Comparative analysis of the possibility to reduce vibration transmission through a pipeline compensator by suppressing dynamic forces and pressure pulsations by active methods

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Abstract. The results of experimental studies of joint two-channel active suppression of pressure surges and vibrations of the working medium to reduce vibration transfer through pipeline compensators of power plants with liquid are considered. No data is available in academic periodicals on the use of active methods to solve this problem. The test bench design with excitation and active damping systems of vibrations and pressure surges is given. Efficiency of more than 15 dB from the use of broadband active systems for damping vibration forces transmitted through the compensator and pressure surges of the working medium inside it, both working separately and together in the frequency range from 10 to 275 Hz has been experimentally obtained. In the narrower frequency ranges from 10 to 350 Hz, active damping up to 25 dB has been obtained.

1. Formulation of the research problem
Transmission of vibrations through pipelines from power plants may exceed transmission over its supporting structures. It is important for oil and gas pipelines located at pumping stations, for power and transport engineering, shipbuilding. Reducing the vibration of the installation itself is limited by the technological level, nature of operating procedures, weight, size characteristics and cost. To reduce the vibration transfer using vibration-isolating compensators, it is necessary to reduce the structural stiffness of the compensators and reduce the forces created by pressure surges of the working medium inside the compensator in a wide frequency range. The transfer of vibrations through compensators of pipelines with a fluid is characterized by its increase by 100 to 1000 times compared with a compensator without fluid in a wide frequency range. The compensator sets up pressure surges and dynamic forces generated by surges in the course of its vibration deformation. Therefore, a decrease in the structural stiffness of the compensator often does not lead to a decrease in the vibration transfer through it at these frequencies [1, 2]. To reduce of vibration transfer through the compensator, it is possible to use broadband active damping systems of both dynamic forces in its structure and pressure surges of the working medium inside the compensator. In the available scientific publications, no papers were found to solve this problem, although there are the works on active damping of vibrations and pressure pulsations in pipelines [1]. According to the Agreement No. 16-19-10292 with the
Russian Science Foundation, relevant experimental and computational studies were conducted, the results of which have been presented in this paper.

2. Test bench

Reducing the vibration transfer through compensators can be obtained by using active vibration protection systems (AVS) [1, 2], creating effects in antiphase to the original from the operation of the unit. In our case, these are the vibration forces acting on the output flange of the compensator and the active system for damping pressure pulsations. A diagram of such an AVS and a test bench is shown in figure 1.

Plate 1 is installed on vibration force sensors 2 on foundation 3. Compensator 5 with a tube 4 filled with water is attached to plate 1. Pressure pulsations P in the tube and compensator are caused by the piston 6 and measured by hydrophones 7. Piston 6 is actuated by an electro-dynamic vibrator Vp with power amplifier PAp. The pulsations P create the dynamic force Fp, which causes the plate 1 vibration. The vibrator Vv with the power amplifier PAv causes the structural actuation of the vibration. It creates a vibration force Fv affecting plate 1 via compensator.

Figure 1. Systems of excitation and active suppression of vibration forces and pressure surges on the AVS bench. CD- control device, PA- power amplifier, V- vibrator, SG- signal generator, SA-signal analyzer. Indices: c- compensation, p- pulsations, v-vibration.

The required signals to the power amplifiers of the excitation channels of vibration and pulsation are supplied from the signal generator SG. The force Q acting on the foundation 3 is measured by sensors 2. The vibrator Vc with the power amplifier PAc creates a compensating force Fc. It reduces
the force $Q$ measured by sensor 2. The signal on PAc and Vc is generated by the CD control system based on the signal processing from force sensors 2. All vibrators are vibration-proof from plate 1 and base 3. Piezoceramic oscillator 8 with power amplifier Paps creates compensating pressure surges. The signal to the oscillator 8 is formed by the CD control system based on the signal processing from the hydrophone 7.

The regulator in the feedback loop was created based on combinations of Butterworth, Chebyshev bandpass filters, elliptical, as well as resonant links (resonators). The parameters of the control device and filters were set by the computer in the MATLAB environment via the RS-485 interface. The control and signal processing were carried out by means of Bruel & Kjaer SA-type 3560-B-140 multichannel analyzer. The filter characteristics — order, bandwidth, and gain in the feedback — were selected to maximize damping without losing system stability. Using a combination of several filters of the first and second orders, effective regulators were built to suppress the dynamic forces and pressure surges in wide frequency range. To increase stability and efficiency, resonators with suppression at resonant frequencies were added to wideband filters, at which they were observed to grow intensively with increasing gain in the feedback loop as it approaches the stability boundary.

3. Results of vibration forces and pressure surges active suppression

As a result of the research, a broadband reduction of dynamic forces and pressure surges has been obtained by means of AVS, working both separately and together in the frequency range from 10 to 275 Hz by 15 dB or more as seen on figure 2. An active vibration protection system decreases both source signal 1 and noise 3.

Figure 2. a) – active suppression of the vibration force $Q$ downstream from a pipeline compensator $\phi100$ mm with fluid. b) damping efficiency $\Delta Q$, dB (ratio of force before and after damping). AVS with a set of elliptic filters $\Delta f=40$-90 Hz, filter order $n=1$, $\Delta f=70$-120 Hz, $n=2$, $\Delta f=100$-180 Hz, $n=2$, $\Delta f=160$-220 Hz, $n=2$, $\Delta f=200$-260 Hz, $n=2$ and resonator at frequency $f=145$ Hz, phase 0. 1 - source signal, 2 - damping, 3 - noise.

The efficiency of the pulsation $P$ active suppression in figure 3 (a) and the corresponding suppression efficiency $\Delta Q$ under the plate (b) at a resonance of 310 Hz with a 335 Hz resonator in
channel P with phase $\pi/2$ on the resonance of a fluid column 310 Hz amounted to 26 dB in both ripple and force determined by the pulsation. Simultaneous active suppression of second-order pulsation P by means of elliptic filters in the 110-160 Hz band a) and the force Q in the 120-140 Hz band with simultaneous excitation of the force and pulsation is shown in figure 4. Both damping channels operate efficiently, nearly no affecting each other. In all cases, a further AVS efficiency growth the was restrained by the self-oscillations with an increase in the gain factors in the feedback loop. One way to increase the efficiency was the insertion of suppression links to the regulator.

Figure 5 presents the results of simultaneous active suppression of the pulsation P by an elliptic filter of the second order $n = 2$ and damping of the vibrational force Q by the same filter in the same frequency range $f = 120-140$ Hz. The excitation of pulsations and force were caused by a piston with white noise in the range of 200 Hz. With the active damping of only the pulsation, P decreases by 3 times (figure 5a). Figure 5b shows an additional two-fold decrease in Q (efficiency $\Delta Q = 5$ dB) when damping is turned on in channel Q (in addition to the results of damping the force Q by damping the pulsations in channel P). Additional damping P when applying additional damping in channel Q is not observed (figure 5c). A decrease in force does not affect the pulsations, but is added to the effect of a decrease in force due to the suppression of pulsations.

![Figure 3](image-url)  
Figure 3. Reduction of the pulsation P (a) and the corresponding reduction of the power $\Delta Q$ (b) at 310 Hz resonance with a 335 Hz resonator in channel P with phase $\pi/2$. 1-source signal, 2-damping.
Figure 4. Simultaneous active damping of second-order pulsation $P$ by means of elliptic filters in the 110-160 Hz range a) and the force $Q$ in the 120-140 Hz range b). Excitement by pulsation and force. 1-source signal, 2-suppression.
Figure 5. Simultaneous active suppression of the force Q and pulsations P by an elliptical filter of the second order n = 2 in the frequency range f = 120-140 Hz. Excitation by pulsation of white noise in the 200 Hz range. a – pulsation damping only, b - additional reduction and efficiency of damping Q when damping Q is turned on in addition to damping in channel P, c - change in P when additional damping is added in channel Q. 1 - initial signal, 2 - damping, 3 – hindrance.

Conclusion
Studies and comparative analysis of the options for actively damping pressure pulsations and vibrational forces in compensators with liquid have shown the following. Active damping allows both their separate and simultaneous reduction of ten or more orders of magnitude in a sufficiently wide frequency ranges. Active damping of the vibrational forces transmitted to the foundation behind the compensator practically does not affect the value of pressure pulsations. At the same time, the active suppression of pressure surges reduces significantly the vibrational forces transmitted to the foundation behind the compensator, if these forces are determined by pressure surges. Therefore, when choosing an active damping method, information is needed, which reason is decisive in the occurrence of vibration of the pipeline and its foundation behind the compensator: structural dynamic forces, pressure pulsations or both together. In any case, active damping of forces on the foundation will be effective. Active damping of pressure pulsations can be used additionally if necessary.

Acknowledgments
This research was supported by RSF (project No 16-19-10292).

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Original Russian Text published in Pis'ma v Zhurnal Tekhnicheskoi Fiziki, 2018, Vol. 44, No. 24, pp.38–44. (http://www.iaeme.com/IJCIET/issues.asp?JType=IJCIET&VType=9&IType=10)