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Oscillations of manometric tubular springs with rigid end

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Abstract. The paper presents a mathematical model of attenuating oscillations of manometric tubular springs (MTS) taking into account the rigid tip. The dynamic MTS model is presented in the form of a thin-walled curved rod oscillating in the plane of curvature of the central axis. Equations for MTS oscillations are obtained in accordance with the d'Alembert principle in projections onto the normal and tangential. The Bubnov-Galerkin method is used to solve the equations obtained.

1. Introduction

The use of mechanical pressure gauges in some cases remains uncontested and is regulated by regulatory documentation. The need to carry out activities aimed at increasing the vibration protection of MTS serving as sensitive elements of mechanical pressure gauges is caused by an increase in the requirements for the reliability and precision of instruments for measuring pressure. The main methods of protection are detuning from resonance frequencies and vibro-damping with fluid. Studies in this area are performed in [1-4]; however, in these works the effect of the MTS tip, which significantly affects frequency characteristics, is neglected.

2. Materials and methods

The tips can be used to increase the stroke of elastic elements [5-6].

Typically, for a number of springs, the same tips are used at different pressures, so depending on the thickness of the tube, the mass ratio of the tip and the mass of the tube varies over a wide range (line 2 of Table 1).

A comparison of the experimental and calculated values of the frequencies of free oscillations [1] showed that deviations may be from 10 to 80% (see line 5 of Table 1) and increase with increasing tip mass.

In [2], the mass of the tip is considered by introducing the decreasing coefficient of the frequency of free oscillations determined by the experimental method. The influence of the mass of the tip on the parameters of the oscillations attenuation and, as a result, on the attenuation rate of oscillations has not been theoretically studied to date.

The mathematical model used allows us to take into account the influence of the viscosity of the damping fluid and the mass of the MTS tip and is represented as a curved rod that oscillates in the plane of curvature of the central axis (Fig. 1).

Table 1. Value of frequencies of natural oscillations of steel springs

№	Rated pressure, MPa	0.1	0.4	1.0	2.5	4.0	6.0	10.0
1	Wall thickness, mm	0.2	0.3	0.4	0.6	0.7	0.8	1
2	m_{tip}/M_{tubes}	0.965	0.523	0.197	0.132	0.116	0.101	0.08
3	Frequency (experiment), Hz	49.3	95.5	157	197	234	251	277
4	Frequency (calculation without regard to the tip), Hz	89.3	160	198	235	274	275	301
5	Deviation, %	81.1	67.5	26.1	19.3	16.6	9.6	8.7
6	Frequency (calculation with regard to the tip), Hz	52.8	101.1	161.1	213.5	237	264.3	290.3
7	Deviation, %	7.1	5.9	2.6	8.4	1.3	5.3	4.8

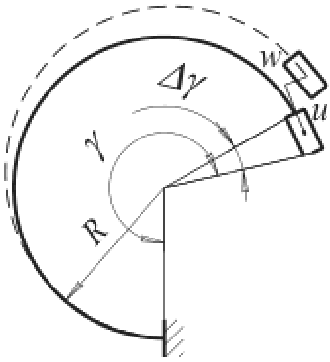

Figure 1. Curved rod.

Figure 2 shows the tangential - τ and the normal - n displacement of an infinitesimal element.

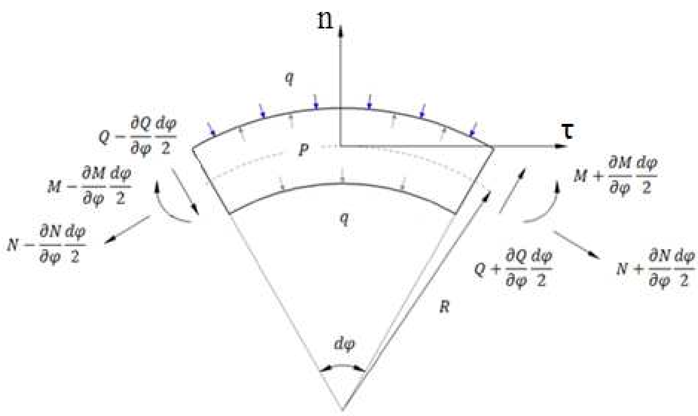

Figure 2. Infinitesimal rod element.

where q – the force of resistance to movement in a fluid (only the normal component is taken into account): $q_n = \beta v_n$; $q_\tau = 0$); β – the coefficient of damping fluid resistance; Q – the transverse force; N – the longitudinal force; M – the bending moment; R – the radius of curvature of the central axis; $d\varphi$ – the angle of an infinitesimal element cut from a curved rod.

Projecting all the forces, including the forces of inertia, on axes τ and n, let us obtain the system of equations of the MTS motion:

$$\begin{gathered} m(\varphi)\frac{\partial^2 w}{\partial t^2} + \left(1 + \frac{b}{R}\right)\beta\frac{\partial w}{\partial t} - \frac{\partial Q}{R\partial\varphi} + \frac{N}{R} = 0, \\ m(\varphi)\frac{\partial^2 u}{\partial t^2} - \frac{Q}{R} - \frac{\partial N}{R\partial\varphi} = 0, \end{gathered} \tag{1}$$

where $m(\varphi)$ – the apparent density (mass).

The following boundary conditions are imposed on this system of differential equations. At the base of the manometric spring, at $\varphi = 0$, the longitudinal and transverse displacements, as well as the angle of rotation of the tube cross-section, are zero (the main boundary conditions).

At the free end ($\varphi = \gamma$), near the concentrated mass, the MTS distortion can be neglected. This assumption leads to the following boundary conditions at the free end: $M(\gamma) = 0$; $N(\gamma) = -m_n\frac{\partial^2 u}{\partial t^2}$; $Q(\gamma) = -m_n\frac{\partial^2 w}{\partial t^2}$ (natural boundary conditions).

The initial conditions do not matter, since only the frequency characteristics of MTS are determined.

The problem is solved by the Bubnov-Galerkin method. Displacements *u* and *w* are given as:

$$\begin{gathered} u = \psi_1(\varphi)a_1(t) + \psi_2(\varphi)a_2(t) + \cdots \psi_n(\varphi)a_n(t) = \sum_{i=1}^{n}\psi_n(\varphi)a_n(t) \\ w = \zeta_1(\varphi)b_1(t) + \zeta_2(\varphi)b_2(t) + \cdots \zeta_n(\varphi)b_n(t) = \sum_{i=1}^{n}\zeta_n(\varphi)b_n(t) \end{gathered} \tag{2}$$

where $a_1, a_2 \dots a_n, b_1, b_2 \dots b_n$ – unknown functions of variable t;

$\psi_1, \psi_2 \dots \psi_v, \zeta_1, \zeta_2 \dots \zeta_v$ – basic functions of variable φ.

Power functions are used as basis functions:

$$\begin{gathered} \psi_i(\varphi) = \varphi^i; \quad i = 1, \dots, n\,. \\ \zeta_j(\varphi) = \varphi^{j+1}; \quad j = 1, \dots, n. \end{gathered} \tag{3}$$

Such system of functions is complete; the functions are linearly independent and satisfy the main boundary conditions.

To estimate the convergence of the solution, the authors studied the change in the values of oscillations attenuation parameters for a different number of functions included in the displacement approximation.

To obtain satisfactory results, it was sufficient to preserve five basic functions.

Calculations of natural oscillations frequencies taking into account the rigid tip are given in lines 7 and 8 of Table 1. As can be seen from the table, deviation from the experimental data is from 4 to 8%, which indicates a good accuracy of the results obtained.

The results of studies of the influence of the tip mass on the frequency of attenuating oscillations and the oscillation attenuation coefficient are presented as surfaces depending on the viscosity of the damping fluid and the mass of the tip for a sample with parameters R=30mm, γ, =30 degrees, h=0.2mm, a=5 mm, 2 mm (Fig.3).

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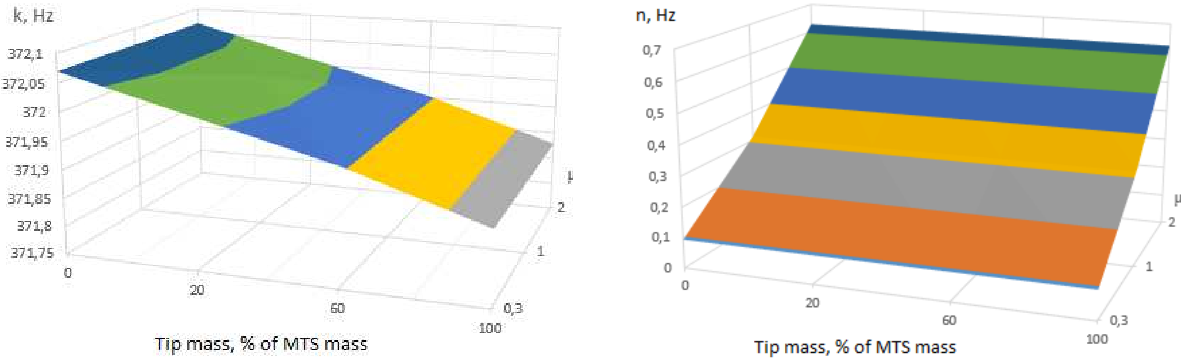
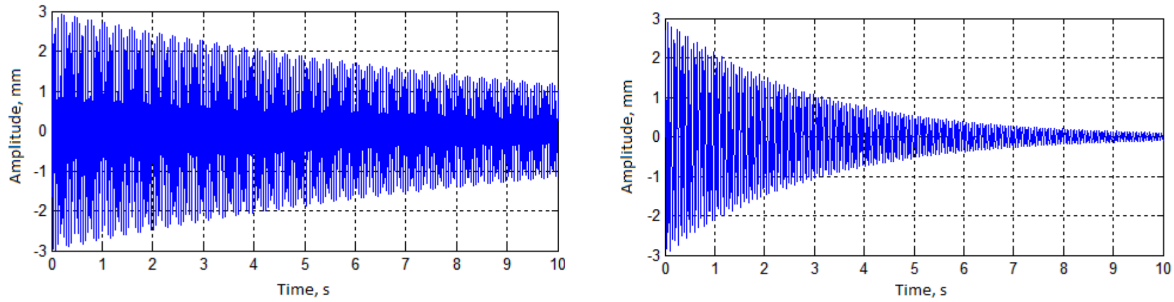


a) Frequency of oscillations b) Coefficient of attenuation

Figure 3. Changes in the oscillations attenuation parameters

Analysis of the presented dependences shows that an increase in the mass of the tip leads to a decrease in the frequency of attenuating oscillations and a slight decrease in the oscillation attenuation coefficient.

A decrease in the attenuation coefficient leads to an increase in the attenuation rate; the results of the investigation of springs with a tip and without it are shown in Fig.4



a) Without a tip b) With a tip

Figure 4. Amplitudes of oscillation attenuation

Since the change in the mass of the tip affects the oscillation process, the limiting value of damping fluid viscosity must undergo a change, at which aperiodic motion begins to take place. Below, Table 2 provides viscosity values at which aperiodic movement begins with and without a tip.

Table 2. Dynamic viscosity value

Dynamic viscosity value, Pas	Sample number	
	1	2
Without a tip	1000	1200
With a tip	1200	1400
Deviation, %	16.7	14.3

The obtained results show that as the mass of the tip increases, the limiting value of dynamic viscosity decreases, but it is possible to increase the mass only to a certain limiting value. Table 3 shows mass values as a percentage of the MTS mass at which the system will cease to oscillate.

Table 3. Dynamic viscosity value

Sample number	1	2
MTS mass, g	2.84	13.23
Tip mass, g	13.3	66.7
Tip mass, %	368.3	404.2

3. Conclusion

Numerical experiments confirm a significant influence of the tip mass on the MTS oscillation process. An increase in the mass of the tip leads to an increase in the total mass of the system, and this in essence is one of the methods of vibration protection - vibro-damping. Thus, an increase in the mass of the tip can be used as an independent method of vibration protection, and in combination with the method of detuning from resonance frequencies and vibro-damping of oscillations with a fluid, thereby increasing the effectiveness of protection.

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