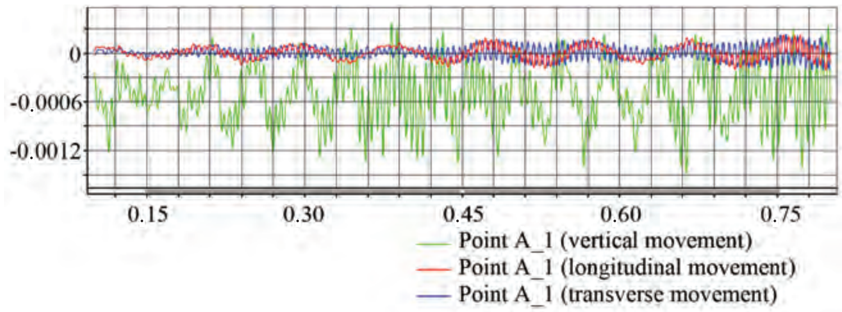

Fig. 17. Plate movement at the point A–1

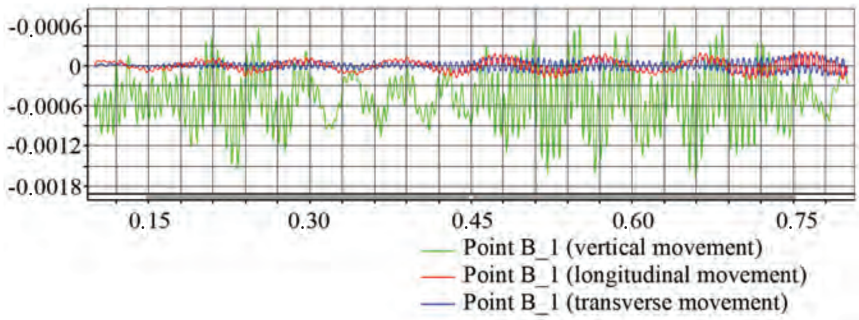

Fig. 18. Plate movement at the point B–1

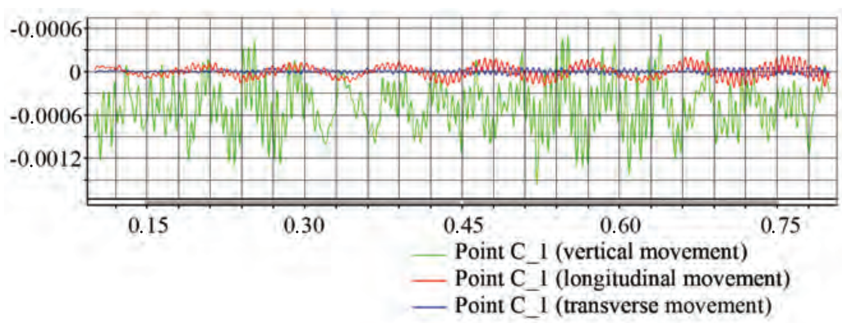

Fig. 19. Plate movement at the point C–1

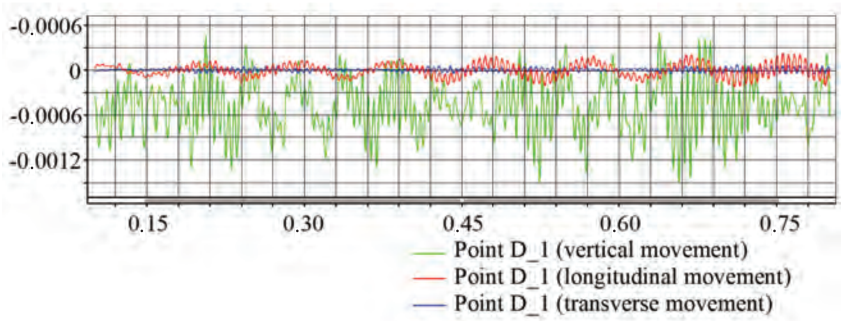

Fig. 20. Plate movement at the point D–1

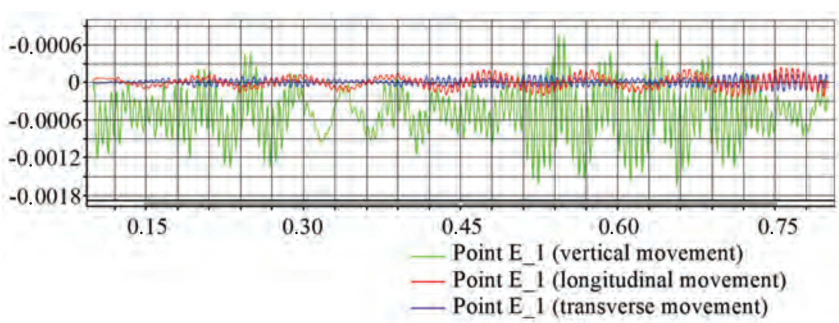

Fig. 21. Plate movement at the point E–1

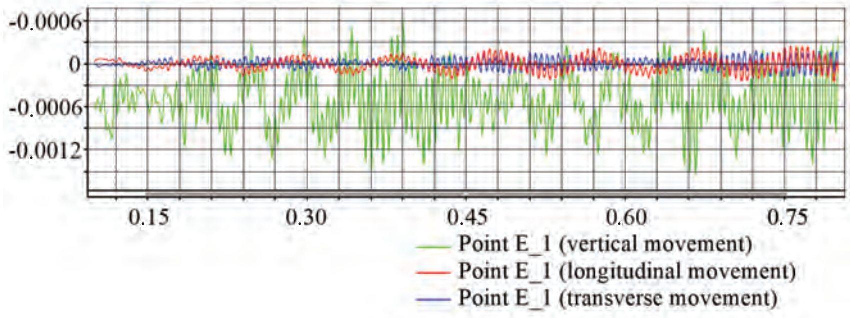

Fig. 22. Plate movement at the point F—1

It is worth noting the novelty of the implementation of the regimes. This is especially evident at the points shown in Fig. 19–22. The carried out researches and the analysis of results have been put in a basis at designing shape-forming frame and a substantiation of places of an arrangement of vibrators on the form.

6. Discussion of the research results of a vibration machine movement with a multimode oscillation spectrum

The research results indicate the presence of complex vibration forms of the shape-forming frame of the vibration machine. This is a fundamentally new result. It consists in the fact that in calculations of vibration machines it is suggested to take into account not only the output numerical values of the amplitude-frequency regime of the exciter of oscillations, but also the shape of the oscillations, which is realized in this case. Shape-forming frames, when in contact with the mixture, generate oscillations different from the fundamental ones. A multimode spectrum of the mixture oscillations is realized. As a result, the energy consumption is reduced by 25 % and the process of forming a flat plate is 30 %. The presence of multimode is confirmed by the graphs of the vibrational record (Fig. 17 point A, Fig. 18 point B). The use of such effects depends on the specific design scheme of the vibration machine and the type of future product. The disadvantages of this study include the use of specific volume elements of the frame of the vibration machine (Fig. 2). In further studies, it is necessary to optimize the structure of the elements by the criterion of minimizing stresses and deformations in these elements. This study has limitations in terms of considering a specific scheme for the location of exciters. More research is needed in other schemes of application of the force of exciters. However, it may be difficult to register landslide deformations of the compaction mixture. It is the landslide deformations, since the numerical values of such deformations will be distorted by spatial oscillations. In this case, their numerical values may be unreliable. A possible solution to such problems is evaluation of the effectiveness of applying external forces in different directions by applying the photographic fixation of the process of settling the mixture during its compaction. This method will give a qualitative assessment of the process parameters and fix the compaction time. The most effective method is additional experimental experiments to determine the strength of the molded specimen on special presses. Comparing the strength values obtained under different compaction regimes, one can evaluate their effect and establish rational parameters.

According to the scheme, the exciter frequency, the structure of the frame, and the points of application of the force vibratory actions to it are important. For products of smaller dimensions (for example, paving plates), different approaches to the implementation of multimode resonance oscillations are needed. Such studies are planned as a continuation of the topic under consideration on the idea of direct energy transfer to the product. The ideology of the implementation of regimes can be successfully applied in road construction for the construction of concrete roads, the construction of which is gradually expanding in various states of the European Union, including Ukraine.

7. Conclusions

1. The design scheme of a vibration machine with a multimode oscillation spectrum of shape-forming surfaces is substantiated. A finite-element model is created by approximating all load-bearing elements of the frame with beam end elements, and the shaping surfaces by plate elements.

2. The basic forms and frequencies of oscillations that are realized at 18.79 Hz, 18.89 Hz and 19.71, respectively, are determined. The form of oscillations with a frequency of 182.5 Hz is studied, at which there is a complicated form of oscillations. The realization of this form of oscillations provides resonance to the shape-forming plates, and the vibrations of the bearing structures of the vibration plant are insignificant.

3. The amplitude-frequency spectrum of the plant oscillations at the excitation frequency of 182.5 Hz is determined, at which the oscillation amplitudes are realized 0.0002...0.0005 m. The investigated mode of oscillations of the computational model is a confirmation of the multimode and can be applied for the design of a highly efficient vibration machine.

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