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OPTIMIZATION OF UNIFORMLY STRESSED STRUCTURES OF CYLINDRICAL TANKS IN CAD

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Запропоновано метод проектування рівнонапружених вузлів циліндричних резервуарів, що містять плоскі круглі пластини змінної товщини, форма діаметральних перерізів яких моделюється рівнянням Гаусса. Для рішення рівняння вигину цих пластин використовуються вироджені гіпергеометричні функції Куммера та Уіттекера. Метод випробуваний в реальному проектуванні з позитивним техніко-економічним ефектом

Ключові слова: рівнонапружені деталі, функції Куммера і Уіттекера, пластини змінної товщини, САПР

Предложен метод проектирования равнонапряженных узлов цилиндрических резервуаров, содержащих плоские круглые пластины переменной толщины, форма диаметральных сечений которых моделируется уравнением Гаусса. Для решения уравнения изгиба этих пластин используются вырожденные гипергеометрические функции Куммера и Уиттекера. Метод испытан в реальном проектировании с положительным технико-экономическим эффектом

Ключевые слова: равнонапряженные детали, функции Куммера и Уиттекера, пластины переменой толщины, САПР

1. Introduction

In the process of creating new parts of a machine, a designer must always aim at reaching maximum effectiveness of the project. The requirement for the uniformly stressed parts is one of the constituents of this effectiveness. This ensures maximum value of the ratio “resistance/mass” and, accordingly, minimum material intensity of product as a whole.

Unfortunately, to attain complete equality of stresses at all points of the part is impossible even for a static problem. It is explained by different influence of load on the separate elements of the parts that have a complex shape, heterogeneity of their material and other special features of design and technology. Therefore, any attempt at designing uniformly stressed parts will only be a way of approaching the maximum of effectiveness.

It is known that, for example, in the cases of apparatuses, which work under pressure, the weakest element, from the point of view of non-uniformity of the distribution of stresses, is a flat bottom, due to which it is necessary to increase its thickness by 3–5 times in comparison with the thickness of the wall.

This problem is proposed to be solved by replacing a flat bottom with the one, which has variable thickness from its center to the periphery. The calculation of this thickness, which ensures the closest approach of the bottom under load to the uniformly stressed state, is a relevant direction of studies.

2. Literature analysis and problem statement

The majority of machine-building parts and nodes contain technological cavities and openings. They are not necessary for the fulfillment of functional “duties” by the parts and give these parts bizarre and often technologically unfavorable shapes. Designers, for example, intuitively substitute in the objects of design a round continuous rolled metal with a pipe, a round pipe with an elliptical one, and remove a part of the “body” of gears, structural panels and a lot more [1].

It is manifested most vividly in the parts, intended for transport objects hence, for example, there are complex shapes of the parts of aircraft and rocket fuselage [2] and automobile bodies [3]. Reliability of the critical parts – turbine blades [4] and of many others – depends on the correctly calculated shape. The purpose of this complication is to obtain a uniformly stressed part or a unit, which, as it is known [5, 6], makes it possible to obtain the most favorable ratio of the mass of a part and its stressed-strained state (SSS).

An additional example of such structures is vessels, which work under pressure, their housings and bottoms [7]. The most common elements of the existing housing structures of vessels are plates and shells with constant thickness In the majority of cases (both at the planar-stressed state and with the bend), the fields of stresses occurring in them are substantially heterogeneous [8]. Consequently, minimization of mass of the housing constructions without use of the elements of variable thickness in them is practically impossible.

In certain cases, a palliative solution to this problem is found. For example, walls of large vertical cylindrical tanks for storing petroleum products are produced with a gradually changing thickness [9, 10]. A similar solution must also be based on calculation of the plates of variable thickness, the contour of which is elastically attached to the cylindrical wall of a tank.

According to conditions of loading such objects, the most stressed section is the place of the joint (most frequently, of welding) of the shell and the bottom [11]. Calculations of strength show that it is in this place that thickness of the bottom must be the largest, which leads to a rather technologically unfavorable solution: to make bottoms of the vessels in the form of round plates with the thickness variable from the center to the edge [12–14].

It is natural to assume that the structure of such a complex article must be obtained as a result of complex calculations by equations of strength of materials that are non-uniform differential equations of the second order [15]. Solutions for such equations for particular objects are written down in the form of sum of general and particular solutions, that is, it consists of two linearly independent functions [16].

However, such approaches do not ensure optimization of the structures of vessels, since the parts and nodes of equal stress, obtained in this case, do not warrant simultaneous achievement of minimum mass of the would-be object [17]. At the same time, there is mathematical apparatus of hyper-geometric functions, with the help of which a similar problem can be solved [18].

For simultaneous achievement of the uniformed stress and minimum mass, it is necessary to develop a new method, which would consider such concept of effective optimization and a pattern of bending the plates of different thickness for the implementation of this method.

3. The aim and the tasks of the study

The aim of this study is a decrease in metal intensity of constructions at the stage of automated design by means of creating uniformly stressed structural elements with the retention of their reliability indices due to rational redistribution of the used materials inside an element.

To achieve this aim, the following tasks were to be solved:

– to develop a method for the optimization of shape of a round plate with variable thickness, which involves transition from the fixed thickness of plate in its center to its fixed volume;

– to develop a model of the bend of a round plate with variable thickness in the form of exponential Gauss function, which considers dependence of thickness in the center of a plate on its volume.

4. Development of mathematical provision for designing spatial uniformly stressed parts

4. 1. Method for designing the shape of uniformly stressed nodes of conjugation of structural elements

Let us consider round plates of radius R, either having one flat and one concave surfaces (Fig. 1, *a*), or having both concave surfaces (Fig. 1, *b*).

A change in thickness of the plate in radial direction r in a rather general case can be described by Gauss function [8]:

$$\delta(r)=\delta_0\exp(-nr^2/6R^2), \tag{1}$$

where δ_0 is the thickness of the plate in the center at r=0.

Parameter n in equation (1) determines intensity of the change in thickness of a round plate in radial direction. In the circular direction, thickness remains constant, that is, shape of the plate is assumed to be axisymmetric.

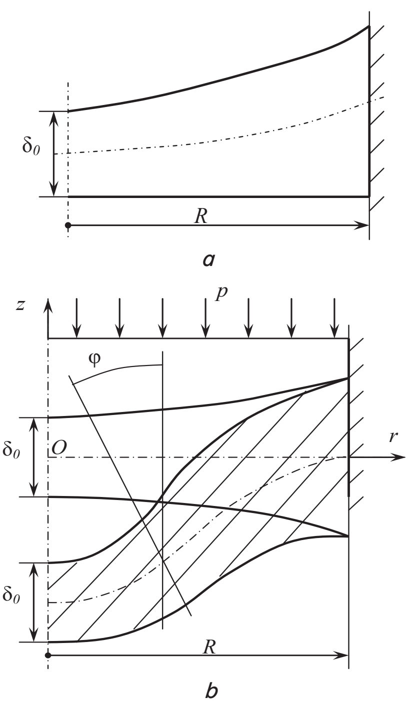

Fig. 1. Round plate of variable thickness, clamped along contour r=R: *a* – flat-concave form of diametrical cross-section; *b* – biconcave form of diametrical section in the initial state and in the state deformed by load p (shaded)

A plane-concave shape of cross-section of the plate is obtained, if we plot δ(r) from the flat lower surface; a biconcave form is obtained if we plot sizes 0,5δ(r) on both sides of the plane z=0. The shape of diametrical section, obtained in this way, is sufficiently general since plates with the concave surfaces (Fig. 2, curves 4, 5) can be described by function (1) with positive values of parameter n.

Shapes of surfaces with parameter n<0 may be recommended for round plates, bent by transverse load p at the rigid clamping of their contour, when the maximum bending moment influences the contour (Fig. 1, *b*). With the hinged fixing of the contour, maximum bending moment occurs in the center of the plate and the shape with maximum thickness in the center, when n>0, becomes preferable.

With the optimization of the shape of diametrical cross-section of a round plate, we strived for the minimization of its mass, determined by the volume of the used material.

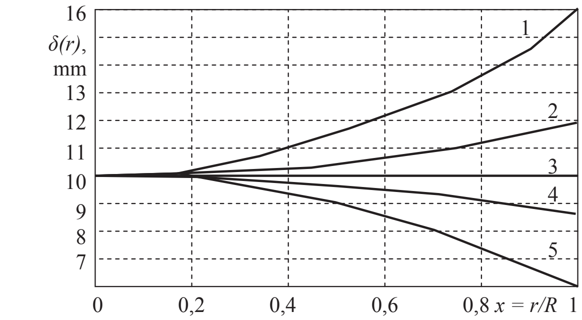

Fig. 2. Graphs of thickness dependence of round plate on its relative radius x=r/R at δ_0=10 mm for certain values of parameter n: n=–3 (1); n=–2 (2); n=0 (3); n=+2 (4); n=+3 (5)

For a rigidly clamped plate, this aim is achieved by the displacement of material from the underloaded central zone to the periphery with retention of constant volume of plate V_0. Formula, which determines a change in thickness of this plate in the radial direction, is obtained from (1) in the form

$$\delta_V(x)=\frac{V_0}{\pi R^2}\cdot\frac{n}{6\left[1-\exp(-n/6)\right]}\exp\left(-\frac{nx^2}{6}\right). \qquad (2)$$

The second cofactor in (2) has uncertainty at n=0. We should accept for this value

$$\lim K(n)\Big|_{n\to 0}=\lim\frac{n}{6\left[1-\exp(-n/6)\right]}\Bigg|_{n\to 0}=1. \qquad (3)$$

A change in thickness in the radial direction, determined by function (2) for certain values of parameter n≤0, is represented in Fig. 3.

Let us note that as a result of exponentiality of dependence (2), at n<0, thickness on the contour of plate grows much more intensively than it diminishes in the center: relation $\delta(1)/\delta(0)=\exp(-n/6)$. For example, at n=–10, thickness on the contour of a plate is 5,3 times larger than in the center.

Differential equation of axisymmetric bend of this plate at the uniformly distributed load (pressure) p relative to the angle of rotation of normal to the median surface ϕ has the second order [5]:

$$\frac{d^2\phi}{dx^2}+\left(\frac{1}{x}-nx\right)\frac{d\phi}{dx}-\left(\frac{1}{x^2}+\mu n\right)\phi=-\bar{p}x\exp\left(\frac{nx^2}{2}\right). \qquad (4)$$

Dimensionless parameter $\bar{p}$ contained in the right side of equation (4) depends on the accepted mathematical model of diametrical cross-section of the plate:

– with equation of the shape of the plate in the form (1) we have:

$$\bar{p}=\bar{p}_\delta=6(1-\mu^2)\frac{pR^3}{E\delta_0^3}; \qquad (5)$$

– with equation of the shape of the plate in the form (2) we have:

$$\bar{p}=\bar{p}_V=6(1-\mu^2)\frac{p}{E}\left[\frac{6\pi R^3}{V_0}\cdot\frac{1-\exp(-n/6)}{n}\right]^3, \qquad (6)$$

where p is the intensity of uniformly distributed load; E, μ are the modulus of elasticity and the Poisson coefficient of the plate's material.

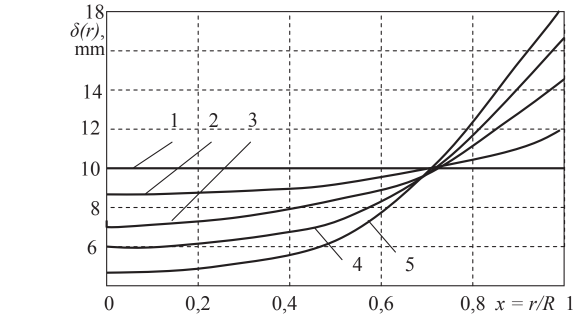

Fig. 3. Graphs of thickness dependence of round plate on its relative radius x=r/R at $V_0/(\pi R^2)$=10 mm for certain values of parameter n: n=0 (1); n=–2 (2); n=–4 (3); n=–6 (4); n=–8 (5)

Integral of non-uniform equation (2) consists of two components, particular and general solutions.

A particular solution takes the following form:

$$\phi_0=-\frac{\bar{p}x}{(3-\mu)n}\exp\left(\frac{nx^2}{2}\right), \qquad (7)$$

where $\bar{p}$ is determined by formula (5) or (6) depending on the method of assigning the shape of diametrical cross-section of the plate.

General solution of homogeneous equation (4) (when the right part equals zero) is assigned in [5] by the power series, which presents certain computational difficulties for practical calculations. In the present work, solution of this homogeneous equation is represented with the help of the degenerate hypergeometric Whittaker functions $M_{k,\gamma}(z)$, $W_{k,\gamma}(z)$ [17]:

$$\phi_1(x)=\frac{\exp(0{,}25nx^2)}{x}\left[C_1M_{k,\gamma}\left(\frac{1}{2}nx^2\right)+C_2W_{k,\gamma}\left(\frac{1}{2}nx^2\right)\right], \qquad (8)$$

where $k=(1-\mu)/2$, $\gamma=1/2$; C_1 and C_2 are arbitrary constants.

Eigenfunctions of solution (8) are equal:

$$F_1(x)=x^{-1}\exp(0{,}25nx^2)M_{k,\gamma}\left(\frac{1}{2}nx^2\right), \qquad (9)$$

$$F_2(x)=x^{-1}\exp(0{,}25nx^2)W_{k,\gamma}\left(\frac{1}{2}nx^2\right). \qquad (10)$$

Examples of plots of functions (9) and (10) at values n=3 and the Poisson coefficient μ=0,3 are represented in Fig. 4.

Function $F_2(x)$ unlimitedly increases at x→0, therefore, for a round plate, (5) assumed C_2=0. For an annular plate, constants C_1 and C_2 can be determined from the boundary conditions.

Whittaker functions in certain cases complicate analysis of the obtained solutions; therefore, in the paper, it was proposed to replace them with the Kummer functions [16]:

$$M_{k,\gamma}(z)=e^{-z/2}z^{1/2+\gamma}M\left(\frac{1}{2}+\gamma-k;1+2\gamma;z\right), \quad (11)$$

$$W_{k,\gamma}(z)=e^{-z/2}z^{1/2+\gamma}U\left(\frac{1}{2}+\gamma-k;1+2\gamma;z\right). \quad (12)$$

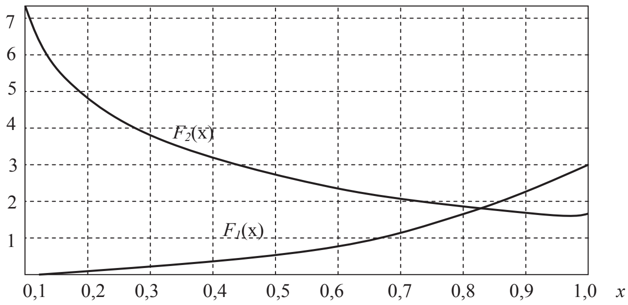

Fig. 4. Examples of plots of eigenfunctions (9) and (10)

After substitution of functions (11) and (12) in (8), we obtain:

$$\phi_1(x)=x^{\gamma-1/2}\left[C_1M\left(\frac{1}{2}+\gamma-k;1+2\gamma;\frac{1}{2}nx^2\right)+\right.$$
$$\left.+C_2U\left(\frac{1}{2}+\gamma-k;1+2\gamma;\frac{1}{2}nx^2\right)\right]. \quad (13)$$

At the given above values of parameters γ, k and particular solution (7), angle of rotation of normal to the median surface of the plate is determined in the form:

$$\phi(x)=\phi_0(x)+\phi_1(x)=$$
$$=\frac{nx}{2}\cdot\left[C_1M\left(\frac{1+\mu}{2};2;\frac{1}{2}nx^2\right)+C_2U\left(\frac{1+\mu}{2};2;\frac{1}{2}nx^2\right)\right]-$$
$$-\frac{\overline{p}x}{(3-\mu)n}\exp\left(\frac{nx^2}{2}\right). \quad (14)$$

Solution in the form (11) may be used for annular plates, that is, in cases when relative radius of their contours $x_1\le x\le 1$. For solid plates, for example, for a bottom or a cover of cylindrical container, it is necessary to assume $C_2=0$, since the solution with the Kummer function of the second order

$$\frac{nx}{2}\cdot U\left(\frac{1+\mu}{2};2;\frac{nx^2}{2}\right)$$

increases indefinitely at $x\to 0$. Consequently, we will search for the solution of the problem of the bend of a continuous round plate with variable thickness in the form:

$$\phi(x)=C_1\frac{nx}{2}\cdot M\left(\frac{1+\mu}{2};2;\frac{1}{2}nx^2\right)-\frac{\overline{p}x}{(3-\mu)n}\exp\left(\frac{nx^2}{2}\right). \quad (15)$$

It is obvious that solution (15) satisfies the required condition $\phi|_{x=0}=0$.

If dependence of the angle of rotation of normal to the median surface of the plate $\phi(x)$ is determined, we will find the equation of this surface by the integration:

$$w(r)=-R\int\phi(x)dx+C_0=$$
$$=C_0+\frac{C_1R\left[F_1(x)+F_2(x)\right]}{(1-\mu)(3-\mu)}-\frac{\overline{p}Rx\exp(0{,}5nx^2)}{(3-\mu)n^2}, \quad (16)$$

where

$$F_1(x)=\left[3-\mu(4-\mu)-nx^2(3-\mu)\right]M\left(\frac{1+\mu}{2};2;\frac{nx^2}{2}\right), \quad (17)$$

$$F_2(x)=\left[3+\mu(2-\mu)\right]M\left(\frac{3+\mu}{2};2;\frac{nx^2}{2}\right). \quad (18)$$

Let us proceed to the models, which contain loaded plates. In this case, radial and peripheral bending moments are determined by formulas:

$$M_r=\frac{D_1(x)}{a}\left(\frac{d\phi(x)}{dx}+\frac{\mu}{x}\phi(x)\right), \quad (19)$$

$$M_t=\frac{D_1(x)}{a}\left(\frac{\phi(x)}{x}+\mu\frac{d\phi(x)}{dx}\right), \quad (20)$$

where the flexural rigidity of plate with variable thickness, which corresponds to (1), takes the form:

$$D_1(x)=\frac{E\delta_0^3}{12(1-\mu^2)}\exp\left(-\frac{nx^2}{2}\right). \quad (21)$$

As a result, we obtain formula for the radial bending moment (14).

Derivative of function (12) takes the following form:

$$\frac{d\phi}{dx}=C_1\frac{n}{2}\left[(1+\mu)M\left(\frac{3+\mu}{2},2,\frac{nx^2}{2}\right)-\mu M\left(\frac{1+\mu}{2},2,\frac{nx^2}{2}\right)\right]-$$
$$-\frac{\overline{p}(1+nx^2)}{(3-\mu)n}\exp\left(\frac{nx^2}{2}\right). \quad (22)$$

Considering (12) and (22), we will find

$$M_r=\frac{D_1(x)}{an}\left\{C_1\left[M_2+\left(\frac{n^2x}{2}-\mu\right)M_1\right]-\right.$$
$$\left.-\frac{\overline{p}(1-x+nx^2)}{(3-\mu)n}\exp\left(\frac{nx^2}{2}\right)\right\}, \quad (23)$$

where

$$M_1=M\left(\frac{1+\mu}{2},2,\frac{nx^2}{2}\right),$$

$$M_2=(1+\mu)M\left(\frac{3+\mu}{2},2,\frac{nx^2}{2}\right).$$

From (13) and (18) we obtained expressions for the angle of rotation of normal and the bending moment on the contour of the plate

$$\phi(1)=C_1\frac{n}{2}M\left(\frac{1+\mu}{2},2,\frac{1}{2}n\right)-\frac{\bar{p}}{(3-\mu)n}\exp\left(\frac{n}{2}\right), \tag{24}$$

$$M_r(1)=$$
$$=\frac{D}{a}\left[C_1\left[M_2(1)+(0,5n^2-\mu)M_1(1)\right]-\frac{\bar{p}x^2}{(3-\mu)}\exp\left(\frac{n}{2}\right)\right]. \tag{25}$$

For designing a uniformly stressed plate, after determining bending moments (19) and (20), radial stresses are calculated:

$$\sigma_r(x)=\frac{6M_r(x)}{\delta_V(x)}. \tag{26}$$

Then we find value of parameter n, at which:

$$\sigma_r(0)=\sigma_r(1). \tag{27}$$

Integration constant C_1 is determined from the conditions of plate fixing along contour x=1. As a result, a scheme of the proposed method for the optimization of shape of a round plate with variable thickness, implying transition from the fixed thickness of the plate in its center to its fixed volume, takes the form, represented in Fig. 5.

4. 2. Model of bend of a round plate with variable thickness

As can be seen from Fig. 5, work of the method begins with designing a model of the bend of a round plate with variable thickness. The proposed model in the form of exponential Gauss function makes it possible to determine the optimum (uniformly stressed, having minimum mass) shape of a continuous round plate with arbitrary fixing along external contour. The model makes it possible to reflect this fixing ranging from absolutely free (hinged support) to absolutely rigid (clamped). Let us examine absolutely rigid fastening. For this case, we will find integration constant C_1 from (24) on condition of fixing the plate on contour x=1:

$$C_1=\frac{2\bar{p}}{(3-\mu)n^2}\cdot\frac{\exp(0,5n)}{M(0,5(1+\mu),2;0,5n)}. \tag{28}$$

We substitute value of C_1, obtained by (28), in (24), and we receive solution to rigid fastening:

$$\phi(x)=\frac{\bar{p}x}{(3-\mu)n}\times$$
$$\times\left[\frac{\exp(0,5n)}{\exp(0,5nx^2)}\cdot\frac{M(0,5(1+\mu),2;0,5nx^2)}{M(0,5(1+\mu),2;0,5n)}-1\right]\exp\left(\frac{nx^2}{2}\right). \tag{29}$$

To prove adequacy of the model of rigid fastening in the form of exponential Gauss function, let us conduct a computational experiment.

For this purpose, let us plot a graph of function dependence (29) on argument x and parameter n (Fig. 6).

In Fig. 6, it is evident that the function of angle of rotation $\phi(X)$ at any value of parameter n at x=0 and at x=1 is equal to zero. This corresponds to boundary conditions of rigid fixing at the axisymmetric deformation of plate, which confirms adequacy of the accepted model and provides for the opportunity of its correct use in the method for the optimization of shape of a round plate with variable thickness (Fig. 5).

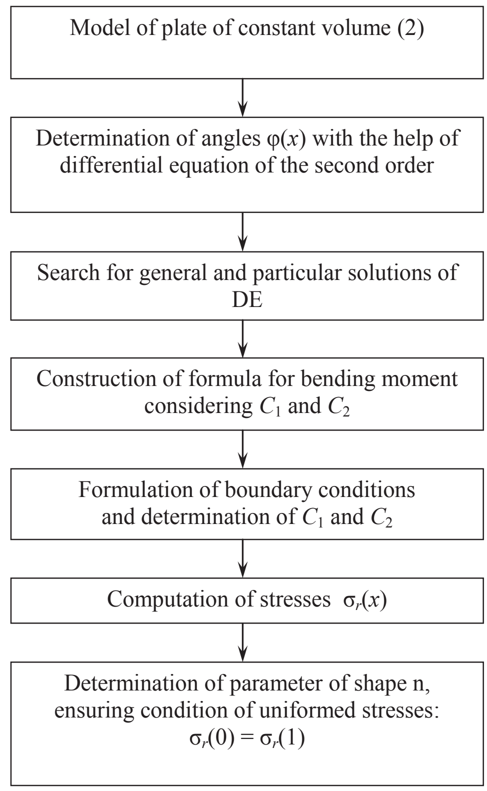

Fig. 5. Structure of method for the optimization of shape of a round plate

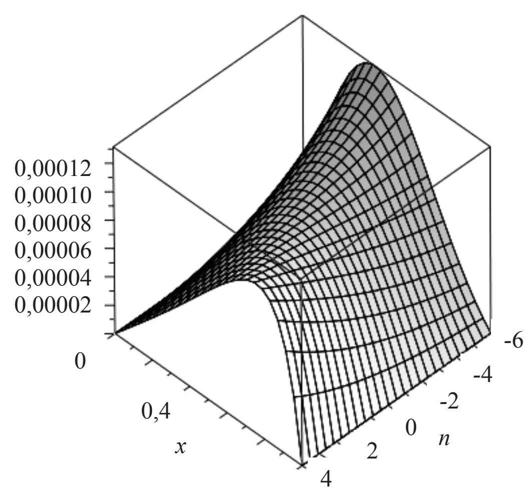

Fig. 6. Graph of function (29) versus argument x and parameter n

5. Discussion of the proposed method for designing uniformly stressed nodes of cylindrical tanks

Result of the study is the confirmation of effectiveness of using a method for the optimization of shape of a round plate with variable thickness, which implies transition from the fixed thickness of plate in its center to its fixed volume. The result is also complemented by the model of bend of a round plate with variable thickness in the form of exponential Gauss function. The model considers dependences of thickness in the center of a plate on its volume, used within the framework of the proposed method.

A positive effect of the study is supported by practical implementation of its results in real CAD, which is employed in the production of vessels that work under pressure. For this purpose, testing of the described method of designing uniformly stressed nodes of conjugated cylindrical and flat housing elements, based on modeling the shape of the latter by hypergeometric functions, was carried out at PAO "Berdichev Machine Building Plant "Progress" (Ukraine).

As the object of design, we selected a vessel, intended for receiving, storage and delivery of concentrated sulfuric and nitric acids at the warehouses of water-treatment plants, the set-up of which includes ion-exchange filters. As a result of practical tests of the new method of design, we obtained a structure of the bottom of a tank, the cross-section of which corresponds to the scheme, given in Fig. 1, that is, thickness of this bottom increases to its edges. The shape of this section is described by expression (2) at the values of thickness of the original (flat) bottom δ_0=0,008 m, radius of the bottom R=0,4 m and parameter n=–2. By integrating (2) under these conditions, we will obtain volume of the new bottom, equal to $9{,}32 \cdot 10^{-3}$ m^3, which in comparison with the volume of the original cylindrical bottom, which was equal to $\pi R^2 \delta_0 / 4 = 10{,}05 \cdot 10^{-3}$, saves 7,15 % of the volume (as well as mass), with the retention of strength characteristics of the bottom as a whole.

6. Conclusions

1. We developed a method for the optimization of shape of a round plate with variable thickness implying transition from the fixed thickness of a plate in its center to its fixed volume, which made it possible to ensure the uniformed stress of the plate. The method is different from the known ones by the fact that it analytically solves two problems: it finds a family of "uniformly stressed shapes" of a part (there is infinite multiple of them at the assigned initial data) and selects a part of the lowest mass.

This ensures maximum effectiveness of the process of optimization of shape and makes it possible to achieve maximum technical and economic effect.

2. A hyper-geometric universal model of bend of a round plate with variable thickness in the form of exponential Gauss function was proposed. The model considers dependence of thickness in the center of a plate on its volume, which allowed us to use it within the framework of the proposed method for the optimization of shape. The model makes it possible to reflect fixing of the bottom in the shell of a vessel that works under pressure in the range from absolutely free (hinged) to absolutely rigid (clamped). For this purpose, in the first case, it is assumed that hardness of the shell is equal to zero, and in the second case, it is equal to infinity. For real objects, the model makes it possible to assign the value of hardness in this range arbitrarily.

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Досліджено вібрації короба грохоту із плоским поступальним рухом короба, збуджені кульовим автобалансиром. 3D моделюванням динаміки грохоту отримано закон усталених вібрацій короба у числовому виді. За допомогою програмного пакета для статистичного аналізу Statistica вібрації ідентифіковано як двочастотні. Розбіжність між результатами 3D моделювання і знайденим законом двочастотних коливань не перевищує 1-го відсотка

Ключові слова: віброзбудник, двочастотні вібрації, 3D моделювання, дебаланс, резонансна вібромашина, автобалансир, грохот

Исследованы вибрации короба грохота с плоским поступательным движением короба, возбуждаемые шаровым автобалансиром. 3D моделированием динамики грохота получен закон установившихся вибраций короба в числовом виде. С помощью программного пакета для статистического анализа Statistica вибрации идентифицированы как двухчастотные. Расхождение между результатами 3D моделирования и найденным законом двухчастотных колебаний не превышает 1-го процента

Ключевые слова: вибровозбудитель, двухчастотные вибрации, 3D моделирование, дебаланс, резонансная вибромашина, автобалансир, грохот

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RESEARCH BY A 3D MODELLING OF THE SCREEN BOX FLAT TRANSLATORY VIBRATIONS EXCITED BY A BALL AUTO-BALANCER

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1. Introduction

Among vibration machines, the most potentially productive are mechanisms that combine in themselves advantages of resonant and dual-frequency vibration machines [1–4]. The resonant operating mode provides vibrations of big sizes of the box and weight at the minimum expenditure of energy and the minimum loads on the drive parts [1, 2]. An operating mode of dual frequency or more ensures increased productivity and performance of additional technology processes [3, 4].

It is suggested to excite dual-frequency vibrations of the screen box with various kinematics of motion by a passive auto-balancer [5]. In different fields of production, screens with a flat translational motion of the box are widely applied [2]. In this regard, it is essential to check the possibility of exciting dual-frequency vibrations by a passive auto-balancer at such kinematics of the box motion.

2. Literature review and problem statement

Among vibration machines, the most power-effective are resonant [1]. The resonant mode of vibrations provides a possibility of using a small drive to set in motion large boxes of screens at a minimum expenditure of energy [2].

A further increase of energy efficiency and productivity of resonant vibration machines is provided with using in them two [3] and more [4] frequency vibroexciters.

In study [5], it is suggested to excite resonant dual-frequency vibrations of the screen box by a method of using a passive auto-balancer. According to the method, the auto-balancer is mounted on a rigid shaft, and the shaft is installed in a rigid support on the screen box. When the shaft is rotated at superresonance velocities, there appear slow and fast harmonic oscillations of the box. Meanwhile, the slow vibrations are vibrations of the box at its own resonant fre-