International Journal of Civil Engineering and Technology (IJCIET)
Volume 9, Issue 10, October 2018, pp. 351–359, Article ID: IJCIET_09_10_036
Available online at https://iaeme.com/Home/issue/IJCIET?Volume=9&Issue=10
ISSN Print: 0976-6308 and ISSN Online: 0976-6316

TEST RESULTS OF THE DIGITAL ACTIVE SYSTEM FOR DYNAMIC FORCES REDUCTION AND PRESSURE PULSATIONS DAMPING IN PIPELINE COMPENSATORS

A.V. Kiryukhin
Kaluga State University named after K.E. Tsiolkovsky, 26, st. Stepan Razin Kaluga, 248000, Russia
Scientific Production Company " Turbokon ",43, st. Komsomolskaya rosha, Kaluga, 248010, Russia

O.O. Milman
Kaluga State University named after K.E. Tsiolkovsky, 26, st. Stepan Razin Kaluga, 248000, Russia
Scientific Production Company " Turbokon ",43, st. Komsomolskaya rosha, Kaluga, 248010, Russia

A.V. Ptakhin
Scientific Production Company " Turbokon ",43, st. Komsomolskaya rosha, Kaluga, 248010, Russia

M.O. Korljakova
Moscow State Technical University named after N.E. Bauman, Kaluga branch, 248000 Kaluga, Str. Bazhenov, 2

ABSTRACT

Developmental active system with a digital control device for simultaneous wideband damping of dynamic forces and pressure pulsations in the pipeline compensators with liquid has been experimentally investigated. Excitation and compensation of vibration and dynamic forces were carried out with electro dynamical vibrators, pressure pulsations piezo electric emitters. The possibility of using standard bandpass filters of Butterworth, Chebyshev, elliptic of various orders types as regulators in the feedback loop has been investigated. Dynamic forces reduction and pressure pulsation damping up to 32 dB in the frequency ranges from 10 to 400 Hz has been obtained.

Keywords: Vibration, Pressure Pulsation, Dynamic Force, Pipeline Compensator, Frequency, Active Vibroprotective System (AVS).

Cite this Article: A.V. Kiryukhin, O.O. Milman, A.V. Ptakhin and M.O. Korljakova, Test Results of the Digital Active System for Dynamic Forces Reduction and Pressure Pulsations Damping in Pipeline Compensators, *International Journal of Civil Engineering and Technology (IJCIET)*, 9(10), 2018, pp. 351–359.
https://iaeme.com/Home/issue/IJCIET?Volume=9&Issue=10

1. INTRODUCTION

1.1. Vibration Transfer Features through Compensators with Liquid

Unloaded compensators are considered. They do not have any expansion forces from the internal pressure between the connecting flanges. Due to the design, these forces are closed to the power elements inside the compensators. Figure 1 shows an angular unloaded compensator 1 based on rubber-cord shells of diaphragm type 2. Vibration transfer through the compensator 1 via the elastic elements 2 and liquid from the pump 3 to the pipeline 4 and the foundation 5 may exceed the transfer through the support vibration isolation 6 (Figure 1) [1-4]. The matter is that vibrational rigidity of most compensators with liquid significantly grows with an increase in the frequency. The transient vibration rigidity of the compensator $C_v(f)$ at the frequency f is defined as the ratio of the dynamic (vibrational) force $F_d(f)$ at the fixed compensator outlet to the vibration amplitude A (f) at its inlet (Figure 1) in the corresponding direction:

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

As in expression (1) values are complex, for the comparison of variants module $C_v(f)$ was used. The increase in the vibration rigidity with respect to the static one (f =0) may be of the order or more in the wide frequency range for most compensators [1, 3, 4] with frequency f increase. The presence of liquid can increase the rigidity of the compensator. Figure 2 shows the dependence of transitional vibration rigidity in direction Z on the frequency and the presence of water for the angular compensator 250 mm in diameter, shown in Figure 1. $C_v(f)$ increases more than a hundred times in the frequency range of up to 650 Hz. The presence of water increases the rigidity by more than an order of magnitude.

Vibratory rigidity of the support vibration isolation 6 is low and has little dependency on the frequency [4]. At frequencies above several tens of hertz, dynamic forces F_d, transferred by compensators, can greatly exceed forces F_a, transferred by the support vibration isolation 6 (Figure 1). Research of various compensators showed that the lower the structural rigidity, the greater the effect of water is. Therefore, the decrease of compensator structural rigidity does not solve the problem of reducing the transfer of vibration, which is determined by pressure pulsations. Research of the pulsations and vibrations interaction in the compensators in [4] has shown that the compensator is a source of pressure pulsations and dynamic forces.

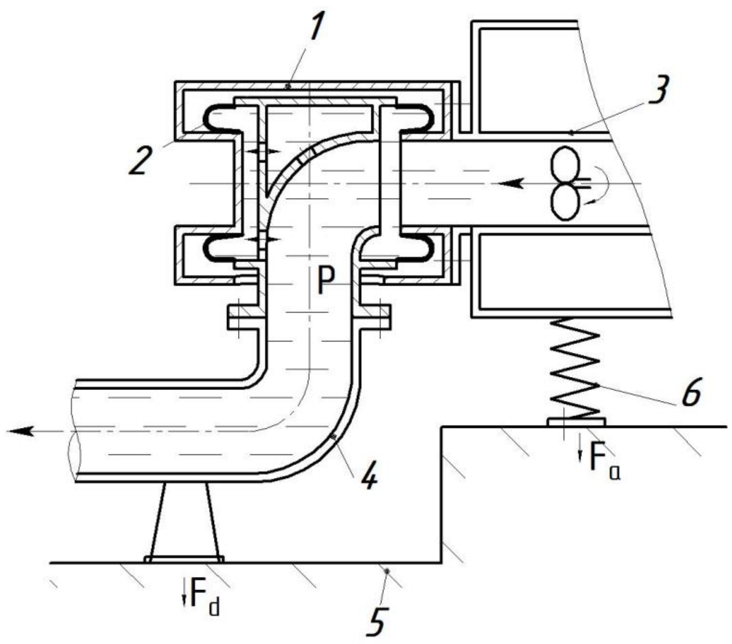


Figure 1 An angular unloaded compensator 1 with elastic elements based on rubber-cord shells of diaphragm type 2 between the pump 3 and the pipeline 4

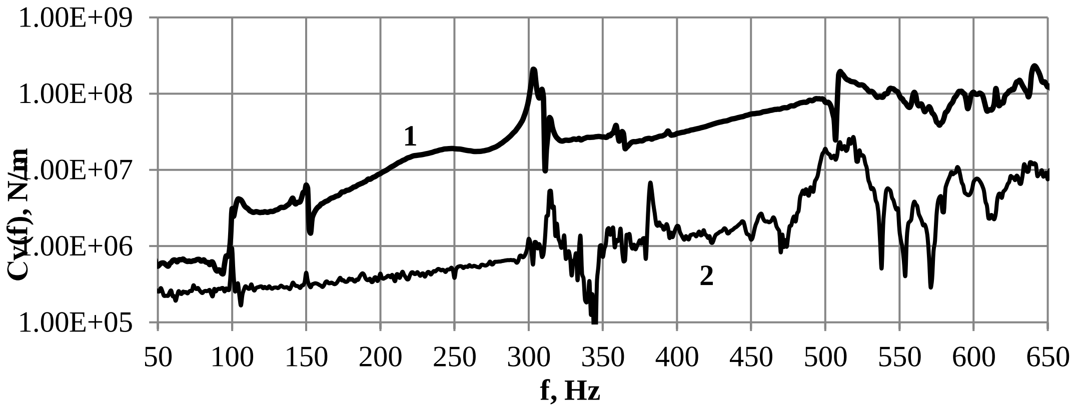


Figure 2 Vibration transitional rigidity of an angular compensator of 250 mm diameter (figure 1) with water (1) and air (2) in the direction **Z**

Therefore, the problem of reducing the vibration transfer through the pipeline compensators with liquid is important in power and transport engineering, shipbuilding, and for oil and gas pipelines at pumping stations. Effectiveness of passive design techniques for reducing dynamic forces and pulsations in compensators is limited by strength, dimensions and physics of working processes in plants and compensators. Reduction of vibration transfer through them can be obtained with the help of active vibroprotective systems (AVS). [1, 5-8].

2. TEST BENCH OF DYNAMIC FORCES AND PRESSURE PULSATIONS ACTIVE DAMPING IN COMPENSATORS

Active systems create vibration effects that compensate for the initial ones from the plant operation. AVS can reduce the discrete components f in the vibration spectra and the random fluctuations in the frequency range. The authors were unable to find works on the active methods of reducing the vibration transfer through the pipeline compensators with the liquid.

Scheme of the AVS test bench for dynamic forces and pressure pulsations suppression in compensators with liquid is shown in Figure 3.

Plate 1 is mounted on the dynamic force sensors 2 on the foundation 3. The compensator 5 with the pipe 4 and the water is mounted on the plate 1. Pressure pulsations P in the pipe and the compensator are created by piston 6 and are measured by hydrophones 7. The piston 6 is excited by an electrodynamic vibrator Vp with the power amplifier PAp. Pulsations P create dynamic force Fp, exciting vibration of plate 1. The structural excitation of vibration is carried out by the vibrator Vv with PAv power amplifier. It creates dynamic force Fv, acting through the compensator on the plate 1. Random or sinusoidal signal of a predetermined frequency to the power amplifiers are fed by the signal generator SG. The total vibration force Fd from plate 1 to the foundation 3 is measured by force sensors 2.

The vibrator Vc with a power amplifier PAc creates a compensating force Fc. It reduces the force Fd measured by sensor 2. The signal on PAc and Vc is generated by a multi-channel control system CD based on signal processing from the force sensors Fd. All vibrators are vibroisolated from plate 1 and foundation 3 by means of an elastic suspension with an intrinsic frequency of less than 1 Hz. Piezo electric emitter 8 with power amplifier PAp creates compensating pressure pulsations. The signal to the emitter 8 is formed by the control system CD based on the signal processing from the hydrophone 7.

The signal from the feedback sensor, the force sensor 2 or the hydrophone 7 is fed to one of the analog-digital converter inputs of the control system 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.

3. TEST RESULTS

The possibility of designing an effective controller in a feedback loop based on the standard band pass filters of various orders, Butterworth, Chebyshev, elliptic and resonant links (resonators). The filter parameters were set from the computer in the MATLAB environment through the RS-485 interface. The signals were processed by a multichannel SA analyzer. The parameters of the filter and amplification in the feedback loop were selected to obtain maximum damping without loss of the system stability. Because of the research based on bandpass filters of the first and second order, regulators for damping dynamic forces and pressure pulsations in a wide frequency range were built. At the resonance frequencies that determine the stability (their intensive growth was observed with increasing gain coefficient in the feedback loop as it approached the stability boundary). To increase stability and efficiency, resonators with damping were added to broadband filters. They were placed at resonant frequencies, where an intensive growth of vibrations was observed with increasing gain coefficient in the feedback loop as it approached the stability boundary.

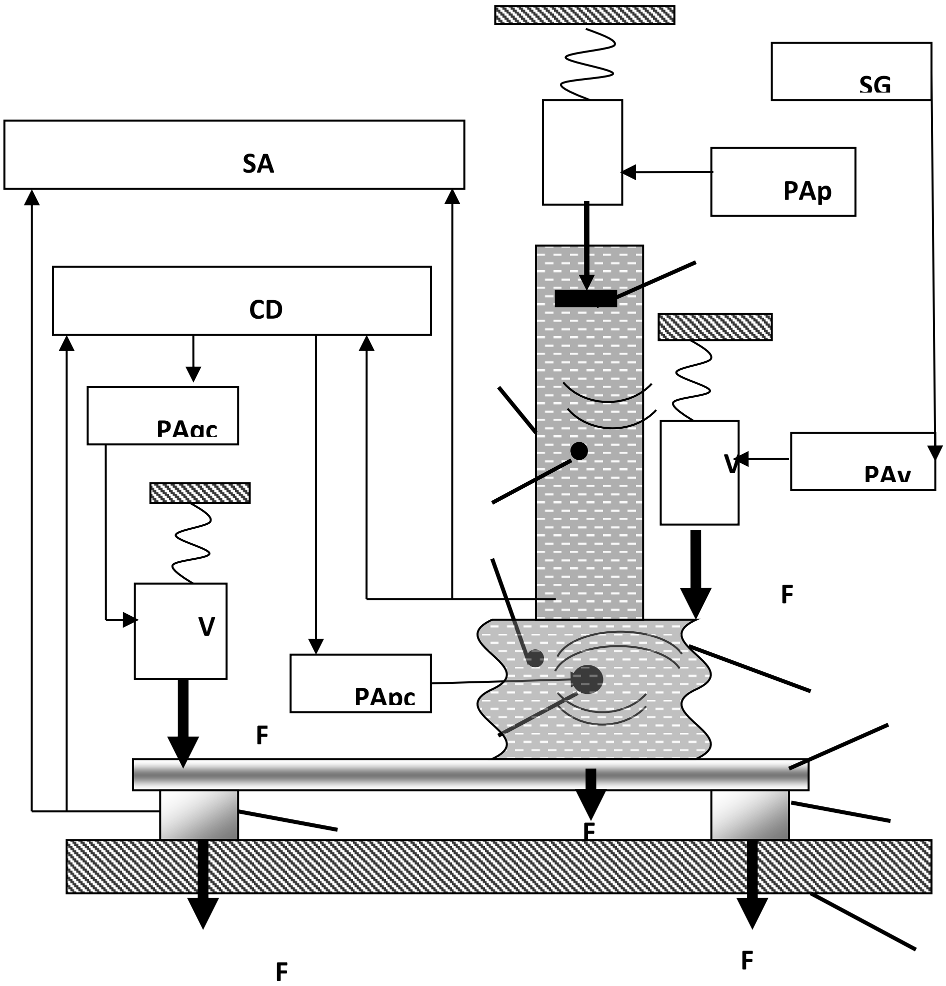


Figure 3 Test bench of a two-channel AVS. CD - control device, PA - power amplifier, V - vibrator, SG - signal generator, SA - signal analyzer. Indices: c - compensation, p - pulsations, v – vibration

The results of the forces $F_d(f)$ active damping under the plate with the help of a resonator with a central frequency of 120 Hz are shown in figure 4. At a frequency of 120 Hz the force is reduced by 30 times. The width of the range, where the force $F_d(f)$ decreases by half or more, is from 80 to 150 Hz. Wideband damping of forces $F_d(f)$ under the plate with the help of five elliptical filters and a resonator at a frequency of 145 Hz decreases the force by almost an order of magnitude, and the width of the damping range Δf is almost five octaves, from 10 to 280 Hz (Figure 5). Active damping also suppresses disturbance 3 from foundation vibration at frequencies of 20-40 Hz, 50, 78, 100 and 150 Hz. The filter at the boundaries of the bandpass reverses the phase of the signal. Therefore, zones of negative efficiency appear below and above the damping range. The use of Butterworth and Chebyshev filters showed almost the same, but lower efficiency. Options with filters of higher orders were less effective compared with those described above.

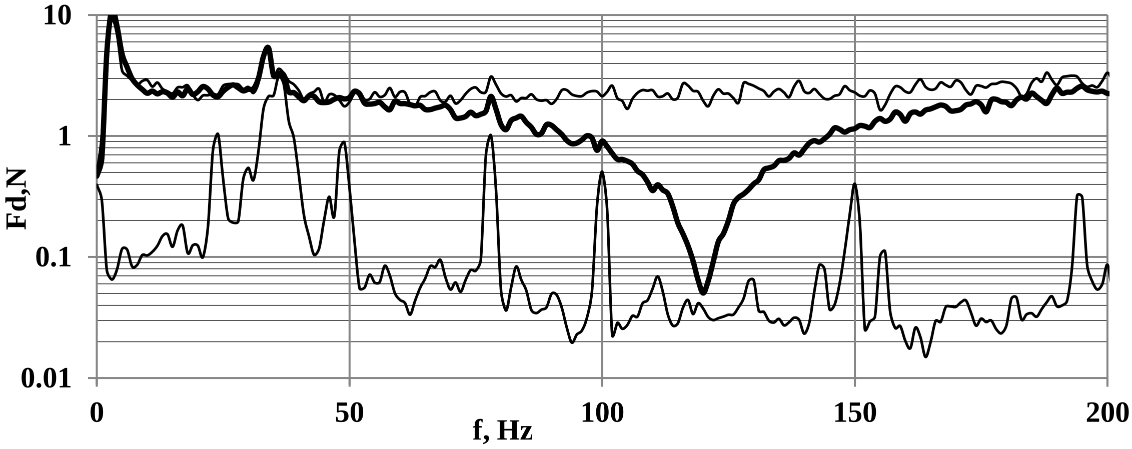


Figure 4 Active damping by a resonator at 120 Hz. 1-initial signal, 2- damping, 3- disturbance

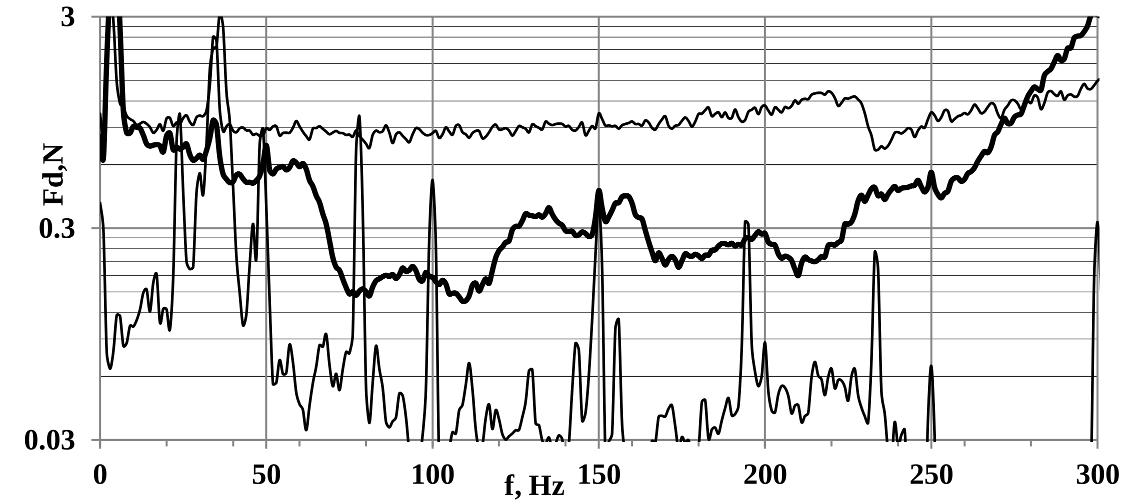
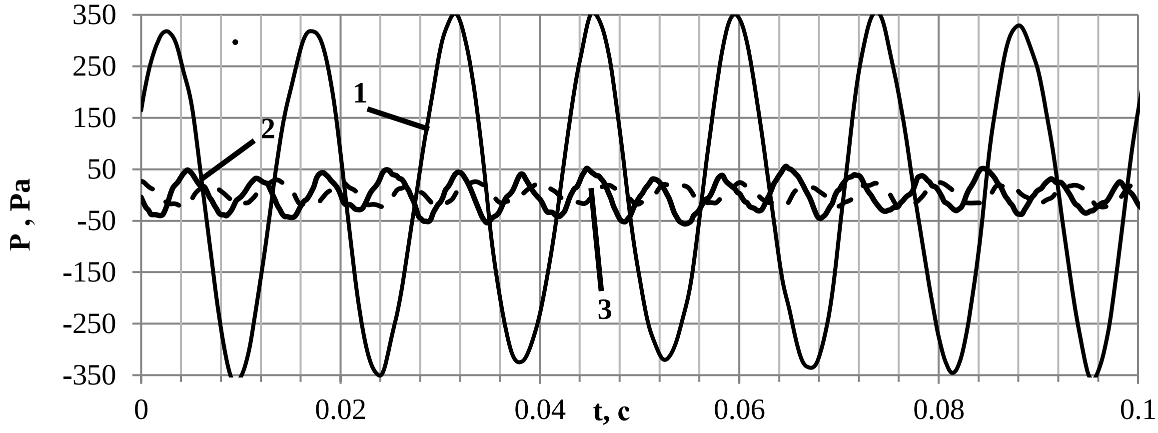


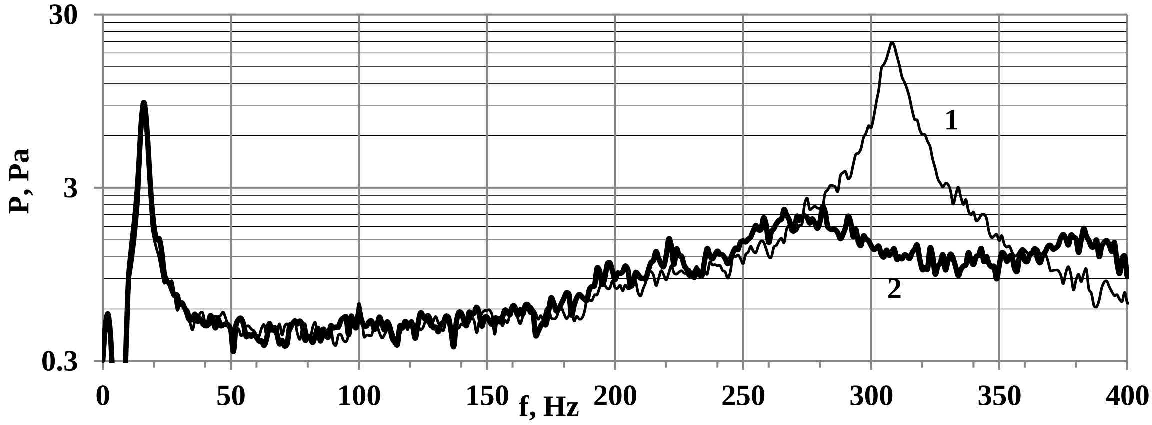
Figure 5 Excitation by force and active damping by force. Elliptical filters Δf = 40-90 Hz, filter order n = 1, 70-120 Hz, n = 2, 100-180 Hz, n = 2, 160-220 Hz, n = 2, 200-260 Hz, n = 2, the resonator is 145 Hz, phase 0. 1- the initial signal, 2-damping, 3-disturbance

Upon excitation of the pulsations active pressure pulsations P damping at a discrete frequency of 70 Hz at the bottom of the pipe with sinusoidal signal is shown in Figure 6a in time range. The initial signal is reduced by an order of magnitude, practically to the level of disturbance. Figure 6b shows the damping of pulsation P results at 310 Hz resonance with a 335 Hz resonator in the P channel with the π/2 phase. The signal at resonance is reduced by 20 times. Figure 6c shows the results of pulsation P damping at 310 Hz resonance with an elliptical filter of the 1st order. The signal at the resonance is reduced by more than 30 times, the damping range is from 100 to 400 Hz. Dynamic forces Fd in this case are equal to the forces Fp from pressure pulsations (Figure 4). Their reduction practically repeats the decrease of pressure pulsations P.

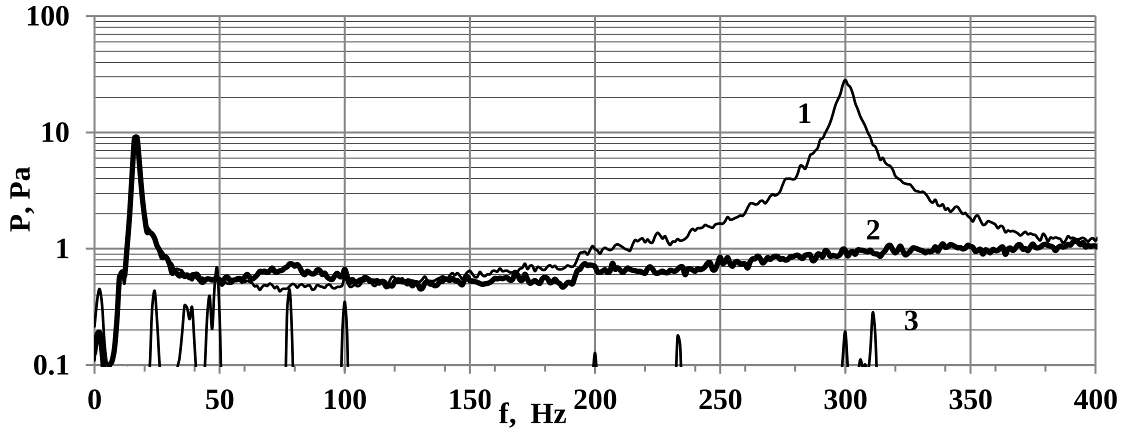
The results of simultaneous active pulsations P damping with an elliptic filter of the second order in the range from 110 to 160 Hz and the forces $F_d(f)$ under the plate with a second order elliptical filter in the range 120-140 Hz with the simultaneous excitation of forces and pulsations are shown in Figures 7a and 7b. The efficiency of the active damping P is up to 10 dB, the forces $F_d(f)$ damping is up to 15 dB. At the same time, both damping channels work successfully, practically without affecting each other.



a)

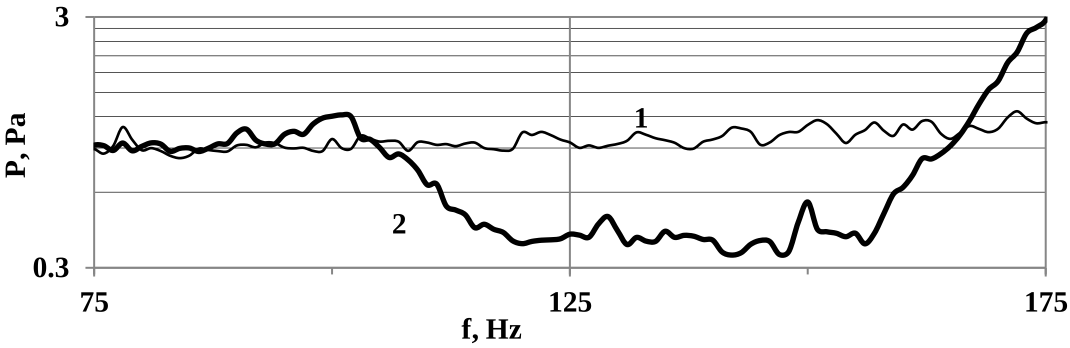


b)

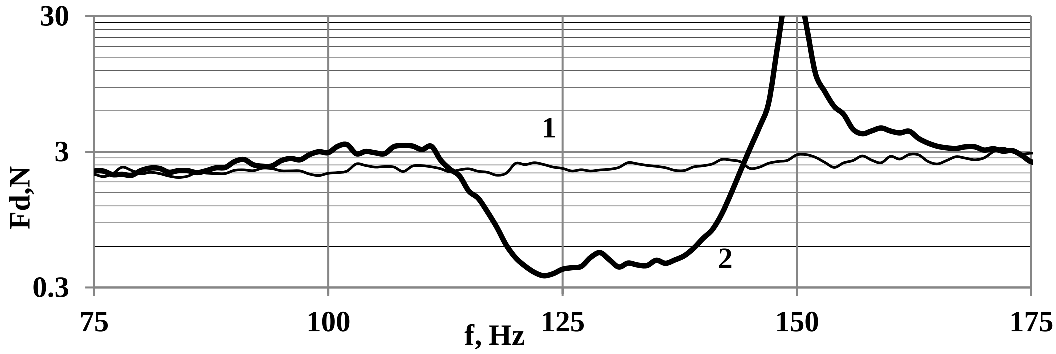


c)

Figure 6 Excitation of pulsations. Active damping results of pulsations P at the bottom of the pipe: a – by a sinusoidal signal with the frequency of 70 Hz; b - at a 310 Hz resonance by a 335 Hz resonator in the P channel with the π/2 phase; c - by an elliptical filter of the first order. 1-initial signal, 2-damping, 3-disturbance



a)



b)

Figure 7 Simultaneous excitation 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.

6. CONCLUSIONS AND RECOMMENDATIONS

1. Reduction of vibration transfer through the pipeline compensators with liquid by active methods has been practically unexplored today. This task is important for the vibration isolation of equipment in pipelines in power and transport engineering, shipbuilding, for oil and gas pipelines.
2. The appearance of pressure pulsations inside the compensator at its vibration deformation causes an increase in its vibration rigidity in relation to the static one in tens and hundreds times in a wide frequency range with increasing frequency. Accordingly, the transfer of vibration through the compensator also increases.
3. A system of dynamic forces and pressure pulsations active damping in the pipeline compensators with liquid has been created and researched. Controllers in its feedback loop in the form of standard digital bandpass filters of Butterworth, Chebyshev, elliptic types and resonance links allowed to reduce simultaneously dynamic forces and pressure pulsations by an order of magnitude or more for separate discrete frequencies and over wide frequency ranges from tens to hundreds Hz.
4. The increase in the efficiency of AVS is limited by the appearance of self-oscillations with increasing gain coefficients in the feedback loop. One of the ways to increase efficiency is to introduce damping units into the controller.

5. Further research of combined active damping of pressure pulsations and dynamic forces in pipeline compensators should be aimed at improving control algorithms and developing methods for calculating pulsations and dynamic forces transferred by compensators with liquid as part of the pipeline.

ACKNOWLEDGEMENTS

The paper was written on account of the Russian Science Foundation (the RSF) Grant. Agreement No. 16-19-10292 dated May 12, 2016 between the Russian Science Foundation and Kaluga State University named after K.E. Tsiolkovsky.

REFERENCES

[1] Kiryukhin, A. V., Milman, O. O., Ptakhin, A. V. Reduction of vibration transmission from power plants by active methods. Thermal energy, 12, 2017, pp. 58-66.

[2] Ganiev, R. F. Nonlinear resonances and catastrophes. Reliability, safety, noiselessness. Moscow: Dynamika, 2013, pp. 505-542.

[3] Popkov, V. I. and Popkov, S. V. Oscillations of mechanisms and structures. St. Petersburg: "Sudarinja", 2009.

[4] Kiryukhin, A. V., Milman, O. O., Ptakhin, A. V., Serezhkin, L. N. and Kondratev, A. V. Development and Calculation-Experimental Analysis of Pressure Pulsations and Dynamic Forces Occurrence Models in the Expansion Joints of Pipelines with Fluid. International Journal of Applied Engineering Research 12(19), 2017, pp. 8209-8216.

[5] Kuznetsov, R. N., Kiryukhin, A.V., Fedorov, V. A. Belogubtsev, E. S, Mikhailov, C. G., Pudokin A. A. and Smagin, D. A. Problems and preliminary results of testing systems for active damping of low-frequency signals in water and air environments. Fundamental and Applied Hydrophysics, 4(3), 2011, pp. 93-107.

[6] Kiryukhin, A. V, Tikhonov, V. A., Chistyakov, A. G. and Yablonsky, V. V. Active vibration protection - purpose, principles, state. 5. Active vibro - and noise insulation of pipelines. Patent research. Problems of mechanical engineering and automation, 3, 2013, pp.125-131.

[7] Khot, S. M., Yelve, N. P., Tomar, R., Desai, S. and Vittal, S. Active vibration control of cantilever beam by using PID based output feedback controller. Journal of Vibration and Control, 18(3), 2012, pp. 366-372.

[8] Omidi, E. and Mahmoodi, N. Vibration suppression of distributed parameter flexible structures by Integral Consensus Control. Journal of Sound and Vibration, 364, 2016, pp. 1-13.