UKRAINIAN JOURNAL OF MECHANICAL ENGINEERING AND MATERIALS SCIENCE
Vol. 9, No. 3, 2023

Pavlo Maistruk

Department of Robotics and Integrated Mechanical Engineering Technologies, Lviv Polytechnic National University, 12, S. Bandery str., Lviv, Ukraine, e-mail: pmaystruk@gmail.com, ORCID 0000-0003-0662-1935

OPTIMIZATION OF THE SHAPE AND DIMENSIONS OF THE CONTINUOUS SECTION OF THE DISCRETE-CONTINUOUS INTER-RESONANCE VIBRATING TABLE

Received: August 22, 2023 / Revised: September 28, 2023 / Accepted: September 26, 2023

https://doi.org/10.23939/ujmems2023.03.010

Abstract. Energy-efficient technologies are an important aspect of the development of mechanical engineering. Therefore, the creation of highly efficient vibration technological equipment is an urgent task. There are discrete-continuous inter-resonance vibration machines that have high values of dynamic amplification of oscillations. Rectangular plates or rods are used as the reactive mass of such vibrating machines. However, the rectangular shape of the plate may not be the optimal shape for achieving maximum energy efficiency. To conduct experimental studies of alternative plates with a variable cross-section to determine the optimal shape of the reactive mass of the vibration machine. Methodology. The selection of alternative options of plates with a variable cross-section, which would satisfy the necessary conditions of fastening and the value of the natural frequency of oscillations, was carried out. Experimental studies were carried out on a sample of an inter-resonance vibrating table. The value of the power supply voltage at which loads of different masses were separated from the working body of the vibrating table for each of the plate samples was compared. Findings (results) and originality (novelty). For the first time, experimental studies of the energy efficiency of inter-resonance vibration machines with plates with a variable cross-section installed as a reactive mass were conducted. It was found that the rhomboid shape of the plate is optimal when using it as a continuous section in a vibration machine with an electromagnetic drive. It was determined that the use of diamond-shaped plates as the reactive mass of the vibrating machine can improve the energy efficiency of the inter-resonance vibrating equipment. For further analysis of plates with a variable cross-section as a reactive mass of an inter-resonance vibration machine, it is necessary to calculate and compare their lumped inertia-stiffness parameters.

Keywords: continuous section, discrete-continuous oscillating system, inter-resonance vibration machine, variable cross-section, natural frequency of oscillations.

Introduction

Implementation of energy-efficient technical solutions is an important aspect of technological equipment modernization. In the field of vibration technologies, single-mass and dual-mass vibration machines, which are easy to design and operate, are the most common. Despite the significantly better energy efficiency of three-mass discrete inter-resonance vibration machines, they have not become widely used. The methodology of creating three-mass discrete vibration machines has evolved into the introduction of a new class of structures, the basis of which are discrete-continuous oscillating systems. Therefore, there is a need to develop calculation and design methods for this type of vibration equipment.

Problem statement

To ensure energy-efficient inter-resonance modes of operation of vibrating equipment, it is necessary that the oscillating masses of the system have certain values of inertia-stiffness and frequency parameters [1]. The disadvantage of highly efficient inter-resonance oscillating systems is the need for ultra-light reactive mass. The use of complex and overall structures is categorically impossible. Therefore, it is most expedient to use continuous sections as reactive mass [2]. The continuous section, which is a flexible body and is hinged in the intermediate mass of the vibrating machine, optimally combines inertial and rigid parameters. A study of hybrid discrete-continuous inter-resonance oscillating systems was carried out, where a rectangular plate was used as a continuous section. However, an assumption was made [3] that the rectangular shape of the continuous section considered in the works [2, 4] is not capable of maximally realizing the dynamic potential of such systems.

Analysis of modern information sources on the subject of the article

The analysis of plates and beams is a common topic of research in various fields of science [5–8]. Usually, these research data refer to plates of rectangular shape. However, in works [9–11] plates and shells with a variable cross-section are considered. In particular, in [9], differential equations for free bending vibrations of straight beams with a variable cross-section were solved using Bessel functions. The article [10] investigates the free oscillations of a multilayer symmetric sandwich beam with a variable cross-section, which is based on the Winkler model and is made of functionally graded materials. In article [11], in contrast to [10], plates with a variable cross-section are attached to rigid rotating disks. V. De Biagi et al. [12] considered optimization approaches for plates with variable cross-section for applications in civil engineering. In this work, the displacement of the beam was obtained from the solution of the differential equation of the elastic line, taking into account the variability of inertia and neglecting any shear contribution. The Cross-Axis Flexural Pivot, containing beams whose width and thickness vary along the axis according to linear or parabolic functions, was analyzed in [13]. At the same time, different types of beam loading were considered, each of which was represented as a chain of two Beam-Constraint Model elements. As an alternative to beams with a variable cross-section, an experimental study of the influence of round holes on the strength of rectangular beams was carried out in the article [14]. These experiments took into account the change in the position, number, and diameter of the holes. The article [15] presents a one-dimensional finite-element model for calculating the nonlinear dynamic behavior of thin-walled composite beams with variable cross-sections under an arbitrary external dynamic load.

Statement of purpose and tasks of research

Considering the need to implement energy-efficient discrete-continuous inter-resonance vibration machines, the purpose of this work is to find the optimal form of the continuous section to be used in the vibration table.

The main material presentation

To achieve the goal, a solid-state model of a discrete-continuous inter-resonance vibrating table with an electromagnetic drive is proposed, shown in Fig. 1. The vibrating table is designed for compaction of concrete mixtures, which is used in the formation of concrete and reinforced concrete products. It can also be used in foundry production to compact the molding mixture during the mold-making process. Due to vibration, when the remaining air and water come to the top, the mixture fills the cavities, ensuring high strength of the products.

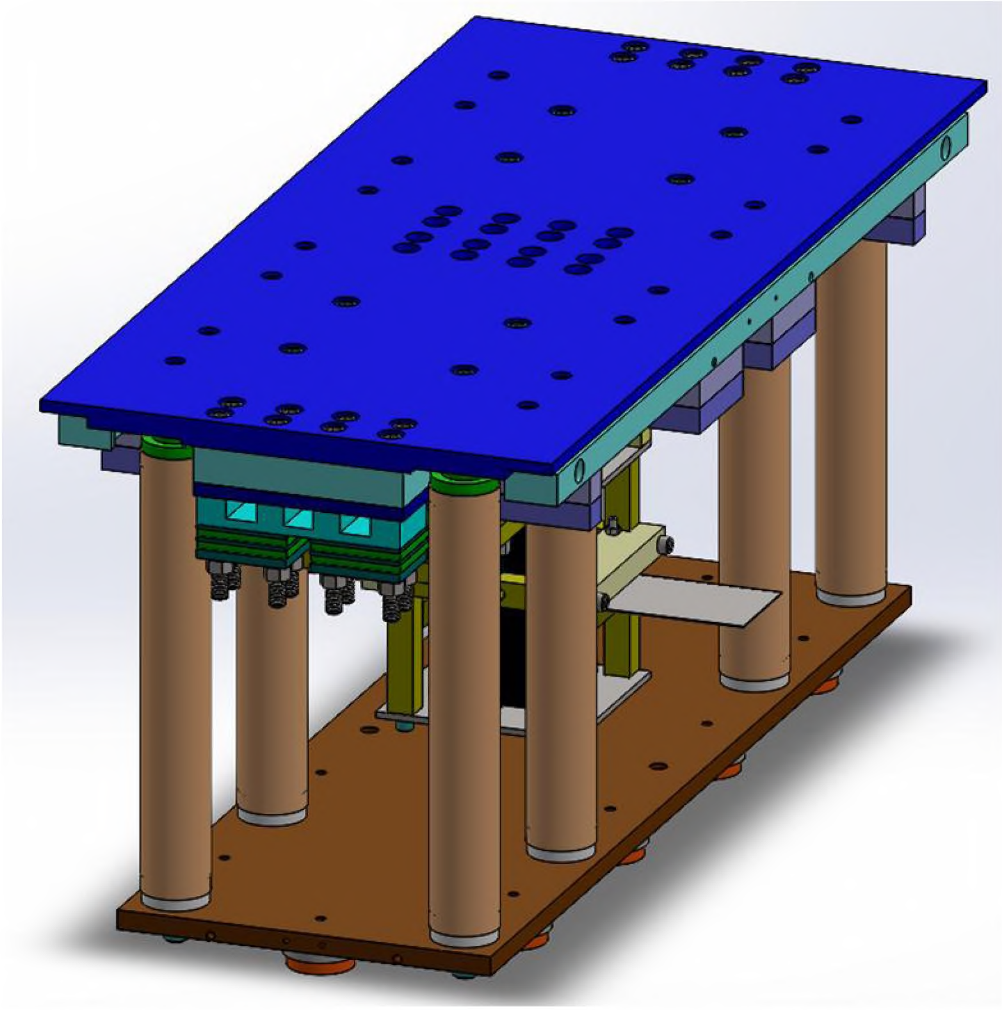

Fig. 1. The solid-state model of the discrete-continuous inter-resonance vibrating table

The vibrating table with an electromagnetic drive contains the first 1 and the second oscillating masses 2, connected to each other by an elastic unit 4. The third oscillating mass 3 is presented in the form of a plate and is connected to the second non-deformable oscillating mass 2 by means of hinge supports 5. The third oscillating mass 3, being a flexible body (continuous section), is endowed with both inertial and rigid characteristics. The entire structure is symmetrical and is attached to the fixed base 6 with the help of vibration isolators 7. The exploded view of the discrete-continuous interresonance vibrating table is shown in Fig. 2.

The most important parameter of a continuous section during the synthesis of a discrete-continuous system is its natural frequency of oscillations ω_n. Correct selection of the natural frequency that makes it possible to obtain a highly efficient operating mode of discrete-continuous inter-resonance vibration machines.

To ensure the inter-resonance mode of operation of the vibrating machine, it is necessary that the first natural frequency of oscillations of the continuous section is close in value to the partial frequency of the reactive mass of the same discrete oscillating system $(\omega_n \approx \omega_p)$. The partial frequency ω_p can be set using an expression [1]:

$$\omega_p = \sqrt{c_{23} / m_3}, \tag{1}$$

where c_{23} – stiffness of the elastic unit connecting the intermediate and reactive masses of the identical discrete oscillating system; m_3 – the inertial parameter of the reactive mass of the identical discrete oscillating system.

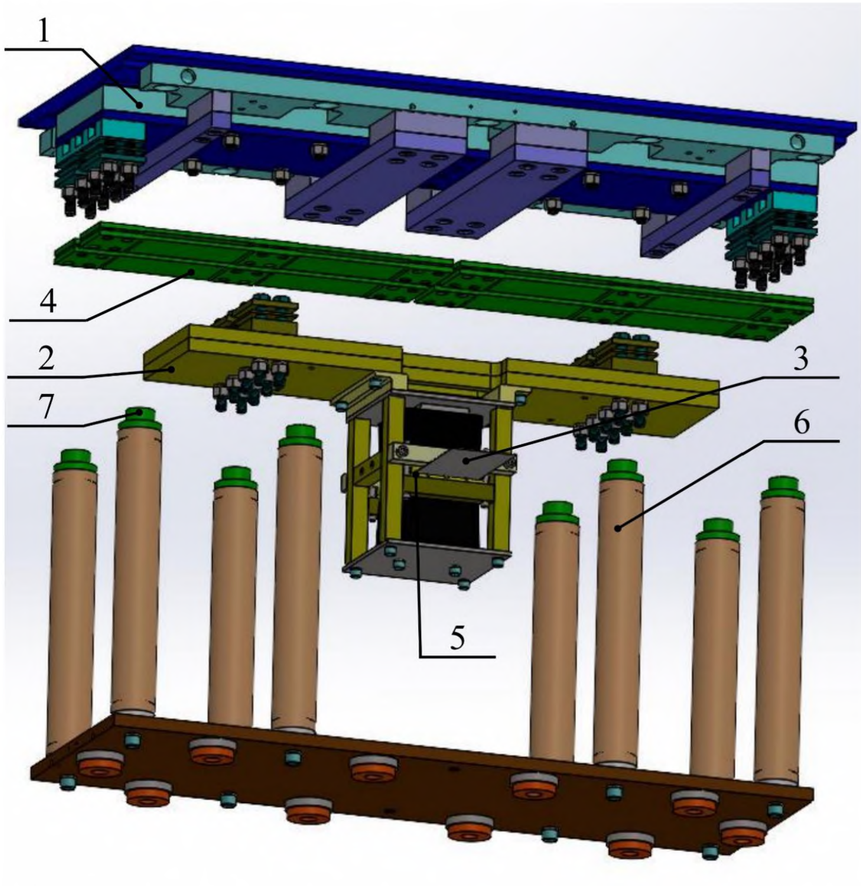

Fig. 2. Exploded view of the discrete-continuous inter-resonance vibration table:
1 – active mass; 2 – intermediate mass; 3 – continuous section (plate); 4 – elastic unit connecting active and intermediate masses; 5 – hinged supports; 6 – fixed base; 7 - vibration isolators

It is technologically necessary that the operating mode of the vibrating table is at a frequency of forced oscillations of around $\nu = 50\,Hz$. In this case, the electromagnets, which are the drive of this vibration machine, can be powered by the electrical network. Taking into account the values of the active $m_1 = 118\,kg$, and intermediate $m_2 = 45\,kg$ oscillating masses, and using the well-known algorithm for calculating a discrete three-mass mechanical oscillating system [1], the range of permissible values of the reactive mass m_3 can be found from inequality:

$$0 < m_3 < \frac{-m_2 (m_1 + m_2)\left(1 - \Lambda^2\right)^2}{m_2 \left(1 - \Lambda^2\right)^2 - 4 m_1 \Lambda^2}, \tag{2}$$

where $\Lambda = \dfrac{\Omega_{n1}}{\Omega_{n2}}$ – the ratio of natural circular frequencies of the system. Values of natural circular frequencies are $\Omega_{n1} = 306\,rad/s$ and $\Omega_{n2} = 333\,rad/s$. By substituting the data into Eq. (2), we determine that the reactive mass can be in the range $0 < m_3 < 0{,}447\,kg$. Given the above conditions, we accept $m_3 = 0{,}36\,kg$.

The stiffness c_{23} can be found from expression [1]:

$$c_{23} = \frac{(m_1 + m_2 + m_3) m_2 \Omega_{n2}^2 \left(\Lambda^2 + 1\right) - \mathrm{H}}{2 \cdot (m_2 + m_3)(m_1 + m_2 + m_3)} \cdot m_3, \tag{3}$$

and with the available system parameters is $c_{23} = 3{,}5 \cdot 10^4\, N/m$.

By substituting the values of mass m_3 and stiffness c_{23} in Eq. (1), the value of the partial frequency of the reactive mass is obtained $\omega_p = 311{,}8\ rad/s$ or $\nu = 49{,}6\ Hz$. Therefore, the natural frequency of the plate should be within $\omega = 49...49{,}6\ Hz$.

The second important parameter of the continuous section is the dimensions for its fastening in the intermediate mass. The scheme of fixing the continuous section (plate) is shown in Fig. 3. Since the plate is fixed in the intermediate mass with the help of a hinged connection at four points along the perimeter, the required dimensions for fixing should be $L_2 = 176\ mm$, $h = 87{,}5\ mm$.

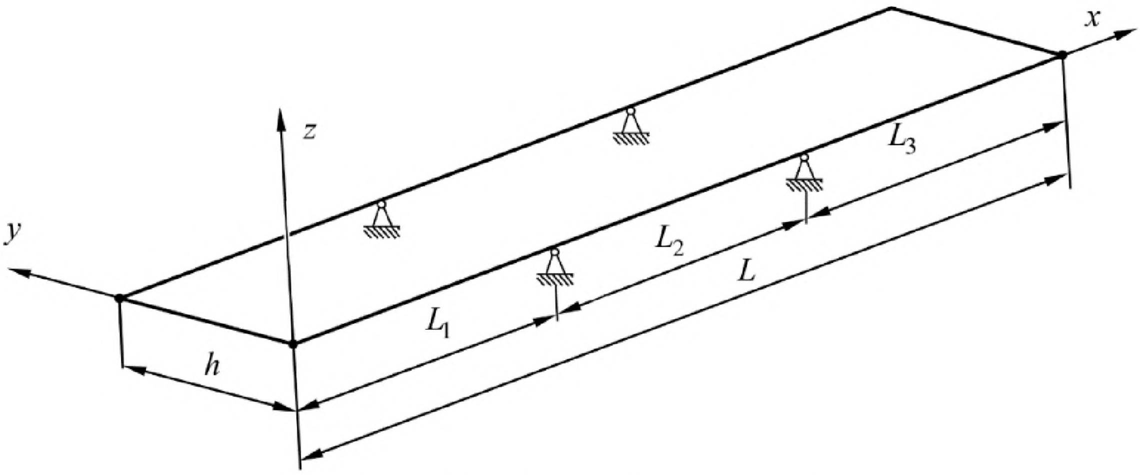

Fig. 3. The calculation scheme of the fastening of the plate

From Fig. 3, it can be observed that the fastening of the elastic plate is quite specific. This is related to the design of the intermediate mass and the location of the hinges.

To optimize the shape of the continuous section of the vibrating table, consider a proper rectangular plate, as well as the following alternative options for plates with a variable cross-section, shown in Fig. 4: parabolic concave plate; parabolic convex plate; X-shaped plate; diamond-shaped plate.

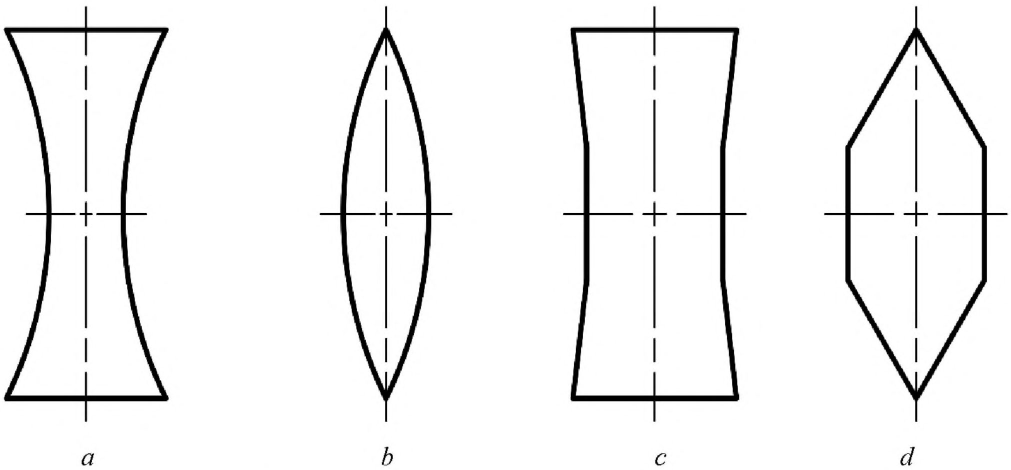

Fig. 4. The investigated alternative types of plates: a – parabolic concave; b – parabolic convex; c – X-shaped; d – diamond-shaped

All investigated types of plates must fulfill the above two conditions. The calculation of natural frequencies of plate oscillations was carried out in the ANSYS Workbench software product. The results of the calculations are presented in Fig. 5–8.

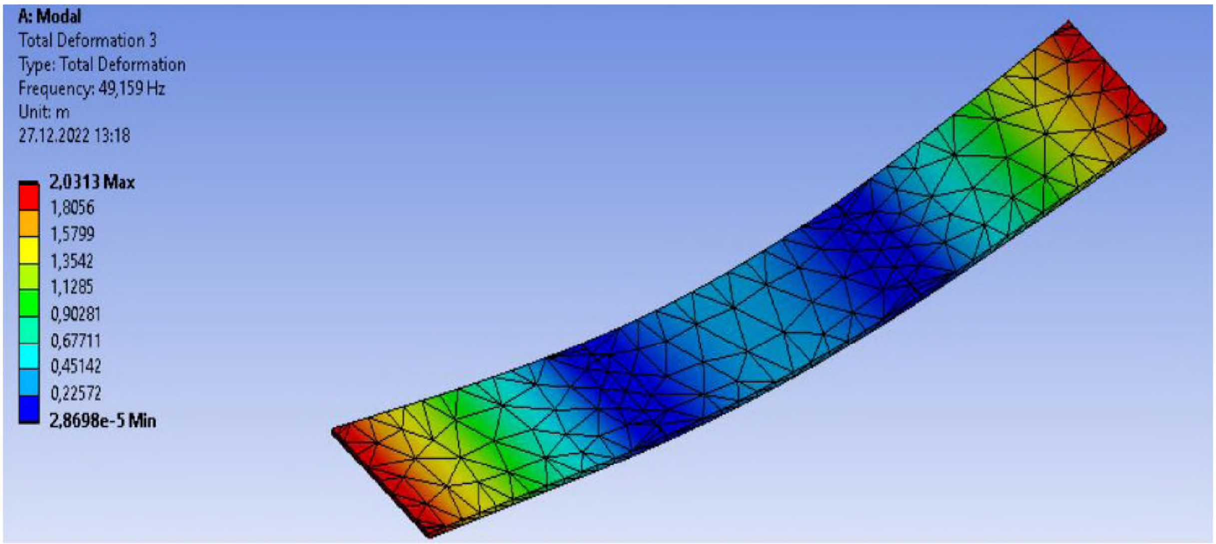

Fig. 5. The result of calculating the first natural frequency of oscillations of a parabolic concave plate

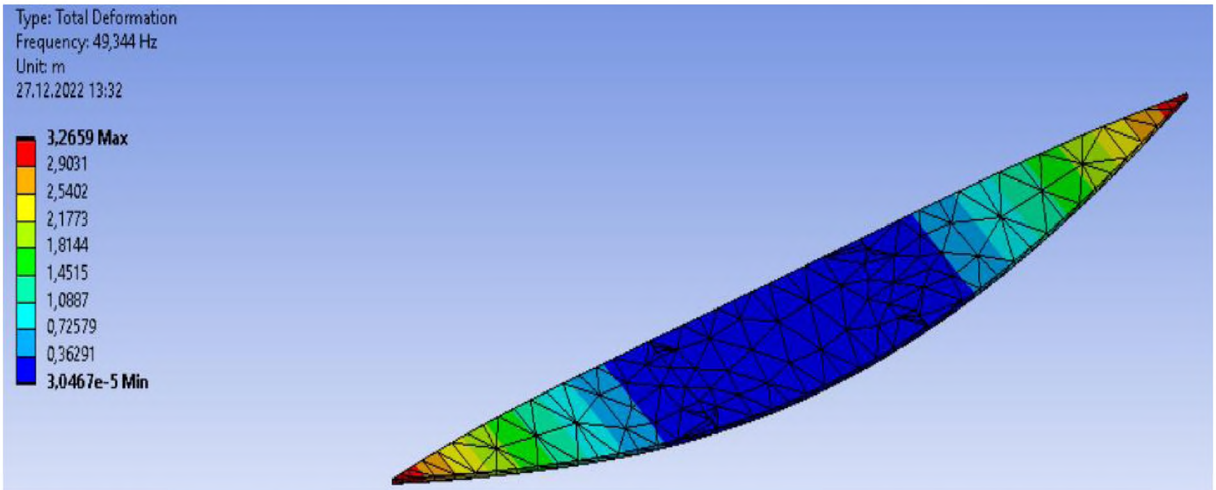

Fig. 6. The result of calculating the first natural frequency of oscillations of a parabolic convex plate

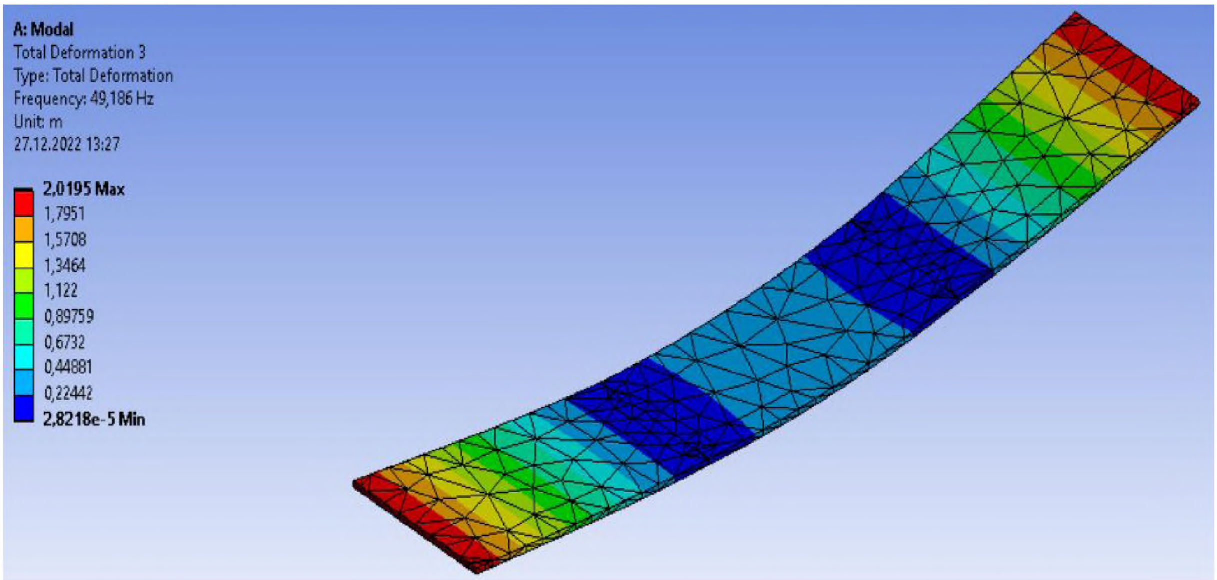

Fig. 7. The result of calculating the first natural frequency of oscillations of a X-shaped plate

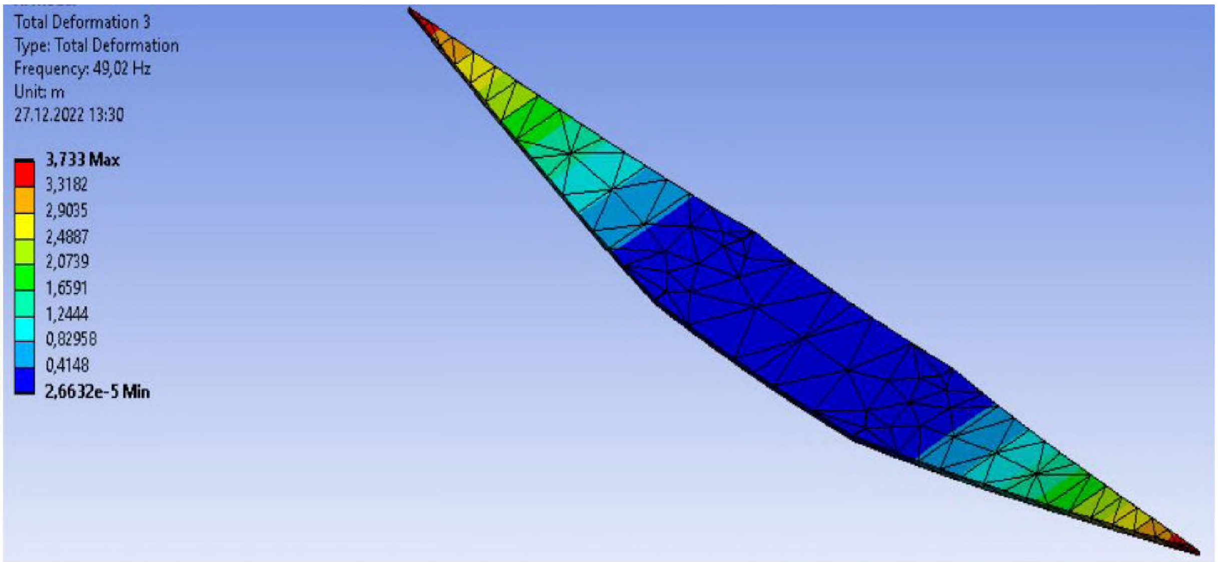

Рис. 8. The result of calculating the first natural frequency of oscillations of a diamond-shaped plate

As can be seen from Fig. 5 – 8, all types of plates fulfill the condition that the first frequency of oscillations of the continuous section must be in the range $\omega = 49...49{,}6$ *Hz*.

With the help of calculations in ANSYS Workbench, optimal geometric parameters were selected for all types of investigated plates. The drawing of plates with a variable cross-section is shown in Fig. 9.

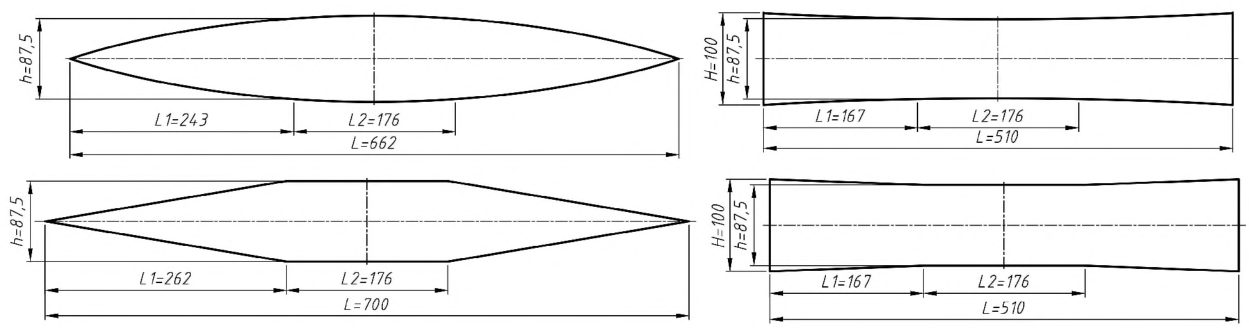

Fig. 9. Geometric dimensions of plates with a variable cross-section

All designed and depicted in Fig. 9 plates have the necessary fastening dimensions.

During the research, the experimental samples of the plates were alternately installed in the intermediate mass of the vibrating table. The vibrating table was connected to the electrical network through a later, which changes the value of the supply voltage, that is transmitted to the electromagnets. Loads of different sizes were placed on the working body of the vibrating table. The weights of the loads are given in Table 1.

Table 1

Masses of loads on the working body of the vibrating table

№ of load	1	2	3	4
Weight, g	70	141	198	263

For each load on the working body of the vibrating table, the value of the voltage at which its separation from the working body began to occur was determined. This technique was used for each of the investigated plates. Oscillations of the plates during the operation of the vibrating table are shown in Fig. 10.

Fig. 10. Oscillations of plates during operation of electromagnets:
a – parabolic concave; b – parabolic convex; c – X-shaped; d – diamond-shaped

Experimental studies were carried out with single fixed plates of various shapes. In the course of the study, the mass of loads on the working body of the vibrating machine also changed. Table 2 shows the dependence of the supply voltage, at which the separation loads from the working body began to occur, on the weight of the loads for different forms of plates.

Table 2

The dependence of the supply voltage, at which the separation loads from the working body began to occur, on the weight of the loads for different forms of plates

Weight of load, g	Supply voltage, V				
	Rectangular plate	Diamond-shaped plate	Parabolic concave plate	X-shaped plate	Parabolic convex plate
70	48	40	42	58	60
141	55	42	45	64	65
198	60	45	49	68	70
263	65	47	51	74	75

It can be seen from Table 2 that the lowest supply voltages, at which separation of loads of different masses occurs, are observed in the case when a diamond-shaped plate is installed as the reactive mass. Somewhat larger values are observed in the case of fixing a parabolic convex plate in the vibrating table. The highest voltage values were recorded when working as a continuous section of a vibrating machine of a parabolic concave plate.

On the basis of the data obtained as a result of experimental studies, graphs of the dependence of the voltage, at which loads of different masses are separated from the working body of the vibrating machine, from the mass of the weights, are constructed for each plate (Fig. 11). Dots indicate experimental results recorded during the experiment.

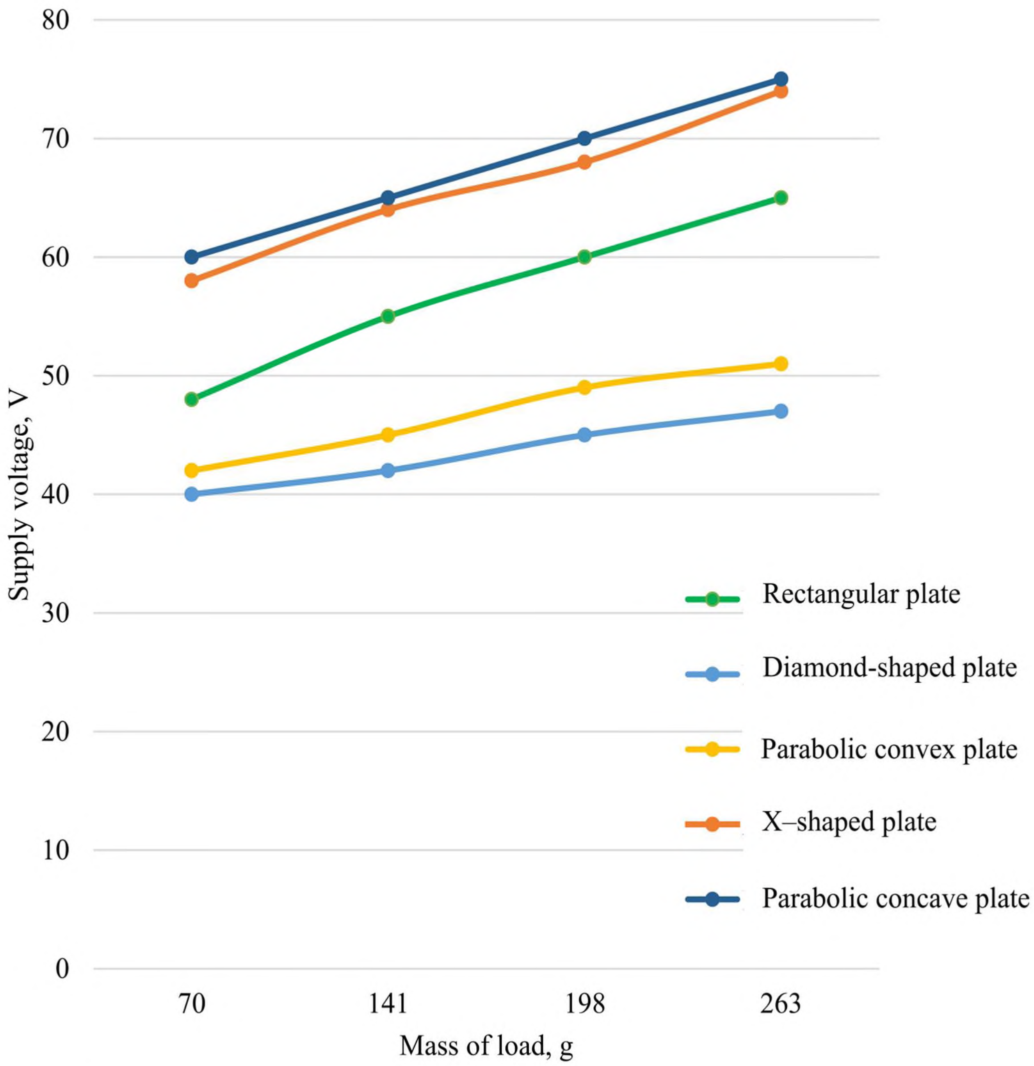

Fig. 11. The graph of the dependence of the supply voltage, at which the separation loads from the working body began to occur, on the weight of the loads for different forms of plates

Summarizing the data in Fig. 11, it can be concluded that the diamond-shaped plate has the highest energy efficiency among the tested samples. In general, the tendency is obvious when the energy efficiency of the vibrating table increases with the increase of the plate wing (distance L_1).

Therefore, as a result of the experimental studies carried out in this paper, it can be concluded that the diamond shape of the plate is optimal for this type of use of a continuous section in a vibration machine with an electromagnetic drive.

Conclusions

Implementation of energy-efficient technical solutions is an important aspect of technological equipment modernization. In the field of vibration technologies, the trend of creating three-mass discrete inter-resonance vibration machines has evolved into the introduction of a new class of designs, the basis of which are discrete-continuous oscillatory systems. The introduction of a continuous section as a reactive mass of vibration machines made it possible to more effectively use inter-resonance modes of operation. Basically, rectangular plates are used as a continuous section. However, taking into account the idea of introducing plates with a variable cross-section as a reactive mass, in this work, experimental studies of the effectiveness of using different types of plates in a vibration table with an electromagnetic drive were carried out. For this, the conditions that the plates must meet were established, namely: the first natural frequency of oscillation, which must be in the range $\omega = 49...49{,}6\,Hz$ and the fastening dimensions $L_2 = 176\,mm,\ h = 87{,}5\,mm.$ Five types of plates are offered: rectangular, parabolic concave, parabolic convex, X-shaped, and diamond-shaped plates. Experimental studies of the power supply voltage at which the separation of loads of different weights occurs during the operation of different types of plates have been carried out. It was found that the diamond-shaped plate has the highest energy efficiency among the tested samples. In general, the tendency is obvious when the energy efficiency of the vibrating table increases with the increase of the plate wing (distance L_1).

References

[1] O. Lanets, *Osnovy rozrakhunku ta konstruyuvannya vibratsiynykh mashyn. Knyha 1. Teoriya ta praktyka stvorennya vibratsiynykh mashyn z harmoniynym rukhom robochoho orhana [Fundamentals of analysis and design of vibrating machines. Book 1. Theory and Practice of Development of Vibratory Machines with Harmonic Motion of the Working Element Body]*. Lviv, Ukraine: Lviv Polytechnic Publishing House, 2018. [in Ukrainian].

[2] O. S. Lanets, O. Yu. Kachur, V. M. Korendiy. "Classical approach to determining the natural frequency of continual subsystem of three-mass inter-resonant vibratory machine", *Ukrainian Journal of Mechanical Engineering and Materials Science*, Vol. 5(3–4), pp. 77–87, 2019.

[3] O. Lanets, R. Kachmar, P. Maistruk, I. Derevenko, A. Hordieiev. "Approximate calculation of natural frequencies of oscillations of the diamond-shaped plates of the discrete-continuous inter-resonance vibrating table", *IOP Conference Series: Materials Science and Engineering*, Vol. 1277, pp.1–7, 2023.

[4] P. Maistruk, O. Lanets, V. Stupnytskyy, "Approximate Calculation of the Natural Oscillation Frequency of the Vibrating Table in Inter-Resonance Operation Mode", *Strojnícky časopis - Journal of Mechanical Engineering*, Vol. 71(2), pp. 151–166, 2021.

[5] H. Kozbur, O. Shkodzinsky, I. Kozbur, N. Gashchyn. "Prediction of the boundary states for thin-walled axisymmetric shells under internal pressure and tension loads", *Strojnícky časopis – Journal of Mechanical Engineering*, Vol. 70(1), pp. 57–68, 2020.

[6] R. Vescovini, L. Dozio, M. D'Ottavio, O. Polit. "On the application of the Ritz method to free vibration and buckling analysis of highly anisotropic plates", *Composite Structures*, Vol. 192, pp. 460–474, 2018.

[7] A. Borković, G. Radenković, D. Majstorović, S. Milovanović, D. Milašinović, R. Cvijić. "Free vibration analysis of singly curved shells using the isogeometric finite strip method", *Thin-Walled Structures*, Vol. 157, pp. 107–125, 2020.

[8] K. Żur, M. Arefi, J. Kim, J. Reddy. "Free vibration and buckling analyses of magneto–electro–elastic FGM nanoplates based on nonlocal modified higher–order sinusoidal shear deformation theory", *Composites, Part B: Engineering*, Vol. 182, pp. 107–601, 2020.

[9] M. Boiangiu, V. Ceausu, C.D. Untaroiu, "A transfer matrix method for free vibration analysis of Euler-Bernoulli beams with variable cross section", *Journal of Vibration and Control*, vol. 22(11) pp. 2591–2602, 2016.

[10] E. Demir, H. Çallioğlu, M. Sayer, "Vibration analysis of sandwich beams with variable cross section on variable Winkler elastic foundation", *Science and Engineering of Composite Materials*, Vol. 20(4), pp. 359–370, 2013.

[11] S. Zolkiewski, "Vibrations of beams with a variable cross-section fixed on rotational rigid disks", *Latin American Journal of Solids and Structures*, Vol. 10, pp. 39–57, 2013.

[12] V. De Biagi, B. Chiaia, G. Carlo Marano, A. Fiore, R. Greco, L. Sardone, R. Cucuzza, G. L. Cascella, M. Spinelli, N. D. Lagaros. "Series solution of beams with variable cross-section", *Procedia Manufacturing*, Vol. 44, pp. 489–496, 2020.

[13] P. Bilancia, M. Baggetta, G. Hao, G. Berselli. "A variable section beams based Bi-BCM formulation for the kinetostatic analysis of cross-axis flexural pivots", *International Journal of Mechanical Sciences*, Vol. 205, 106587, 2021.

[14] Y. De Santis, D. P. Pasca, A. Aloisio, A. Stenstad, K.-C. Mahnert, "Experimental, analytical and numerical investigation on the capacity of composite glulam beams with holes", *Engineering Structures*, Vol. 285, 115995, 2023.

[15] A. Elkaimbillah, B. Braikat, F. Mohri, N. Damil. "A one-dimensional model for computing forced nonlinear vibration of thin-walled composite beams with open variable cross-sections", *Thin-Walled Structures*, Vol. 159, 107211, 2021.