Збірник наукових праць. Галузеве машинобудування, будівництво
Academic journal. Industrial Machine Building, Civil Engineering
http://journals.nupp.edu.ua/znp
https://doi.org/10.26906/znp.2021.57.2593

UDC 666.9.033

Computer modeling of the stress-strain state of vibrating machine frames with spatial vibrations

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The article analyzes the stress-strain state of vibrating platforms with vertical and horizontal vibration exciters. The stiffness of the frames under the action of both vertical and horizontal loads is considered. Deflections in elements of the metal frame were calculated. The type of stress distribution in frame elements was studied. Places of stress concentration are determined for these structures of vibration platform the metal frame. The values of these stresses are much smaller than the calculated resistance of steel, which makes it possible to optimize the frame design by reducing its metal capacity. The metal capacity of the frame can be reduced by changing the thickness of the frame elements by 20%.

Keywords: stress-strain state, vibrating platforms, vibration exciter, spatial vibrations, computer simulation

Комп'ютерне моделювання напружено-деформованого стану рам вібраційних установок з просторовими коливаннями

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Виконано аналіз напрямків використання різних конструкцій вібромайданчиків для ущільнення бетонних сумішей при виробництві плоских панельних елементів. Удосконалення конструкцій вібраційних площадок та їх елементів на етапі проектування можливе за рахунок моделювання роботи під навантаженням, починаючи від стадії не завантаженої конструкції і закінчуючи її повним завантаженням. Нами розроблені моделі рам вібраційних площадки та віброзбуджувачів коливань розроблених в КБ «ВІБРОТЕХНІКА» Національного університету «Полтавська політехніка імені Юрія Кондратюка». Найпростіший спосіб удосконалення конструкцій вібраційних площадок та їх елементів на етапі проектування можливе за рахунок моделювання роботи під навантаженням, починаючи від стадії не завантаженої конструкції і закінчуючи її повним завантаженням. Також за рахунок вивчення переміщень і деформацій елементів рам вібраційних площадок можливо виявляти місця концентрації напружень та змінювати конструктивні рішення. Визначення напружено-деформованого стану рами вібраційної площадки є непростою задачею, складність якої пов'язана з цілою низкою факторів, таких як: збільшення кількості скінченних елементів, використання різних типів скінченних елементів у одній чисельній моделі, механічних властивостей гумових опор площадки, моделювання різних видів навантажень (статичних та динамічних), моделювання зварних швів просторової рами. Досліджено характер розподілу напружень у елементах рам. Визначено місця концентрації напружень для даних конструкцій металевих рам віброплощадок. Величини цих напружень значно менша за розрахунковий опір сталі, що дає можливість виконати оптимізацію конструкції рами, шляхом зменшення її металоємності. Металоємність рами можна зменшити змінивши товщину елементів каркасу рами на 20%.

Ключові слова: напружено-деформований стан, віброплатформи, віброзбуджувач, просторові коливання, комп'ютерне моделювання

Introduction

Nowadays, computer modeling, the selection and use of equipment that would best meet the technological production during the vibration compaction scheme of flat reinforced concrete products is an increasingly important task.

Improving the design of vibration platforms at the design stage is possible by modeling the stress-strain state. By means diagrams of movement distribution and stresses deformation of metal designs of vibrating platforms at various operating modes can be defined.

Determination of the stress-strain state of the frame vibrating platforms is related to various factors: the number of finite elements, different types of finite elements for one model, static and dynamic loads for the model, welded joints for the frame of the vibrating platform [1-4].

Review of the research sources and publications

Increasing the efficiency of the work process consists in finding constructive solutions for creating machines with a variable amplitude-frequency oscillation mode.

The identification of new parameters during the operation of compaction machines and the consideration of these parameters during process modeling is a mandatory component of modeling. An effective method is the use of continuous models that take into account the propagation of vibration waves [5].

The proposed approach is the basis for determining the real distribution of amplitudes and frequencies of oscillations and the use of multimode effects [6-7].

A model of the structure to be researched and created can transfer energy (the maximum amount) from the working body to the environment. This process occurs due to the frequency spectrum with different components of energy, and resonance phenomena of the system by choosing the operating mode that coincides with the elastic-inertial and elastic-viscous properties of the machine and the environment [8].

Problem statement

The work aims to study the distribution of stresses and deformations in structural elements of vibration machines that will allow analyzing and identifying places of stress concentration, which allows optimization of design solutions.

With the existing methods of research and mathematical modeling of vibrating machine-environment systems, the problem of specifying the elements of the prestatic and dynamic action of the load arises.

Thus, the formulation of the problem of computer modeling consists of the construction of a system that reflects the real operation of the vibrating machine-environment system.

Basic material and results

Improvement of the designs of vibration machines is possible by modeling the operation of this machine under load, from a completely unloaded design to a full load [5, 9]. The study of deformations and displacements of structural elements of vibrating machines allows to identify stress concentrators and optimize design solutions with the help of finite element analysis (Fig. 1) [10].

Movements in the direction of the lower planes of the vibration supports are limited along the X, Y, Z axes, which simulate the fastening of the structure to the foundation.

Vibration exciters on the frame are conventionally shown, and their effect on the frame is determined by the driving force. The load on the frame of the vibrating machine from the metal form and the concrete mixture is modeled by a uniformly distributed load.

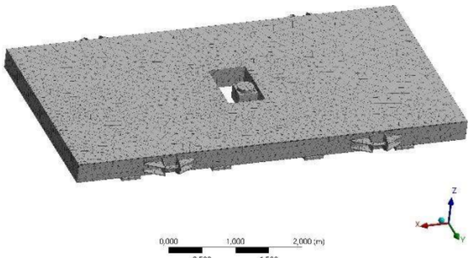

Figure 1 – Design solutions using finite element analysis for the vibration platform

Given generalized coordinates in all nodes $\square_1$, nodal intersections, the set of which for a given element is written in the form of a matrix:

$$\{\lambda\} = \{\lambda_1, \lambda_2, ..., \lambda_N\}^T, \quad (1)$$

where N – number of nodal movements for each element;

T – matrix transposition.

Nodal displacements are the components of the displacement vector of nodes along the coordinate axes, as well as the rotation angles of the element at the nodal points.

The displacements of any point M are approximated by nodal displacements, which are unknown quantities inside each element:

$$u_i = \Phi_{ik}(M)\lambda_k \;\; i = 1,2,3, \; k = 1,2...N; \quad (2)$$

the same in matrix notation $\{u\}=\{\Phi\}\{\lambda\}$ and vector form:

$$\vec{u} = \Phi_{ik}\vec{e}_k\lambda_k = \{\vec{\Phi}\}\{\lambda\}, \quad (3)$$

where $\Phi_{ik}(M)$ values are functions of the shape of the element and show the relationship between the displacements of point M and nodal displacements; polynomials are used as a shape function;

$\{\vec{\Phi}\} = \{\vec{\Phi}_1, \vec{\Phi}_2, ..., \vec{\Phi}_N\}$, $\vec{e}_i$ –unit vectors.

Relations (8.3) are substituted into the body equilibrium equation, from which the nodal displacements $\{\lambda\}$ for each element are determined.

When using the finite element method, it is most convenient to obtain the body balance equation based on the principle of possible displacements. Let $\vec{u}$ is the field of displacements of points deformable body under the action of external loads applied to it. By setting each point a small displacement $\delta\vec{u}$ allowed by the constraints imposed on the body (movement is possible). According to this principle, the increase in the work of internal forces is equal to the work of external forces on possible displacements, i.e.:

$$\delta U = \delta W \tag{4}$$

Denoting as $\vec{q}$ the external load distributed over the volume of the body V, and as $\vec{p}$ – the load distributed over its surface S.

Let's obtain:

$$\delta W = \int_V \vec{q} \cdot \delta\vec{u} dV + \int_S \vec{p} \cdot \delta\vec{u} dS. \tag{5}$$

The expression for the work of internal forces has the form:

$$\delta U = \int_V \sigma \cdot \delta\varepsilon dV, \tag{6}$$

where $\sigma = \sigma_{ij}\vec{e}_i\vec{e}_j$ – stress tensor;

$\varepsilon = \varepsilon_{ij}\vec{e}_i\vec{e}_j$ – strain tensor;

$\vec{e}_i$ – unit unit vectors, $i, j = 1, 2, 3$.

Then relation (4) takes the form:

$$\int_V \sigma \cdot \delta\varepsilon dV = \int_V \vec{q} \cdot \delta\vec{u} dV + \int_S \vec{p} \cdot \delta\vec{u} dS. \tag{7}$$

In the case of small deformations of the body:

$$\varepsilon = \nabla\vec{u}, \tag{8}$$

where $\nabla\vec{u} = \frac{1}{2}\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\vec{e}_i\vec{e}_j$ – tensor operator;

$i, j = 1, 2, 3$; x_1, x_2, x_3 – coordinate axes directed along the unit vectors $\vec{e}_1, \vec{e}_2, \vec{e}_3$.

Substituting (2) into (8), let's obtain an expression for the components of the strain tensor in terms of the nodal displacements:

$$\varepsilon_{ij} = \frac{1}{2}\left(\frac{\partial \Phi_{ik}}{\partial x_j} + \frac{\partial \Phi_{jk}}{\partial x_i}\right)\lambda_k \tag{9}$$

or in matrix form:

$$\{\varepsilon\} = \{B\}\{\lambda\}, \tag{10}$$

where

$\{B\} = \{\nabla\vec{\Phi}\} = \left\{\frac{1}{2}\left(\frac{\partial \Phi_{ik}}{\partial x_j} + \frac{\partial \Phi_{jk}}{\partial x_i}\right)\right\}$ – matrix associates deformations with nodal displacements.

The relationship between the components of stress and strain tensors for an elastic body is expressed by a law:

$$\sigma_{ij} = D_{ijkl}\varepsilon_{kl}, \tag{11}$$

where D_{ijkl} – the elastic constants of the body, $i, j, k, l = 1, 2, 3$, or in matrix form $\{\sigma\} = \{D\}\{\varepsilon\}$.

Substituting expression (10), let's find the dependence of the stress tensor on nodal displacements:

$$\{\sigma\} = \{D\}\{B\}\{\lambda\}. \tag{12}$$

After transformations, let's obtain the equation of equilibrium of an elastic body containing the movement of its points:

$$\int_V D\nabla\vec{u} \cdot \delta(\nabla\vec{u}) dV = \int_V \vec{q} \cdot \delta\vec{u} dV + \int_S \vec{p} \cdot \delta\vec{u} dS. \tag{13}$$

Let's apply relation (13) to a finite element with some volume V_e bounded by the surface S_e, and find:

$$\delta\lambda_i\left\{\int_{V_e} \nabla\vec{\Phi}_i \cdot D\nabla\vec{\Phi}_j \cdot \lambda_j dV - \int_{V_e} \vec{q}\vec{\Phi}_i dV - \int_{S_e} \vec{p}\vec{\Phi}_i dS\right\} = 0, \tag{14}$$

where $i, j = 1, 2, \ldots, N$.

Since $\delta\lambda_i$ — arbitrary nonzero values, the last equality requires all expressions in curly braces to rotate to zero. From these conditions, let's obtain a system of linear algebraic equations expressing the equilibrium conditions for a finite element:

$$\{K\}\{\lambda\} = \{f\}, \tag{15}$$

where $K_{ij} = \int \nabla\vec{\Phi}_i \cdot D\nabla\vec{\Phi}_j dV$ – stiffness matrix of the element, which, using relations (10) and (12), can also be written in the form: $\{K\} = \{B\}^T\{D\}\{B\}$;

$f_i = \int_{V_e} \vec{q} \cdot \vec{\Phi}_i dV + \int_{S_e} \vec{p} \cdot \vec{\Phi}_i dS$ – vector of nodal forces of the element, where $i, j = 1, 2, \ldots, N$.

The set of equations (15) for all elements is supplemented by the equations of ligatures imposed on the body (limiting conditions), which are a system of equations for the equilibrium of the body under consideration, written in a form similar to (15):

$$\{\bar{K}\}\{\bar{\lambda}\} = \{\bar{f}\}, \tag{16}$$

where $\{\bar{K}\}$ is called the global body stiffness matrix;

$\{\bar{\lambda}\}$ and $\{\bar{f}\}$ – vectors of nodal displacements and forces of the whole body.

Equations of the type (16) are used for strength analysis of structures under static load. From their solution, the vector of nodal displacements is determined, then, according to relation (2), one can find the displacement of the points of the body, and according to (8)–(11) or (13) – deformations and stresses. From (15) it is easy to obtain the equation of motion of the element. By introducing, according to the d'Alembert principle, the volumetric forces of inertia into the integral for the nodal forces (15):

$$\vec{q}^{in} = -\rho\frac{\partial^2\vec{u}}{\partial t^2} = -\rho\vec{\Phi}_j \cdot \vec{\lambda}_j \tag{17}$$

let's obtain a system of equations:

$$\{M\}\{\ddot{\lambda}\}+\{K\}\{\lambda\}=\{f\}, \qquad (18)$$

where $M_y=\int\vec{\rho}\vec{\Phi}_i\vec{\Phi}_j dV$ mass matrix of the element;
ρ – material density;
$\ddot{\lambda}$ – the second time derivative of the vector of nodal displacements.

In the presence of viscous resistance forces in the system, proportional to the speeds of the points, the matrix of damping coefficients {B} is introduced into (18), after which the equations of motion take the form:

$$\{M\}\{\ddot{\lambda}\}+\{B\}\{\dot{\lambda}\}+\{K\}\{\lambda\}=\{f\}. \qquad (19)$$

When studying the problems of elastic resistance of structural elements, the equilibrium equation is added taking into account the change in the geometry of the body in the deformed state:

$$\left[\{\bar{K}\}-\beta\{\bar{K}_d\}\right]\{\bar{\lambda}\}=0 \qquad (20)$$

where by means of a matrix of geometric stiffness $\{\overline{K}_d\}$, called differential in MSC.vN5W, the work of external forces due to a change in the geometry of the body is taken into account;
β –loading parameter.

Equating to zero the determinant of the system (20) det, $\det\left[\{\bar{K}\}-\beta\{\bar{K}_d\}\right]=0^0$ let's find the value of the load parameter β_1, β_2, ... at which there are nontrivial displacements for nodal displacements $\{\lambda\}$, i.e. new forms of body balance appear, different from the original. Such values β, called critical, show how many times the critical load $F^{\kappa p}$, at which the loss of resistance of the original form of body balance occurs, is greater than the current load F:

$$F_i^{\kappa p}=\beta_i F,\quad i=1,2,\ldots \qquad (21)$$

The first lowest critical load F_1^{kp} is usually of practical interest, since only it will be realized during the operation of the structure. To study the working process of energy-saving machines and their structural elements for the implementation of specific technological work processes, the following research sequence was established:

– justification and development of calculation schemes for the machine;
– analysis of the basic forms, amplitudes, and frequencies of vibration of the structure;
– determination of modes, rational parameters to create a new highly efficient design.

A vibration platform with a vertical vibrator (fig. 2) (horizontal vibrator (fig. 3)) is divided into finite elements (fig. 1). For it, a displacement distribution diagram was obtained (fig. 4) in the X, Y, Z directions and a stress distribution diagram (fig. 5). The largest total displacements of the frame elements (fig. 4) are observed in the outer frame elements, their maximum value is 0.0615 mm.

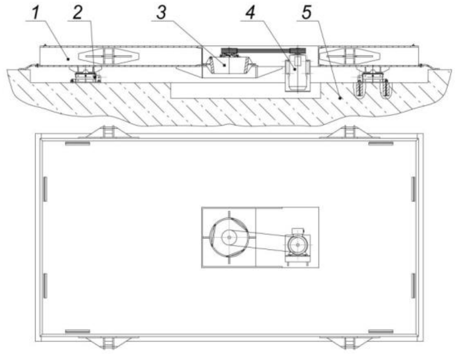

Figure 2 – Vibration platform with a vertical vibrator VPGP-6.3×3.6 for the formation of slabs:
1 – movable frame, 2 – elastic support, 3 – vibration exciter of torsional vibrations, 4 – electric motor, 5 – foundation

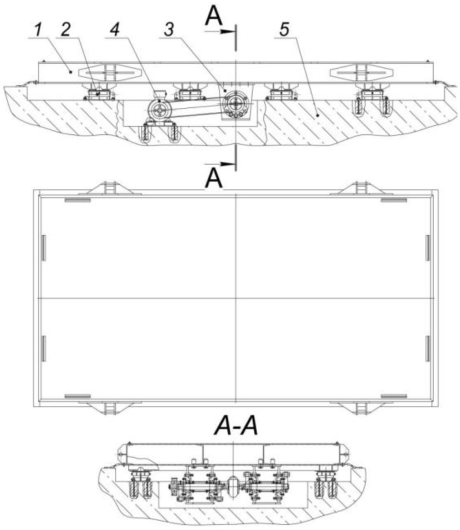

Figure 3 – Vibration platform with a horizontal vibrator VPGP-6.3×3.6 for the formation of slabs:
1 – movable frame, 2 – elastic support, 3 – vibration exciter of torsional vibrations, 4 – electric motor, 5 – foundation

Movement in the vertical direction – along the z-axis, has both positive and negative values (fig. 5), which indicates the presence of small deflections (0.0569 mm) on the cantilever sections of the frame and local bends on the side channels of the frame.

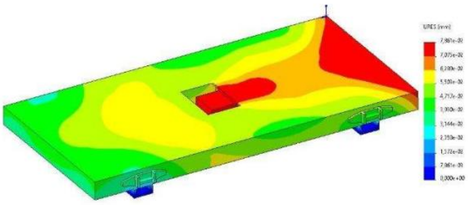

Figure 4 Distribution of total displacements frames with a vertical vibrator

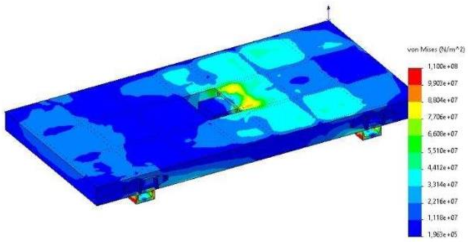

Figure 5 – Distribution of equivalent stresses frames with a vertical vibrator

The highest values of the equivalent stress reach 10.6 MPa (fig. 5). They are observed in the places where vertical and horizontal channels are welded at the place where the vibrator is attached. The values of these stresses are less than the maximum standard stresses, which indicates that the strength of the welded seams is ensured.

In the direction of the X axis, the distribution of displacements is circular in nature, with a maximum (0.042 mm) at the place of load application; the outermost parts of the frame are the least deformable

The distribution of stresses in the frame elements is uneven. The concentration of stresses is observed in the places of welding of elements, but the stress raisers have small values of 4.12 MPa in comparison with the ultimate strength of steel.

The stress in the direction of the X axis is distributed circularly, similar to the distribution of displacements. Moreover, the highest stress values are observed in the two nearest cells to the place of load application.

The structure of the vibration platform VPGP-6.3×3.6 (fig. 3) includes a movable frame with overall dimensions in plan 6.28×3.66, made of channels and a steel sheet made of calm-melting steel st.3sp. In the windows of the moving frame, vibration plates with welded-on M30 nuts are rigidly attached. On the moving frame there are rigid transverse stops for fixing the form with the concrete mixture. The vibration exciter is bolted to the movable frame. With the help of a V-belt transmission, the vibration exciter shaft rotates from an electric motor with a power of 30 kW. The windows of the movable frame are closed with metal covers that prevent particles of the concrete mixture from entering the platform's vibration drive. The movable frame rests on 8 rubber-metal supports attached directly to the foundation of the vibration platform. The supports are attached to the moving frame with the help of protrusions on the supports entering the mounting holes in the moving frame.

The vibration platform model was loaded with the same uniformly distributed load, but the design and direction of vibration movement were changed.

For this option, a deformed diagram, a displacement distribution diagram (fig. 6) and a stress distribution diagram (fig. 7) were also obtained

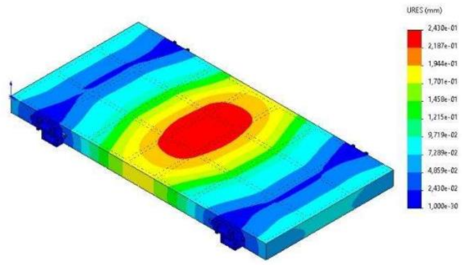

Figure 6 – Distribution of total displacements frames with a horizontal vibrator

The most total displacements of the frame elements are 0.0753 mm (the area is marked in red in the diagram). Such deflections occur in the side channels of the frame on the cantilever sections free from supports. In the center of symmetry of the frame, local deflections of up to 0.0608 mm are also observed.

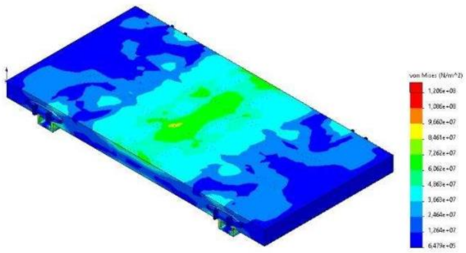

Figure 7 – Distribution of equivalent stresses frames with a horizontal vibrator

Equivalent stresses have a maximum value of 16.6 MPa. These values can be seen in the channels located above the frame supports and in the welds at the vibrator attachment points.

Conclusions

The study of the operation of vibrating frames with horizontal and vertical vibration exciters, in particular, their stress-strain state, shows that the stresses in the main structural elements of the frame are much smaller than the calculated resistance of steel. This ensures the strength of the structure of the vibrating frame.

From the point of view of the structure rigidity, the deflections of the main elements of the frame are less than permissible, which ensures rigidity both in space and in the plane of the frame and the shape of the vibrating platform.

The analysis of the stress-strain state of frames with horizontal vibration exciters allowed us to determine the maximum deflections of 0.0753 mm observed in the cantilever sections of the frame and in the places where the vibrators are attached to the frame, where the deflections have a sinusoidal appearance.

The maximum stresses are 18.3 MPa and 14.5 MPa in the vertical and horizontal directions, respectively. Stress concentrators can be observed in the welds in the area of attachment of the vibration exciter and in the places where elastic supports are attached.

Analysis of the stress-strain state of frames with vertical vibration exciters indicates a circular distribution of stresses in the zone of vertical vibrators location. We can see the highest stress values of 13.8 MPa in the area where the vibration exciter is located. The values of these stresses are much smaller than the calculated resistance of steel, which makes it possible to optimize the frame design by reducing its metal capacity. The metal capacity of the frame can be reduced by changing the thickness of the frame elements by 20%.

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