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VIBRATIONS OF A NONHOMOGENEOUS MEDIUM-CONTACTING CYLINDRICAL SHELL STIFFENED WITH RINGS AND SUBJECTED TO ACTION OF COMPRESSIVE FORCE

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Abstract- The non-homogeneity of a cylindrical shell in thickness may be taken into account by two different methods by introducing sandwich and non-homogeneity function. In this paper the inhomogeneity was taken into account by accepting the Young modulus and the density of the material as a function of coordinate changing in thickness.

Keywords: Three-Dimensional Functional, Nonhomogeneity, Liquid Medium, Variation Principle.

I. INTRODUCTION

Investigation of strength characteristics of medium-contacting constructions and structural elements subjected to the action of external force with regard to nonhomogeneity of distribution of material, influence of medium and compressive force is of great importance.

Parametric vibrations of uniform cylindrical shells with regard to nonhomogeneity in thickness were studied in the papers [1-4]. Using the variational principle in solving the problem, for finding vibration frequencies of the considered system a frequency equation was structured and studied depending on physico-geometrical parameters characterizing the system, the characteristic curves were constructed on the force-frequency plane.

In this papers [5-7], for studying free and forced vibrations of a fluid-filled cylindrical shell stiffened with bars and subjected to the action of compressive force in the axial direction, physical mathematical model was constructed. For the cases of axially symmetric and asymmetric cases of vibrations, the frequency equation of the system was structured and approximate roots were found. The influence of geometrical, physical-mechanical parameters characterizing the system on the found frequencies were studied.

At the same time, the forced vibrations of the system were considered, displacements of the cylinder in resonance frequencies and near it, were calculated. By introducing optimization parameters, the optimal variant of geometrical sizes of the cylindrical shell and the number of bars were found. In the paper, natural vibrations of the system of cylindrical shell, continuum stiffened with non-homogeneous rings and subjected to the action of compressive force are studied.

The non-homogeneity of a cylindrical shell in thickness may be taken into account by two different methods by introducing sandwich [1] and non-homogeneity function. In the paper the inhomogeneity was taken into account by accepting the Young modulus and the density of the material as a function of coordinate changing in thickness.

II. PROBLEM STATEMENT

For taking into account the nonhomogeity of the cylindrical shell in thickness, we will use a three-dimensional functional. In this case, the total energy of the cylindrical shell is as follows:

$$U=\frac{1}{2}\iint\int_{-h/2}^{h/2}\left(\sigma_\alpha e_\alpha+\sigma_\beta e_\beta+\tau_{\alpha\beta}e_{\alpha\beta}+\right.$$
$$\left.+\rho(z)\left(\left(\frac{\partial u}{\partial t}\right)^2+\left(\frac{\partial \vartheta}{\partial t}\right)^2+\left(\frac{\partial w}{\partial t}\right)^2\right)\right)d\alpha d\beta dz \quad (1)$$

where

$$\sigma_\alpha=\frac{T_1}{h}+\frac{12M_1}{h^3}z$$
$$\sigma_\beta=\frac{T_2}{h}+\frac{12M_2}{h^3}z \quad (2)$$
$$\tau_{\alpha\beta}=\frac{s}{h}+\frac{12H}{h^3}z$$

The nonhomogeneity is taken into account by different methods. One of them is to accept the Young modulus and the density of the material as a function of coordinate changing in thickness [1]: $E=E(z)$, $\rho=\rho(z)$. We assume that the Poisson ratio is constant. In this case stress-strain relations are written as follows:

$$e_\alpha=\frac{1}{E(z)}\left(\sigma_\alpha-\nu\sigma_\beta\right)$$
$$e_\beta=\frac{1}{E(z)}\left(\sigma_\beta-\nu\sigma_\alpha\right) \quad (3)$$
$$e_{\alpha\beta}=\frac{2(1+\nu)}{E(h)}\sigma_{\alpha\beta}$$

Taking into account relations (2) and (3), and equality

$$\iint \int_{-h/2}^{h/2} \left(\rho(z)\left(\frac{\partial u}{\partial t}\right)^2 + \left(\frac{\partial \vartheta}{\partial t}\right)^2 + \left(\frac{\partial w}{\partial t}\right)^2 \right) d\alpha d\beta dz =$$

$$= \iint \left(\rho_0 \left(\left(\frac{\partial u}{\partial t}\right)^2 + \left(\frac{\partial \vartheta}{\partial t}\right)^2 + \left(\frac{\partial w}{\partial t}\right)^2 \right) - \right.$$

$$\left. -2\rho_1 \left(\frac{\partial^2 w}{\partial x \partial t} \cdot \frac{\partial u}{\partial t} + \frac{\partial^2 w}{\partial y \partial t} \cdot \frac{\partial \vartheta}{\partial t} \right) + \rho_2 \left(\left(\frac{\partial^2 w}{\partial x \partial t}\right)^2 + \left(\frac{\partial^2 w}{\partial y \partial t}\right)^2 \right) \right) d\alpha d\beta$$

in Equation (1), we can write:

$$V = \frac{1}{h} \iint \left\{ T_1 \left[\frac{1}{E_0}(2T_2 - \nu T_1) + \frac{12}{E_1 h^3}(M_2 - \nu M_1) \right] + \right.$$

$$+ T_2 \left[-\frac{\nu T_2}{E_0} + \frac{12}{E_1 h^3}(M_1 - \nu M_2) \right] + \tag{4}$$

$$+ 2(1+\nu) s \left(\frac{s}{E_0} + \frac{12H}{E_1 h^3} \right) + \frac{72}{E_2 h^6} \times$$

$$\left. \times \left(2M_1 M_2 - \nu M_1^2 - \nu M_2^2 + 2(1+\nu) H^2 \right) \right\} d\alpha d\beta$$

Let us write the total energy of the rings:

$$V_1 = \frac{1}{h} \sum_{j=1}^{k_2} \int_{y_1}^{y_2} \left[E_j F_j \left(\frac{\partial \vartheta_j}{\partial y} - \frac{w_j}{R} \right)^2 + \right.$$

$$+ E_j J_{xj} \left(\frac{\partial^2 w_j}{\partial x^2} + \frac{w_j}{R^2} \right)^2 + E_j F_{zj} \left(\frac{\partial^2 u_j}{\partial y^2} - \frac{\varphi_{kpj}}{R} \right)^2 +$$

$$\left. + G_j J_{kpj} \left(\frac{\partial \varphi_{kpi}}{\partial y} + \frac{1}{R} \frac{\partial u_j}{\partial y} \right)^2 \right] dy + \tag{5}$$

$$+ \sum_{j=1}^{k_2} \rho_j F_j \int_{y_1}^{y_2} \left[\left(\frac{\partial u_j}{\partial t} \right)^2 + \left(\frac{\partial \vartheta_j}{\partial t} \right)^2 + \right.$$

$$\left. + \left(\frac{\partial w_j}{\partial t} \right)^2 + \frac{J_{kpj}}{F_j} \left(\frac{\partial \varphi_{kpj}}{\partial t} \right)^2 \right] dy$$

The influence of medium on the cylindrical shell is replaced by external forces q_x, q_y, q_z. The work done by these forces in displacements of points of the shell is

$$A_0 = -\int_0^{x_1} \int_0^{2\pi} \left(q_x u + q_y \vartheta + q_z w \right) dx dy \tag{6}$$

Let us write potential energy generated in the shell and bar because of influence of compressive stresses σ_x:

$$\Pi = -\frac{\sigma_x h}{2} \int_0^{\xi_1} \int_0^{2\pi} \left(\frac{\partial w}{\partial \xi} \right)^2 d\xi d\theta - \frac{\sigma_x F_c}{2R} \sum_{i=1}^{k_1} \int_0^{\xi_1} \left(\frac{\partial w}{\partial \xi} \right)^2 \Bigg|_{\theta=\theta_i} d\xi \tag{7}$$

The total energy of the considered system will consist of the sum

$$W = V + V_1 + A_0 + \Pi \tag{8}$$

In Equations (1)-(8), u, ϑ, w are displacements of the shell, u_i, ϑ_i, w_i are displacements of the points of the bar, $E\nu$ are modulus of elasticity of the cylindrical shell's material and the Poisson ratio, respectively, R, h are radii and thickness of the cylindrical shell, E_i is the modulus of elasticity of longitudinal bar, F_i is the area of cross section of the longitudinal bar, G_i is the elasticity modulus of the longitudinal bar in shear, I_{yi}, I_{kpi} are inertia moments of the cross-section of the longitudinal bar, k_1 is the amount of longitudinal bars, q_x, q_y, q_z are the components of pressure force influencing on the cylindrical shell as viewed from medium, and $\rho_i = \int_{-h}^{h.} \rho(z) z^i dz$, $\frac{1}{E_i} = \int_{-h}^{h.} \frac{z^i dz}{E(z)}$, $\xi = \frac{x}{R}$, $\theta = \frac{y}{R}$.

It is considered that the rigid contact conditions between the shell and rings are satisfied:

$$\begin{aligned} & u_j(y) = u(x_j, y) + h_j \varphi_j(x_j, y) \\ & \vartheta_j(x) = \vartheta(x_j, y) + h_j \varphi_2(x_j, y) \\ & w_j(x) = w(x_j y);\ \varphi_j = \varphi_2(x_j, y) \\ & \varphi_{kpj}(x) = g j_1(x_j, y) \end{aligned} \tag{9}$$

The system of motion equations of the medium in cylindrical coordinates are written as follows [8]:

$$\begin{aligned} & (\lambda_s + 2\mu_s) \frac{\partial \theta}{\partial r} - \frac{2\mu_s}{r} \frac{\partial \omega_x}{\partial \varphi} + 2\mu_s \frac{\partial \omega_\varphi}{\partial x} - \rho_s \frac{\partial^2 s_x}{\partial t^2} = 0 \\ & (\lambda_s + 2\mu_s) \frac{1}{r} \frac{\partial \theta}{\partial \varphi} - 2\mu_s \frac{\partial \omega_r}{\partial x} + 2\mu_s \frac{\partial \omega_x}{\partial x} - \rho_s \frac{\partial^2 s_\varphi}{\partial t^2} = 0 \\ & (\lambda_s + 2\mu_s) \frac{\partial \theta}{\partial x} - \frac{2\mu_s}{r} \frac{\partial}{\partial r}(r\omega_\varphi) + \frac{2\mu_s}{r} \frac{\partial \omega_r}{\partial \varphi} - \rho_s \frac{\partial^2 s_r}{\partial t^2} = 0 \end{aligned} \tag{10}$$

where, S_x, S_{gj}, S_r are the components of displacement vector of medium, λ_s, μ_s are Lame coefficients of medium, ρ_s is the density of medium, x, r, ϕ are longitudinal, radial peripheral coordinates $a_t = \sqrt{\frac{\lambda_s + 2\mu_s}{\rho_s}}$, $a_b = \sqrt{\frac{\mu_s}{\rho_s}}$.

Volume extension θ and $\omega_x, \omega_\varphi, \omega_r$ component are calculated by the following expressions:

$$\theta = \frac{\partial s_r}{\partial r} + \frac{s_r}{r} + \frac{1}{r} \frac{\partial s_\varphi}{\partial \varphi} + \frac{\partial s_x}{\partial x};\ 2\omega_x = \frac{1}{r} \left[\frac{\partial (r s_\varphi)}{\partial r} - \frac{\partial s_r}{\partial \varphi} \right]$$

$$2\omega_\varphi = \frac{\partial s_r}{\partial x} - \frac{\partial s_x}{\partial r};\ 2\omega_x = \frac{1}{r} \frac{\partial s_x}{\partial \varphi} - \frac{\partial s_\varphi}{\partial x}$$

The stresses generated in medium are expressed by displacements s_x, s_φ, s_r as follows:

$$\begin{aligned} & \sigma_{rx} = \mu_s \left(\frac{\partial s_x}{\partial r} + \frac{\partial s_r}{\partial x} \right) \\ & \sigma_{r\varphi} = \mu_s \left[r \frac{\partial}{\partial r} \left(\frac{s_\varphi}{r} \right) + \frac{1}{r} \frac{\partial s_r}{\partial \varphi} \right] \\ & \sigma_{rr} = \lambda_s \left(\frac{\partial s_x}{\partial x} + \frac{1}{r} \frac{\partial (r s_r)}{\partial r} + \frac{1}{r} \frac{\partial s_\varphi}{\partial \varphi} \right) + 2\mu_s \frac{\partial s_r}{\partial r} \end{aligned} \tag{11}$$

When studying the vibrations of medium and a visco-elastic cylindrical shell stiffened with bars we will consider two cases: a) the inertial influence of medium on vibration process is weak; b) in studying the vibration process inertial influence of medium may not be ignored.

in the case a)

$$\begin{aligned} s_x &= \left[\left((-kr)\frac{\partial I_n(kr)}{\partial r} - 4(1-\nu_s)kI_n(kr)\right)A_s + kI_n(kr)B_s\right]\times \\ &\times \cos n\varphi \cos kx \sin \omega t \\ s_\theta &= \left[-\frac{n}{r}I_n(kr)B_s - \frac{\partial I_n(kr)}{\partial r}C_s\right]\times \\ &\times \sin n\varphi \sin kx \sin \omega t \\ s_r &= \left[-k^2 r I_n(kr)A_s + \frac{\partial I_n(kr)}{\partial r}B_s + \frac{n}{r}I_n(kr)C_s\right]\times \\ &\times \cos n\varphi \sin kx \sin \omega t \end{aligned} \tag{12}$$

in the case b)

$$\begin{aligned} s_x &= \left[A_s k I_n(\gamma_e r) - \frac{C_s \gamma_t^2}{\mu_t} I_n(\gamma_t r)\right]\times \\ &\times \cos n\varphi \sin kx \sin \omega t \\ s_\theta &= \left[-\frac{A_s n}{r}I_n(\gamma_e r) - \frac{C_s nk}{r\mu_t}I_n(\gamma_t r) - \frac{B_s}{n}\frac{\partial I_n(\gamma_t r)}{\partial r}\right]\times \\ &\times \sin n\varphi \sin kx \sin \omega t \\ s_r &= \left[A_s \frac{\partial I_n(\gamma_e r)}{\partial r} - \frac{C_s k}{\mu_t}\frac{\partial I_n(\gamma_t r)}{\partial r} + \frac{B_s n}{r}I_n(\gamma_t r)\right]\times \\ &\times \cos n\varphi \sin kx \sin \omega t \end{aligned} \tag{13}$$

The system of motion equations of medium (10) is complemented by contact conditions. We will assume that in the deformation process, the tangential surfaces of the cylindrical shell and medium shift with respect to another one but are not separated. In this case at the $x = x_1$ and $x = x_2$ the conditions $\sigma_{xx} = 0$; $s_\varphi = s_r = 0$ should be satisfied.

The condition of equality of normal displacement components

$$s_x = u;\ s_\varphi = \vartheta;\ s_r = w \quad (r = R) \tag{14}$$

The conditions of equality of pressure forces

$$q_x = -\sigma_{rx},\ q_y = -\sigma_{r\varphi},\ q_z = -\sigma_{rr} \quad (r = R) \tag{15}$$

Thus, the forces influencing on the cylinder as viewed from the medium may be determined by means of the obtained expressions. As a result, the solution of the stated problem is reduced to joint integration of total energy (8) of the construction consisting of medium-filled cylindrical shell stiffened with discretely distributed rings within the boundary conditions (14) and (15) of the system of motion Equations (10) of the medium.

III. PROBLEM SOLUTION

Represent the expressions of pressure components q_x, q_y, q_z in the following way:

$$q_x = -\left(\tilde{A}_x u_0 + \tilde{B}_x \vartheta_0 + \tilde{C}_x w_0\right)\cos k\varphi \cos\frac{\pi x}{l}\sin \omega t$$

$$q_y = -\left(\tilde{A}_\varphi u_0 + \tilde{B}_\varphi \vartheta_0 + \tilde{C}_\varphi w_0\right)\sin k\varphi \sin\frac{\pi x}{l}\sin \omega t$$

$$q_z = -\left(\tilde{A}_r u_0 + \tilde{B}_r \vartheta_0 + \tilde{C}_r w_0\right)\cos k\varphi \sin\frac{\pi x}{l}\sin \omega t$$

In Equations (12), (13), (17), (18) A_s, B_s, C_s are unknown constant numbers, *k*, *n*, γ_e, γ_t are wave numbers, I_n is a modified, *n*th order, first kind Bessel function, $\gamma_e^2 = k^2 - \mu_e^2$, $\gamma_t^2 = k^2 - \mu_t^2$, $k^* = kR$, ω is unknown frequency. In Equation (16), as the coefficients of the unknowns $\gamma_l^* = \gamma_l R$, $\gamma_t^* = \gamma_t R$, $\mu_t^* = \mu_t R$ are bulky, we do not cite them here u_0, ϑ_0, w_0.

In Equation (8) $u, \vartheta, w, T_1, T_2, M_1, M_2, S, H$ are variation quantities. Let's determine stationary value of functional (8). Therefore,, we use the Rietz method. We will look for unknown quantities in the following way:

$$\begin{aligned} u &= \cos\frac{\pi x}{l}\sin(k\varphi)(u_0\cos\omega t + u_1\sin\omega t) \\ \vartheta &= \sin\frac{\pi x}{l}\cos(k\varphi)(\vartheta_0\cos\omega t + \vartheta_1\sin\omega t) \\ w &= \sin\frac{\pi x}{l}\sin(k\varphi)(w_0\cos\omega t + w_1\sin\omega t) \\ T_1 &= \sin\frac{\pi x}{l}\sin(k\varphi)(T_{10}\cos\omega t + T_{11}\sin\omega t) \\ T_2 &= \cos\frac{\pi x}{l}\cos(k\varphi)(T_{20}\cos\omega t + T_{21}\sin\omega t) \\ S &= \sin\frac{\pi x}{l}\sin(k\varphi)(S_{10}\cos\omega t + S_{11}\sin\omega t) \\ M_1 &= \cos\frac{\pi x}{l}\sin(k\varphi)(M_{10}\cos\omega t + M_{11}\sin\omega t) \\ M_2 &= \sin\frac{\pi x}{l}\sin(k\varphi)(M_{20}\cos\omega t + M_{21}\sin\omega t) \\ H &= \cos\frac{\pi x}{l}\cos(k\varphi)(H_{10}\cos\omega t + H_{11}\sin\omega t) \end{aligned} \tag{17}$$

If we substitute the Equation (13) in functional (8), we get a function dependent on the variables u_0, u_1, ϑ_0, ϑ_1, w_0, w_1, $T_{10}, T_{11}, T_{20}, T_{21}$, S_{10}, S_{11}, M_{10}, M_{12}, $M_{20} M_{22}$, H_{10}, H_{11}. The stationarity condition of the obtained function is determined from following system:

$$\begin{aligned} &1)\frac{\partial J}{\partial u_0} = 0;\ 2)\frac{\partial J}{\partial u_1} = 0;\ 3)\frac{\partial J}{\partial \vartheta_0} = 0;\ 4)\frac{\partial J}{\partial \vartheta_1} = 0;\ 5)\ \frac{\partial J}{\partial w_0} = 0 \\ &6)\frac{\partial J}{\partial w_1} = 0;\ 7)\frac{\partial J}{\partial T_{10}} = 0;\ 8)\frac{\partial J}{\partial T_{11}} = 0;\ 9)\frac{\partial J}{\partial T_{20}} = 0;\ 10)\frac{\partial J}{\partial T_{21}} = 0 \\ &11)\frac{\partial J}{\partial S_{10}} = 0;\ 12)\frac{\partial J}{\partial S_{11}} = 0;\ 13)\frac{\partial J}{\partial M_{10}} = 0;\ 14)\frac{\partial J}{\partial M_{11}} = 0 \\ &15)\frac{\partial J}{\partial M_{20}} = 0;\ 16)\frac{\partial J}{\partial M_{21}} = 0;\ 17)\frac{\partial J}{\partial H_{10}} = 0;\ 18)\frac{\partial J}{\partial H_{11}} = 0 \end{aligned} \tag{18}$$

As the system (18) is homogeneous, for the existence of its nonzero solution, the principal determinant should be equal to zero. As a result, we get the frequency equation:

$$\det\left\|a_{ij}\right\| = 0 \ , \ i, j = 1{,}18 \qquad (19)$$

Equation (19) was studied by numerical method. The following values were taken for the parameters of medium and shell:

$h^* = \frac{h}{R} = 0.25\times10^{-2};\ \nu = 0.3;\ F_j = 5.75\ \text{mm}^2;$

$I_{xj} = 19.9\ \text{mm}^4;\ I_{kpj} = 0{,}48\ \text{mm}^4;$

$E_j = E = 6.67\times10^9\,\text{H/m}^2;$

$\alpha = 0.5;\ \rho_j = 0.26\times10^4\,\text{N.san}^2/\text{m}^2;$

$a_e = 2{,}25a_t;\ a_t = 308\ \text{m/san};\ E_0 = E;\ \rho_0 = \rho_j.$

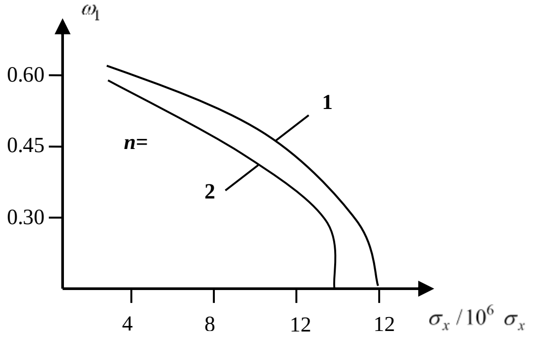

Figure 1. Dependence of frequency ω_1 on compressive force σ_x

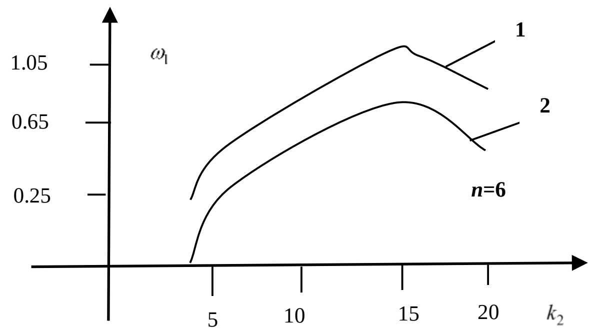

Figure 2. Dependence of the frequency ω_1 on the amount of rings

Two cases of inhomogeneity functions were considered as linear

$$E(z) = E_0\left[1+\alpha\left(\frac{z}{h}\right)\right],\ \rho(z) = \rho_0\left[1+\alpha\left(\frac{z}{h}\right)\right]$$

and parabolic

$$E(z) = E_0\left[1+\alpha\left(\frac{z}{h}\right)^2\right],\ \rho(z) = \rho_0\left[1+\alpha\left(\frac{z}{h}\right)^2\right].$$

where, the Young modulus is α nonhomogeneity parameter. Note that in the case of linear function $|\alpha|<1,$, in the case of parabolic principle α is any number, and $\omega_1 = \sqrt{\frac{(1-\nu^2)\rho_0 R^2\omega^2}{E}}$.

IV. CONCLUTIONS

The result of calculations was given in Figure 1 in the form of dependence of frequency parameter on compressive stress, in Figure 2 in the form of dependence of frequency parameter on the amount of rings. The line of nonhomogeneity laws corresponds to curves 1, parabolic change cases of nonhomogeneity laws correspond to curves 2. The calculations show that the number of vibration frequencies corresponding to linear case of inhomogeneity laws is greater than the number of vibration frequencies corresponding to the case of parabolic change. As is seen from Figure 1, as the value of compressive stress increases frequency of vibrations decreases and tends to zero. As is seen from Figure 2, the frequency of natural vibrations of the system at first increases and gets maximum value and again decreases. This is explained by the fact that as the number of rings increases, the mass also increases and inertial influence to the vibration process of the system amplifies.

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