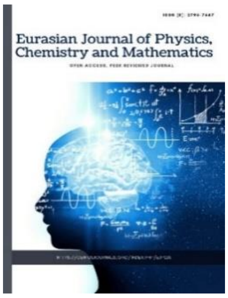

Specific Vibration Of A Fluid-Conducting Toroidal Shell

1.Toyirov Gayrat Fozilovich — Bukhara Engineering and Technology Institute, Bukhara, 105017 Uzbekistan

2.Jurayev Shukhrat Isroilovich — 2..Bukhara State Pedagogical Institute. Bukhara, 200100 Uzbekistan

ABSTRACT

In this work, the problem of specific vibration of the fluid-conducting toroidal shell is solved. The object of the research is to evaluate the ideal fluid and the composite toroidal shell with rheagogic properties, their characteristic (or special) vibrations and dynamic characteristics. Research methods. The integral differential equations with eigenvalues are reduced to the system of ordinary differential equations. To solve these equations, freezing method, separation of variables method, Muller's method, and Godunov's orthogonal projection methods were used. The reliability of the obtained results is based on the correct formulation of the spectral limit problem, the rigor of the derived mathematical expressions, the use of based solution methods and the evaluation of the accuracy of the solutions, as well as comparisons with the solutions of other mathematical problems.

Keywords: toroidal shell, ideal fluid, composite material, deformation, integro-differential equation.

Introduction

If the shell consists of a curved axis, studying its vibrations becomes more complicated. Since there is no theory for the study of a curved shell, it has been studied on a curved stern surface. In the investigation of vibrations in the curved section of shells, we can see that a number of works approached from the point of view of the boom theory studied straight booms under impact load by theoretical and experimental methods [1,2,3]. The analysis of curvilinear sturgeons is much less studied. He introduced equations in the form of Tymoshenko's equation for curved lines [4] and obtained dispersion lines for continuous wave motion. Theoretical and experimental works are published in the following [5]. This work is devoted to elastic wave dispersion in a spiral waveguide. In his work [6], he published his research results for different types of springs. Later, in the works of [8,9,10], wave propagation in curved sturgeons was studied. And finally, in the works of [11], Morley's equations are presented as the most optimal way of checking wave propagation in curved sterns [12], and numerical calculations are performed on several similar examples; in the scientific research of [13], the theoretical works created by [14,15] are summarized. The results of rigging devices consisting of straight and curved sections are given. Also, theoretical and experimental results are compared.

Statement of the problem and solution methods

To compare the calculation results of a straight and curved pipeline, we consider the equation of motion of a cylindrical shell through which liquid flows [1]. We can reduce the equilibrium equations of the shell element to a single displacement equation. For this T_2, Q_2, S we eliminate the stresses, use the relations between strains and stresses, as well as displacements and strains. When considering small displacements, discarding all non-linear parts, we arrive at the following equation:

$$\frac{\partial^3 u}{\partial \xi^3} + h_\nu^2 \frac{\partial^3}{\partial \theta^3}\left(\tilde{\vartheta}_2 + \frac{\partial^3 \tilde{\vartheta}_2}{\partial \theta^3}\right) - \frac{r \cdot p}{Eh}\frac{\partial^3 \tilde{\vartheta}_2}{\partial \theta^3} + \frac{r \cdot p}{Eh}\left(\rho h + \frac{p_{жс} \cdot r}{2}\right) \times$$
$$\times\left(-\frac{\partial^3 u}{\partial \xi \partial t^2} + \frac{\partial^3 \vartheta}{\partial \xi \partial t^2} + \frac{\partial^3 \omega}{\partial \theta^2 \partial t^2}\right) + \frac{\partial^2}{\partial \theta^2}\left[\frac{r}{Eh}\left(p_{жс} \cdot V^2 + p\right)\frac{\partial^3 \omega}{\partial \xi^2}\right] + \quad (1)$$
$$+ \frac{2 p_{жс} r^2 V}{Eh}\frac{\partial^3}{\partial \theta^2 \partial t}\left(\frac{\partial w}{\partial \xi}\right) = 0$$

in this

$$h_\nu = \frac{h}{r\sqrt{12\left(1 - \nu^2\right)}},$$

(1) starting to solve , it should be mentioned that we write all the displacement and torsion angles by ω. Let's isolate the variables and represent ω as a string:

$$\omega = \sum_n^\infty \sum_m^\infty f_{mn}(\xi) e^{i\omega t} \cos m\theta \qquad (2)$$

In this ω- rotational frequency of free oscillation. In that case

$$\vartheta_1 = \sum_n^\infty \sum_m^\infty \frac{1}{m} f_{mn}(\xi) e^{i\omega t} \sin m\theta; \quad u = -\sum_n^\infty \sum_m^\infty \frac{1}{m^2} f'_{mn}(\xi) e^{i\omega t} \cos m\theta;$$
$$\vartheta_2 = -\sum_n^\infty \sum_m^\infty \frac{m^2 - 1}{m} f_{mn}(\xi) e^{i\omega t} \sin m\theta; \quad \tilde{\vartheta}_1 = -\sum_n^\infty \sum_m^\infty \frac{1}{m^2} f'_{mn}(\xi) e^{i\omega t} \cos m\theta \qquad (3)$$

(2) and (3) the (1) putting into the equation and using the algorithm of the Bubnov Galyorkin method

$$\int_0^{2\pi} L^* \cos k\theta d\theta = 0 \quad (k = 1,2,\ldots,m),$$

In this L^* - (2) the left side of the equation, $f_{mn}(\xi)$-we get a system of equations to determine the function we are looking for. It should be noted here at m≠k the coefficients of this system will be zero:

$$f_{mn}^{IV}(\xi_0) + a_{mm} f''_{mn}(\xi_0) + b_{mm} f'_{mn}(\xi_0) + c_{mm} f_{mn}(\xi_0) = 0 \qquad (4)$$

In this f_{mn} -variable function $\xi_0 = \xi\sqrt{h_\nu}, 0 < \xi_\theta < l_0; \quad l_0 = \frac{\sqrt{h_\nu L}}{r}$

when changing in the interval $0 < \xi_\theta < l_0; \quad l_0 = \frac{\sqrt{h_\nu L}}{r}$ will be.

(4) coefficients of the system of equations are determined by the following expressions.

$$a_{mn} = \frac{r^2}{h_\nu^2 Eh}\left(\rho h + \frac{p_{жc}\cdot r}{2}\right)\omega^2 + m^4\cdot\frac{r}{Ehh_\nu^2}\left(p_{жc}V^2 + p\right);$$

$$b_{mm} = i\,\frac{2p_{жc}\cdot r^2\cdot V}{Ehh_\nu^2}m^4\omega;$$

$$c_{mm} = m^4\left(m^2-1\right)\left(m^2 + p_* - 1\right) - \frac{r^2}{Ehh_\nu^2}\left(\rho h + \frac{p_{жc}\cdot r}{2}\right)\left(m^2+m^4\right)\omega^2 \qquad (5)$$

In this

$$p_* = 12\left(1-\nu_2\right)\frac{p}{E}\left(\frac{r}{h}\right)^3$$

We write the system of homogeneous differential equations (4) in matrix form using the operator designations:

$$\begin{vmatrix} L_{11} & 0 & 0 & . \\ 0 & L_{22} & 0 & . \\ 0 & 0 & L_{33} & . \\ . & . & . & . \end{vmatrix}\cdot\begin{vmatrix} f_{1n} \\ f_{2n} \\ f_{3n}\end{vmatrix} = 0 \qquad (6)$$

In this L_{mn}- linear operator, differential and f_{mn} determines the algebraic process (operation) on the function:

$$L_{mm} = \frac{d^4(\ )}{d\xi_0^4} + a_{mm}\frac{d^2(\ )}{d\xi_0^4} + b_{mm}\frac{d(\ )}{d\xi_0} + c_{mm}(\ ) = 0 \qquad (7)$$

f_{mn} (ξ_0) function, the appearance of which depends on the boundary conditions in the cross-section of the shell, we consider the case where both cross-sections are hinged. f_{mn} (ξ_0)we look at the function in the form of a trigonometric series, we assume that it satisfies the preliminary boundary condition:

$$f_{mn}\left(0,l_0\right) = 0; \quad f''_{mn}\left(0,l_0\right) = 0 \qquad (8)$$

The solution

$$f_{mn}\left(\xi_0\right) = D_{mn}\sin\lambda_n\xi_0 \qquad (9)$$

we look for it in appearance. In this $\lambda_n = \dfrac{n\pi r}{L\sqrt{h_\nu}}$ (3.8) satisfies. Substituting the solution (9) into (8) and using the Bubnov-Galyorkin method again (0, l_0) applying it in the limit of integration, we get a system of n homogeneous algebraic equations. The condition of non-participation of trivial solutions causes the determinant of the system coefficients to be equal to zero:

$$\begin{vmatrix} A_{11} & \delta_{11} & B_{12} & \delta_{12} & B_{13} & \delta_{13} \\ B_{21} & \delta_{21} & A_{22} & \delta_{22} & B_{23} & \delta_{23} \\ B_{31} & \delta_{31} & B_{32} & \delta_{32} & A_{33} & \delta_{33} \\ . & . & . & . & . & . \\ . & . & . & . & . & . \end{vmatrix} = 0 \qquad (10)$$

in this

$$A_{nn} = \lambda_n^4 - a_{mm}\lambda_n^2 h_\nu + c_{mm}; \quad B_{nj} = \lambda_n \sqrt{h_\nu}\, b_{mm} \quad (j = 1,2,\ldots,n)$$

$\delta_{nn} = l_0 / 2;\quad \delta_{nj} = 0$ when $n = j$ or $n,\, j$ if even, or if odd: otherwise $\quad \delta_{nj} = \dfrac{2nl_0}{\pi(n^2 - j^2)}$

(10) by opening the detector, we get free oscillation frequency spectra.

The analysis of the obtained results shows that the influence of the Coriolis force on the frequency of free oscillation is small. Therefore, we will not take into account the influence of these forces from now on. In this case, the expressions for determining the square of the frequency take the following form.

$$\omega^2 = \frac{Eh_\nu^2\left[\lambda_n^4 - m^4 \dfrac{r}{Ehh_\nu^2}\left(p_{ж} \cdot V^2 + p\right)\lambda_n^2 h_\nu + m^4\left(m^2 - 1\right)\left(m^2 + p_* - 1\right)\right]}{r^2\left(p + \dfrac{p_{ж} \cdot r}{2h}\right)\left(\lambda_n^2 h_\nu + m^2 + m^4\right)} \quad (11)$$

(11) terms in the second parentheses in the denominator of the expression reflect the different effects of the components of the inertial force. All calculation processes of the above frequency are presented in [2]. That is, the first tone of the free vibration overlaps with the bending vibration of the hammer. For this case (11) after substitutions,

$$\omega = \frac{\pi^2 n}{L^2\sqrt{\dfrac{pF + p_{ж}F_{ж}}{EI}}}\sqrt{n^2 - \frac{V^2 \cdot L^2\left(p_{ж}F_{ж}\right)}{\pi^2 EI}} \quad (12)$$

we write in the form. This result overlaps with the solutions of [3] found without considering the fluid pressure. Formula (12) is not valid for the class of pipes of medium length. Because it does not take into account the effect of deformation of the pipe profile. It takes into account only the component of the inertia forces in the bending plane. Further analysis shows that the internal pressure of the liquid has a significant effect on the frequency of free oscillation of a sufficiently thin-walled pipe (in m>1).

Algorithms and programs, a series of eigensolutions obtained, show that they are correct compared to the known results in [4]. The considered toroidal shell is subject to a single problem on the boundary contours of the shell $u = \vartheta = \omega = w_\alpha = 0$ given In this case, the minimum frequency corresponds to the shape of the axis of symmetry of the vibration.

The value of this frequency depending on the given values is determined by (12). In the work of [4], the following formula was given to determine the natural frequency of a curved cylindrical shell taking into account the effect of fluid

$$\frac{\omega_m}{\omega} = \frac{1}{\sqrt{1 + \dfrac{x_n \rho_{ж} S_1}{\rho h}}} \quad \text{or} \quad \omega = \omega_{ж}\sqrt{1 + \frac{x_n \rho_{ж} S_1}{\rho h}} \quad (13)$$

in this , $S_1 = \dfrac{2\pi\alpha}{n}$; n - meridian node numbers; h- thickness; α - shell radius, x_n - coefficient value (S / L) depends on.

If $0 < S / L \leq 2{,}0$, $1{,}2 < x_n < 0{,}2$. Taking into account the effect of fluid plays the role of attaching masses from the analysis of known results. So, in this case, the frequency takes on an increased value (empty shell pressure). To date, pipelines made of polyethylene are widely used for the transportation of gas, oil, and petroleum products. Pipe material made of polyethylene Pe-80 and Pe-100, elastic modulus E=500 MPa, Poisson's coefficient, outer diameter up to 1200 mm, ratio of pipe wall thickness to middle surface radius $\frac{h}{r}=\frac{1}{8}\div\frac{1}{12}$ up to 0.8 MPa internal hydrostatic pressure, designed. The curved part of the pipelines is in the form of a thin-walled toroidal shell made of sections of polyethylene pipe with an outer diameter of up to 630 mm. The dynamic calculation of such pieces will have to be done based on the theory of shells. Therefore, the determination of the specific vibration frequency of a curved section of a pipeline made of a polyethylene pipe is carried out according to the above method. In curved sections of polyethylene pipeline, which has a small modulus of elasticity compared to steel, the first three specific vibrations $m=1,2,3$ according to the form ω_{mn} examination of the frequency shows that the fluid flow rate has a significant effect on the natural vibration frequency. Calculation of the frequencies carried out according to the quality of the polyethylene pipe given in [132], the relative thickness of the wall $\frac{h}{r}=\frac{1}{12{,}5}$, The pipeline was made for a curved section with an outer diameter of 630 mm. $\frac{r}{R}=\frac{1}{10}$ and $\frac{1}{50}$ The specific vibration of the pipe was analyzed when the water velocity changes from zero to the relative curvature of the fluid flow. The test results are presented in table 1 and figure 1 graphically. The analysis of the test results shows that the tradition of changes in vibration frequencies established in steel pipelines is also preserved for polyethylene. It can be seen from the graph in Figure 1 that it has a large curvature $\frac{r}{R}=\frac{1}{10}$ the tube vibration frequency is significantly higher than the frequency of the tube with small curvature $\frac{r}{R}=\frac{1}{10}$. $\frac{r}{R}=\frac{1}{50}$ for, $\omega=0{,}25\ Гц$. The main conclusion of the analysis of the results of the analysis of the bending vibration natural frequency of the polyethylene pipe flowing through the liquid is that the natural vibration frequency of these pipelines is much more dependent on the influence of the fluid velocity than that of the steel pipelines. In this ω_{mn} decrease in vibration frequency, change in current speed from 0 to $40\frac{м}{c}$ reaches up to 20%. It is necessary to take this into account in dynamic calculations of pipelines. It can be seen from Figure 1 that the damping coefficient ω_{mn}^{I} for $u=30\ м/c$ takes the maximum value for.

So, at this value, the vibration of the body decreases to a maximum. Similarly, in pipelines made of steel, the largest specific vibration frequency is the frequency of the first vibration form ω_{1n} ($m=1$ for).

The deformation of the contour of the pipe cross-section does not take part in it. These frequencies correspond to the Sturgeon theory of pipeline calculations.

1 - table. Correlation of the velocity change of the flowing liquid with the specific frequency

$\frac{r}{R}=\frac{1}{20},\frac{h}{r}=\frac{1}{40}$ $\mu=4{,}1$		Velocity of the flowing liquid (м/с) for ω_{mn} (Гц)		
Form of vibration	Frequencies	$v=0$	$v=20$	$v=30$
$m=1$	ω_{11}	6,67	6,47	5,56
	ω_{12}	7,45	6,82	6,01
	ω_{13}	8,74	7,91	7,51
$m=2$	ω_{21}	4,65	6,19	2,52
	ω_{22}	5,55	6,85	3,75
	ω_{23}	6,02	5,51	4,69

$m=3$	ω_{31}	2,23	1,73	0,29
	ω_{32}	34,04	2,55	1,53
	ω_{33}	5,54	4,91	3,25
$m=4$	ω_{41}	7,02	5,79	4,77
	ω_{42}	8,19	6,95	5,62
	ω_{43}	10,27	8,86	6,92

The smallest natural frequency of a bending vibration in a curved section is a shell-shaped vibration (m=2 and 3 for) corresponds to the deformed contour of the cross section. For the case when the sinusoidal longitudinal forms one half-wave (for n=1). Conducted special vibration tests of a curved piece of polyethylene pipeline show that the frequency of vibration along the investigated shell shape (m,n=1,2,3) for the modulus of elasticity of polyethylene is 400 times smaller than that of steel, and from a practical point of view, all the real geometric dimensions of the piece are significant. small (compared to the corresponding steel pipeline dimensions see Figure 1). The pipeline is considered dangerous due to the fact that a resonance state may appear at a lower frequency of the specific vibration. Therefore, the condition of special frequency construction with external excitation frequencies for the polyethylene pipeline requires careful investigation.

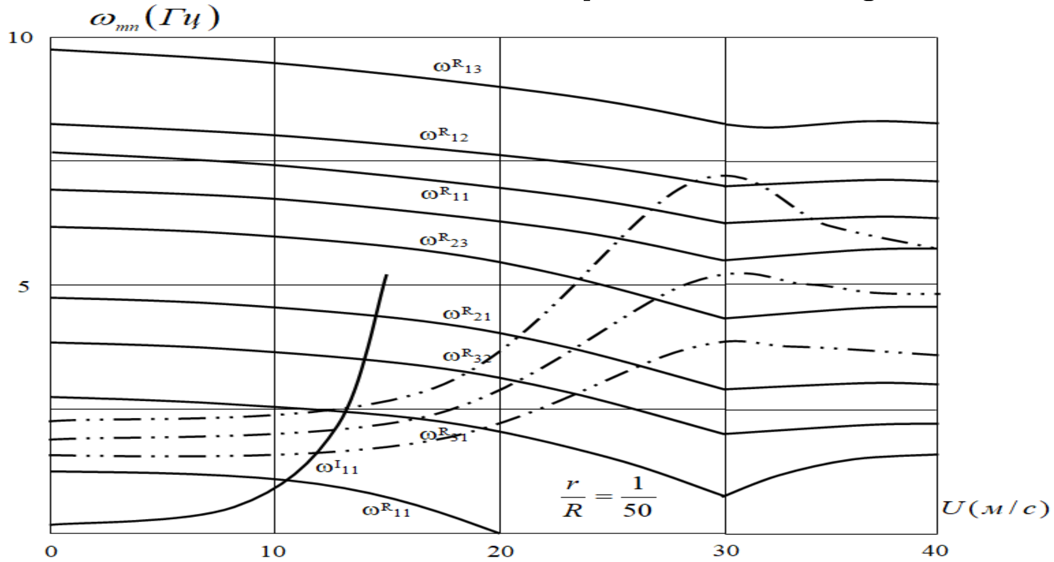

Fig. 1. Dependence of the speed of the flowing fluid on the frequency of the specific bending vibration

According to the normative document, the condition for frequency otstroyka (construction) for lower vibration frequencies:

$$\frac{\omega_{\min}}{\Omega} \geq 1{,}28 \quad \text{or} \quad \frac{\omega_{\min}}{\Omega} \leq 0{,}69 \qquad (14)$$

(3) (14) will appear. In this $\omega_{\min}$ – pipeline natural oscillation lower frequency? Ω external excitation frequency. For pipelines above ground, in special cases, the effect of wind force can serve as an external driver. The lower frequency of bending vibration, when it becomes zero, also leads to the loss of pipeline priority. Not at all, when calculating the curved part of the polyethylene pipeline, the first form of vibration (m,n=1) and the flow rate of the liquid v=20m/c, relative curvature at $\frac{r}{R}=\frac{1}{50}$ frequency $\omega_{11}=0$ (See the dashed part of Figure 1). This means that for such a pipeline, the velocity becomes critical and loses its priority at v=20m/c. To check, the parameter P*0 depends on the characteristic vibration frequency of the curved section of the pipeline from the velocity of the flowing fluid. This was done only to check the dependence of the vibration frequency of the hydrodynamic pressure generated by the fluid movement.

Literature

1. 1.Базаров М.Б. Сафаров И.И., Шонин Ю.М Численное моделирование колебаний диссипативно -неоднородных и однородных механических систем. -Новосибирск: Сибирского отд. РАН, 1996 -189с.
2. Векслер Н. Д. Информационные проблемы гидроупругости.- Таллин: 1982. -180 с.
3. Гринченко В.Т., Мелешко В.В. Гармонические колебания и волны в упругих телах, - Киев: Науково думка, 1981.-283с.
4. Гузь А.Н., Кубенко В.Д., Черевко М.А. Дифракция упругих волн. -Киев : Науково думка, 1978 .-308с.
5. 5.Голоскоков Д.П. Уравнения математической физика . Решение задач в системе MAPLE.- СПб: Питер,2004.-539с.
6. Жумаев З.Ф., Сафаров И.И. О разрушении тоннеля при сильных движениях земли. Международная конференция по сейсмостойкому строительству. С-Петрбург, 2000, с. 71-78
7. Крауклис П.В., Молотков Л.А. Низкочастотные волны Лэмба в цилиндрических и сферических слоях , расположенных в упругой среде // Зап. Научных семинаров ЛОМИ АН СССР.1972. Т. 25 .-с.101-110.
8. Купрадзе В. Д. Методы потенциала в теории упругости. - М.:1963.- 472 с.
9. Kulmuratov N.R. **,** Safarov I.I. Oscillations of curved pipeline under the action of variable internal pressure //World Journal of Engineering Research and Technology WJRET. Wjret, 2018, Volume 4, Issue 6. **- P. 125-139.**
10. **Kulmuratov N.R.,** Safarov I.I., Teshayev M.K., Kuldashov N.U. Interaction of Nonstationary Waves on Cylindrical Body **//** Journal Applied Mathematics, 2019, 10, 435-447.
11. **Kulmuratov N.R.,** Safarov I.I., Teshayev M.K., Kuldashov N.U. Diffraction of Surface Harmonic Viscoelastic Waves on a Multilayer Cylinder with a Liquid **//** Journal Applied Mathematics, 10, 468 -484.
12. Кульмуратов Н.Р., Гайбуллаев З.Х., Каюмов С. Исследование механизмов потерь энергии в упругих механических системах // Узбекский журнал «Проблемы механики». - Тошкент, 2003. - №3 - С.32-34. (05.00.00, №6).
13. Кульмуратов Н.Р. Собственные колебания кусочно-однородных цилиндрических тел, находящихся в упругой среде // «Замонавий илм - фан ва технологияларнинг энг мухим муаммолари» Республика илмий-Амалий конференцияси маколалар тўплами. Жиззах: 2004. – С.14-17.

14. Кульмуратов Н.Р., Жакыпбекова Р., Гайбуллаев З.Х. Собственные колебания жесткого цилиндрического включения в упругой среде // Материалы Международной научно-технической конференции «Современные проблемы и перспективы механики».- Ташкент, 2006. – С. 277- 278.
15. Кулмуратов Н.Р., Сафаров И.И., Тешаев М.Х., Кулдашов Н.У. Дифракции гармонических волн на вязкоупругой пространственной цилиндрической оболочке с жидкостью // Материалы XXI Международной конференции по вычислительной механике и современным прикладным программным системам (ВМСППС), Крым. Алушта. 2019. - С.344-346.