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Original Article

Simulation of Forced Oscillations of Pressure Monitoring Devices

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Abstract — *Harsh working conditions, as well as vibrations of technological process units and unstable load intensity, impose high standards on overpressure monitoring devices that ensure the required measurement accuracy and trouble-free operation. Using manometer gauges today is a mandatory requirement for monitoring overpressure. Manometric tubular springs have become widespread in various fields of technology. Therefore, the issue of determining the forced fluctuations of the manometric tubular springs becomes more significant. The article presents a mathematical model of forced oscillations of the manometric tubular springs based on Lagrange equations of the second kind. A program has been developed in MATLAB based on the proposed model, which allows determining the required characteristics of pressure monitoring devices to exclude the possible occurrence of resonance. The presented model can be successfully used to calculate standard manometric tube designs since it is a classical approach to solving vibrations problems of mechanical systems.*

Keywords — *Manometric tubular spring, forced oscillations, forced oscillation frequencies, Lagrange equations of the second kind, MATLAB.*

I. INTRODUCTION

The main development vector of the oil and gas industry is oriented to the Arctic and offshore area. Transportation of energy carriers from deposits to consumers is planned to be carried out using sea and pipeline transport. Construction areas are extremely complex and require high control during the operation of infrastructure facilities, which is reflected in [1]-[16].

One of the main technological processes parameters in the oil and gas industry is the operating pressure. Abnormal operation conditions, as well as vibrations of technological process units and unstable load intensity, impose high requirements on the pressure measuring devices, which in turn must have high reliability and the required accuracy.

According to the regulatory documentation, all-digital pressure measurement sensors must be duplicated by analogue pressure gauges, and in some cases, using pressure gauges is a single option way to monitor excessive pressure.

Manometric tubular springs (MTS), in which the property of a curved tube with a non-circular cross-section is used to deform with increasing internal pressure, are employed as sensitive elements in pressure gauges.

Currently, besides pressure gauges, MTS have found application in various fields of technology [17]-[19].

The strength and frequency characteristics of tubular spring vibrations, the effects of the shape, cross-sectional dimensions, and basic geometric dimensions of the MTS, as well as the process of MTS vibration damping by a liquid, are analyzed in a large number of works [20]-[26].

However, the issue related to taking into account internal pressure pulsations remains unexplored. The present article solves the problem of forced oscillations of the MTS using the Lagrange equation of the second kind.

II. METHODS

The MTS is considered as a mechanical system with two degrees of freedom, with two generalized coordinates. Let us denote the relative change in the main angle of the MTS as φ and the increase in the vertical component of the cross-section of the MTS as w.

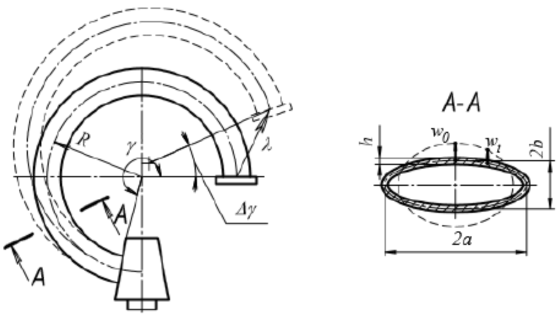

Fig. 1 The dynamic model of MTS

The mathematical model represents a system of equations based on the Lagrange equation of the second kind. In the case under consideration, exciting forces *F(t)* act at the points of the system in addition to the forces having potential.

$$\begin{cases} \dfrac{d}{dt}\left(\dfrac{\partial T}{\partial \dot{\varphi}}\right) - \dfrac{\partial T}{\partial \varphi} + \dfrac{\partial U}{\partial \varphi} = \dfrac{\partial A}{\partial \varphi}, \\ \dfrac{d}{dt}\left(\dfrac{\partial T}{\partial \dot{w}}\right) - \dfrac{\partial T}{\partial w} + \dfrac{\partial U}{\partial w} = \dfrac{\partial A}{\partial w}, \end{cases} \qquad (1)$$

Where t is the time, T is the kinetic energy, U is the potential energy, A is the generalized work of internal pressure pulsation.

Earlier in [19, 20], expressions were obtained for determining the potential and kinetic energies. The expression for the potential energy has the form:

$$U = \frac{Eh\gamma}{2R(1-\mu^2)}\left[\frac{A_1}{m^2}w^2 - 2b\frac{A_2}{m}\varphi w + b^2 A_3 \varphi^2 + \frac{\chi^2}{12}\frac{n}{m^2}w^2\right], \quad (2)$$

where E is the elasticity modulus, μ is the Poisson's ratio, h is the wall thickness,

γ is the central angle, R is the radius of the central axis, A_1, A_2, A_3, m, n, b, χ are the coefficients depending on the MTS cross-section shape.

The expression for the kinetic energy has the form:

$$T = 2\rho h R^3 \gamma\left(\frac{\gamma^3}{3} - 4\sin\gamma + 2\gamma\sin\gamma + 2\gamma\right)B_1\dot{\varphi}^2 + 2hR\gamma\left(\frac{B_2}{m^2} + \frac{B_3}{K_б^2 m_б^2}\right)\dot{w}^2, \quad (3)$$

Where ρ is the density of the material, B_1, B_2, B_3, $K_б$, $m_б$ are the coefficients depending on the cross-section shape.

The work of the internal pressure forces pulsation can be represented in the following form:

$$A = q\Delta f R\gamma \quad (4)$$

where q is the internal pressure distribution law, which can be represented as $p \cdot \sin(k_p t + \delta)$, Δf is the change in the area bounded by the middle line of the cross-section outline, represented as $\frac{2wa}{m}\left(1 - \frac{b^2}{a^2}\right)n$, where the notation is the same as in formula (2), γ is the central angle, R is the radius of the central axis.

In the final form, the generalized work of the internal pressure pulsation will have the following form:

$$A = p \cdot \sin(k_p t + \delta)\frac{2wa}{m}\left(1 - \frac{b^2}{a^2}\right)nR\gamma \quad (5)$$

Having calculated the partial derivatives of the potential and kinetic energies, as well as the generalized work of the internal pressure pulsation, we obtain a system of differential equations describing forced oscillations of the system:

$$\begin{cases} a_1\ddot{\varphi} + c_1\varphi + c_3 w = 0 \\ a_2\ddot{w} + c_2 w + c_3\varphi = d_1\sin(k_p t + \delta) \end{cases} \quad (6)$$

where a_1 and a_2 are the inertia coefficients, c_1, c_2, c_3 are the stiffness coefficients.

$$a_1 = 4\rho h R^3\gamma\left(\frac{\gamma^3}{3} - 4\sin\gamma + 2\gamma\sin\gamma + 2\gamma\right)B_1, \quad (7)$$
$$a_2 = 4hR\gamma\left(\frac{B_2}{m^2} + \frac{B_3}{K_б^2 m_б^2}\right),$$
$$c_1 = \frac{Eh\gamma}{2R(1-\mu^2)}(2b^2A_3),$$
$$c_2 = \frac{Eh\gamma}{2R(1-\mu^2)}\left(\frac{A_1}{m^2} + \frac{\chi^2}{12}\frac{n}{m^2}\right),$$
$$c_3 = \frac{Eh\gamma}{2R(1-\mu^2)}\left(-2b\frac{A_2}{m}\right),$$
$$d_1 = p\frac{2wa}{m}\left(1 - \frac{b^2}{a^2}\right)nR\gamma.$$

The general solution of the system of differential equations (6) is the sum of the general integral of the corresponding system of homogeneous equations (8) and the partial integral of the system (6).

$$\begin{cases} a_1\ddot{\varphi} + c_1\varphi + c_3 w = 0, \\ a_2\ddot{w} + c_2 w + c_3\varphi = 0. \end{cases} \quad (8)$$

When integrating the resulting system of differential equations, partial solutions can be represented as:

$$\varphi = I_1\sin(kt + \beta), \quad w = I_2\sin(kt + \beta), \quad (9)$$

Let us introduce a coefficient μ, equal to the ratio of generalized coordinates:

$$\mu = \frac{w}{\varphi} = \frac{I_2}{I_1}. \quad (10)$$

Thus, we receive:

$$\begin{aligned} \varphi &= I_1\sin(kt + \beta), \\ w &= \mu I_1\sin(kt + \beta), \\ \ddot{\varphi} &= k^2 I_1\sin(kt + \beta) \\ \ddot{w} &= k^2\mu I_1\sin(kt + \beta) \end{aligned} \quad (11)$$

Substituting (11) into (8) and dividing both equations by $I_1\sin(kt + \beta)$ we get:

$$\begin{cases} a_1k^2 + c_1 + c_3\mu = 0, \\ a_2k^2\mu + c_2\mu + c_3 = 0. \end{cases} \quad (12)$$

Solving the system (12) with respect to μ, we obtain a characteristic equation for determining the frequencies of free oscillations k:

$$a_1a_2k^4 + (a_1 + a_2)c_2k^2 + c_2^2 - c_3^2 = 0 \quad (13)$$

Determining the oscillation frequencies, we have obtained two values of μ, corresponding to each of the oscillations. For each oscillation, the ratio of amplitudes will have a certain meaning, μ, which will depend on the parameters of the system under consideration and in no way depend on the initial conditions.

The general solution to system (8) can be obtained by summing up the particular solutions. The final oscillatory motion of the system will represent a superposition of simple harmonic oscillations with different frequencies:

$$\varphi = I_1^{(1)}\sin(k_1t+\beta_1) + I_1^{(2)}\sin(k_2t+\beta_2),$$
$$w = \mu_1 I_1^{(1)}\sin(k_1t+\beta_1) + \mu_2 I_1^{(2)}\sin(k_2t+\beta_2). \quad (14)$$

Constants $I_1^{(1)}, I_1^{(2)}, \beta_1, \beta_2$ are determined from the initial conditions.

The partial solution to the system of equations (6), determining the forced oscillations of the system, has the form:

$$\varphi = 0,$$
$$w = J_2\sin(k_pt+\delta), \quad (15)$$

Substituting (15) into (6) and dividing both equations by $\sin(k_pt+\delta)$, we obtain:

$$\begin{cases} c_3J_2 = 0 \\ (a_2{k_p}^2 + c_2)J_2 = d_1 \end{cases} \quad (16)$$

Substituting the values of J_2 found by the solving system (16), into equations (15), we obtain a general solution of the system of equations (6):

$$\varphi = I_1^{(1)}\sin(k_1t+\beta_1) + I_1^{(2)}\sin(k_2t+\beta_2), \quad (17)$$
$$w = \mu_1 I_1^{(1)}\sin(k_1t+\beta_1) + \mu_2 I_1^{(2)}\sin(k_2t+\beta_2) + J_2\sin(k_pt+\delta).$$

However, at resonance (in the case when $k_p = k_1$ or $k_p = k_2$), expression (15) will not be a particular solution of the system (6). A particular solution at resonance can be obtained by turning to the main coordinates of the system (8). The general solution at $k_p = k_1$ will have the form:

$$w = \frac{\mu_1(H_1+\mu_1H_2)}{2k_pa_1^*}t\sin\left(k_pt+\delta-\frac{\pi}{2}\right) + \frac{\mu_2(H_1+\mu_2H_2)}{a_2^*(k_2^2-{k_p}^2)}\sin(k_pt+\delta). \quad (18)$$

The first component of expression (18) contains the variable t presented in explicit form; this component will increase indefinitely over time, which confirms the occurrence of resonance.

The developed mathematical model is implemented in MATLAB as a program with a graphical interface. The choice of MATLAB is because firstly, it is a high-level programming language with understandable mathematical logic, and secondly, it is widely used in various fields of science and technology, as well as has a large database of already implemented libraries and algorithms. The interface of the developed program is shown in Fig. 2.

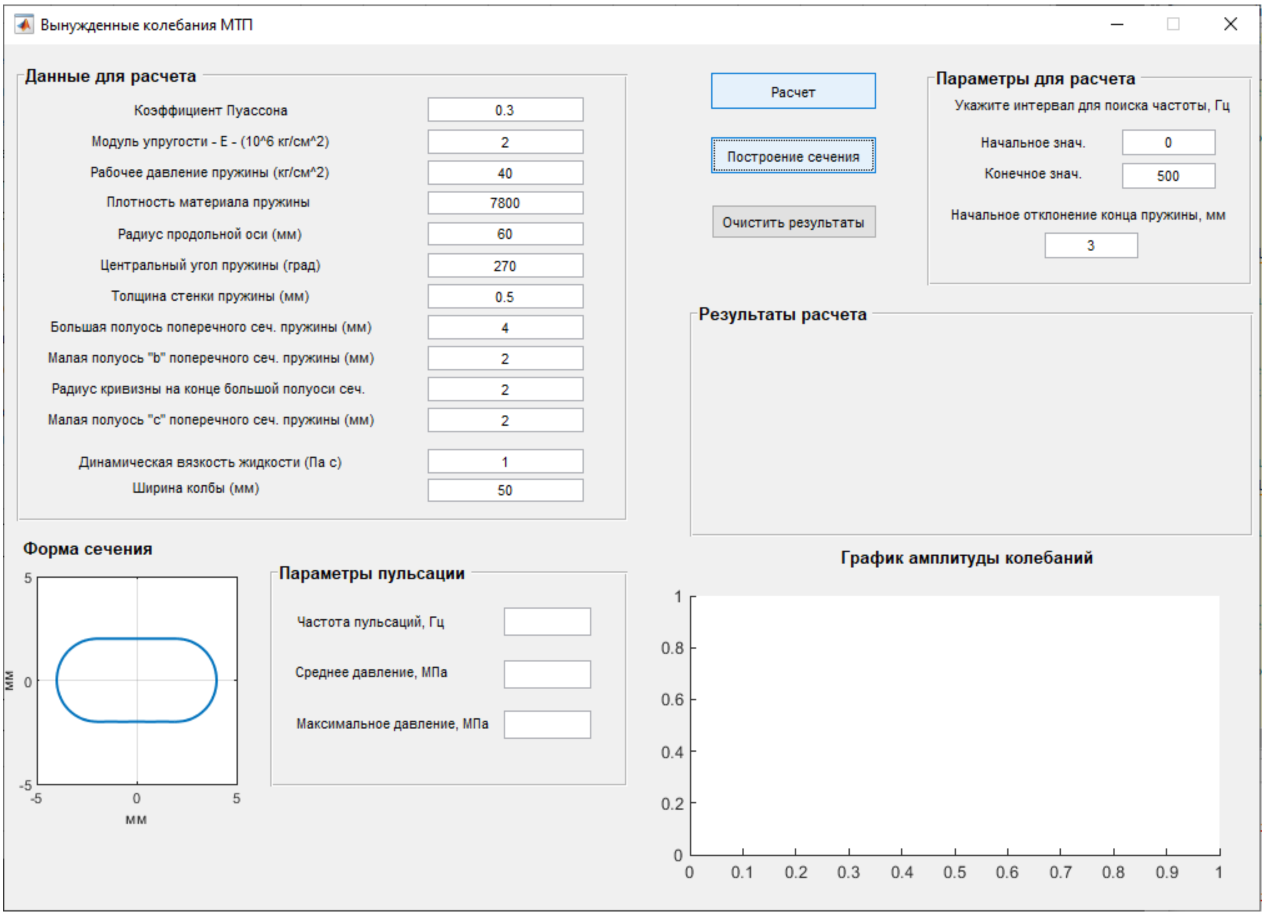

Fig. 2 The graphical interface of the program for calculating forced oscillations of the MTS

III. RESULTS

Employing the developed program, we estimated the effect of internal pressure pulsations on the movings of the free end of the MTS. The pressure was variable with periods equal to 1, 2.5, and 5 periods of natural oscillations. The results are shown in Fig. 3.

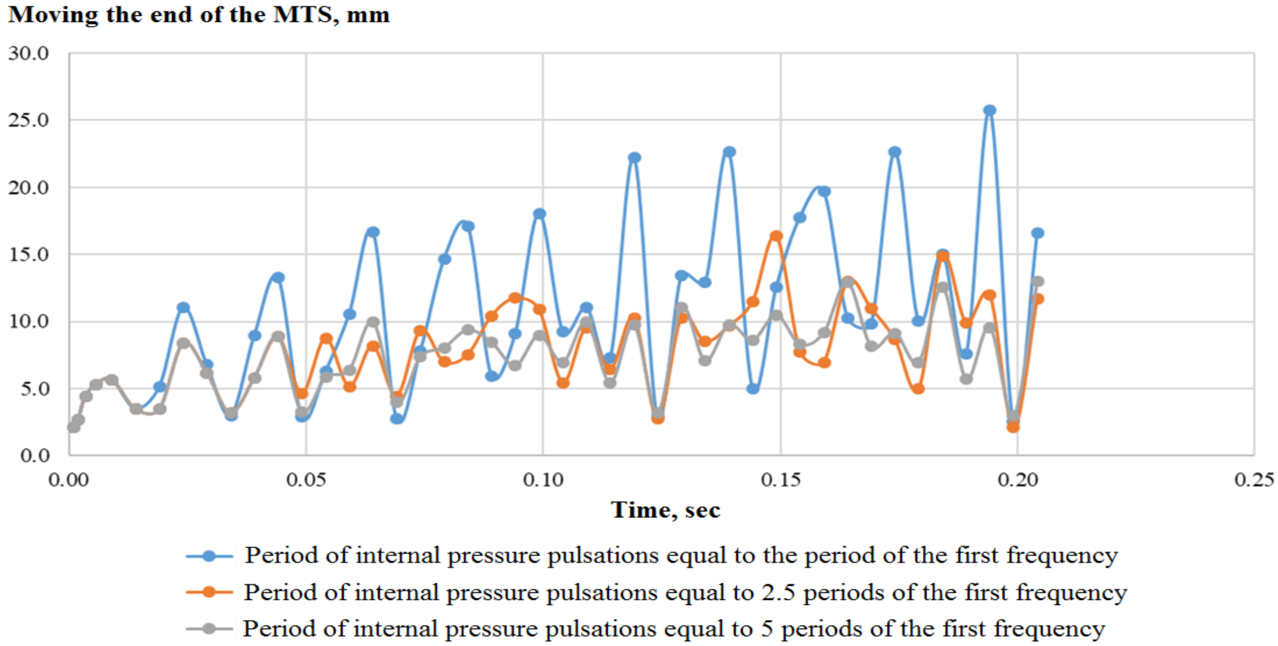

Fig. 3 Moving free end of the MTS, caused by pressure pulsations

As can be seen from the graph, when the period of pulsations is greater than the period of natural oscillations of the MTS, oscillations of the free end of the MTS occur, which allows indicating the presence of pulsations. When the pressure pulsates with a frequency equal to the first frequency of free oscillations of the MTS, a monotonous increase in moving is observed, indicating the presence of a resonance phenomenon. The operation of pressure gauges in such a mode will not allow correctly monitoring the state of the technological process and, as a result, will not be able to ensure the reliable and safe operation of the installations.

IV. DISCUSSION

A mathematical model of forced oscillations of MTS has been developed based on Lagrange equations of the second kind. Employing the developed program, it is possible to determine the required characteristics of pressure monitoring devices that exclude the possibility of resonance occurrence.

The presented model can be successfully applied to calculate standard MTS designs since it is based on a classical approach to solving problems of mechanical systems vibrations. However, for non-standard designs, such as MTS with a changing cross-sectional shape along the length of the tube, a changing wall thickness of the MTS, etc., using this model will be difficult and requires certain adjustments.

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