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REDUCTION OF VIBRATION FORCES AND PRESSURE PULSATIONS IN THE PIPELINE COMPENSATORS WITH PASSIVE AND ACTIVE METHODS

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ABSTRACT

In the energy and transport engineering, in oil and gas transportation systems, it is often necessary to reduce the vibration transfer through the pipelines. The use of compensators in pipelines is not enough for effective vibration isolation of equipment. There are no papers on the issue of reducing the vibration transfer through compensators with liquid, including active methods. The research carried out by the authors has demonstrated, that due to the interaction of the compensator structure vibration and pressure pulsations, with the growth of the frequency the vibration transfer through the compensator and its rigidity can grow by orders of magnitude.

On the basis of experimental studies of the vibration transfer through the compensators with liquid, physical and analytical calculation models of compensators structure vibration and resulting pressure pulsations interaction have been created. The comparison of results of the vibration transfer calculations and pressure pulsations through the compensators based on bellows, rubber-shells, new types of compensators with thin-layer rubber-metal elements with the experiment has demonstrated the accuracy of the models.

Analytically and experimentally it has been proved the existence of the frequency range, wherein the dynamic (vibration) forces, transferred through compensator *structure, are compensated with the pressure pulsations forces. Vibration transfer decreases ten and more times compared to the compensator structure without fluid. A physical explanation of this phenomenon has been given and methods of its use in practice have been suggested.*

A two-channel active vibroprotective system (AVS) with a digital control device for broadband combined damping of dynamic forces and pressure pulsations, that determine the vibration transfer through the pipeline compensators with liquid, has been experimentally investigated. The efficiency was up to 32 dB at active damping of forces and up to 25 dB at the pressure pulsations damping in the frequency range from 10 to 350 Hz. Directions for further research have been determined.

Keywords: Plant, Vibration, Pressure Pulsation, Dynamic Force, Pipeline Compensator, Vibration Rigidity, Frequency, Active Vibroprotective System.

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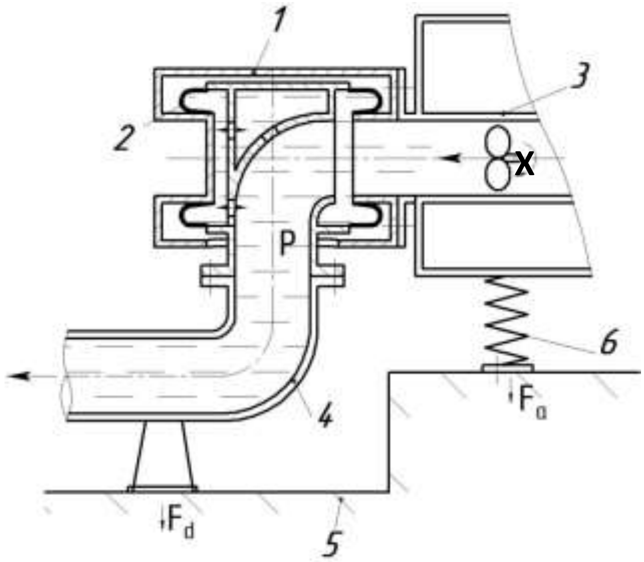
1. INTRODUCTION

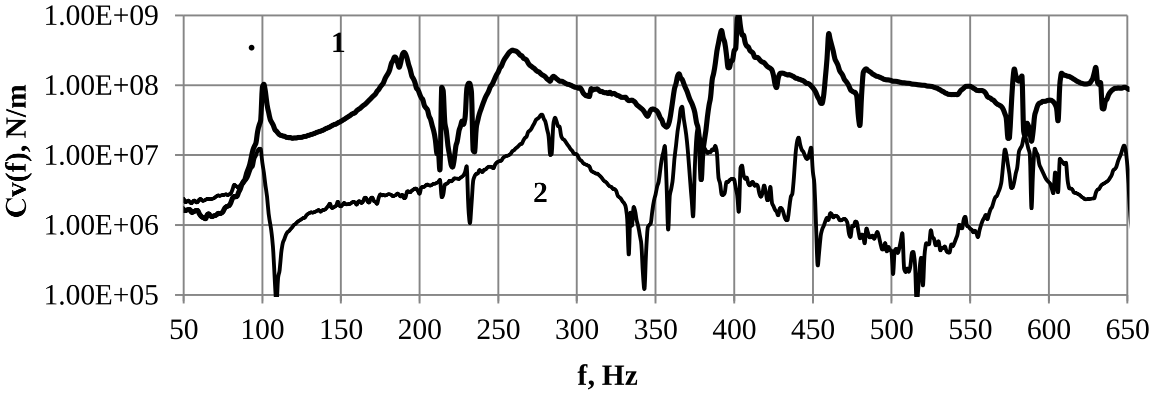
1.1. Frequency Dependencies of Vibration Transfer through Compensators

The analyses of publications show that there are practically no research papers on the issue of reducing the vibrations transfer through the compensators with liquid, including active methods. There are no corresponding physical and mathematical models and methods of calculation. The authors' studies [1] have shown the next phenomena. The interaction of the compensator structure vibrations and working fluid pulsation increases with frequency growth, and the vibration transfer through the compensator can grow by orders of magnitude respectively. This is true for almost all compensators and complicates the creation of new compensator designs with low vibration transfer. This problem is important today in transport and power engineering.

The static pressure unloaded compensator designs are considered. Figure 1a shows the angular unloaded compensator 1 based on diaphragm type rubber-cord shells (RCS) 2 between the pump 3 and the pipeline 4. Figure 2a shows the appearance of two types of unloaded compensators based on bellows with screeds. In the unloaded compensators, there are no strut forces between the connecting flanges. These forces for high-pressure and high-diameter compensators can reach thousand kN. Due to the design of the compensator, these forces are closed to the power elements inside the compensator.



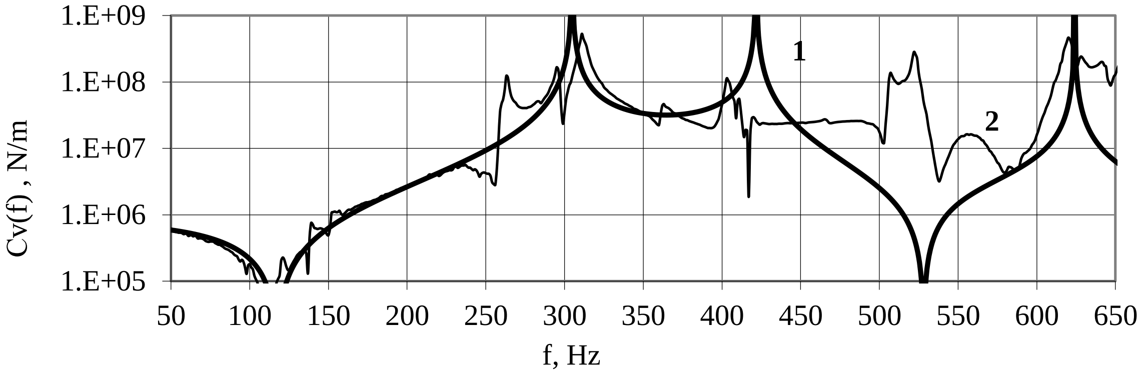
a)



b)

Figure 1 a - an angular unloaded compensator 1 with RCS of diaphragm type 2 with a pump 3 and a pipeline 4; b - its transfer vibration rigidity with water (1) and air (2) in the axial direction X

a)



b)

Figure 2 a- the appearance of statically unloaded compensators of angular type A and direct flow type B; b - comparison of calculated 1 and experimental 2 vibration rigidity $C_v(f)$ of the compensator type A DN100 mm with water

The vibration transfer from the power plant or pump 3 through the compensator 1 of the pipeline 4 with the liquid on the foundation 5 (the force F_d) often exceeds its transfer through the support vibration isolation 6 (force Fa, figure 1a) [1]. The methods for vibration transfer reducing through the pipeline with passive dampers by reducing the medium pressure pulsations were considered in [2, 3]. To reduce the transfer of vibration through the pipeline structure, elastic junctions (compensators) are used [4]. Vibration isolation quality of the compensator can be characterized through the value of its transfer vibration rigidity $C_v(f)$. The value $C_v(f)$ determines the dynamic force $F_{d,}$ transferred by the compensator 1 from the active hydraulic device, e.g., the pump 3, through the pipeline 4 to the foundation 5 (Figure 1a). It is defined as the ratio of the dynamic force $F_d(f)$ on the braked (fixed) output of the compensator to the vibration amplitude A (f) at its input:

$$C_v(f) = F_d(f)/A(f). \quad (1)$$

At perturbance frequency f=0 vibration rigidity is equal to the static one. With decreasing $C_v(f)$, vibration isolation improves. In expression (1), all the quantities are complex. To compare the variants, the rigidity modulus is used. Experiments [1, 4, 5, 6] show that with increasing frequency for compensators with a liquid, the increase in the vibration rigidity with respect to the static one can be of the order and more in a wide frequency range. The presence of a liquid can significantly increase the rigidity of the compensator. This is due to the interaction of the liquid and the compensator structure when they oscillate. As an example, Figure 1b shows the influence of deformation frequency f and the presence of water on the experimentally measured axial transfer vibration rigidity of the angular unloaded compensator with RCS (Figure 1a). With increasing frequency $C_v(f)$ is increased by three orders of magnitude. The presence of water increases the rigidity compared to structural rigidity by almost two orders of magnitude over a wide frequency range up to 650 Hz. The experiments carried out on different types of compensators based on the RCS, bellows, rubber-cord sleeves, with the diameter of up to 750 mm, have shown that the less structural rigidity, the more effect of the water influence is. Reducing the structural rigidity of the compensators does not solve the problem of reducing vibration transfer, which is determined by pressure pulsations. The compensator itself is a powerful source of pressure pulsations and dynamic forces.

2. PHYSICAL AND MATHEMATICAL MODELS AND COMPENSATOR CALCULATION RESULTS

There are no published data on the reasons for significant growth with the frequency of vibratory rigidity of the pipelines compensators with water. The paper [4] just pointed out the dependence of the experimental transfer vibration rigidity of compensators on the frequency and described the methods of their experimental determination. The complex matrix [4], which describes the transfer of forces and pulsations through the compensator, has the dimension of 13*13. Experimentally determining the elements of this matrix is a difficult task. It is unclear which of them and how is necessary to change for vibrations transfer reduction at a given frequency. Therefore, in the papers [5, 6], the authors developed and proved experimentally physically visual models and calculation analytical dependencies. They describe the occurrence of pressure pulsations and the dynamic forces caused by them, which determine the transfer of vibration through the compensators in the frequency range up to hundreds of hertz. Dynamic forces F_d from the pressure pulsation in the compensator are connected with the oscillation frequency f through a quadratic dependence.

Analysis of the vibrational deformation schemes of various types of unloaded compensators in the paper [6] shows, that their deformation is accompanied by a working fluid flow between the internal local volumes without changing the total compensator volume. The greater the moving mass of the fluid and the higher its acceleration, the greater the resulting pressure pulsations are. Acting on the surface of the compensator, pressure pulsations create dynamic forces transferred to the pipeline and the foundation. In general terms, for all models transfer vibration rigidity of compensators, caused by pressure pulsations, has the following form:

$$C_v(f) = F_d(f)/A(f)= (2\pi f)^2 \rho k_g, \quad (2)$$

where k_g is a coefficient determined only by the geometry of the compensator and the boundary conditions on its flanges. The transfer vibration rigidity $C_v(f)$, due to the pressure pulsation in the compensator, is proportional to the density of the working fluid ρ and to the square of the circular deformation frequency. These models allowed constructing of analytical dependencies for calculation of pressure pulsations, vibrational forces and transfer vibration rigidity of various compensators. The correspondence of calculation and experiment in Figure 2b for an unloaded compensator of type A on the basis of bellows (Figure 2a) to frequencies of the order of 650 Hz can be considered good.

3. STRUCTURAL FORCES COMPENSATION BY FORCES FROM PRESSURE PULSATIONS

Compensators based on bellows, sleeves, RCS and thin-layer rubber-metal elements were experimentally investigated and calculated. For all designs, a noticeable decrease ("fall") of the vibration rigidity was observed compared to structural rigidity over a sufficiently wide low-frequency range caused by the presence of water. For the type A compensator (Figure 2a) with bellows with a diameter of 100 mm, a "fall" is observed at frequencies from 50 to 150 Hz (Figure 3). The maximum rigidity reduction by 100 times compared with the structural rigidity occurs at a frequency of 110 Hz. At frequencies above the "fall", water increases rigidity compared to the structural component.

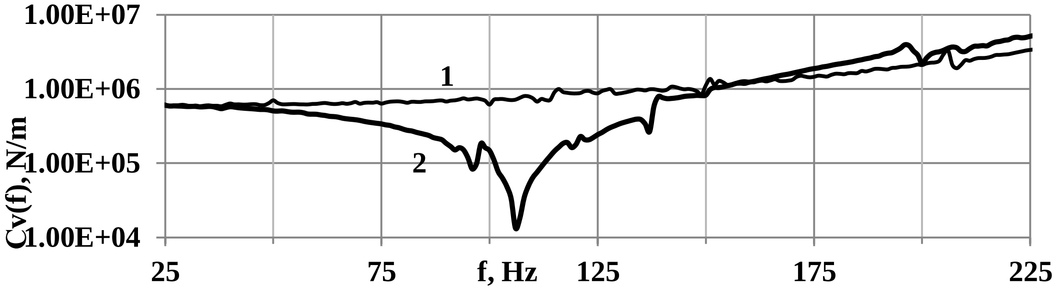


Figure 3 Effect of water on the vibration rigidity $C_v(f)$ of the compensator type A DN100 mm, Figure 2: 1 air (structural rigidity), 2 water

A computational and experimental study of the causes of the appearance of a "fall" in the $C_v(f)$ dependence showed the following. The structural component of the total force $F_d(f)$ from the deformation of the elastic elements of the compensator acts in phase with the displacement A (f) of its input (movable) flange. It was shown in papers [5, 6] that pressure pulsations P appear in the phase with the acceleration of the flange and in the antiphase with its displacement A (f). Therefore, the forces from the pulsation and structure are subtracted. Forces from pulsation grow in proportion to the square of the frequency. At the central frequency of the "fall", these forces cancel each other out and give the minimum value of the total force $F_d(f)$ and the rigidity of $C_v(f)$. Up to the central frequency of the "fall", the phase shift of the total force with respect to the vibration of the flange is zero and is determined by the transmission of vibration through the structure. Above the "fall" frequency, the phase changes by 180° and is determined by the pulsations of the liquid.

The carried out research allows constructive measures to minimize the transfer vibration rigidity of the compensator in a given frequency range in the low-frequency region. This can be the reverse or blade frequency of the pump or compressor. The way to adjust the frequency of the "fall" may be to change the structural rigidity of the compensator. The higher it is, the higher the frequency of the "fall" is. Another way is to change the height of the oscillating column of liquid in the compensator, for example, due to the length of the section between the bellows in Figure 2a. The longer it is, the lower the frequency of "fall" is. The second method does not affect the static rigidity of the compensator.

4. USING ACTIVE METHODS

Further reduction of vibration transfer through the compensators can be carried out by active vibration protection systems (AVS) [1, 7-10]. AVS create compensating dynamic forces, for example, acting on the structures connected to the installation in antiphase to the forces acting from the installation. Active systems for suppressing pressure pulsations act analogously. AVS can reduce both discrete components in the vibration ranges and random oscillations in the frequency range. There are no works in the literature to reduce the transfer of vibration through compensators of pipelines with liquid by active methods. Below we consider the results of testing the experimental wide range two-channel AVS of simultaneous suppression of forces and pressure pulsations in compensators. AVS and the stand scheme is shown in Figure 4.

The plate 1 is mounted on the dynamic force sensors 2 on the foundation 3. The compensator 5 with the pipe 4 and water is mounted on the plate 1. The piston 6 is excited by an electro-dynamic vibrator V_p with a power amplifier PAp. It creates pressure pulsations P in the pipe and compensator, measured by hydrophones 7. Acting on the internal surfaces of the pipe and compensator, pulsations create a dynamic force F_p, exciting the vibrations of plate 1. The channel of structural excitation of vibration consists of an electrodynamic vibrator V_v

with a power amplifier PAv. It creates a dynamic force F_v acting through the compensator on the plate 1. Signals to the power amplifiers are fed from the signal generator SG in the form of a broadband random signal or a sinusoidal signal of a predetermined frequency. The vibration force F_d of the plate 1 on the foundation 3 is measured by the force sensors 2.

The V_c vibrator with power amplifier PAfc creates a compensating force Fc. It acts on the plate 1 and reduces the force F_d measured by sensor 2. The signal on PAfc and V_c is generated by a multi-channel control device CD based on signal processing from the force sensors F_d. All vibrators are vibroprotected from plate 1 and foundation 3 by means of an elastic suspension with an intrinsic frequency of less than 1 Hz. A piezoelectric actuator 8 with a power amplifier PApc produces compensating pressure pulsations. The signal to the actuator 8 is formed by the control device CD based on the signal processing from the hydrophone 7.

In the force compensation and pressure pulsation channels, the signal from the feedback sensor, the force sensor 2 or the hydrophone 7, is transmitted to one of the inputs of the A/D converter of the control device CD. The signal is digitized and transmitted to the processor module CD, where a narrowband or broadband compensating signal is generated using a digital controller. Through the digital-to-analog converter, the signal is fed to the appropriate power amplifier.

The possibility of constructing (synthesizing) an effective regulator in a feedback loop based on standard bandpass filters of various orders, Butterworth, Chebyshev, elliptic, and resonant links (resonators) was experimentally investigated. The study showed that based on band filters of the first and second orders an effective regulator can be built to damp dynamic forces and pressure pulsations in a wide frequency range. The parameters of the control device were set from the computer in the MATLAB environment via the RS-485 interface. The signals were monitored and processed by a multichannel signal analyzer SA of the 'Puls' type from Brüel & Kjær.

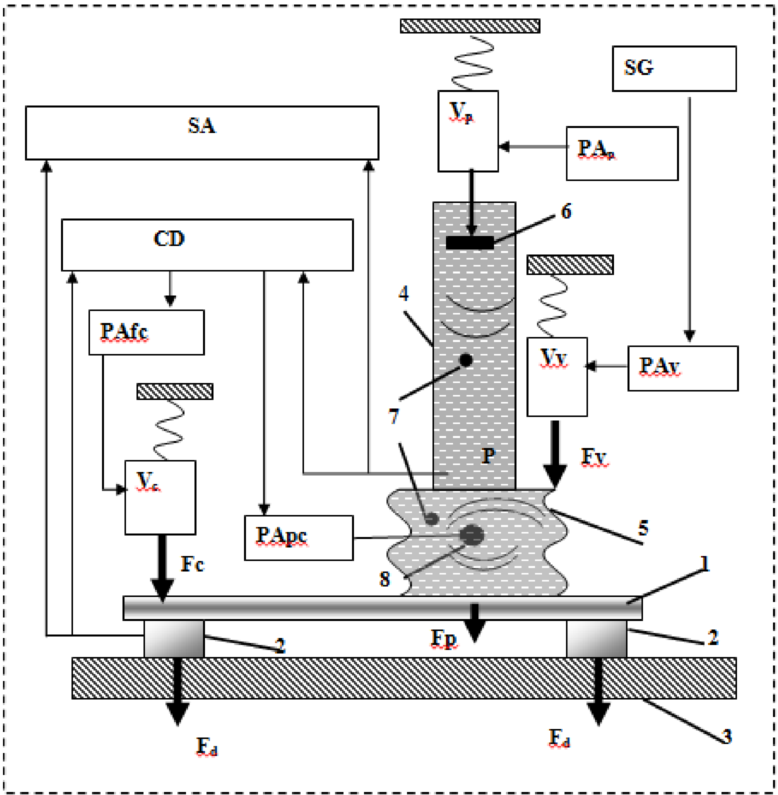


Figure 4 Stand of two-channel AVS. CD-control device, PA-power amplifier, V-vibrator, SG-signal generator, SA-signal analyzer. Indices: c - compensation, p - pulsations, v -vibration, f – force.

The characteristics of the regulator (filter) and the gain coefficient in the feedback circuit were chosen to obtain maximum damping without loss of system stability. The results of the active damping of forces $F_d(f)$ under the resonator plate with a central frequency of 120 Hz are given in Figure 5. The damping of the force at a frequency of 120 Hz reaches a value of 32 dB. The width of the range, where the force $F_d(f)$ decreases by half or more, is from 80 to 150 Hz.

Figure 6 shows the damping of the force simultaneously by two elliptical filters: first order 30-70 Hz, second order 50-120 Hz and by two resonators 130 Hz and 160 Hz. The decrease in the dynamic force reaches 15 dB, and the width of the active damping range is 10-170 Hz. At the same time, active damping suppresses signals from interference 3 coming from the foundation at frequencies of 20-40 Hz, 50, 78, and 100 Hz.

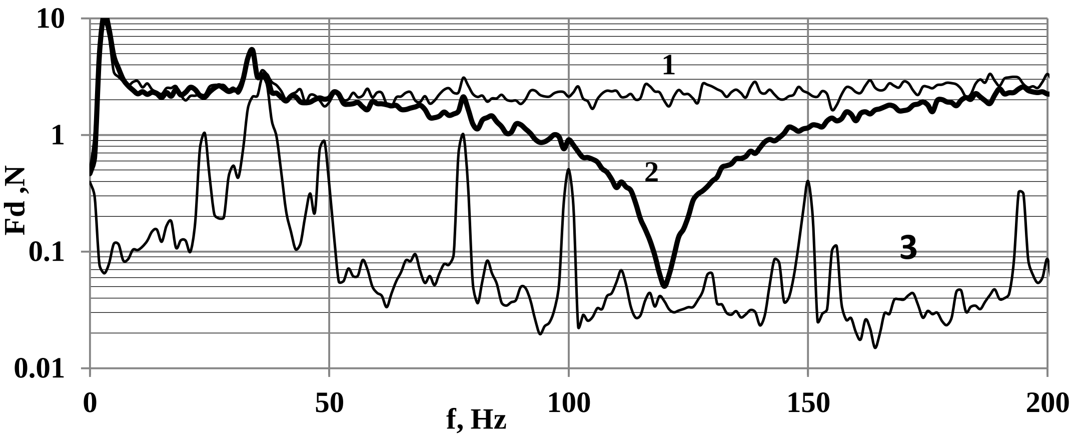


Figure 5 Active damping the dynamic force by a resonator at a frequency of 120 Hz 1-initial signal, 2-damping, 3- interference

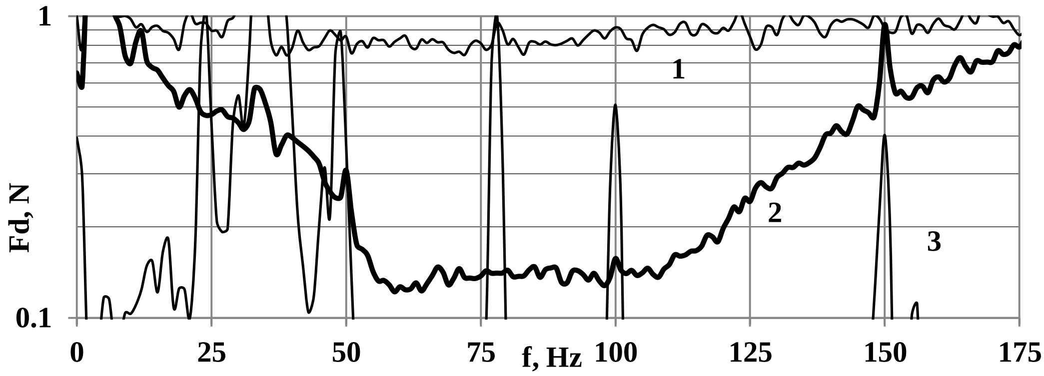


Figure 6 Active damping the dynamic force by elliptical filters of the first order of 30-70 Hz, the second order of 50-120 Hz, and two resonators 130 Hz and 160 Hz. 1-initial signal, 2- damping, 3-interference

Figure 7 shows the results of the pulsations P active damping at the first resonance in the liquid column in the pipe 4 (Figure 4) at a frequency of 310 Hz by a resonator with the frequency 335 Hz and the phase $\pi/2$. Active damping reduces the amount of pulsations P almost twenty times. Reducing strength $F_d(f)$, acting from pulsations on plate 1, is also 26 dB at a frequency of 310 Hz.

The results of simultaneous active damping of pulsations P by a second-order elliptical filter in the range 110-160 Hz and forces below the plate $F_d(f)$ by a second-order elliptical filter in the range 120-140 Hz with simultaneous excitation of forces and pulsations are shown in Figures 8a and 8b. The active damping efficiency P is up to 10 dB, the forces $F_d(f)$ is up to 15 dB. At the same time, both damping channels work successfully, practically without affecting each other.

It should be noted that when using band-pass filters at frequencies below and above the active damping range, zones of negative efficiency appear (Figures 6, 7, 8). They are caused by the fact that the regulator (filter) at the boundaries of its work reverses the phase of the signal, and instead of damping amplifies the original signal.

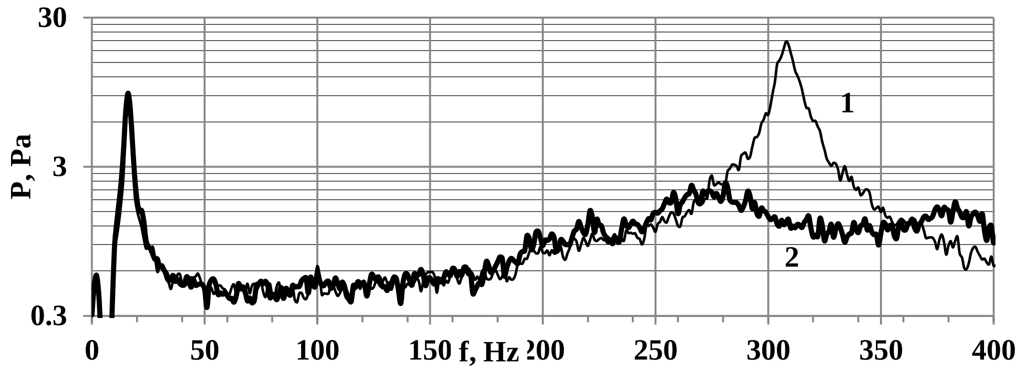
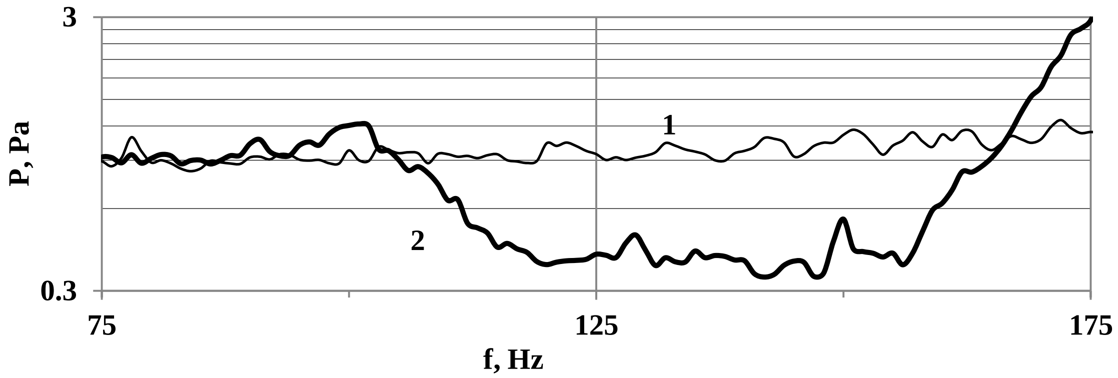
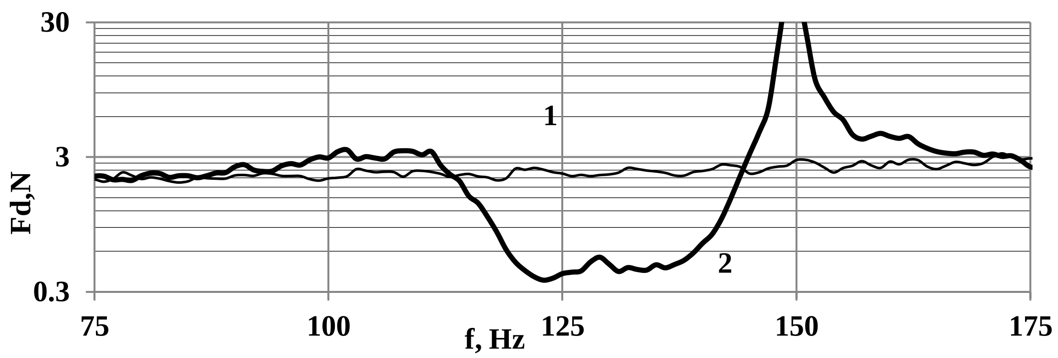


Figure 7 Combined damping of pulsation P and force $F_d(f)$ at 310 Hz resonance by a 335 Hz resonator in the P channel with the $\pi/2$ phase. 1-initial signal, 2 – damping



a)



b)

Figure 8 Simultaneous damping by pulsation and force. Combined damping by second-order elliptic filters of pulsation P in the range P110-160 Hz. a) and the force $F_d(f)$ in the range 120-140 Hz. b). 1 - initial signal, 2 - damping

In order to increase the stability and efficiency of AVS, narrowband filters of the "resonant link" type (resonators) were added to broadband filters. They were located at frequencies at which an intensive growth of the control signal was observed with an increase in the gain coefficient in the feedback when approaching the stability boundary. Resonators increased the efficiency of damping and dissipation in the system. Variants with filters of higher orders turned out to be less effective than those shown. With the expansion of the damping range with the same type of regulator, the maximum efficiency of the ABC is reduced.

To estimate the importance of the obtained results of active damping up to 10 or more times in wide frequency ranges, we shall point out the following. In the power engineering, increase in the vibro velocity overall level of the equipment from 4 mm/s to 7 mm/s (an increase of less than twice or 6 dB) results in a reduced time of the equipment service life, and at 11 mm/s (an increase of fewer than 3 times, or 10 dB), the equipment must be stopped.

5. CONCLUSIONS AND RECOMMENDATIONS

1. The reduction of the vibration transfer through the compensators of pipelines with liquid, including by active methods is practically unexplored today. This problem is important for equipment vibration isolation from the foundation and the environment along the pipeline in the energy and transport mechanical engineering, shipbuilding, as well as for oil and gas pipelines in pumping stations.
2. For most existing compensators of pipelines with an increase in vibration frequency, there is an increase in the vibration rigidity compared with the static one which is tens and hundreds of times over a wide frequency range. It is caused by the occurrence of pressure pulsations inside the compensator during its vibrational deformation, as well as by structural resonances of the elastic elements in the compensator.
3. Physical models and calculated analytical dependencies have been developed which describe the occurrence of pressure pulsations and dynamic forces in pipeline compensators and determine vibration transfer therethrough in the frequency range up to hundreds of hertz. Their validity is confirmed by the correspondence of calculations and experiments.
4. Pressure pulsations, dynamic forces, vibration rigidity of the compensator, due to pressure pulsations, are related to the oscillation frequency of the compensator by a quadratic dependence. With increasing frequency, they can increase by three or four orders of magnitude compared with low frequencies. This may serve as a diagnostic sign of the predominance of this rigidity model in the compensator in its experimental research.
5. It has been analytically shown and experimentally confirmed that there is the frequency range in which vibration forces transferred by the structure of the compensator are compensated by forces from the pressure pulsations. The decrease of the vibration rigidity of the compensator and the corresponding reduction in the vibration transfer can reach ten or more times compared with the transfer along the compensator structure without fluid. At frequencies above this range, the presence of liquid increases the vibration transfer through the compensator. A physical explanation of the phenomenon is given. Constructive ways have been suggested which reduce vibration transfer through the compensator at the required frequency range in a low-frequency region by changing the structural rigidity of the compensator and changing the height of the liquid column in the compensator.
6. An experimental two-channel AVS has been investigated. When standard band pass filters of various orders, Butterworth, Chebyshev, elliptical, as well as resonant links

(resonators) are used as regulators in the feedback loop, it provides a simultaneous reduction of vibration forces and pressure pulsations by an order of magnitude both at discrete frequencies and in sufficiently wide frequency ranges from tens to hundreds of herz.

7. The increase in the AVS efficiency with broadband damping is limited by the appearance of auto-oscillations with increasing gain coefficients in the feedback loop. One of the ways of increasing efficiency may be the introduction of dissipation units into the controller.
8. Further studies of the simultaneous active damping of the pressure pulsations and dynamic forces in pipeline compensators must be aimed at improving the control algorithms, the development of computational methods of pulsations and dynamic forces transferred by the compensators with liquid in the pipeline, including methods of finite elements.

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