Physical features of fluid and structure interaction inside power unit pipeline vibration-isolating expansion joints

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Abstract. The calculation and experimental studies of vibration transmission through the pipeline compensators with liquid were carried out. No publications have been found among the scientific papers published on this issue. It has been demonstrated that the influence of structural resonances and the interaction of working medium pulsations with the structure increases vibration transmission through the compensator by two to three orders of magnitude or more with increasing frequency. The physical and mathematical models of this phenomenon are described, and a comparison of calculation and experiment is given. The ways of using the found interaction of medium pulsation and the structure to reduce vibration transmission by ten or more times within a certain frequency range are shown.

Introduction
At gas compressor and oil pumping stations, in power engineering, especially in the transport one, it is often necessary to reduce vibration transmission through pipelines. For this, compensators are used, which acting simultaneously as compensators for thermal expansions are vibration isolators. The experimental studies of various compensators carried out by the authors revealed an increase in the vibrational rigidity of compensators for pipelines with liquid (poorly compressible working medium) with an increase in frequency by several orders of magnitude. In this case, vibration transmission through the pipeline exceeds significantly the transmission through the supporting isolation of the installation. Below the results of studies of this phenomenon, carried out in the framework of the Russian Science Foundation grant under Agreement No. 16-19-10292 of May 12, 2016 between the Russian Science Foundation and KSU named after K.E. Tsiolkovsky, have been considered. In order to reduce the volume of the article, references are made to intermediate publications on the grant.

1. Influence of fluid
The experiments showed a significant effect of increasing the frequency $f$ and the presence of liquid inside the pipeline joints on the amount of vibrational transition rigidity (hereinafter simply rigidity) $C(f)$. It is defined as the ratio of the vibration force $Q(f)$ transmitted to the foundation at the fixed output of the compensator to the vibration amplitude $A(f)$ at its input at the frequency $f$, and it is a complex value \cite{1-3}. At zero frequency, it is equal to the static rigidity of the compensator. Figure 1 shows a diagram of a pipeline compensator based on rubber-cord shells (RCS) 1. Its total rigidity $C(f)$ with water (1) and structure with air (2) in the axial direction is shown in figure 2. The total rigidity increases by three orders of magnitude with increasing frequency from zero to 400 hertz.
The presence of water increases the total rigidity as compared to the structural rigidity by 100 times. The effect of water grows with frequency.

Other types of compensators based on bellows, rubber-cord sleeves and RCO tested by the authors have a similar pattern of rigidity dependence on frequency and the presence of fluid inside (see, for example, figure 7 for a compensator with bellows). This phenomenon adversely affects the effectiveness of vibration isolation through pipelines of various power plants from foundations, for example, on oil and gas pipelines, in the energy and shipbuilding industries.

Figure 2. Rrigidity $C(f)$ with water (1) and air (2) of a compensator with an RCS, shown in figure 1.

2. Physical and calculation models of the formation of rigidity
The authors were unable to find scientific publications, describing the effect of the fluid in the anti-vibration compensator pipelines in the transmission of vibration and vibration forces through the compensator. As part of the research under the Agreement No. 16-19-10292 of May 12, 2016 with the Russian Science Foundation, relevant experimental and computational studies of this phenomenon were carried out.
The appearance of the unloaded straight-through compensator based on the bellows. Figure 3.

Figure 4. The axial deformation scheme of the straight-flow bellows compensator shown in figure 3. A is the axial displacement, S is the cross-sectional area of the small (side) bellows, 2S is the area of the large (medium) bellow, \( \Delta V_1 \) is the volume change of the small bellow, \( \Delta V_2 \) is the volume change of the large bellow, 1 - pump movable pipeline, 2 - small bellow, 3 - large bellow, 4 - rigid bilateral thrust, 5 - stationary pipeline, 6 - working medium (water). Arrows indicate the flow of water between the bellows.

The typical designs of compensators unloaded by static pressure with diameters of up to 750 mm based on bellows, rubber-cord sleeves, RCS, and developed experimental designs were tested. In the unloaded compensator the thrust forces from internal pressure affect the elements of its structure. The appearance of such a compensator based on bellows is shown in figure 3. Figure 4 shows a diagram of its axial deformation. During axial deformation of such a compensator, there is no change in the total internal volume and separating forces between the external flanges from the internal pressure. At the same time, there is a change in local internal volumes and a flow of medium occurs between them. It turned out that the smaller the structural rigidity, the greater the influence of water. Therefore, reducing structural rigidity does not solve the problem of reducing vibration transmission, which is determined by pressure pulsations. The compensator is a source of pressure pulsations and vibration forces. For various types of existing compensators [1,2], physical and computational models were developed for the occurrence of pressure pulsations and corresponding vibration forces during vibration deformation.

In the development of models and subsequent calculations based on them, the inextensibility of the compensator elastic elements under the action of internal pressure was assumed (at the same time they provided deformation due to bending and shear). Their rigidity was taken to be static. The kinematic excitation of the movable input flange with a fixed output flange was considered. The experimentally determined sound speed in the liquid inside compensator and connected pipelines was 1250 m/s. The problem was solved in a linear formulation, when the vibration amplitudes were significantly less than the characteristic gaps in the compensator. Three models of the occurrence of pulsations were obtained and researched. The first was determined by compensator oscillations, as a solid moving with a liquid. The second was determined by the flow of a liquid column between the local cavities during the vibration deformation of the compensator. As an example, Figure 4 shows the scheme of fluid flow between the large and two small bellows of the compensator shown in Figure 3. The third model describes the displacement of fluid from the slots (corrugations of bellows and annular cavities with a small height and a large area). Figure 5a shows the design scheme for models 2 and 3. For model 2, the pressure pulsation P is determined by the flow of the liquid mass \( M_{w2} \) along the piston wall at its
height $h_p$ during its vibration $X_p$. For model 3, the pressure pulsation $P_3$ under the piston on the piston area $S_{w3}$ is determined by the acceleration of a part of the mass $M_{w3}$ when it flows out of the slit with a height $h$ under the piston. The smaller the slit height $h$ in comparison with the piston diameter and its height, the more significant the influence of model 3. Analysis of compensator deformation schemes, a description of the models discovered, and the results of experimental and computational studies are given in detail in the following publications [1,2].

For the transient vibration rigidity of the compensator $C(f)$, determined by the fluid, for all three models, the general expression was found

$$C(f) = \frac{Q}{A} = \omega^2 \cdot \rho \cdot k_{\text{geom}}$$ (1)

where $k_{\text{geom}}$ is coefficient determined only by the geometry of the compensator, $Q$ is vibration force at the outlet of the compensator, $A$ is vibration amplitude at its input, $\omega$ is angular oscillation frequency, $\rho$ is fluid density. The magnitude of rigidity $C(f)$ of the pulsation pressure in the compensator due to its vibration deformation is proportional to the working fluid density and the square of the frequency of deformation. The latter may be a diagnostic sign of the predominance of just such a model of rigidity in experimental studies.

3. Comparison with experiment

The validation test of three models developed was carried out on experimental test benches and showed good correspondence of experimental and calculated data. The calculated and experimental data shown in figure 5b for the most complex third model (fluid displacement from the slits) agree well up to frequencies of 1000 hertz.

![Figure 5. Schemes of models 2 (piston) and 3 (piston near the wall) of pressure pulsation appearance in the compensator a) and comparison of the calculated and measured rigidity $C(f)$ for model 3 with a piston diameter of $D_p$, 50 mm and a slit $h = 1$ mm in the pipe $D = 80$ mm b).](image)

The use of the developed models for calculating the rigidity of various designs of tested compensators made it possible in some cases to develop analytical calculated dependences for the pressure and rigidity pulsations $C(f)$. Calculations based on them showed fairly good correspondence with the experiment. For example, for an unloaded compensator with a diameter of 100 mm based on the bellows, the calculation coincides quite well with the experiment for frequencies of up to 450 hertz, figure 6.
4. Results application to reduce vibration rigidity of compensators.

Figure 7 shows the experimentally obtained reduction in rigidity \( C(f) \) of a bellow-based unloaded compensator with water 2 at a frequency of 110 Hz by two orders of magnitude relatively to rigidity without water 1. Calculations and additional experiments showed that this was because of water pulsations in antiphase with forces transmitted through the compensator structure. According to expression (1), the pulsation forces grow in proportion to the square of the frequency. At the center frequency of the “dip” of 110 Hz, these forces are mutually compensated and give the minimum value of the transient vibration rigidity \( C(f) \). Up to the center frequency of the “dip,” the phase shift of the total force relative to the vibration of the flange is zero, and is determined by the transmission of vibration through the structure. Above the “dip” frequency, the phase changes by 180° and is determined by the fluid pulsation.

Figure 7. The effect of water on the vibration rigidity \( C(f) \) of a double bellows expansion joint with a diameter of 100 mm. 1- air (structural rigidity), 2 water.

This allows minimization of the transient vibration rigidity of the compensator at a given frequency in the low-frequency range by means of design measures. The way to adjust the “dip” frequency can be a change in the structural rigidity of the compensator. The higher the structural rigidity, the higher the frequency of the “dip”. Another way is to change the height of the oscillating fluid column in the compensator, for example, by increasing the mass of the oscillating fluid (the length of the compensator). The higher the oscillating liquid column, the lower the frequency of the “dip”. The second method does not affect the static rigidity of the compensator.

Conclusion

Calculated and experimental studies of vibration transmission by various designs of unloaded compensators with liquid showed the following. The component of their vibrational rigidity, which is due to the flow of the working medium inside the compensator, is proportional to the density of the medium and its vibrational acceleration, which is proportional to the acceleration of the moving parts of the compensator. Therefore, the rigidity component with water is increasing as the squared
frequency rate of the compensator deformation. The proportionality coefficient depends only on the
design of the compensator.
Three physical models of this phenomenon have been discovered. Their analytical description made it
possible to obtain good agreement with the experiment for the frequencies of the order of magnitude
equal to 1000 Hz. The use of the models discovered in calculating the vibrational rigidity of relatively
simple designs of compensators, for example, based on bellows, showed a good agreement between
the calculation and the experiment. The use of these models to reduce the vibrational rigidity of the
compensator in a certain frequency range due to the mutual compensation of elastic forces and
pressure pulsations has been proposed and verified experimentally. For more complex designs,
numerical methods should apparently be applied considering the physical models discovered. This
requires additional computational studies using the experimental data obtained in the research.

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