https://doi.org/10.33271/nvngu/2022-2/037

N. M. Fidrovska[1],
orcid.org/0000-0002-5248-273X,
Ye. D. Slepuzhnikov[2],
orcid.org/0000-0002-5449-3512,
V. O. Shevchenko[1],
orcid.org/0000-0001-8707-1837,
D. V. Legeyda[3],
orcid.org/0000-0002-8983-0822,
S. V. Vasyliev[2],
orcid.org/0000-0002-6602-8765

1 – Kharkiv National Automobile and Highway University, Kharkiv, Ukraine, e-mail: nfidrovskaya@ukr.net
2 – National University of Civil Defence of Ukraine, Kharkiv, Ukraine
3 – National University of Construction and Architecture, Kharkiv, Ukraine

DETERMINATION OF THE STABILITY OF A THREE-LAYER SHELL OF A TRAVELING WHEEL WITH LIGHT FILLER

Purpose. Development of a calculation methodology for three-layer cylindrical shell stability, which will significantly improve the calculation practice for these structures regarding the determination of critical external pressure.

Methodology. When determining the critical external pressure, the method of variational calculation using the Euler equation of the mixed variational problem was used. To determine three-layer cylindrical shell stability, the factors of significant influence on its strength and stability were taken into consideration, namely the reduced modulus of a three-layer wall elasticity. Bending stiffness D_h was substituted with flexural stiffness of three-layer shell with account for the shear deformation.

Findings. The current situation of the three-layer cylindrical shell stability issue is investigated. Using the variational calculation methods via the Euler equation of the mixed variational problem an equation is composed of equality condition of inner and external force action of an orthotropic structure, which is under the state of neutral equilibrium with radial displacement. The previously obtained equation for radial displacements having been taken and applied to the system potential energy per unit of length equation, an equation for the critical pressure determination is determined. The analytical solution obtained was proposed for the structure of the crane travelling wheel with an elastic insert. P_{cr} = 1267 MPa was obtained. The allowable wheel pressure on rail for the crane travelling wheels is adopted to be within 250 MPa, i. e. the available stability margin is n_c = 1267/250 = 5.1. As we can see, the stability margin is more than sufficient.

Originality. A new methodology for the three-layer cylindrical shell under external pressure calculation is developed. A quantitative assessment of the crane travelling wheel with flexible insert critical pressure is carried out.

Practical value. A determination methodology for critical pressure of a three-layer cylindrical structure under external pressure is created.

Keywords: *three-layer shell, external pressure, elasticity modulus, critical pressure, stability, travelling wheel*

Introduction. The structure designing is based on special theoretic and experimental research methods. The most relevant tasks that arise when developing new structures are determining the minimum weight parameters at a specified load. For cylindrical shells, not only the issues of strength but the issues of stability are essential as well.

Multilayer structures, mainly three-layer plates and shells, are utilized in various fields of technology, such as construction, shipbuilding and aircraft industry, rocket and space engineering and machinery.

The three-layer structure is composed of two strong external layers connected using filler. The filler is a material with less strength than the external layers, yet it provides the shell with elastic properties.

Literature review. Reinforced and multilayer shells are quite often used in modern machines, yet the methodologies for their strength and stability calculation have no proper development [1, 2]. The layered-fibrous structure is an important feature of composites since it is characterized by the availability of reinforced elastic layers of various physical and mechanical properties [3, 4]. Yet resolving the practical issues of designing remains a relevant aspect of the methodological support [5, 6].

The dynamic properties of three-layer cylindrical shells depending on the intermediate layer thickness are studied in the paper [7]. This allows obtaining the vibration frequency of such a structure under various initial conditions. Yet the issues related to the multilayer structure strength and stability remain to be unresolved.

The strength issues of the three-layer cylindrical shell of finite length with constant inhomogeneity in the temperature field are solved in the paper [8]. The studies allowed drawing a conclusion about the inhomogeneity occurrence reasons of the stress-strain state of materials. But hereby, the task of strength and stability of this type of shell at a variable temperature field has not been resolved due to objective difficulties in resolving such a task using analytical methods.

The load impulse action on a three-layer cylindrical shell in an elastic medium is studied in the paper [9]. The authors proposed a dynamical damping mechanical and mathematical model of such cylindrical shell in an elastic medium. But hereby, analysis of this kind of load effect on the strength and stability of the three-layer shell itself has not been conducted.

The solution of the three-layer cylindrical structure strength problem is obtained in the paper [10]. This solution allows determining stress in the shell depending not only on its geometrical parameters but on the filler shear modulus as well, which improves the designing plausibility and operation of this kind of structure. Yet the issue of the three-layer cylindrical shell stability remains to be unresolved regardless of the fact that for thin-wall cylindrical shells the issues of stability are paramount.

The vibration characteristics of a multilayer cylindrical shell with supporting rings, loaded with an internal pressure at different fixing of the shell ends are considered in the paper [11]. The study showed that the availability of supporting rings increases the natural oscillation frequency. Herewith, the fact of local stress occurrence in the places of welding supporting rings that have quite high values and effect on the cylindrical shell stress-strain state was not taken into account.

Analysis of vibration of a multilayer structure with supporting rings is conducted in the paper [12]. The study showed that the oscillation frequency is decreased with decreasing the ratio of the shell length to its radius. In addition, the fact of oscillation frequency decreasing with increasing the central layer thickness is determined. However, the authors did not take into account the local stresses that emerge in the supporting rings welding-on places and the effect of these stresses on the static and dynamic loads during these shells' activity.

Based on the theory of shallow shells, a method for determining the critical external pressure for orthotropic and isotropic cylindrical shells is obtained in the paper [13]. Significant effect of the shell loading rate on the critical dynamic pressure is proven. But the analytical value for determining the critical pressure is not obtained in the paper. The method for determining critical external pressure is obtained using the numerical method only, which impairs the calculation practice. In addition, the three-layer structure stability is not considered.

Multilayer structure calculation with account for the creep is considered in the paper [14]. The authors prove that creep of the middle layer has a positive effect on the stress-strain state of the whole structure. At the same time, the effect of other factors on the cylindrical shell stressed state is not considered. A numerical calculation method is obtained, which significantly decreases the value of calculation.

Unsolved aspects of the problem. As we can see from the literature sources [1—14], the stability of a three-layer cylindrical shell under external pressure was not considered by the authors.

Purpose. The objective of the study is to develop calculation methodology for three-layer shell stability, which significantly improves the calculation practice for these structures regarding the determination of external critical pressure for three-layer shells.

To reach the goal one has to solve the following tasks:

- determining the factors affecting the three-layer cylindrical shell stability;
- determining the critical pressure value for the considered type of shells with account for the shell layers geometrical and elastic properties as well as the law of external pressure distribution.

Presentation of the main material. To determine the critical pressure at elastic action of the cylindrical shell wall, the formula of Mises is used, but his theory was further supplemented.

When determining the critical external pressure, the method of variational calculation using the Euler equation of the mixed variational problem was used.

To determine the multilayer structure stability one has to account for the factors of significant influence on its strength and stability as much as possible.

First of all, this is the reduced modulus of three-layer wall elasticity. Bending stiffness Dh is substituted with flexural stiffness of three-layer shell with account for the shear deformation.

$$D_{sh} = D_1 + D_2 + \frac{D_{12}}{1+\dfrac{2D_{12}hn^2(1+w^2)^2}{(2h+\delta_{sl})^2 R^2 G_f}}, \tag{1}$$

where $D_1 = \dfrac{E\delta_{s1}^3}{12\left(1-\mu^2\right)}$; $D_2 = \dfrac{E\delta_{s2}^3}{12\left(1-\mu^2\right)}$;

$$D_{12} = \frac{2E\delta_{s1}\delta_{s2}(h+0.5\delta_{ws})^2}{12(1-\mu^2)\delta_{ws}}, \quad w = \frac{m\pi}{L};$$

$$\omega = \frac{\dfrac{m\pi}{L}}{\dfrac{n}{R}} = \frac{\dfrac{\pi R}{n}}{\dfrac{L}{m}},$$

where m is the amount per half-waves along the shell; L is the shell length; N is the amount per half-waves around the shell.

To determine the three-layer cylindrical shell stability, the equivalent thickness of the wall is replaced with the aggregate thickness of bearing layers

$$2\delta_{ws} = \delta_{s1} + \delta_{s2}, \tag{2}$$

where δ_{n1}, δ_{n2} are thicknesses of the bearing layers.

It is assumed that the external pressure, which affects the shell, is distributed according to the formula

$$p(x) = p_0(1 - ax^2). \tag{3}$$

Under radial compression the shell displacements are radial

$$w = \xi(x)\cos n\varphi. \tag{4}$$

Where $\xi(x)$ is the function of cylindrical shell radial deflection.

Under continuous pressure growth up to pcr, the action of external load is converted into the radial pressure potential energy.

The condition for equality of the orthotropic structure inner and external forces action, which is in the state of neutral equilibrium with radial displacement, is

$$U = \int_0^L \Gamma dx = 0, \tag{5}$$

where

$$\Gamma = \oint\left[\frac{1}{2}m_\varphi x_\varphi + \frac{1}{2}\sigma_x\delta_c\varepsilon_x - \frac{1}{2}p_{cr}R_{wx\varphi}\right]Rd\varphi, \tag{6}$$

where m_φ are transverse momenta of bend

$$m_\varphi = D(x_\varphi + \mu x_x),$$

where x_φ is curvature change in the mean surface in the circumferential direction

$$x_\varphi = -\frac{1}{R}\left(\frac{d^2w}{d\varphi^2} + w\right).$$

D and R are, respectively, the elasticity modulus and radius of the shell mean surface; x_x is curvature change in the mean surface in the generatrix direction

$$x_x = -\frac{d^2w}{dx^2},$$

where μ is Poison's ratio; σ_x are additional normal axial stresses,

$$\sigma_x = E\varepsilon_x,$$

where E is elasticity modulus; ε_x is relative axial deformation.

$$\varepsilon_x = \frac{du}{dx} = -\frac{R}{n^2}\frac{d^2\xi\left(x\right)}{dx^2}\cos n\varphi.$$

After substituting these values we obtain

$$\Gamma = [\theta_1\xi^2(x) + \theta_2\theta_3 - \theta_4 p_{cr}\xi^2(x)]\pi R, \tag{7}$$

where $\theta_1 = \dfrac{D_h(n^2-1)^2}{2R^4}$; $\theta_2 = \dfrac{E\delta_l R^2}{2R^4}$; $\theta_3 = \left(\dfrac{d^2\xi\left(x\right)}{2R^4}\right)^2$;

$\theta_4 = \dfrac{(n^2-1)}{2R}$.

The previously obtained equation for the three-layer shell deflection function is as follows [10]

$$\xi(x) = e^{k_n x}\left(C_1\sin k_n x + C_2\cos k_n x\right) + \frac{k}{k_n^4}p_0\theta_5, \tag{8}$$

where

$$C_1 = \frac{p_0\alpha\cos\varphi}{x^4\left(\alpha^2-\beta^2\right)}\cos\beta\frac{L}{2};$$

$$C_2 = \frac{p_0 \alpha \cos\varphi}{x^4(\alpha^2 - \beta^2)\left(e^{\alpha\frac{L}{2}} - e^{-\alpha\frac{L}{2}}\right)\sin\beta\frac{L}{2}};$$

$$k_n^4 = \frac{\frac{2.28}{G_f R^2 F} - \frac{33.44}{D_t}}{4ER\left(\frac{33{,}266.89}{G_f^2 R^4 F^2} - \frac{58923}{D_t^2}\right)};$$

$$k = \frac{\frac{103.2}{D_t} - \frac{39.71}{G_f R^2 F}}{2\pi R^2\left(\frac{33{,}266.89}{G_f^2 R^4 F^2} - \frac{58{,}923}{D_t}2\right)}; \quad x^4 = \frac{4E\delta_{ws}}{D_t R^2\left(\delta_{ws} + 1\right)};$$

$$2v^2 = \frac{E\delta_{ws}}{R^2 h G_{ws}\left(2\delta_{ws} + 1\right)};$$

$$\alpha = \sqrt{\frac{x^2 + v^2}{2}}; \quad \beta = \sqrt{\frac{x^2 - v^2}{2}}; \quad \theta_5 = \left(1 - ax^2\right).$$

The solution to equation (7) is

$$p_{cr} = \frac{\left[\frac{D_{sh}\left(n^2 - 1\right)}{2R^4} + \frac{E\delta_{ws} R\tau_2}{2n^4\tau_1}\right]2R}{n^2 - 1}, \tag{9}$$

where

$$\tau_1 = \gamma_1 + \gamma_2 + \gamma_3 + \gamma_4 - \gamma_5 - \gamma_6; \tag{10}$$

$$\gamma_1 = \frac{(e^{2k_n L} - 1)}{4k_n}(C_1^2 + C_2^2);$$

$$\gamma_2 = \frac{(e^{2k_n L} - 1)[\cos 2k_n L - 1] + \sin 2k_n L}{4k_n}\theta_6;$$

$$\theta_6 = C_2^2 - C_1^2;$$

$$\gamma_3 = 2C_1 C_2 \frac{(e^{2k_n L} - 1)[\sin 2k_n L - 1]}{4k_n};$$

$$\gamma_4 = \frac{p_0 k (e^{k_n L} - 1)}{k_n^5}\left\{C_1\theta_7 + C_2\theta_8\right\};$$

$$\theta_7 = [\sin k_n L - (\cos k_n L) - 1];$$

$$\theta_8 = [(\cos k_n L - 1) + \sin k_n L];$$

$$\gamma_5 = \frac{akC_1(e^{k_n L} - 1)}{k_n^5}\left[\theta_9 \sin k_n L - \theta_{10}(\cos k_n - 1)\right];$$

$$\theta_9 = L^2 - \frac{1}{k_n}; \quad \theta_{10} = L^2 + \frac{1}{k_n};$$

$$\gamma_6 = \frac{akC_2(e^{k_n L} - 1)}{k_n^5}\theta_{11} - \theta_{12};$$

$$\theta_{11} = \left[\left(L^2 - \frac{1}{k_n}\right)(\cos k_n L - 1)\left(L^2 + \frac{1}{k_n}\right)\sin k_n L\right];$$

$$\theta_{12} = \frac{k^2 p_0^2 L}{k_n^8}\left(\frac{aL^2}{3} - 1\right);$$

$$\tau_2 = \gamma_7 - \gamma_8 + \gamma_9 + \gamma_{10} - \gamma_{11}; \tag{11}$$

$$\gamma_7 = C_1^2 k_n^3 (e^{2k_n L} - 1)\left[1 + 2(\cos 2k_n L - 1) + \theta_{13}\right];$$

$$\theta_{13} = \sin 2k_n L;$$

$$\gamma_8 = C_1 C_2 k_n (e^{2k_n L} - 1)\left[\sin 2k_n L - (\cos 2k_n L - 1)\right];$$

$$\gamma_9 = \frac{4k^2 a^2 p_0^2 L}{k_n^8};$$

$$\gamma_{10} = C_2^2 k_n^3 (e^{2k_n L} - 1)\left[1 + 2(\cos 2k_n L - 1) + \theta_{14}\right];$$

$$\theta_{14} = \sin 2k_n L;$$

$$\gamma_{11} = \frac{8kap_0}{k_n^3}(e^{2k_n L} - 1)C_2\left[\sin k_n L - (\cos k_n L - 1)\right].$$

An analytical solution is obtained for determining the external critical pressure for multilayered cylindrical shells under external pressure.

Let us perform a calculation for a three-layer cylindrical shell (Figs. 1–3) with the following parameters: $R = 100$ mm, $L = 80$ mm, the material of the external layers is cast iron SCh 18–36 with elasticity modulus $P = 1.4 \cdot 10^5$ MPa.

The inner layer material is rubber mix 7–7130 with elasticity modulus $P_{fil} = 80.1$ MPa, $h = 20$ mm, $\delta_{n1} = 22$ and $\delta_{n2} = 45$ mm.

From (9), the first constituent is much greater than the second one, so in engineering calculation, with the purpose of simplicity, only the first constituent can be taken into consideration, and then $P_{cr} = 1{,}267$ MPa is obtained. The allowable wheel pressure on rail for the crane travelling wheels is adopted to be within 250 MPa, i. e. the available stability margin is $n_c = 1{,}267/250 = 5.1$. As we can see, the stability margin is more than sufficient.

For further research, a travelling wheel with an elastic ring was proposed [15]. Earlier experiments showed that this kind of wheel has a number of advantages compared with solid wheels [16].

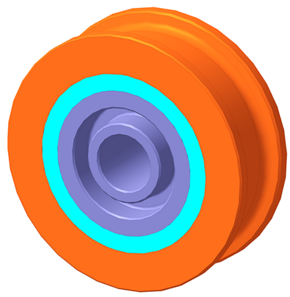


Fig. 1. Crane travelling wheel with elastic ring located in the rim

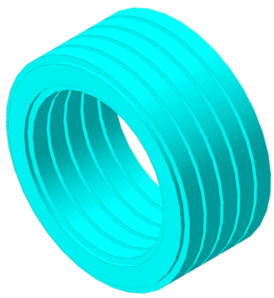
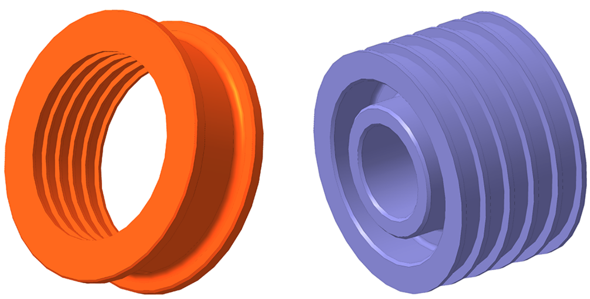


Fig. 2. Elastic ring



a *b*

Fig. 3. Structure of a crane travelling wheel with elastic ring: a – rim; b – hub

The elastic ring (Fig. 2) is of stepped shape that enters the grooves cut on the wheel inner surface (Fig. 3).

Due to h thick filler, three-layer shells with relatively small weight feature high strength and rigidity, which is explained by a high value of the moment of inertia of the entire wall. Compared with an ordinary single-layer wall, the moment of inertia can increase tenfold.

The effect of the shell external layer thickness on the critical pressure (Fig. 4) as well as the effect of the elastic insert thickness on the critical pressure (Fig. 5) was also studied.

The multilayer shell stability was considered by V. T. Lizin. V. T. Lizin gives the formula of critical pressure for three-layer shells with supported ends

$$\sigma_{cr} = \frac{E}{R^2}\left(\frac{L}{\pi m}\right)^2 + \frac{D_{1p}}{2\delta_{sr}}\left(\frac{\pi m}{L}\right)^2 + \frac{D_1 - D_{1p}}{2\delta_{sr}\left(\frac{L}{\pi m}\right) + \frac{2\delta_{sr} h (D_1 - D_{1p})}{G_m (h+\delta_{sr})}}. \quad (12)$$

Let us perform calculations for the crane travelling wheel given in Fig. 1.

The results of the analysis conducted for the effect of various shell layers thickness effect on the critical stresses are given in Figs. 6–7.

The study conducted allowed for obtaining the calculation methodology of three-layer shell stability.

Results. Comparing the obtained critical pressure calculation results according to the methodology of V. T. Lizin, one can conclude that the otherwise stability margin will amount to 100,480/250 = 401.9. This stability margin value is quite doubtful, the external layers material under this kind of stress loses its structure.

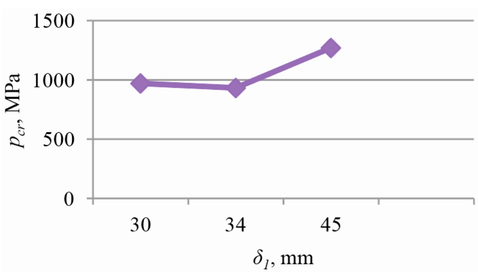


Fig. 4. Effect of the cylindrical shell external layer thickness on the critical pressure

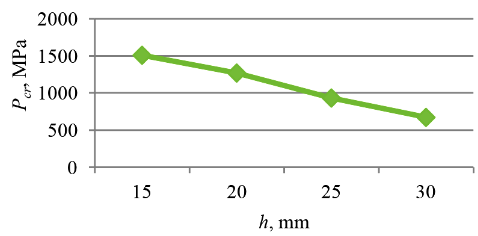


Fig. 5. Effect of the elastic insert thickness on the three-layer cylindrical shell critical pressure

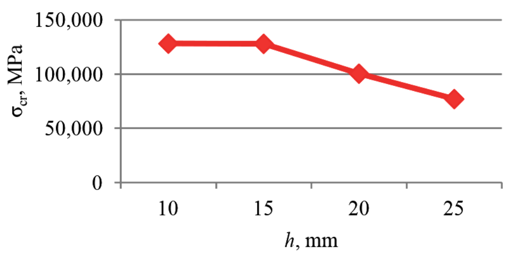


Fig. 6. Graph of the three-layer shell filler thickness effect on the critical pressure

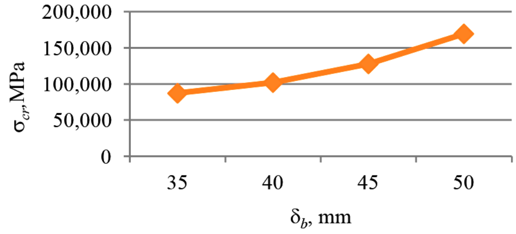


Fig. 7. Graph of the three-layer shell inner layer thickness effect on the critical pressure

In addition, V. T. Lizin's calculation does not take into account the law of external pressure distribution over the cylindrical shell, while the proposed methodology accounts for this point.

The calculation proposed in this paper can be applied to a three-layer cylindrical structure, whose distributed external pressure is in accordance with (7). In case if the distribution law differs, a new value of shell deflection should be obtained, i. e. the (8), and the critical pressure shall be determined as per the methodology proposed.

The obtained methodology is universal since it allows calculating the three-layer cylindrical shell, which is under external pressure.

Conclusion.

1. A new methodology for calculating three-layer cylindrical shell stability with account for geometrical and elastic properties of all layers as well for the law of external pressure distribution is presented in the paper.

2. For the travelling wheel proposed, which consists of three layers, the critical pressure will amount to 1,267 MPa, while the maximum pressure occurring in the zone of the travelling wheel is 2.7 MPa. As we can see, the proposed crane wheel structure is a very strong structure from the point of view of stability.

References.

1. Pavliuk, A. V. (2017). Dynamics of three-layer cylindrical shells elliptical cross-sectionwith a longitudinal-transverse discrete ribbed filler. *Physics and Chemistry of Solid State, 18*(2), 243-248. https://doi.org/10.15330/pcss.18.2.243-248.

2. Ghamkhar, M., Naeem, M., Imran, M., & Soutis, C. (2019). Vibration analysis of a three-layered FGM cylindrical shell including the effect of ring support. *Open Physics, 17*(1), 587-600. https://doi.org/10.1515/phys-2019-0060.

3. Kairov, O. S., Vlasov, O. I., & Latanska, L. O. (2017). Natural vibrations of structurally inhomogeneous multilayer orthotropic cylindrical shells made of composite materials. *Visnyk Zaporizkoho Natsionalnoho Universytetu. Fizyko-matematychni nauky*, (2), 57-65.

4. Larin, O., Morozov, O., Nazarenko, S., Chernobay, G., Kalynovskyi, A., Kovalenko, R., ..., & Pustovoitov, P. (2019). Determining mechanical properties of a pressure fire hose the type of "T". *Eastern-European Journal of Enterprise Technologies, 6*/7(102), 63-70. https://doi.org/10.15587/1729-4061.2019.184645.

5. Otrosh, Y., Kovalov, A., Semkiv, O., Rudeshko, I., & Diven, V. (2018). Methodology remaining life time determination of the buildingstructures. *MATEC Web of Conferences, 230*, 1-7. https://doi.org/10.1051/matecconf/201823002023.

6. Mohamed, Ali J., Alsubari, S., & Aminanda, Y. (2016). Hygrothermoelastic analysis of orthotropic cylindrical shells. *Latin American Journal of Solids and Structures, 13*, 573-589. https://doi.org/10.1590/1679-78252249.

7. Surianinov, M., Yemelianova, T., & Lazarieva, D. (2018). Analytical and computer research of stability of three layer shells, supported by stiffness ribs. *International Journal of Engineering & Technology, 7*(4), 6797-6800. https://doi.org/10.14419/ijet. v7i4.24672.

8. Shah, A., Ali, A., Naeem, M., & Arshad, S. (2012). Vibrations of three-layered cylindrical shells with FGM middle layer resting on Winkler and Pasternak foundations. *Advances in Acoustics and Vibration, 2012*, 1-11. https://doi.org/10.1155/2012/507031.

9. Leonenko, D. V., & Starovoitov, E. I. (2015). Impulsive action on the three-layered circular cylindrical shells in elastic media. *Izvestia*

Saratov University, 15(2), 202-209. https://doi.org/10.18500/1816-9791-2015-15-2-202-209.
10. Fidrovska, N. M., Slepuzhnikov, E. D., & Perevoznyk, I. A. (2019). Determining the strength of a three-layer cylindrical shell. *Mashynobuduvannia, 24*, 17-23. https://doi.org/10.32820/2079-1747-2019-24-17-23.
11. Isvandzibaei, M., Jamaluddin, H., & Hamzah, R. (2013). Effects of ring support and internal pressure on the vibration behavior of multiple layered cylindrical shells. *Advances in Mechanical Engineering, 2013*, 1-13. https://doi.org/10.1155/2013/791917.
12. Ghamkhar, M., Naeem, M., Imran, M., Kamran, M., & Soutis, C. (2019). Vibration frequency analysis of three-layered cylinder shaped shell with effect of FGM central layer thickness. *Scientific Reports, 9*, 1-13. https://doi.org/10.1038/s41598-018-38122-0.
13. Dubrovin, V. M., & Butina, T. A. (2014). Modeling the dynamic stability of a cylindrical shell under the action of external overpressure. *Nauka i innovatsii, 6*, 1-14. https://doi.org/10.18698/2308-6033-2014-6-1237.
14. Chepurnenko, A. S., Yazyiev, B. M., & Lapina, A. P. (2018). Calculation of a three-layer cylindrical shell taking into account the creep. *Stroitelstvo i arhitektura, 4*(21), 14-18. https://doi.org/10.29039/article_5c35ed650 acfc0.48169070.
15. Fidrovska, N. M., Slepuzhnikov, E. D., Larin, O. O., Varchenko, I. S., Lipovyi, V. O., Afanasenko, K. A., & Harbuz, S. V. (2020). Increase of operating reliability of the travel wheel using the use of the elastic inserts. *EUREKA: Physics and Engineering, 5*(30), 69-79. https://doi.org/10.21303/2461-4262.2020.001387.
16. Fidrovska, N. M., Slepuzhnikov, E. D., Varchenko, I. S., Harbuz, S. V., Shevchenko, S. M., Chyrkina, M. A., & Nesterenko, V. V. (2021). Determining stresses in the metallic structure of an overhead crane when using running wheels of the new design. *Eastern-European Journal of Enterprise Technologies, 1*/7(109), 22-31. https://doi.org/10.15587/1729-4061.2021.225097.

Визначення стійкості тришарової оболонки ходового колеса з легким заповнювачем

Н. М. Фідровська[1], *Є. Д. Слепужніков*[2], *В. О. Шевченко*[1], *Д. В. Легейда*[3], *С. В. Васільєв*[2]

1 – Харківський національний автомобільно-дорожній університет, м. Харків, Україна, e-mail: nfidrovskaya@ukr.net

2 – Національний університет цивільного захисту України, м. Харків, Україна

3 – Харківський національний університет будівництва та архітектури, м. Харків, Україна

Мета. Розробка методики розрахунку тришарової циліндричної оболонки на стійкість, що значно покращить розрахункову практику таких конструкцій із визначення критичного зовнішнього тиску.

Методика. При визначенні критичного тиску використовувався метод варіаційного розрахунку із застосуванням рівняння Ейлера змішаної варіаційної задачі. Для визначення стійкості тришарової циліндричної оболонки враховувалися ті фактори, що значно впливають на її міцність і стійкість, а саме – наведений модуль пружності тришарової стінки. Жорсткість на вигин D_h була замінена жорсткістю на вигин тришарової оболонки з урахуванням деформації зсуву.

Результати. Досліджено сучасний стан питання стійкості тришарової циліндричної оболонки. Із використанням методів варіаційного розрахунку й рівняння Ейлера змішаної варіаційної задачі складені рівняння умови рівності робіт внутрішніх і зовнішніх сил ортотропної конструкції, що перебуває у стані байдужої рівноваги з радіальним переміщенням. Прийнявши отримане раніше рівняння для радіальних переміщень і підставивши його до рівняння потенційної енергії системи на одиницю довжини, отримали рівняння визначення критичного тиску. Отримане аналітичне рішення було запропоновано для конструкції кранового ходового колеса, що має пружну вставку. Було отримано $p_{кр} = 1267$ МПа. Для кранових ходових коліс допустимий тиск колеса на рейку приймається в межах 250 МПа, тобто маємо запас стійкості рейки $n_c = 1267/250 = 5{,}1$. Як бачимо, запас стійкості більш ніж достатній.

Наукова новизна. Розроблена нова методика розрахунку тришарової циліндричної оболонки під дією зовнішнього тиску. Проведена кількісна оцінка критичного тиску ходового кранового колеса, що має пружну вставку.

Практична значимість. Створена методика визначення критичного тиску тришарової циліндричної конструкції під дією зовнішнього тиску.

Ключові слова: *тришарова оболонка, зовнішній тиск, модуль пружності, критичний тиск, стійкість, колесо ходове*

The manuscript was submitted 13.06.21.