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On the development and practical application of the theory of sound insulation of cylindrical shells

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Abstract

Interest in sound insulation properties of cylindrical shells has been expanding within past five years, their use in reducing environmental noise in various fields was investigated [1, 2, 3]. To fully understand the properties of cylindrical shells we should analyze the mechanical properties of constructions, as well as wave propagation and sound radiation. With this aim in view, theoretical presumptions of different authors on this issue were reviewed. The possibilities of the use of the theory of sound insulation of cylindrical shells and pipelines were elucidated.

The authors set a task of creating sound insulation measures by using the complex of cylindrical shells and pipelines with the elements of other shapes. In particular, it is possible to go over to statistical characteristics. Finally, of interest could be various corrections, conditioned by the ultimate thickness of the shell. In this connection, it is possible to indicate a number of precise solutions of the dynamic theory of elasticity, suitable for shells of any thickness. However, the concrete numerical results here are still too few.

Theoretical solutions carried out in this paper will help to concentrate attention to the use of the properties elucidated in solving practical engineering issues related to reduction of industrial and environmental noise.

Keywords: sound insulation, cylindrical shells and pipelines.

Introduction

The broad application of cylindrical shells as the elements of constructions evokes interest in respect of the calculation of dynamic properties of those shells [1, 2, 3, 4]. As regards the issues of vibrations of pipelines, special interest was focused on the problems relating to the propagation of waves in cylindrical shells [5, 6]. Excitation of vibrations in the shells on the part of the medium, random vibrations, sound insulation and damping and a number of other problems of a more special content was the object of research. Here special attention will be accorded to long lines, since it is the waveguide approach that is adequate at the highest degree to the properties of the construction under study.

Theory

Propagation of waves in the closed cylindrical shell $\left(|\bar{u}| << h << \lambda,\ h << R_r\right)$ on the basis of known equations [7] is given by:

$$\left.\begin{aligned} L_{11}U_1 + L_{12}U_2 + L_{13}U_3 &= \frac{1}{c^2}\frac{\partial^2 U_1}{\partial t^2} \\ L_{21}U_1 + L_{22}U_2 + L_{23}U_3 &= \frac{1}{c^2}\frac{\partial^2 U_2}{\partial t^2} \\ L_{31}U_1 + L_{32}U_2 + L_{33}U_3 &= \frac{1}{c^2}\frac{\partial^2 U_3}{\partial t^2} \end{aligned}\right\} \qquad (1)$$

$$\left.\begin{aligned} &L_{11} = \left(\frac{\partial^2}{\partial t^2} + \frac{1-\sigma}{2R_r^2}\frac{\partial^2}{\partial \varphi^2}\right),\ L_{12} = \frac{1+\sigma}{2R_r}\frac{\partial^2}{\partial z \partial \varphi},\ L_{13} = -\frac{\sigma}{R_r}\frac{\partial}{\partial t}, \\ &L_{21} = \frac{1+\sigma}{2R_r}\frac{\partial^2}{\partial t \partial \varphi},\ L_{22} = \left(\frac{1-\sigma}{2}\frac{\partial^2}{\partial t^2} + \frac{1}{R_r}\frac{\partial^2}{\partial \varphi^2}\right), \\ &L_{23} = -\frac{1}{R_r^2}\frac{\partial}{\partial \varphi},\ L_{31} = \frac{\sigma}{R_r}\frac{\partial}{\partial t},\ L_{32} = \frac{1}{R_r^2}\frac{\partial}{\partial \varphi}, \\ &L_{33} = \left(-\frac{1}{R_r^2} - \frac{h^2}{12}\left[\frac{\partial^2}{\partial t^2} + \frac{1}{R_r^2}\frac{\partial^2}{\partial \varphi^2}\right]^2\right). \end{aligned}\right\} \qquad (2)$$

where $U_1,\ U_2,\ U_3$ are the components of the vector of displacement (see Fig 1), $h,\ R_r,\ c,\ \sigma$ are the parameters of the shell: thickness, radius, sound propagation speed in the shell material, Poisson's coefficient, accordingly

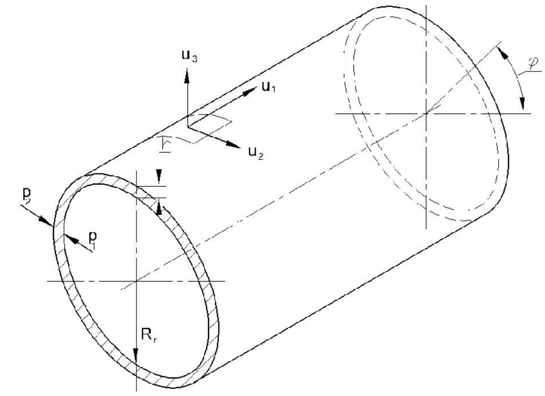


Fig. 1. **Sketch to the choice of designations**

The theory of thin shells was studied by a number of authors [7, 8, 9]. In particular, in the work [9], with the introduction of Young's complex modulus, special damping of vibrations was taken into account. Solution is found in the form of traveling waves $U_i = V_i \exp\left(ikz + in\varphi - i\omega t\right)$. Wave numbers of some lower forms $k = k\left(\bar{\omega}\right)$ are presented in Fig. 2 and 3.

Each traveling wave (of eigen forms of the long duct) is identified at the low frequencies with the wave of a transversal type, shear wave or with the flexural wave, in dependence on the characteristic specificities of deformations and propagation velocity.

There is dependence of eigen frequencies of transversal resonance of the shell on the internal constant pressure. At $p > 0$ the shell gets extended by the internal pressure and frequencies at $p < 0$ the shell gets compressed and when reaching the critical value the loss of stability may occur. The critical values of instability for n^{th} form are according to Euler.

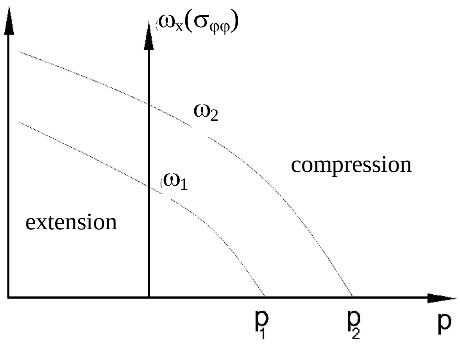


Fig. 2. **Dependence of eigen frequencies of the transversal resonance of the shell on the extension ($\sigma_{\varphi\varphi} > 0$) in the circular direction and compression ($\sigma_{\varphi\varphi} < 0$)**

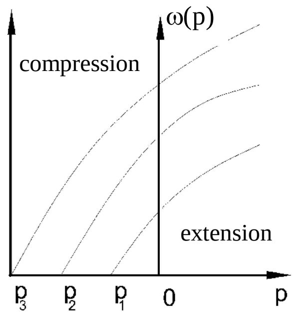


Fig. 3. **Dependence of eigen frequencies of the transversal of the shell on the compression and extension**

Table 1

n	$h^2/12R_r^2 = \mu$							
	$\mu = 10^{-3}$		$\mu = 10^{-4}$		$\mu = 10^{-5}$		$\mu = 10^{-6}$	
	bending	extension	bending	extension	bending	extension	bending	extension
1	0.21	1.41	0.0071	1.41	0.0021	1.41	0.00071	1.41
2	0114	2.24	0.035	2.24	0.0084	2.24	0.0026	2.24
3	0.270	3.16	0.084	3.16	0.032	3.16	0.027	3.16
4	0.490	4.12	0.158	4.12	0.0447	4.12	0.049	4.12
5	0.775	5.10	0.245	5.10	0.100	5.10	0.079	5.10
6	1.12	6.09	0.346	6.08	0.100	6.08	0.104	6.08
7	1.53	7.07	0.49	7.07	0.141	7.07	0.155	7.07
8	2.01	8.07	0.624	8.06	0.200	8.06	0.202	8.06
9	2.54	9.06	0.806	9.06	0.245	9.06	0.256	0.05
10	3.14	10.05	1.0	10.05	0.316	10.05	0.316	10.05

In Table 1 the critical frequencies of the transversal resonance of the closed shell for different μ are presented.

Even though the indicated types of problems are quite of importance for investigation of the propagation of wave energy, further steps in this respect should approximate us to the real object. For example, the significant value of the internal pressure q_0 causes the primary membrane state of stress:

$$\sigma_{\varphi\varphi} = -q_0 \frac{R_r}{h}, \tag{3}$$

where $\sigma_{\varphi\varphi}$ is the hoop tensile stress in a long shell. This circumstance in its turn leads to the considerable increase of resonant frequencies. Actually, the equation of movement of a shell (Eq.1) in this case will contain the following operators [10]:

$$\left.\begin{array}{l}
L_{11} = \left(\frac{\partial^2}{\partial t^2} + \frac{1-\sigma}{2R_r^2}\frac{\partial^2}{\partial\varphi^2}\right),\ L_{12} = \left(\frac{1+\sigma}{2R_r}\frac{\partial^2}{\partial t\partial\varphi} - \frac{q_0R\left(1-\sigma^2\right)}{EhR_r}\frac{\partial^2}{\partial t\partial\varphi}\right), \\
L_{13} = \left(-\frac{\sigma}{R_r}\frac{\partial}{\partial t} + \frac{q_0R_r^2\left(1-\sigma^2\right)}{EhR_r}\frac{\partial}{\partial t}\right), L_{21} = \left(\frac{1+\sigma}{2R_r}\frac{\partial^2}{\partial t\partial\varphi} + \frac{q_0R_r\left(1-\sigma^2\right)}{EhR_r}\frac{\partial^2}{\partial t\partial\varphi}\right), \\
L_{22} = \left(\frac{1-\sigma}{2}\frac{\partial^2}{\partial t^2} + \frac{1}{R_r^2}\frac{\partial^2}{\partial\varphi^2}\right),\ L_{23} = \left(-\frac{1}{R_r^2}\frac{\partial^2}{\partial\varphi^2}\right),\ L_{33} = -\left[1 + \frac{h^2}{12}\left[\frac{\partial^2}{\partial t^2} + \frac{1}{R_r^2}\frac{\partial^2}{\partial\varphi^2}\right]^2 - \right. \\
\left. -\frac{q_0R\left(1-\sigma^2\right)}{EhR_r^2}\left(\frac{\partial^2}{\partial\varphi^2} + 1\right)\right],\ L_{31} = \left(\frac{\sigma}{R_r}\frac{\partial}{\partial t}\right),\ L_{32} = \frac{1}{R_r^2}\frac{\partial}{\partial\varphi}.
\end{array}\right\}, \tag{4}$$

which lead to the following dispersion equation

$$\left.\begin{matrix} \left(\frac{\omega^2}{c^2}-k^2-\frac{1-\sigma}{2R_r^2}n^2\right)\left(-\frac{1+\sigma}{2R_r}kn-\frac{q_0R_r\left(1-\sigma^2\right)}{EhR_r}kn\right)\left(-\frac{\sigma}{R}+\frac{q_0R_r\left(1-\sigma^2\right)}{EhR_r}\right)ik \\ -\left[\frac{1+\sigma}{2R_r}+\frac{q_0R_r\left(1-\sigma^2\right)}{EhR_r}\right]kn\ \left(\frac{\omega^2}{c^2}-\frac{1-\sigma}{2}k^2-\frac{h}{R_r^2}\right)\ \left(-\frac{in}{R^2}\right), \\ \left(\frac{\sigma}{R_r}ik\right)\quad\left(\frac{ih^2}{R^2}\right)\ \left(\frac{\omega^2}{c^2}-\frac{1}{R_r^2}-\frac{h^2}{12}\left[k^2+\frac{h^2}{R_r^2}\right]^2+\frac{q_0R\left(1-\sigma^2\right)}{EhR_r^2}\left(1-n^2\right)\right) \end{matrix}\right\}=0\,. \tag{5}$$

Consequently, it is possible to find the corresponding increase of frequencies (see Fig. 2). Damping of vibrations of the shells was studied, mainly on the basis of notions, related to the concept of the Young's complex modulus. The physical theory of damping did not find the sufficiently broad application [11, 12]. The number of works in this field is quite few [13, 14]. A descriptive method of absorption with the help of introduction of the force of a viscous friction, applied in some works [15], in general is not scientifically well-founded and contradicts the known physical facts [16]. Theoretical issues related to the aerodynamic excitation of shells on the part of the moving medium have been studied insufficiently. Even in the linearized statement, the sufficiently full investigation of the combined movement and acoustic medium is absent. The above-indicated circumstances make analysis of the picture of vibrations of real objects more difficult. The statement of the problems on the interaction of shells with the gaseous medium may be conditionally subdivided into the following conventional groups:

- The forced (random and determined) vibrations of shells.
- Sound insulation of shells.
- Stability of shells in gas flows.

The first series of problems is related with the preliminary solution of a wave equation for the gaseous area, limited by rigid walls. Further, this obtained pressure is introduced into the first part of the equation of the movement of a shell, which takes the following form

$$\begin{aligned} L_{11}U_1+L_{12}U_2+L_{13}U_3&=\frac{1}{c^2}\frac{\partial^2U_1}{\partial t^2}\\ L_{21}U_1+L_{22}U_2+L_{23}U_3&=\frac{1}{c^2}\frac{\partial^2U_2}{\partial t^2}\\ L_{31}U_1+L_{32}U_2+L_{33}U_3&=\frac{1}{c^2}\frac{\partial^2U_3}{\partial t^2}+\frac{p_1-p_2}{\rho c^2h} \end{aligned} \tag{6}$$

Here p_1 and p_2 is the sound pressure on the shell surfaces from inside and outside accordingly (see Fig. 1). The considerable number of problems of that type is given in the book by M.A. Ilgamov [17]. Probably, the indicated type of solution may give the acceptable results when estimating the strength of shells, whereas for the sound insulation purposes it seems to be unacceptable. Issues of sound insulation of shells are exactly conditioned by those subtle effects, in respect of which consideration of interaction is of importance. Problems of interaction of shells and medium were studied by numerous authors [18, 19, 20, 21, 22]. Some of the statements are more distinct, in some of those works clarity and precision of solution are absent. In a general case, solution of such problem is related to the joint consideration of the equation of the movement of a shell (Eq. 2) and of the wave equation for the sound field in the medium under the shell and outside the shell:

$$\begin{aligned} \frac{\partial^2p_1}{\partial t^2}+\frac{1}{r}\frac{\partial}{\partial r}r\frac{\partial}{\partial r}p_1+\frac{1}{r^2}\frac{\partial^2p_1}{\partial\varphi^2}&=\frac{1}{c_0^2}\frac{\partial^2p_1}{\partial t^2},\\ \frac{\partial^2p_2}{\partial t^2}+\frac{1}{r}\frac{\partial}{\partial r}r\frac{\partial}{\partial r}p_2+\frac{1}{r^2}\frac{\partial^2p_2}{\partial\varphi^2}&=\frac{1}{c_0^2}\frac{\partial^2p_2}{\partial t^2}. \end{aligned} \tag{7}$$

where c_0 is the velocity of sound in the medium, at the corresponding boundary conditions on the inner and outer surfaces of a shell, meeting the requirement of the absence of breaks (cavitation) between the shell and the medium

$$\rho_0\frac{\partial^2U_3}{\partial t^2}=-\frac{\partial p_1}{\partial r}=-\frac{\partial p_2}{\partial r},\qquad r=R_r\,, \tag{8}$$

where ρ_0 is the density of the medium.

For mathematical solution of the problem, it is also necessary to set the excitation in the form of the sources of sound situated at the sufficient distance from the shell. Finally, it is important to define the concept of sound insulation in such a way that this concept would reflect the essence of the phenomenon and could be measured. As the coefficient of sound insulation it is possible to take the relation of some value, characterizing the field, in the presence of the shell to the corresponding value when the sound-insulating device (shell) is absent. This definition may be demonstrated by the examples from works [23] and [24]. For the case when the radiator represents the cylindrical dipole, located on the axis of the shell, sound insulation for points, situated outside the shell, will be in a decibel scale:

$$\begin{aligned} R=20\lg\frac{\pi\rho c^2h}{2\rho_0c_0^2R_r}+10\lg\Bigg\{&\left(\frac{1}{\Omega^2\alpha-1}-\Omega^2\alpha+\frac{d^2}{12}+1\right)\dot{I}_1^4\left(\Omega\right)+\\ +&\left[\frac{1}{\Omega^2\alpha-1}-\Omega^2\alpha+\frac{d^2}{12}+1\right]\dot{I}_1\left(\Omega\right)\dot{N}_1\left(\Omega\right)+\frac{2\rho_0c_0^2R_r}{\pi\rho c^2h}\Bigg\}, \end{aligned} \tag{9}$$

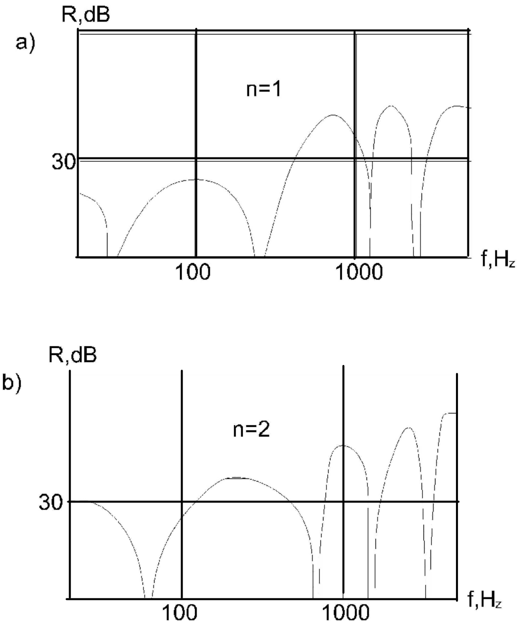
where $\Omega=\dfrac{\omega R_r}{c_0}$; $\alpha=c_0^2/c^2$; $d=h/R_r$; $I_n\left(x\right)$ $N_n\left(x\right)$ are the functions of Bessel and Neumann, accordingly; *R* is the sound insulation.

For the case of the multipole of *n*-th order, the sound insulation will be

$$R = 20\lg\frac{\pi\rho c^2 h}{2\rho_0 c_0^2 R_r} + 10\lg\left\{\left(\frac{1}{\Omega^2\alpha - h^2} - \Omega^2\alpha + \frac{h^2 d^2}{12} + 1\right)\right. \qquad (10)$$

$$\left.\dot{I}_n(\Omega) + \left(\frac{n^2}{\Omega^2\alpha - n^2} - \Omega^2\alpha + 1\right)\cdot\dot{I}_n(\Omega)\,\dot{N}_n(\Omega)\right\}.$$

The corresponding curves are presented in Figs 4 a and b.



Figs 4. **Sound insulation of the cylindrical shell (*h*=2 mm, R_r=300 mm) in air at the multipole (*n*=1 and *n* =2) sources of sound, located on the axis**

Other statements of the problems of sound insulation with the use of statistical specifications are given in works [13, 25].

Issues of the stability of shells in the gas flow with or without consideration of the boundary layer are linked with the joint solution of equations of the theory of shells and equations of mechanics of the moving continuous medium [26]. Here also are possible some or other approximated approaches, however, the numerical results are quite few as a result of awkwardness of solutions [22, 27].

The problems are connected with the simultaneous solution of equations of the movement of shells (Eq.2) and wave equations for the moving medium with the velocity *v:*

- under the shell

$$\nabla^2\varphi_1 = \frac{1}{c_0^2}\left(\frac{\partial}{\partial t} + V\frac{\partial}{\partial x}\right)^2\varphi\,. \qquad (11)$$

- the medium at rest outside the shell

$$\nabla^2\varphi_2 = \frac{1}{c_0^2}\frac{\partial^2}{\partial t^2}\varphi\,. \qquad (12)$$

- at the boundary conditions, reflecting the contact of media with the shell

$$\left(\frac{\partial U_3}{\partial t} + V\frac{\partial U_3}{\partial x}\right) = \frac{\partial\varphi_1}{\partial r} = \frac{\partial\varphi_2}{\partial r}. \qquad (13)$$

In addition

$$p_1 = -\rho_0\left(\frac{\partial\varphi_1}{\partial t} + V\frac{\partial\varphi}{\partial x}\right),\ \ p_2 = -\rho_0\frac{\partial\varphi_2}{\partial t}\,. \qquad (14)$$

Roots of the dispersion equation obtained there have the imaginary and real parts that may take the negative value:

$$U_i = V_i\,\mathrm{e}^{ikx+in\varphi-i\omega t},\ \ I_m\ \ k<0 \qquad (15)$$

this exactly pointing to the possible instability (of a flutter type).

Other statements of problems on vibration of cylindrical shells exist that may find application as regards the issues of dynamics of gas pipelines. Of special importance, for example, may be the problems concerning the propagation of vibrations through the limited shell at the heterogeneous boundary conditions. This problem may serve as the calculation model of excitation of vibrations in the gas pipeline from compressors. Let, for example, excitement be set in the form of the kinematic boundary conditions:

- at $z=0$

$$\begin{cases} U_1(0,\varphi,t) = \Psi_1(\varphi)\mathrm{e}^{-i\omega t},\ U_2(0,\varphi,t) = \Psi_2(\varphi)\mathrm{e}^{-i\omega t} \\ U_3(0,\varphi,t) = \Psi_3(\varphi)\mathrm{e}^{-i\omega t},\ \dfrac{\partial U_3}{\partial t} = \Psi_4(\varphi)\mathrm{e}^{-i\omega t} \end{cases} \qquad (16)$$

- at $z=l$

$$\begin{cases} U_1(l,\varphi,t)=0,\ U_2(l,\varphi,t)=0,\ U_3(l,\varphi,t)=0, \\ \dfrac{\partial U_3}{\partial t}=0. \end{cases} \qquad (17)$$

Expanding the boundary conditions into Fourier series along the angular coordinate φ and dividing variables, we obtain from (Eq.1) the following system of ordinary differential equations with the constant coefficients

$$\left.\begin{aligned} L_{11}^*U_1(z) + L_{12}U_2(z) + L_{13}U_3(z) = 0 \\ L_{21}U_1(z) + L_{22}^*U_2(z) + L_{23}U_3(z) = 0 \\ L_{31}U_1(z) + L_{32}U_2(z) + L_{33}^*U_3(z) = 0 \end{aligned}\right\}, \qquad (18)$$

where

$$L_{11}^* = \left(\frac{\omega^2}{c^2} - \frac{1-\sigma}{2R_r^2}n^2 + \frac{d^2}{dt^2}\right),\ L_{12} = \left(\frac{1+\sigma}{2R_r}in\frac{d}{dz}\right)$$

$$L_{13} = -\frac{\sigma}{R_r}\frac{d}{dz},\ L_{21} = \left(\frac{1+\sigma}{2R_r}in\frac{d}{dt}\right),\ L_{22} = \left(\frac{\omega^2}{c^2} - \frac{n^2}{R_r^2}\frac{1-\sigma}{2}\frac{d^2}{dt^2}\right)$$

$$L_{23} = \left(-\frac{in}{R_r^2}\right),\ L_{31} = \left(\frac{\sigma}{2R_r}\frac{d}{dt}\right),\ L_{32} = \left(\frac{in}{R_r^2}\right)$$

$$L_{33}^* = \left[\frac{\omega^2}{c^2} - \frac{1}{R_r^2} - \frac{h^2}{12}\left(\frac{d^2}{dt^2} - \frac{n^2}{R_r^2}\right)^2\right] \qquad (18a)$$

with boundary conditions:

- at $z=0$

$$U_1(0)=\psi_{1n},\ U_2(0)=\psi_{2n},\ U_3(0)=\psi_{3n},\ \frac{dU_3}{dz}=\psi_{4n}, \qquad (19)$$

- at $z=l$

$$U_1(l)=0,\ U_2(l)=0,\ U_3(l)=0,\ \frac{dU_3}{dz}=0\,, \qquad (20)$$

The general solution of that system of ordinary equations shall be found by common methods (see Fig 5) [5].

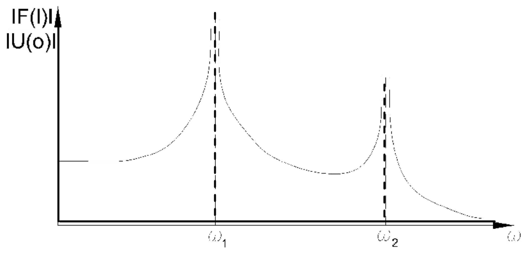


Fig. 5**. Frequency transfer functions of the cylindrical shell: |F(l)| the force of reaction in the sealing; |U(o)| amplitude of excitation at the other end**

In particular, it is possible to go over to statistical characteristics. Finally, of interest here may be different corrections, conditioned by the final thickness of the shell. In this connection, it is possible to point out a number of precise solutions of the dynamic theory of elasticity that are suitable for the shells of any thickness [7, 22]. However, concrete numerical results here have been very few so far.

Conclusions

1. Even though a cylindrical shell is the object of numerous theoretical investigations, the sufficiently precise material, suitable for practical engineering calculations, is still insufficient.

2. Theoretical solutions carried out in this paper will help to concentrate attention to the use of the properties elucidated in solving practical engineering issues related to reduction of industrial and environmental noise.

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Cilindrinių kevalų garso izoliacijos teorijos plėtra ir taikymas praktikoje

Reziumė

Pastaruosius penkerius metus didėja susidomėjimas cilindrinių kevalų garso izoliacijos savybėmis, jų panaudojimu aplinkos triukšmui mažinti [1, 2, 3]. Norint gerai suprasti cilindrinių kevalų savybes, reikia išnagrinėti šių konstrukcijų mechaninius ypatumus, bangų sklidimą ir garso išspinduliavimą. Tam buvo apžvelgtos skirtingos teorinės prielaidos dėl cilindrinių kevalų ir vamzdynų garso izoliacijos panaudojimo galimybių. Autoriai turi tikslą sukurti garso izoliacijos priemones naudodamiesi cilindrinių kevalų ir vamzdynų kompleksu su kitų formų elementais. Straipsnyje aptarti teoriniai sprendiniai padės sutelkti dėmesį į praktinius inžinerinius klausimus, liečiančius gamybinio ir aplinkos triukšmo mažinimą.

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