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прогнозную модель снежно-ледяных образований на горных дорогах [5]. Применение модели позволит разработать предупредительные мероприятия по снегозащите инженерных сооружений, а также принять оперативные решения по расстановке снегоуборочной техники и оптимальному ее использованию, что сократит расходы на зимнее содержание автомобильных дорог.

Выводы

Таким образом можно сформулировать следующий вывод.

Выявлены природные и техногенные факторы, влияющие на эффективность транспортного обеспечения: природные факторы - климатические условия, рельеф местности, характер растительного покрова, свойства снега; техногенные - расположение дороги по отношению к направлению господствующих ветров; особенности конструкции дорожного полотна (в выемках, на нулевых отметках, в насыпях) и приуроченность дороги к различным геоморфологическим элементам, что служит основой для построения оценочно-прогнозной модели снежно-ледяных образований на горных дорогах.

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DEFORMATION OF A CYLINDRICAL SHELL BY EXTERNAL PRESSURE

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ABSTRACT

The article considers the problem of determining the stress-strain state of a thin-walled cylindrical shell under the action of external pressure. Such structures are used in many machines, for example, aircraft casings, submarine casings, drums of hoisting machines.

Equations were derived for circumferential, longitudinal and axial displacements depending on the geometric dimensions and elastic properties of the rope and drum. At the same time, the hypotheses about the non-tightness of the shell in the circumferential direction and the absence of displacement were not accepted. This brings the design scheme much closer to the actual operating conditions of the cylindrical shell.

Keywords: cylindrical shell, external pressure, stress-strain state, rope drum.

Cylindrical shells are the most common forms of closed structures. Therefore, the determination of their stress state under the influence of external pressure is a very urgent task. Different load schemes of structures require different calculation methods. A large number of different shell designs, different conditions of their loading and operation cause a certain difficulty in the analysis of the stress-strain state. The rope drum is considered as a cylindrical shell with variable external pressure, which arises under the action of the rope, which is wound on the drum.

When designing thin-walled structures, the results of theoretical research in the field of structural mechanics and the theory of elasticity are used [1, 2]. But the calculation methods must correspond to the design features for which they will be applied [3–5]. For cylindrical shells, the ratio of the wall thickness to its radius is of great importance, which is not always taken into account when choosing a calculation method [6, 7].

The transverse bending of a cylindrical shell is a rather complex problem, on the solution of which many researchers have worked [8–10].

In work [11], a theory for the calculation of spatial systems was developed, which makes it possible to determine the stress state of various structures. The shell is considered as a multiply statically undefined system. But if we consider it as a system with a stressed state that arises in a thin-walled bank with an under-formed period, then it is possible to determine the normal stresses and shear forces in the elements of cross-sections. In addition, elementary rings can be considered, which are distinguished by planes perpendicular to the axis of the system, while internal forces in longitudinal sections are determined.

That is, the cylindrical shell is considered in the form of a cross system consisting of longitudinal beams-strips and transverse frames-rings. It is clear that the lateral load will simultaneously bend the strip beams and the ring frames.

The deformation compatibility conditions are fulfilled taking into account additional internal forces acting in both transverse and longitudinal directions.

In work [12], a simplification of the solution of the problem is proposed if all unknown internal forces are expressed in terms of the total bending moments:

$$m = m_{fo} + m_{fd}, \quad (1)$$

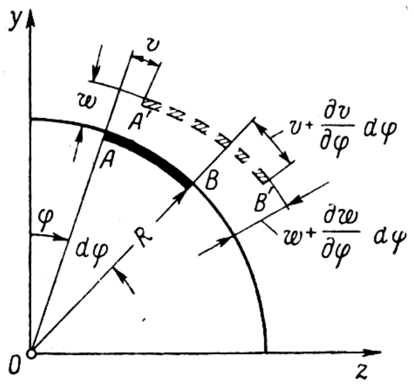
or through the total radius displacement w.

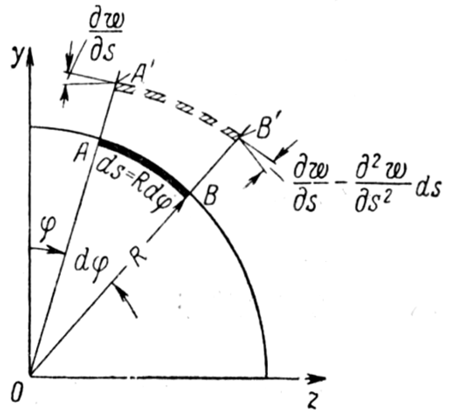
This calculation method is based on the application of Hooke's law and two hypotheses.

In the first hypothesis, it is assumed that there are no shifts in the middle surface of a thin-walled structure. The second hypothesis assumes that the shell is not tight in the circumferential direction.

The presences of tangential forces q_d and hoop normal stresses $\sigma_{\varphi d}$, which are determined by the equilibrium equation, indicate a very close proximity of these hypotheses. This is also indicated by the results of experiments [12].

With a symmetric load of the shell relative to the x axis, its individual elements have radial displacements w, longitudinal u and circular v. Consider an element of a cylindrical shell (Fig. 1), where:





a *b*

Fig. 1. Deformation of the middle surface of the shell
a – radial displacements, b – circumferential displacement

Let us consider a rope drum as a cylindrical shell, loaded with an asymmetric external pressure, which arises from the action of the wound rope. The rope is wound on a drum along a helical line. In most cases, the drum is made profiled. If we decompose the rope tension force into two components – longitudinal and transverse (Fig. 2), then we can write:

$$T_1 = T \cdot cos\beta, \quad (2)$$

$$T_2 = T \cdot sin\beta, \quad (3)$$

where β is the deviation angle.

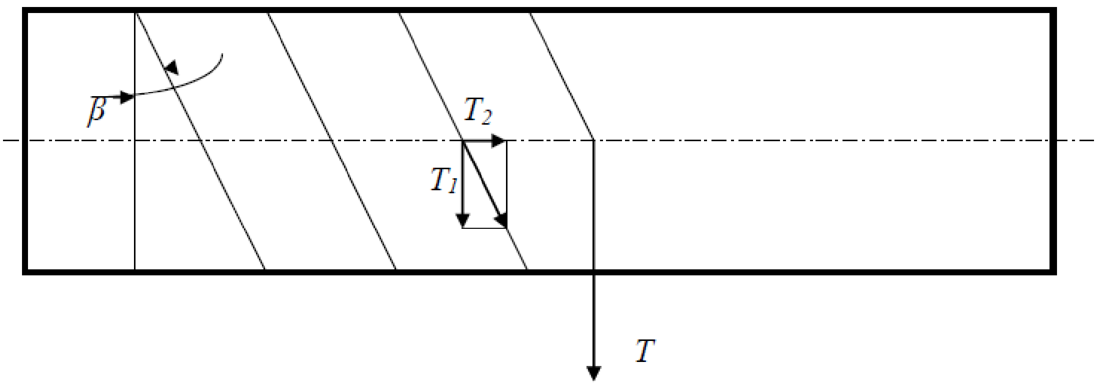


Fig. 2. Design scheme

These forces will act on the crest of the profiled drum; they will be less in the wall itself. That is, if we assume that the forces N_φ and N_x are equal to T_1 and T_2, then this error will go into the safety margin of the drum.

Then we have:

$$\begin{cases} \frac{E\delta}{1-v^2}\cdot\left[\frac{\partial u}{\partial x}+\frac{v}{R}\cdot\left(w+\frac{\partial v}{\partial\varphi}\right)\right] = T\cdot sin\beta\cdot\frac{E\delta}{1-v^2}\cdot \\ \cdot\left[\frac{1}{R}\cdot\left(w+\frac{\partial v}{\partial\varphi}+v\cdot\frac{\partial u}{\partial x}\right) = T\cdot cos\beta\right] \end{cases}. \quad (4)$$

Suppose that the radial displacements change according to the following law:

$$w = f(x)\cdot cosn\varphi, \quad (5)$$

where $f(x)$ is a function that is variable along the length of the shell;

φ is the angle that is measured from the vertical y – axis;

n – 2, 3, 4 – natural numbers.

The rope load will not be constant, it changes depending on many factors: the winding angle φ, the friction coefficient μ, the geometric and elastic properties of the rope and drum.

We accept the following law of rope tension change:

$$T = T_0 \cdot e^{-\frac{k\mu(1-x)}{h} \cdot 2\pi}, \qquad (6)$$

where D_o is the maximum rope tension (at the point where the rope coincides with the drum);

k – coefficient depending on the geometric and elastic properties of the rope and drum;

$$k = \frac{E_k \cdot F_k}{8E} \cdot \frac{\delta}{R}, \qquad (7)$$

where E_k is the modulus of elasticity of the rope;

$F_к$ – cross-sectional area of the rope;

l is the length of the rope winding;

μ is the coefficient of friction between the rope and the drum;

h – pitch of rope winding.

We substitute expressions (5) and (6) into the system of equations (4). After solving, we get:

$$v = \frac{T_0 \cdot e^{k\mu\frac{1-x}{h} \cdot 2\pi R}}{E\partial} \cdot (sin\beta - vcos\beta) \cdot (\varphi - \pi) - \left(\frac{f(x) sinn\varphi}{h}\right), \qquad (8)$$

$$u = \frac{T_0(1-v) \cdot h \cdot e^{-k\mu \cdot \frac{l \cdot 2\pi}{h}}}{2\pi \cdot E \cdot k \cdot \mu \cdot \delta}\left[cos\beta - \frac{v}{1-v^2}(sin\beta - vcos\beta)\right] \cdot \left(l^{\frac{2\pi \cdot k \cdot \mu}{h}} - 1\right). \qquad (9)$$

Thus, we will receive the formulas of all three displacements depending on one function *f (x)*. To find it, we use the theorem of the minimum potential energy.

The work of internal and external forces, which is performed during the transition of the system from the deformed to the initial state, will have the form:

$$\Gamma = \oint\left[\frac{1}{2} \cdot m_\varphi x_\varphi + \frac{1}{2} \cdot m_{xd} x_{x\varphi} + \frac{\delta}{2} \cdot \sigma_{xd} \varepsilon_x + \frac{\delta}{2} \cdot \sigma_\varphi \varepsilon_\varphi - m_{\varphi 0} x_\varphi\right] R d\varphi, \qquad (10)$$

where

$$x_x = -\frac{\partial^2 w}{\partial x^2}, \qquad (11)$$

$$x_\varphi = -\frac{1}{R^2}\left(\frac{\partial^2 w}{\partial x^2} + w\right), \qquad (12)$$

$x_{x\varphi}$ – relative twist angle of the element,

$$x_{x\varphi} = \frac{1}{R}\left(\frac{\partial v}{\partial x} - \frac{\partial^2 w}{\partial \varphi \partial x}\right), \qquad (13)$$

ε_φ – relative deformation in the circumferential direction,

$$\varepsilon_\varphi = \frac{w}{R} + \frac{1}{R} \cdot \frac{\partial v}{\partial \varphi}, \qquad (14)$$

ε_x – relative deformation along the generatrices,

$$\varepsilon_x = \frac{\partial u}{\partial x}, \qquad (15)$$

σ_{xd} - additional normal stresses,

$$\sigma_{xd} = E \cdot \frac{\partial u}{\partial x}, \qquad (16)$$

σ_φ – ring normal stresses,

$$\sigma_\varphi = \frac{R}{\delta} \cdot \left(\frac{\partial Q_{xd}}{\partial x} + \frac{\partial Q_\varphi}{R \partial_\varphi}\right) + \frac{\rho R}{\delta}, \qquad (17)$$

where Q_{xd} and Q_φ are shear forces,

$$Q_{xd} = \frac{\partial m_{xd}}{\partial x} + \frac{\partial m_{\varphi xd}}{R \partial \varphi}, \qquad (17)$$

$$Q_\varphi = \frac{\partial m_\varphi}{R \partial \varphi} + \frac{\partial m_{xd}}{\partial x} \partial x, \qquad (18)$$

where m_{xd} – additional longitudinal bending moments,

$$m_{xd} = D \cdot (x_x + v \cdot x_\varphi), \qquad (19)$$

$m_{x\varphi d}$ and $m_{\varphi xd}$ – additional torques of individual shell elements,

$$m_{\varphi xd} = m_{x\varphi d} = D \cdot (1 - v) \cdot x_{x\varphi}, \qquad (20)$$

ρ – radial load,

$$\rho = \frac{T}{R \cdot h}. \qquad (21)$$

Let us write down the Euler equation for the variational problem:

$$\frac{\partial \Gamma}{\partial f(x)} - \frac{d}{dx}\left(\frac{d\Gamma}{df'(x)}\right) + \frac{\partial^2}{dx^2}\left(\frac{\partial \Gamma}{\partial f''(x)} = 0\right). \qquad (22)$$

Then we obtain the differentiated equations of the fourth degree:

$$f^{IV} - a_1 f''(x) + a_2 f(x) = a_3 l^{-k\mu \cdot \frac{l-x}{h} 2\pi}, \qquad (23)$$

where

$$a_1 = \frac{1}{R^2}\left[\frac{3(n^2-1)}{2} + \frac{(1-v) \cdot (n-2) \cdot n^3 + 1}{n^2}\right], \qquad (24)$$

$$a_2 = \frac{(n^2-1)^2}{R^4}, \qquad (25)$$

$$a_3 = \frac{2(1-v) \cdot T_0 \cdot k^2 \cdot \mu^2 \cdot 4\pi^2 \cdot (sin\beta - v \cdot cos\beta) \cdot (n^2-1)}{E \cdot \delta \cdot h^2 \cdot R}. \qquad (26)$$

Taking into account that $a_2 < a_1$ and a_3, we obtain the solution of equation (23) in the form:

$$f(x) = A \cdot e^{k\mu\frac{l-x}{h} 2\pi} + c_1 e^{\sqrt{a_1 x}} + c_2 e^{\sqrt{a_1 x}} + c_3 x + c_4, \qquad (27)$$

where

$$A=\frac{a_3\cdot h^4}{4\pi^2\cdot k^2\cdot\mu^2\cdot(4\pi^2\cdot k^2\cdot\mu^2-a_1\cdot h^2)}. \tag{28}$$

The coefficients c_1, c_2, c_3 and c_4 are found from the initial conditions:

$$f(x)\big|_{x=0,L}=0, \tag{29}$$

$$\left.\frac{\partial^2 f}{\partial x^2}\right|_{x=0,L}=0. \tag{30}$$

Then the calculated system of equations will have the form:

$$\left\{\begin{matrix} Ae^{-k\mu\frac{l}{h}2\pi}+c_1+c_2+c_4=0 \\ Ae^{-k\mu\frac{|l-L|}{h}2\pi}+c_1 l^{\sqrt{a_1L}}+c_2e^{-\sqrt{a_1L}}+c_3L+c_4=0 \\ A\frac{k^2\mu^2 4\pi^2}{h^2}e^{k\mu\frac{l}{h}2\pi}+a_1(c_1+c_2)=0 \\ A\frac{k^2\mu^2 4\pi^2}{h^2}e^{-k\mu\frac{|l-L|}{h}2\pi}+a_1\left(c_1e^{\sqrt{a_1L}}+c_2 l^{-\sqrt{a_1L}}\right)=0 \end{matrix}\right\}. \tag{31}$$

We solve this system and get:

$$c_1=-A\frac{k^2\mu^2 4\pi^2}{a_1h^2}\left(e^{-k\mu\frac{l}{h}2\pi}+\frac{e^{-k\mu\frac{|l-L|}{h}2\pi}-e^{-k\mu\frac{l}{h}2\pi}}{e^{\sqrt{a_1L}}-e^{-\sqrt{c_1L}}}\right), \tag{32}$$

$$c_2=\frac{Ak^2\mu^2 4\pi^2\left(e^{-k\mu\frac{1}{h}2\pi}\right)}{h^2a_1\left(e^{\sqrt{a_1L}}-e^{-\sqrt{a_1L}}\right)}, \tag{33}$$

$$c_3=\frac{A}{L}\left\{\begin{matrix} \frac{k^2\mu^2 4\pi^2}{a_1h^2}\left(e^{-k\mu\frac{|l-L|}{h}2\pi}-e^{-k\mu\frac{l}{h}2\pi}\right)- \\ -e^{k\mu\frac{l}{h}2\pi}\left(e^{\sqrt{a_1L}}-1\right)-e^{-k\mu\frac{|l-L|}{h}2\pi} \end{matrix}\right\}, \tag{34}$$

$$c_4=Ae^{-k\mu\frac{l2\pi}{h}}\cdot\left(\frac{k^2\mu^2 4\pi^2}{a_1h^2}-1\right). \tag{35}$$

Conclusion. The article considers the rope drum as a cylindrical shell with variable external pressure. The pressure is generated by the rope that is wound around the drum.

When calculating the drum, the friction force acting between the rope and the drum wall is taken into account. When calculating the drum, the hypothesis of the non-tightness of the shell in the circumferential direction and the absence of displacement was not accepted.

Equations are derived for circumferential, longitudinal and axial displacements depending on the geometric dimensions and elastic properties of the rope and drum.

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