Transport and Storage of Hydrocarbons IOP Publishing
IOP Conf. Series: Materials Science and Engineering **154** (2016) 012013 doi:10.1088/1757-899X/154/1/012013

# Prospects of Applying Vibration-Resistant Pressure Gauges in the Oil and Gas Industry

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**Abstract.** The article presents justification for improving vibration protection of pressure gauges used in the oil and gas industry. A mathematical model of manometric tubular spring oscillations in a viscous medium is viewed. By the developed model, the authors have determined the impact of manometric spring geometric characteristics and damping fluid viscosity on oscillation attenuation parameters, as well as provided evaluation of the impact of the cross-sectional shape on the oscillation attenuation rate.

## 1. Introduction

The application and development of pressure gauge constructions, aimed at increasing vibration resistance when working in harsh conditions in the oil and gas industry, are governed by a number of regulatory documents:

- "Regulations on industrial safety of hazardous production facilities where pressurized equipment is used " of 25.03.2014;
- Federal Law "On industrial safety of hazardous production facilities" of 21.07.97 №116-FZ (ed. of 04.03.2013);

"Framework of the federal system of monitoring the critical facilities and (or) potentially hazardous infrastructure facilities of the Russian Federation and dangerous goods" approved by the RF government order of 27.08.2005 №1314-r.

Harsh operational conditions lead to increased measurement error, wear of operating device and, as a result, cause breakdown in production process. At present, vibration damping by a fluid is an effective method of improving protection against vibration. Hereafter, evaluation of the impact of various characteristics on improving vibration protection of vibration-resistant pressure gauges is given.

## 2. Development of a mathematical model and comparison with test values

A sensing element – manometric tubular spring (MTS) – is presented as a curved bar. The motion of an infinitesimal element (figure 1) is composed of the longitudinal $u$ and cross $w$ components. The motion resistance force in the fluid is the distributed load q. Forces from the internal pressure on upper and lower parts of the cross-section of the infinitesimal element, at pressure $P$, are presented as distributed load inside the tube.

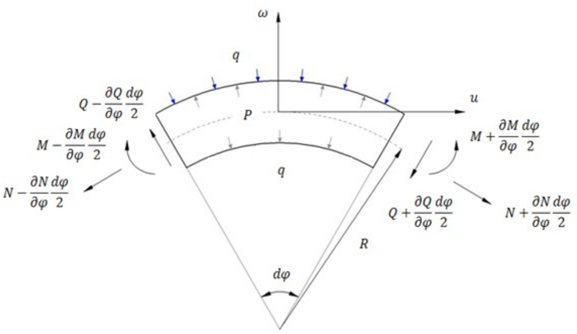


**Figure 1.** Infinitesimal bar element.

The system of differential equations of MTS motion is [1, 2]:

$$m(\varphi)\frac{\partial^2 w}{\partial t^2}+(1+\frac{b}{R})\beta\frac{\partial w}{\partial t}-\frac{\partial^2}{\partial\varphi^2}\{H_R(\frac{\partial u}{\partial\varphi}-\frac{\partial^2 w}{\partial\varphi^2})\}+\{D_R(\frac{\partial u}{\partial\varphi}+w)\}=\frac{4ab}{R}P,$$

$$m(\varphi)\frac{\partial^2 u}{\partial t^2}-\frac{\partial}{\partial\varphi}\{H_R(\frac{\partial u}{\partial\varphi}-\frac{\partial^2 w}{\partial\varphi^2})\}-\frac{\partial}{\partial\varphi}\{D_R(\frac{\partial u}{\partial\varphi}+w)\}=0. \tag{1}$$

where $m(\varphi)$– the mass of the single tube length ($m(\varphi)=\rho S(\varphi)$); $\beta$ – the resistance coefficient of the damping fluid; $a$ – the major semi-axis of the cross-section; $b$ – the minor semi-axis of the cross-section; R – the radius of the central axis curvature; $\partial\varphi$ – the angle of the infinitesimal element cut out from the curved bar; $H_R=\frac{EJ(\varphi)K_k(\varphi)}{(1-\mu^2)R^4}$; $D_R=\frac{ES(\varphi)}{(1-\mu^2)R^2}$ .

At the base of the manometric spring (at the section where the spring is rigidly fixed) - φ=0 longitudinal and cross motion, as well as the rotation angle of the spring cross-section equal zero. At the loose end φ=γ shear and tensile forces, as well as bending moment equal zero, therefore, we obtain boundary conditions:

Essential boundary conditions at φ=0:

$$U(0)=0;\ w(0)=0;\frac{\partial w}{\partial\varphi}(0)=0$$

Natural boundary conditions at φ=γ:

$$\left(\frac{\partial u}{\partial\varphi}(y)-\frac{\partial^2 w}{\partial\varphi^2}(y)\right)=0,\ \left(\frac{\partial u}{\partial\varphi}(y)+w(y)\right)=0\ ,\left(\frac{\partial^2 u}{\partial\varphi^2}(y)-\frac{\partial^2 w}{\partial\varphi^2}(y)\right)=0.$$

Correlation of the results produced by this model and the test results is given in figures 2-3. The comparison has been conducted on 10 steel samples with various geometric characteristics, glycerol being a damping fluid [3].

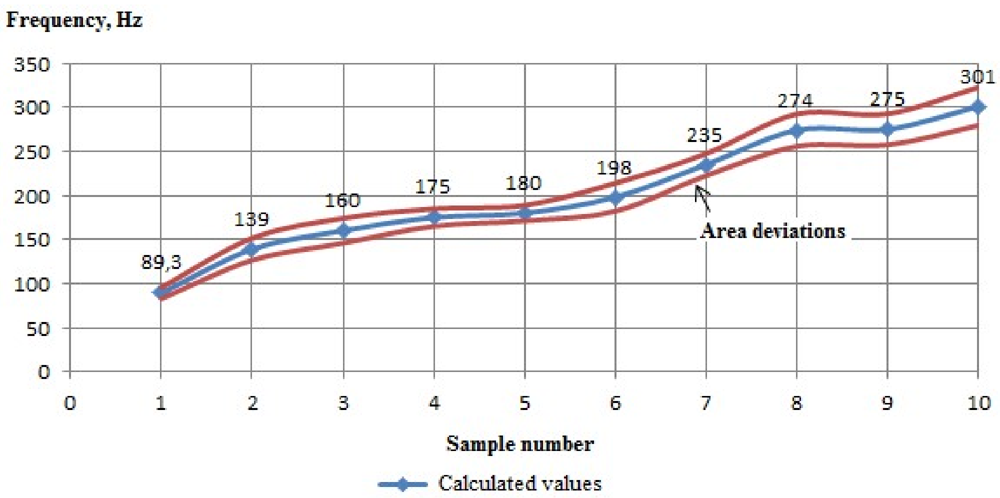


**Figure 2.** Comparison of calculated and test values of oscillation attenuation frequencies.

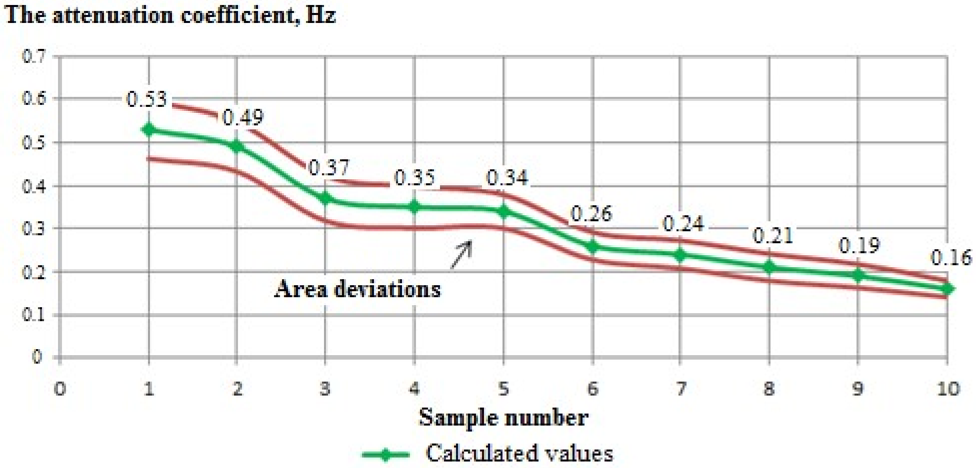


**Figure 3.** Comparison of calculated and test values of attenuation coefficient.

The comparison of theoretical and test values has demonstrated that deviations of frequencies do not exceed 10%, and those of the attenuation coefficient do not exceed 15%. This proves a reasonable accuracy of calculations, thus this model can be applied to calculate vibration-resistant pressure gauges.

## 3. Results and discussion

Using the developed model we have studied various vibration-resistant pressure gauges constructions [4], with the impact of MTS geometric characteristics and damping fluid viscosity on the attenuation coefficient shown in figures 4-7.

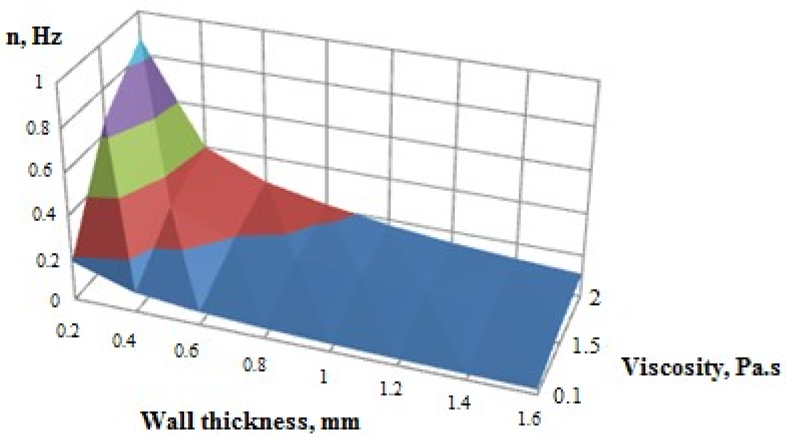


**Figure 4.** Impact of the MTS wall thickness and damping fluid viscosity on the attenuation coefficient.

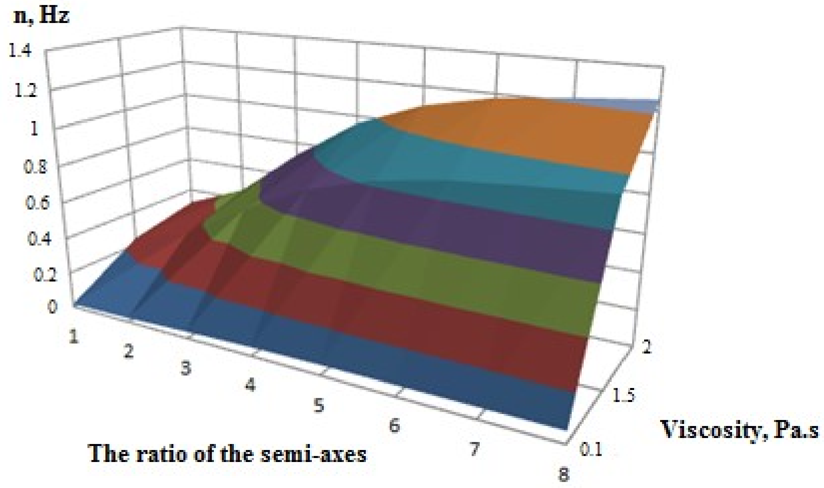


**Figure 5.** Impact of the ratio of the MTS semi-axes and damping fluid viscosity on the attenuation coefficient.

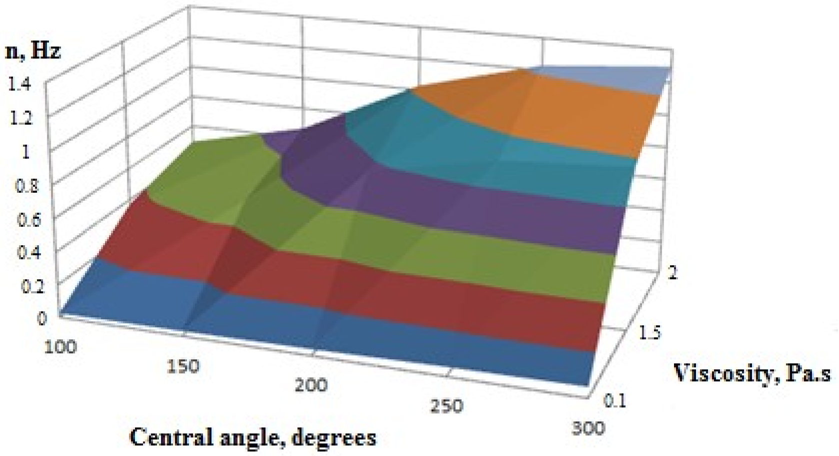


**Figure 6.** Impact of the MTS central angle and damping fluid viscosity on the attenuation coefficient.

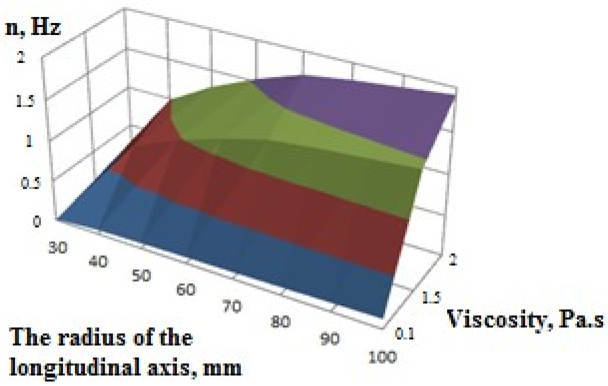


**Figure 7.** Impact of the radius of the MTS longitudinal axis and damping fluid viscosity on the attenuation coefficient.

The impact of MTS geometric characteristics and damping fluid viscosity on the oscillation attenuation frequency is shown in figures 8-11.

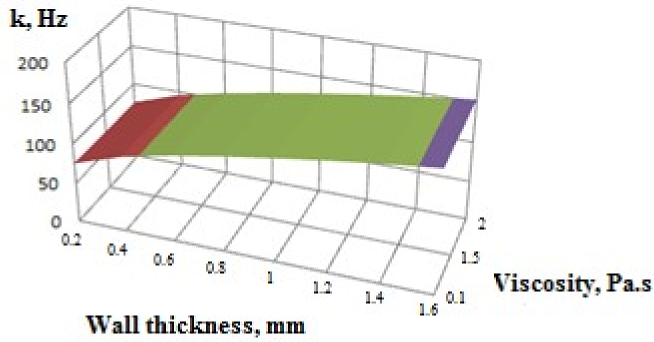


**Figure 8.** Impact of the MTS wall thickness and damping fluid viscosity on the oscillation attenuation frequency.

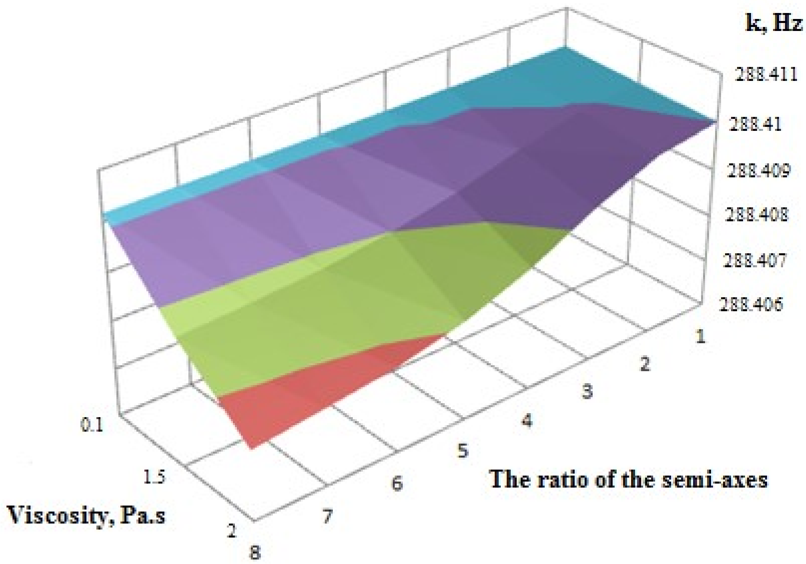


**Figure 9.** Impact of the ratio of the MTS semi-axes and damping fluid viscosity on the oscillation attenuation frequency.

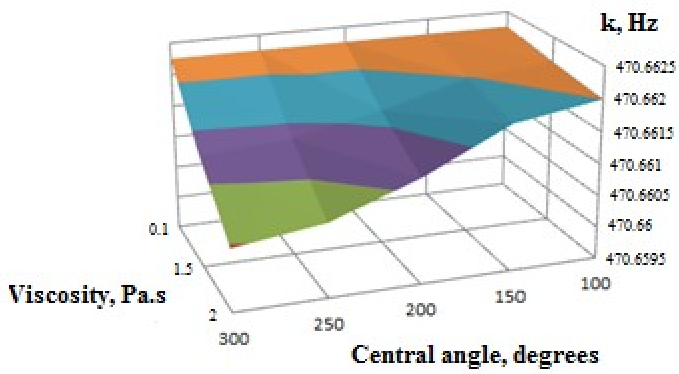


**Figure 10.** Impact of the MTS central angle and damping fluid viscosity on the oscillation attenuation frequency.

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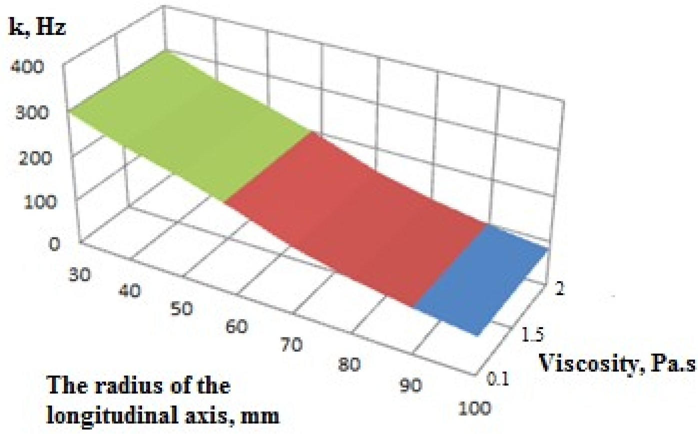


**Figure 11.** Impact of the radius of the MTS longitudinal axis and damping fluid viscosity on the oscillation attenuation frequency.

The study of the impact of the MTS sectional shape on the oscillation attenuation rate is shown in figure 12.

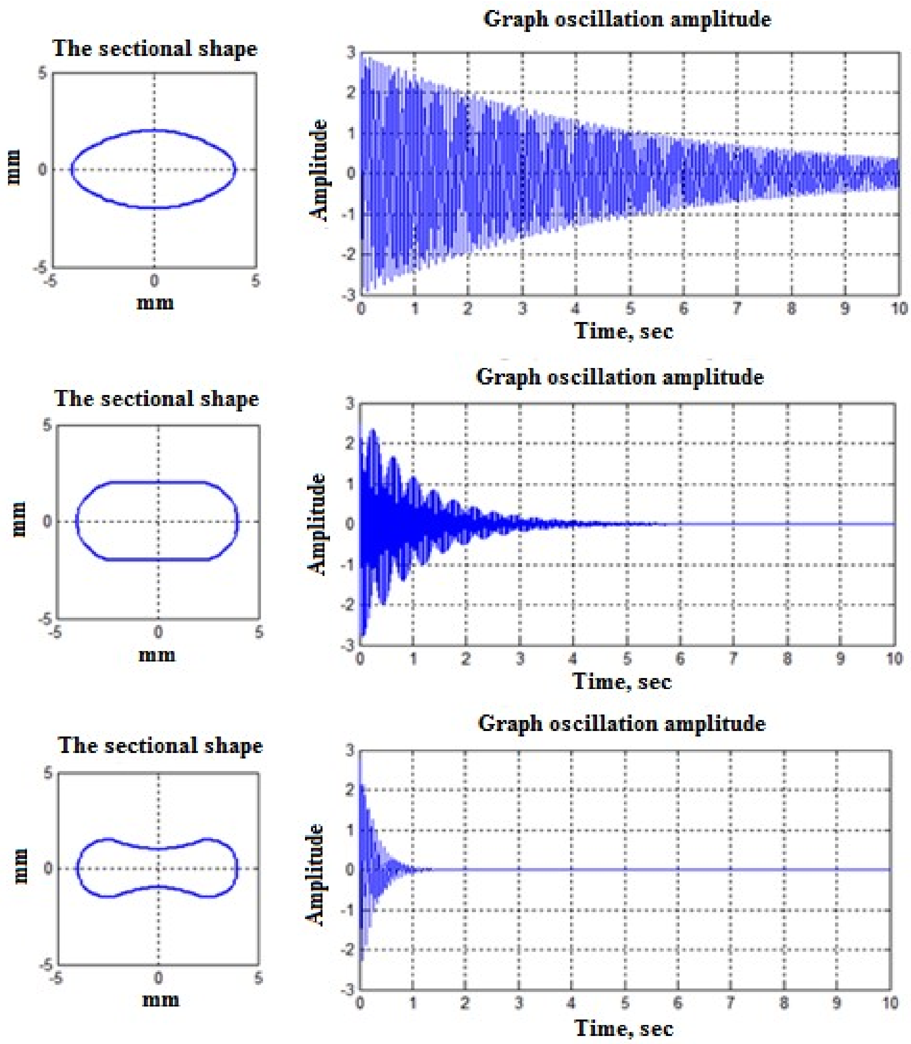


**Figure 12.** Impact of the MTS sectional shape on the oscillation attenuation rate.

## 4. Conclusions

For pressure gauges working within the resonance zone vibration damping by a fluid significantly reduces the time for oscillation attenuation. The study of the impact of the MTS geometric characteristics and damping fluid viscosity on the oscillation attenuation parameters has demonstrated that increasing the wall thickness leads to increasing the oscillation attenuation frequency and decreasing the attenuation coefficient, and increasing the ratio of the semi-axes, central angle и radius of the longitudinal axis leads to decreasing the oscillation attenuation frequencies and increasing the attenuation coefficient. The oscillation attenuation frequencies depend on the geometric characteristics of tubular springs. The attenuation coefficient is determined by dynamic viscosity of the damping fluid and, to a lesser extent, depends on the tubular spring geometric characteristics. A significant impact on the attenuation rate is exerted by the cross-sectional shape, tubes with the elliptical cross-section showing the smallest rate, and with the 8-shaped – the greatest.

## References

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