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Bending Vibrations Polymeric Pipes of Variable Section with Interference inside the Liquid

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Abstract
In work on the basis of the rod theory, bending vibrations of a visco elastic pipeline are considered by the action of an inner ideal fluid.

Keywords: tube, oscillations, fluid, beam, the Dalamber principle, inertia force.

Introduction
Polymer pipes are widely used in many areas of the national economy. In this case, as a rule, during operation they are in contact with a liquid or gaseous medium and are subjected to dynamic influences. Of particular relevance dynamic tasks have in the field of modern robotics with flexible hinge less manipulators, which due to their design features are very sensitive to external loads and capable of breaking under the influence of internal pressure [1,2,3].

Statement of problem and methods of solution.
When investigating flexural vibrations of a tube with a fluid flowing inside, we use a model in the form of an unprismatic beam and a hypothesis of plane cross sections. In this case, for the analysis of oscillations, the differential equations

$$\frac{\partial^2}{\partial x^2}\left[E_0 J \frac{\partial^2}{\partial x^2}\left(w(x,t)-\int_0^t R(t-\tau)w(x,\tau)d\tau\right)\right]=q(x,t), \qquad (1)$$

Where $w(x,t)$ - equation of the elastic axis of the beam with respect to its unreformed state under the action of a transverse specified load $q(x,t)$. Let the bending stiffness be constant along the axis of the tube EJ, weight of pipe length unit m_1 and the mass of the liquid volume m_2, filling the unit length of the pipe. In accordance with the Dalamber principle, the inertial forces arising during oscillations (Fig. 1) can be considered as a transverse load for the beam, then

$$q(x,t)=-m_1\frac{\partial \vartheta_1}{\partial t}-m_2\frac{\partial \vartheta_2}{\partial t},$$

Where ϑ_1 - absolute speed of the pipe element, a ϑ_2 - absolute velocity of a fluid.

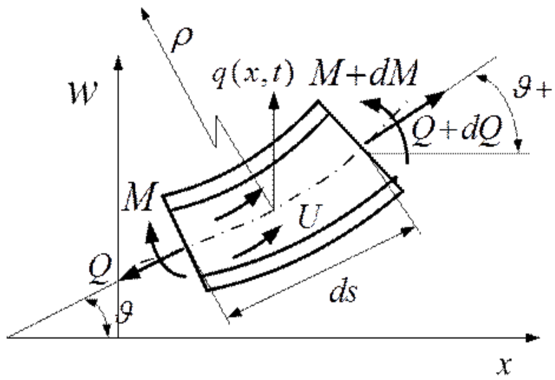

Fig 1: The calculation scheme

For a steady flow, the pressure along the tube axis does not change, and the velocity of the fluid V does not depend on the vibrations of the tube. Then the inertial load can be written in the form

$$q(x,t) = -m_1\frac{\partial^2 w}{\partial t^2} - m_2\frac{\partial^2 w}{\partial t^2} - m_2V^2\frac{\partial^2 w}{\partial x^2} - 2m_2V\frac{\partial^2 w}{\partial x\partial t}. \quad (2)$$

Here the first term is the force of inertia of the pipe element, which arises during its transverse oscillations; since the liquid element performs a complex motion (portable motion with speed V_i element of the pipe, relative - with speed V), the remaining terms in (2) reflect its inertia forces-the inertial force of the mobile motion, the normal component of the inertia force of the relative motion and the inertial force of Carioles, respectively. When calculating the acceleration components of a fluid element, it is taken into account that the curvature of the beam $k = \frac{1}{\rho} = \frac{\partial^2 w}{\partial x^2}$, angle of rotation of the element $\vartheta = \frac{\partial w_1}{\partial x}$, and its angular velocity $\dot{\vartheta} = \frac{\partial^2 w}{\partial x\partial t}$. Substituting (2) in (1), we obtain the differential equation of transverse oscillations of the pipeline axis relative to the initial rectilinear position:

$$\frac{\partial^2}{\partial x^2}\left[E_0J\frac{\partial^2}{\partial x^2}\left(w(x,t) - \int_0^t R_E(t-\tau)w(x,\tau)d\tau\right)\right] = -(m_1+m_2)\frac{\partial^2 w}{\partial t^2} - m_2V^2\frac{\partial^2 w}{\partial x^2} - 2m_2V\frac{\partial^2 w}{\partial x\partial t} \quad (3)$$

The solution of equation (3) can be obtained by one of the approximate analytical methods. We use the Bubnov-Galerkin method, representing the solution of the equation in the form of a product of two function

$$w(x,t) = X(x)e^{i\omega t} \quad (4)$$

Solution (3) must satisfy the four boundary conditions corresponding to the options for securing the ends of the pipeline. The substitution of (4) in (3) allows for the function to obtain the ordinary differential equation:

$$\frac{d^4X}{dx^4} - a_z\omega^2X + c_z\frac{d^2w}{dx^2} = 0, \quad (5)$$

where $a_z = \frac{m_1+m_2}{\bar{E}J}$, , $c_z = \frac{m_2}{\bar{E}J}V^2$, $\bar{E}\varphi = E\left[1 - \Gamma^C(\omega_R) - i\Gamma^S(\omega_R)\right]\varphi$, We seek the solution of (5) in the form $X(x) = \sum_{\kappa=1}^{\infty} A_\kappa \sin(\frac{k\pi x}{l})$, substitution of which into equations (5), followed by multiplication by $\sin(\frac{\pi x}{l}), \sin(\frac{2\pi x}{l})\ldots\ldots$ and integration under boundary conditions from x=0 до x=*l*, leads to a system of linear homogeneous equations with respect to unknown constants $A_1, A_2, A_3\ldots\ldots$ Equating the determinant of the system to zero, we obtain an equation for determining the vibration frequencies of the pipeline ω_κ. In studying the oscillations that arise, the answer to the question of the value of the critical velocity V_{pk} flow (the flow rate of the fluid at which the pipeline may lose static stability). This value can be found from the condition that the first frequency of oscillations is equal to zero (which, in turn, occurs when the term that does not contain the frequency in the equation for determining the frequencies is zero). As an example, consider a section of a viscous elastic pipeline with hinged supports at its ends, flowing at a constant speed $V = 10 м/сек$ ideal incompressible fluid. Average diameter of the pipeline section $D = 0.09 м$, pipeline wall thickness $\delta = 0.025 м$, length of section $l = 1 м$, material density $\rho_1 = 2700 кг/м^3$, elastic modulus $E = 75 ГПа$. Mass of liquid $\rho_2 = 15.64 кг/м^3$. Determine the first two frequencies of the transverse oscillations of the pipeline with a resting and flowing liquid without taking into account the action of the static forces of weight. We use the differential equation (3), where $m_1 = \rho_1\pi D\delta$ - the mass of a unit of pipe length, and $m_2 = \rho_2\pi(D-\delta)^2$ - mass of unit length of liquid. To obtain the solution, we use the Bubnov-Galerkin method. Assuming that $w(x,t) = X(x)e^{i\omega t}$, after substituting the assumed solution in (5) and performing fairly simple transformations, for equation (5) we obtain the coefficients $a = 0.014, b = 0.08V, c = 0.004V^2$. The solution of equation (5) must satisfy the boundary conditions:

When $x = 0$: $X(0) = 0; \frac{d^2X}{dx} = 0,$ at

$$x = l: X(l) = 0; \frac{d^2X}{dx} = 0,$$

We seek a solution of equation (5) in the form

$$X = A_1\sin(\pi x/l) + A_2\sin(2\pi x/l).$$

We substitute this solution in (5) and successively multiply the resulting expression by $\sin(\pi x/l)$ и $\sin(2\pi x/l)$.

The obtained relations are integrated in the interval from $x = 0$ before $x = l$ taking into account the boundary conditions. As a result of the performed operations, we arrive at an algebraic system of two linear homogeneous equations with respect to unknowns A_1, A_2 :

$$A_1\left[(\pi / l)^4 - a\omega^2 - c(\pi / l)^2\right] - A_2(8bi / 3l)\omega = 0;$$
$$A_1\left[(i8b / 3l)\omega\right] + A_2\left[16(\pi / l)^4 - a\omega^2 - 4c(\pi / l)^2\right] = 0$$

Equating the determinant of this system to zero, we obtain an equation of the form $\omega^4 + a_1\omega_2 + a_2 = 0$ for determination of two frequencies of transverse oscillations of the pipeline.

Numerical results

When $V = 0$, $\omega_1 = 24.9 рад / сек, \omega_2 = 98.9 рад / сек$

At $V = 10 м / сек,$

$\omega_1 = 24.2 рад / сек, \omega_2 = 101.7 рад / сек$. The critical flow rate of a liquid arises when one of the vibration frequencies is equal to zero, as is the case when $a_2 = \left[(\pi / l)^2 - c\right]\left[4(\pi / l)^2 - c\right] = 0$, where $c = \frac{m_2}{\bar{E}J} V_{кр}^2$. Hence the value of the minimum critical speed ($V_{кр}$) will be $V_{кр} = \frac{\pi}{l}\sqrt{\frac{\bar{E}J}{m_2}}$. The results of calculations of bending vibrations along the z axis and time t are shown in Fig. 2 and 3.

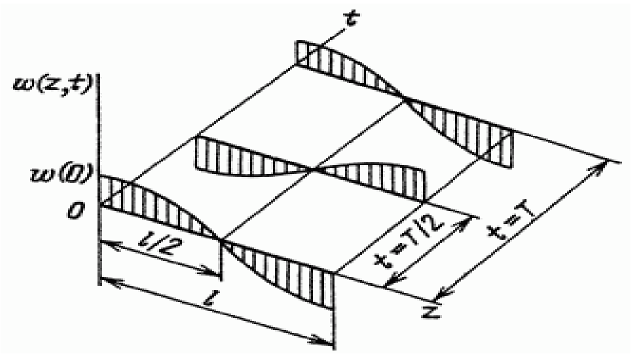

Fig 2: Moving the piping section point as a function of distance and time

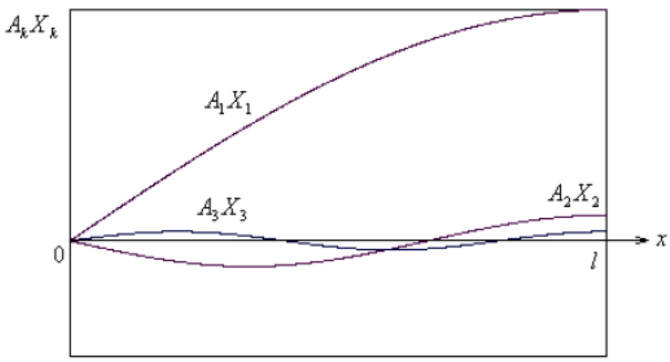

Fig 3: Moving the section point of the cantilever pipeline as a function of distance.

Based on the developed approximate mathematical model of bending vibration movements of the pipeline, its free oscillations were investigated. It is established that with increasing internal pressure, an increase in the amplitude of free bending vibrations simultaneously increases and an increase in the frequency of free rotational vibrations of the tube.

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