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Modeling of the deformed state of the screen box of a heavily loaded vibratory machine

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Моделювання деформованого стану грохотної коробки сильно навантаженої вібраційної машини

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Abstract. *The paper addresses the problem of improving the technical and economic efficiency of heavily loaded vibratory machines through optimization of the load-bearing frame of the screen box. The main elements providing the spatial stiffness of the box structure are tubular transverse tie-beams, which operate under conditions of intensive cyclic bending loads. Analysis of the available literature indicates that these elements belong to the most highly loaded and vulnerable structural components, for which fatigue failure is the dominant failure mechanism. The traditional approach to increasing the strength of structural elements, namely, enlarging the cross-sectional area, leads to an increase in the mass of the screen box and the associated inertial loads, which adversely affects the dynamic characteristics of the vibratory machine. The objective of this study was to improve the structural efficiency of the screen box of a vibratory machine by optimizing the geometry of the tie-beams using a rational material distribution along the length of the element. An approach is proposed that involves varying the outer diameter of the tubular tie-beam according to a parabolic law while maintaining a constant inner diameter. Such an approach makes it possible to reduce the beam mass in regions of low bending moments while simultaneously maintaining or increasing stiffness in the most heavily loaded sections. To evaluate the effectiveness of the proposed solution, a series of numerical experiments was carried out using the finite element method. The simulations included calculations of axial compression, bending under transverse inertial loading, determination of the first natural frequency, and estimation of the critical buckling load for tie-beams with both baseline and parabolic profiles. The obtained results demonstrate that the use of a parabolic profile makes it possible to reduce the mass of the tie-beam by 41%, while the transverse acceleration corresponding to the onset of yielding increases by 55%, and the first natural frequency increases by 18%. Stress distribution maps indicate a more uniform loading of the material in the tie-beam with the parabolic profile. The deformed state of the screen box was also modeled using both the baseline and the optimized tie-beams. The results show that the change in beam geometry has virtually no effect on the dynamic characteristics of the box. The first natural frequency of the structure changes only slightly, while the deformation pattern and the maximum displacement amplitude remain practically unchanged. The obtained results confirm the prospects of applying variable-section tie-beams for improving the energy efficiency and reliability of vibratory machines.*
Keywords: *vibratory machine, screen box, reliability, structural optimization, stress–strain state, natural frequencies, vibration resistance of structures.*

Анотація. *У статті розглядається проблема підвищення техніко-економічної ефективності важко навантажених вібраційних машин шляхом оптимізації несучої рами грохотної коробки. Основними елементами, що забезпечують просторову жорсткість конструкції коробки, є трубчасті поперечні балки, які працюють в умовах інтенсивних циклічних згинальних навантажень. Аналіз доступної літератури вказує на те, що ці елементи належать до найбільш навантажених та вразливих конструктивних елементів, для яких домінуючим механізмом руйнування є втомне руйнування. Традиційний підхід до підвищення міцності конструктивних елементів, а саме збільшення площі поперечного перерізу, призводить до збільшення маси грохотної коробки та пов'язаних з нею інерційних навантажень, що негативно впливає на динамічні характеристики вібраційної машини. Метою цього дослідження було підвищення конструктивної ефективності грохотної коробки вібраційної машини шляхом оптимізації геометрії балок з використанням раціонального розподілу матеріалу по*

довжині елемента. Запропоновано підхід, що передбачає зміну зовнішнього діаметра трубчастої балки за параболічним законом при збереженні постійного внутрішнього діаметра. Такий підхід дозволяє зменшити масу балки в областях з низькими згинальними моментами, одночасно зберігаючи або збільшуючи жорсткість у найбільш навантажених перерізах. Для оцінки ефективності запропонованого рішення було проведено серію числових експериментів з використанням методу скінченних елементів. Моделювання включало розрахунки осьового стиску, згинання під поперечним інерційним навантаженням, визначення першої власної частоти та оцінку критичного навантаження на вигин для стяжних балок як з базовим, так і з параболічним профілями. Отримані результати показують, що використання параболічного профілю дозволяє зменшити масу стяжної балки на 41%, тоді як поперечне прискорення, що відповідатиме початку текучості, збільшується на 55%, а перша власна частота збільшується на 18%. Карти розподілу напружень вказують на більш рівномірне навантаження матеріалу в стяжній балці з параболічним профілем. Деформований стан ситового короба також було змодельовано з використанням як базового, так і оптимізованого стяжних балок. Результати показують, що зміна геометрії балки практично не впливає на динамічні характеристики короба. Перша власна частота конструкції змінюється незначно, тоді як картина деформації та максимальна амплітуда зміщення залишаються практично незмінними. Отримані результати підтверджують перспективність застосування анкерних балок змінного перерізу для підвищення енергоефективності та надійності вібраційних машин.
Ключові слова: *вібраційна машина, грохот, надійність, структурна оптимізація, напружено-деформований стан, власні частоти, вібростійкість конструкцій.*

Introduction

Vibratory machines are widely used in continuous technological processes for the processing of bulk materials in the mining, metallurgical, and construction industries. Such equipment belongs to the class of units with increased requirements for reliability, durability, and stability of technological performance indicators [1, 2]. At the same time, under modern conditions, improving their energy efficiency has become particularly important. This is associated both with the rising cost of energy resources and with the need to reduce the negative environmental impact of industrial activities. In particular, reducing the energy consumption of technological equipment directly contributes to lowering emissions of fuel combustion products during electricity generation, including carbon monoxide (CO), which is an important environmental factor [3, 4].

One of the promising directions for improving the energy efficiency of vibratory machines is the enhancement of the design of their load-bearing elements and the application of new approaches to component shape formation. Traditionally, the strength of structural elements is increased by enlarging their cross-sectional area. However, for vibratory machines this approach has significant limitations. An increase in the mass of components leads to higher inertial loads, which in turn adversely affects the dynamic characteristics of the system. Thus, a contradiction arises between the need to improve the strength and reliability of the structure and the requirement to reduce inertial loads. With increasing vibration frequency of the working body, this contradiction becomes even more pronounced.

Analysis of published data and problem statement

The main load-bearing elements of the screen box of a vibratory machine are the side plates and the tie-beams. The side plates are manufactured from steel sheet blanks reinforced with stiffening ribs. They serve as the supporting base for the installation of screen panels and transmit the inertial forces generated by rotating unbalanced exciters mounted on the box. The spatial stiffness of the structure is ensured by tubular transverse tie-beams with a circular cross-section that connect the side plates to each other. Since the side plates are located in the plane of vibration, the tie-beams are oriented perpendicular to it. As a result, they are subjected to significant periodic bending loads, the direction of which changes with the operating vibration frequency of the vibratory machine. Therefore, cyclic bending is the dominant loading mode for these elements, whereas axial tensile and compressive forces mainly arise due to deformation of the screen box and are not the governing loads [5].

In the study [6], performed using the ANSYS software package, it was established that the tie-beams of a vibratory machine are among the most heavily loaded and structurally vulnerable components of the machine. The authors of [7] determined that fatigue failure is the primary mechanism of their destruction. The results of modal and harmonic analyses made it possible to clearly identify the most critical zones of the structure that are prone to crack initiation and propagation.

In publication [8], an approach combining the analysis of vibratory machine dynamics with crack growth modeling was proposed. The dynamic behavior of the vibratory machine was described using a two-dimensional model developed based on the finite element method, in which the tie-beam was represented by an Euler beam element with a local defect. To estimate the crack growth rate, the Paris model was applied, allowing the variation of system stiffness during damage evolution to be taken into account. It was shown that the amplitude of the excitation force is the dominant factor determining the fatigue life of the tie-beam, while cracks located near the joints with the side plates propagate significantly faster due to increased stress concentration.

A number of studies address the problem of reduced durability of tie-beams associated with the tendency to increase both their dimensions and the dynamic intensity of vibratory machines. In particular, study [9] demonstrated that an increase in productivity and vibration frequency leads to a significant rise in dynamic loads, which results in the formation of cracks in tie-beams. To improve structural reliability, the use of beams with redundant constraints was proposed, providing additional stiffness and the ability to sustain loads even in the case of partial damage to structural elements. Experimental investigations have shown that such a design contributes to efficient dissipation of vibration energy. It was found that in the improved beam the amplitude of the input power flow is manifested mainly in the low-frequency range and decays more rapidly at higher frequencies, which indicates an increase in the vibration resistance of the structure.

In [10], a mathematical model for calculating the dynamic stresses of tie-beams in a vibratory machine with linear oscillations was presented. The authors applied the finite element method integrated with the dynamic model of the vibratory machine, taking into account the influence of the material mass using the mass-flow method. It was demonstrated that traditional quasi-static calculation methods lead to significant errors because they do not account for high-frequency excitation components that may be close to the natural frequencies of the structure. It was established that the tie-beam experiences complex bending vibrations in both horizontal and vertical planes, while the maximum dynamic stresses occur mainly in the end sections of the tie-beams, where fatigue cracks are most likely to form.

The above considerations highlight the necessity of paying particular attention to the problem of fatigue failure of load-bearing elements in vibratory machine structures, especially tie-beams operating under long-term cyclic loading.

Considering the characteristics of spatially distributed inertial loads, improving the vibration resistance of the screen box of a vibratory machine can be achieved not by increasing the cross-sectional area of the transverse beams but, conversely, by rationally reducing their cross-section in the region of maximum deflection. Such an approach makes it possible to reduce the mass of the structure, decrease inertial loads, and improve the dynamic characteristics of the system.

The purpose and objectives of research

The objective of this study was to improve the technical and economic efficiency of vibratory machines through optimization of the load-bearing structure of their screen box by applying new approaches to the geometric design of tie-beams. This approach makes it possible to significantly reduce the mass of the structure while simultaneously improving its vibration and stiffness characteristics.

Materials and methods

The structure of the screen box of the vibratory screen GST-62.MF (Fig. 1) was selected as the object of the study. The box consists of side plates 1, interconnected by tie-beams 2 and two transverse beams 3, in which flange-mounted motor-vibrators 4 are installed and connected to the side plates of the box by flanges.

Considering their structural configuration, the tie-beams of the box can be regarded as statically indeterminate elements clamped at both ends by the elastic side plates. The most critical loading conditions occur when the operating frequency passes through their resonant frequencies. According to the analysis of the dynamic characteristics of vibratory screens [11], operating frequencies that significantly differ from the first natural frequencies of the structural elements of the screen box make it possible to avoid resonant excitation. Such resonant conditions can lead to large vibration amplitudes and significant destructive effects on the box structure and, consequently, reduce the durability and operational reliability of the machine.

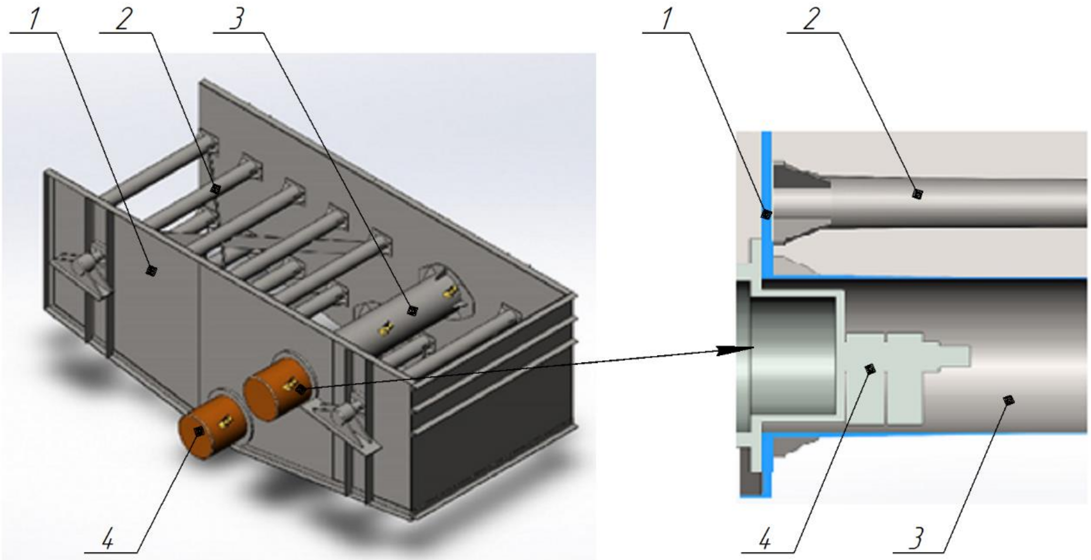

Figure 1 – Structure of the screen box of the vibratory screen GST-62.MF.

An increase in the operating frequency of the screen during the processing of metallurgical charge components generates additional dynamic loads, which necessitates the search for reserves to improve the structural stiffness of the working body and its components, primarily the tie-beams.

One of the possible approaches to reducing the influence of vibration loads is a rational modification of the cross-section of the tie-beam, taking into account the distribution of bending moments along its length. Such an approach involves reducing the mass in regions with small bending moments while maintaining or increasing stiffness in regions where the bending moments reach their maximum values. As a result, the mass of the beam can be reduced without a significant decrease in its bending stiffness, which contributes to an increase in the natural frequencies of the structure. One practical way to implement this approach is to reduce the cross-sectional area of the beam in its central part. Provided that technological manufacturing constraints and longitudinal strength requirements are satisfied, this makes it possible to reduce the amplitude of transverse vibrations of the beam.

Taking this into account, the geometric shape of the surface of a tie-beam with a circular cross-section and a constant inner diameter d can be described as a circular paraboloid, provided that the outer diameter of the section varies from the maximum value D_{max} at the junction with the side plate to the minimum value D_{min} in the plane coinciding with the axis of symmetry of the box, depending on the longitudinal coordinate according to a parabolic law

$$D(x) = 2 \cdot R(x) = 2 \cdot (a \cdot x^2 + b). \qquad (1)$$

Considering the specified boundary conditions imposed on the shape of the generating curve of the tie-beam, the functional coefficients are determined by the following relationships:

$$\begin{cases} a = \dfrac{D_{\max} - D_{\min}}{0{,}5 \cdot L^2}, \\ b = 0{,}5 \cdot D_{\min} \end{cases} \qquad (2)$$

where L – is the length of the tie-beam.

Fig. 2 shows the baseline profile of the tie-beam and the profile obtained according to the parabolic law of diameter variation.

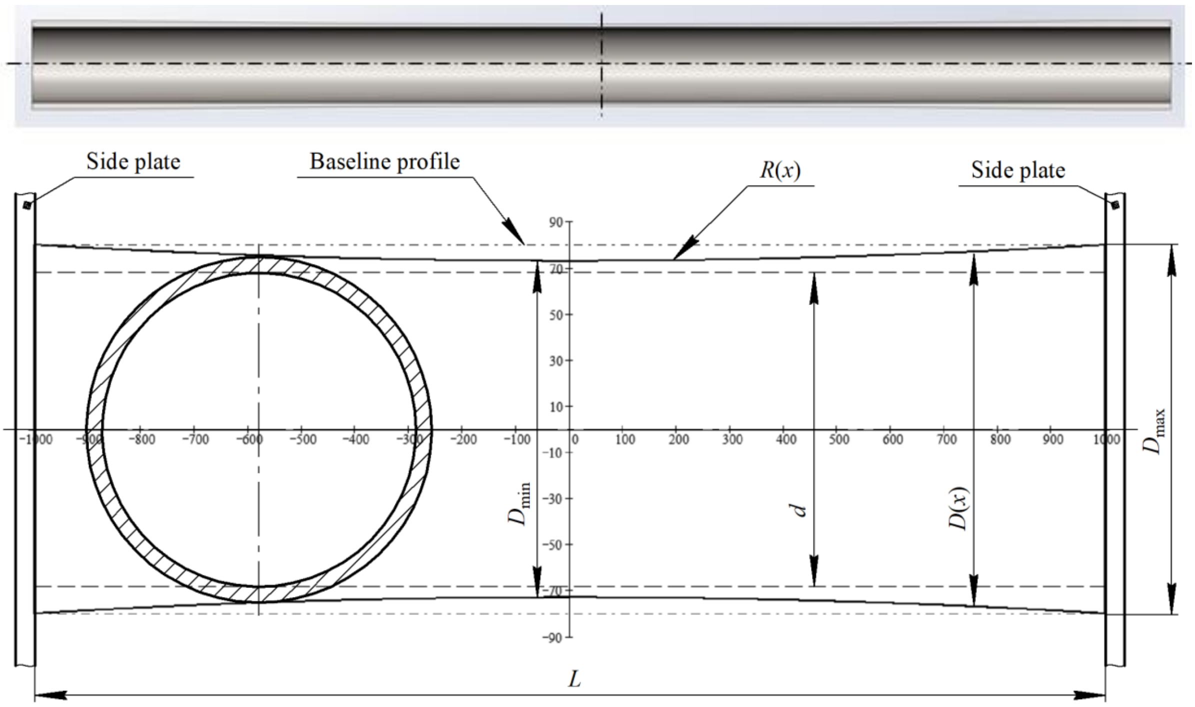

Figure 2 – Baseline profile of the tie-beam and the profile obtained according to the parabolic law of diameter variation (dimensions in mm).

Results of the Research and Their Discussion

To investigate the influence of static and dynamic loads on tie-beams with baseline and parabolic profiles, a series of comparative numerical experiments was carried out.

For each beam configuration, the following analyses were performed in the simulation environment:

axial compression until the stress corresponding to the yield strength of the material was reached at any point of the beam;

bending under transverse inertial loading (maximum transverse acceleration) with clamped ends until the yield strength of the material was reached;

determination of the first natural frequency of the beam with clamped ends;

determination of the critical compressive load corresponding to the buckling of the beam with clamped ends.

As a result of the numerical analysis, stress distribution maps were obtained (Fig. 3), while the corresponding quantitative characteristics are presented in Tab. 1.

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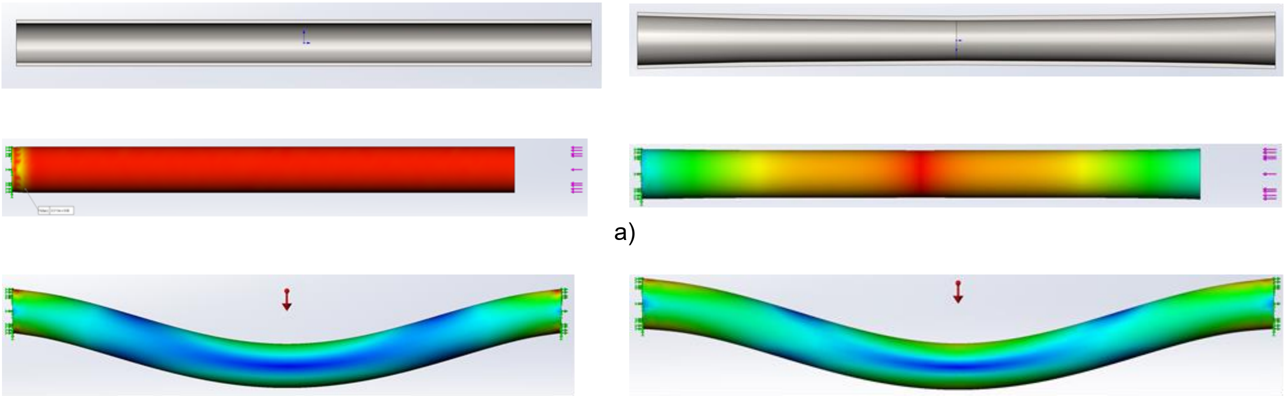
Baseline profile Parabolic profile



a)

b)

Figure 3 – Stress distribution maps: a – axial compression; b – bending at maximum acceleration.

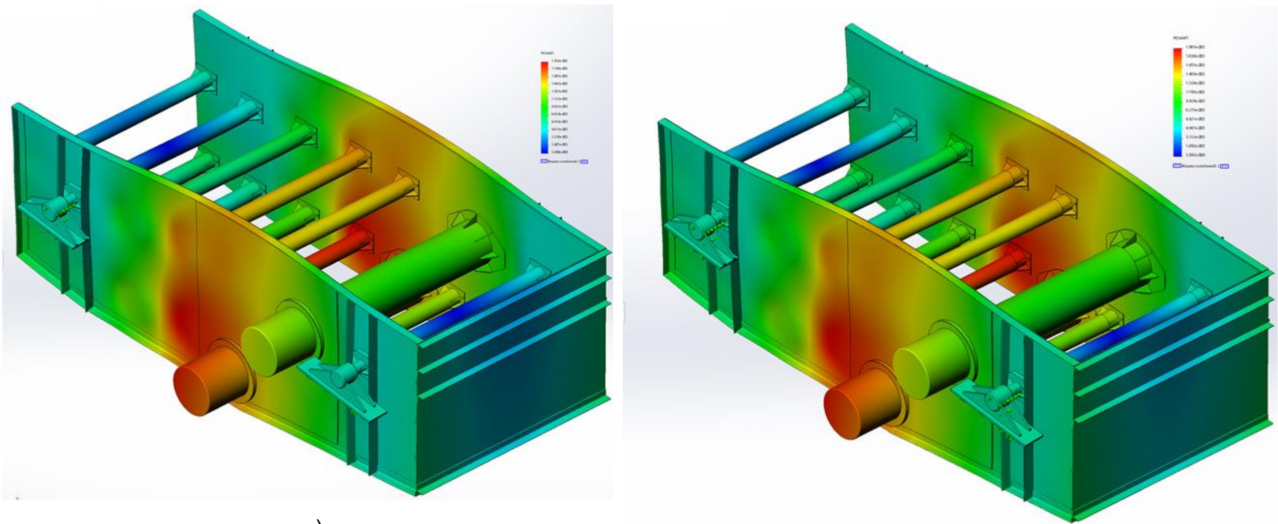
Table 1 – Calculation results for tie-beams with different profiles.

Parameter	Profile		Deviation, %
	baseline	parabolic	
Axial compressive force at the yield limit, kN	1830	770	-58
Transverse acceleration at the yield limit, m/s²	4600	7110	+55
First natural frequency with clamped ends, Hz	218	258,06	+18
Resonant vibration amplitude, mm	0,17	0,226	+33
Critical buckling load for clamped supports, kN	1830	900	-51
Mass of the tie-beam, kg	88,16	52,39	-41

The stress distribution maps demonstrate a more uniform loading of the material in the tie-beam with a parabolic profile, reducing the extent of underloaded regions and increasing the vibration resistance of the structure. Such a distribution contributes to more efficient utilization of the material and improves the operational reliability of the structure.

The analysis of the calculated characteristics of tie-beams with different profiles shows that the tie-beam with a parabolic profile exhibits a significant mass reduction of 41% compared to the baseline design, which provides a potential reserve for reducing the overall mass of the working body. At the same time, the axial compressive force corresponding to the yield limit decreases by 58%, and the critical buckling load decreases by 51%.

The analysis of transverse dynamics indicates that the acceleration corresponding to the onset of yielding for the parabolic profile increases by 55%, while the first natural frequency of vibration increases by 18%. At the same time, the resonant vibration amplitude also increases by 33%.



a) b)

Figure 4 – Deformation distribution map of the screen box of the GST-62.MF vibratory screen:
a – baseline profile of tie-beams; b – parabolic profile of tie-beams.

In the real screen box structure, the boundary conditions of the tie-beams differ from those adopted in the simplified model. The elasticity of the side plates results in limited stiffness of the beam end restraints, while interaction with other beams through the side structural elements affects both the stress distribution and the dynamic characteristics of the system. This necessitates additional investigation of the frequency behavior of the tie-beams directly within the screen box structure.

Fig. 4 presents the deformation map of the screen box of the GST-62.MF vibratory screen.

The obtained results show that the first natural frequency of the screen box structure, compared with the baseline design, changes only slightly, increasing from 12.481 Hz to 12.662 Hz. The maximum resultant displacement amplitude of the box remains the same in both considered cases and equals $1.9 \cdot 10^{-2}$ mm. The configuration of the deformed state of the structure in both variants is practically identical. Analysis of the deformation distribution maps indicates that the character and spatial distribution pattern of the deformations for both screen box models are also essentially the same.

Conclusions

The conducted study has shown that tie-beams are important load-bearing elements of the screen boxes of vibratory machines and are subjected to significant cyclic loads. To improve their operational efficiency, the use of tubular beams with a variable outer diameter following a parabolic law while maintaining a constant inner diameter has been proposed. Such a geometry ensures a more rational distribution of material along the beam length in accordance with the distribution of bending moments.

The results of numerical modeling demonstrate that the use of a beam with a parabolic profile makes it possible to reduce its mass by 41%, while simultaneously increasing the allowable transverse acceleration and the first natural frequency of vibration. Analysis of the stress–strain state indicates a more uniform distribution of stresses within the beam material.It has been established that the use of such beams practically does not change the dynamic characteristics of the screen box of the vibratory machine. This confirms the feasibility of their application for reducing the structural mass and improving the energy efficiency and reliability of vibratory machines.

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