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METALS AND STRENGTH PROBLEMS

Reducing Vibration Transfer from Power Plants by Active Methods

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Abstract—The possibility of applying the methods of active damping of vibration and pressure pulsations for reducing their transfer from power plants into the environment, the seating, and the industrial premises are considered. The results of experimental works implemented by the authors on the active broadband damping of vibration and dynamic forces after shock-absorption up to 15 dB in the frequency band up to 150 Hz, of water pressure pulsations in the pipeline up to 20 dB in the frequency band up to 600 Hz, and of spatial low-frequency air noise indoors of a diesel generator at discrete frequency up to 20 dB are presented. It is shown that a reduction of vibration transfer through a vibration-isolating junction (expansion joints) of pipelines with liquid is the most complicated and has hardly been developed so far. This problem is essential for vibration isolation of power equipment from the seating and the environment through pipelines with water and steam in the power and transport engineering, shipbuilding, and in oil and gas pipelines in pumping stations. For improving efficiency, reducing the energy consumption, and decreasing the overall dimensions of equipment, it is advisable to combine the work of an active system with passive damping means, the use of which is not always sufficient. The executive component of the systems of active damping should be placed behind the vibration isolators (expansion joints). It is shown that the existence of working medium and connection of vibration with pressure pulsations in existing designs of pipeline expansion joints lead to growth of vibration stiffness of the expansion joint with the environment by two and more orders as compared with the static stiffness and makes difficulties for using the active methods. For active damping of vibration transfer through expansion joints of pipelines with a liquid, it is necessary to develop expansion joint structures with minimal connection of vibrations and pulsations and minimal vibration stiffness in the specified frequency range. The example of structure of such expansion joint and its test results are presented.

Keywords: power plant, vibration, pressure pulsation, dynamic forces, air noise, active vibroprotection, pipeline, expansion joint, active vibroprotection system

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Reducing vibration of power plants is important in two aspects. The first is ensuring a vibration strength, without which their operation is not possible. This problem has at present been solved in most cases, except for individual emergencies, such as rotor unbalance due to the blade destruction or initial curvature when rotor and stator are misaligned [1]. The second aspect is ensuring the specified vibration transfer into the seating and into the environment. Sometimes these problems are interconnected, for example, when the vibration causes soil creep under the supports that leads to misalignments of power plant components and can cause an accident. By misalignment is understood changing the rotor position with reference to the stator when they are mutual displaced, which can be caused by a variety of reasons: heat deflections of rotor and stator, displacements of rotor supports, for example, when the heat deformations of the seating supports or due to the soil creep under the supports, including owing to vibration. The misalignment is also a noncoincidence of rotation axes of the rotors of drive turbine and generator, turbine and pump.

The development of modern power engineering is characterized by increasing the demands for transfer of vibration and noise into the seating and environment by operating plants. Reducing the vibration transfer from equipment is possible by decreasing the vibration of the plant itself (in the vibration source) and by vibration-isolating means. Vibration reduction in the source is limited by the technology level, characteristics of operating processes in the plant and its overall dimensions and cost. Thus, at present, without using the vibration-isolating means, it is often impossible to provide the demands for noise radiation into the environment. These demands are constantly becoming tougher in connection with environmental problems and technical specifications of operation of the plants.

The scheme of transfer of vibration and noise from the power plant is presented in Fig. 1. Vibration is emitted along the support connections of the plant—

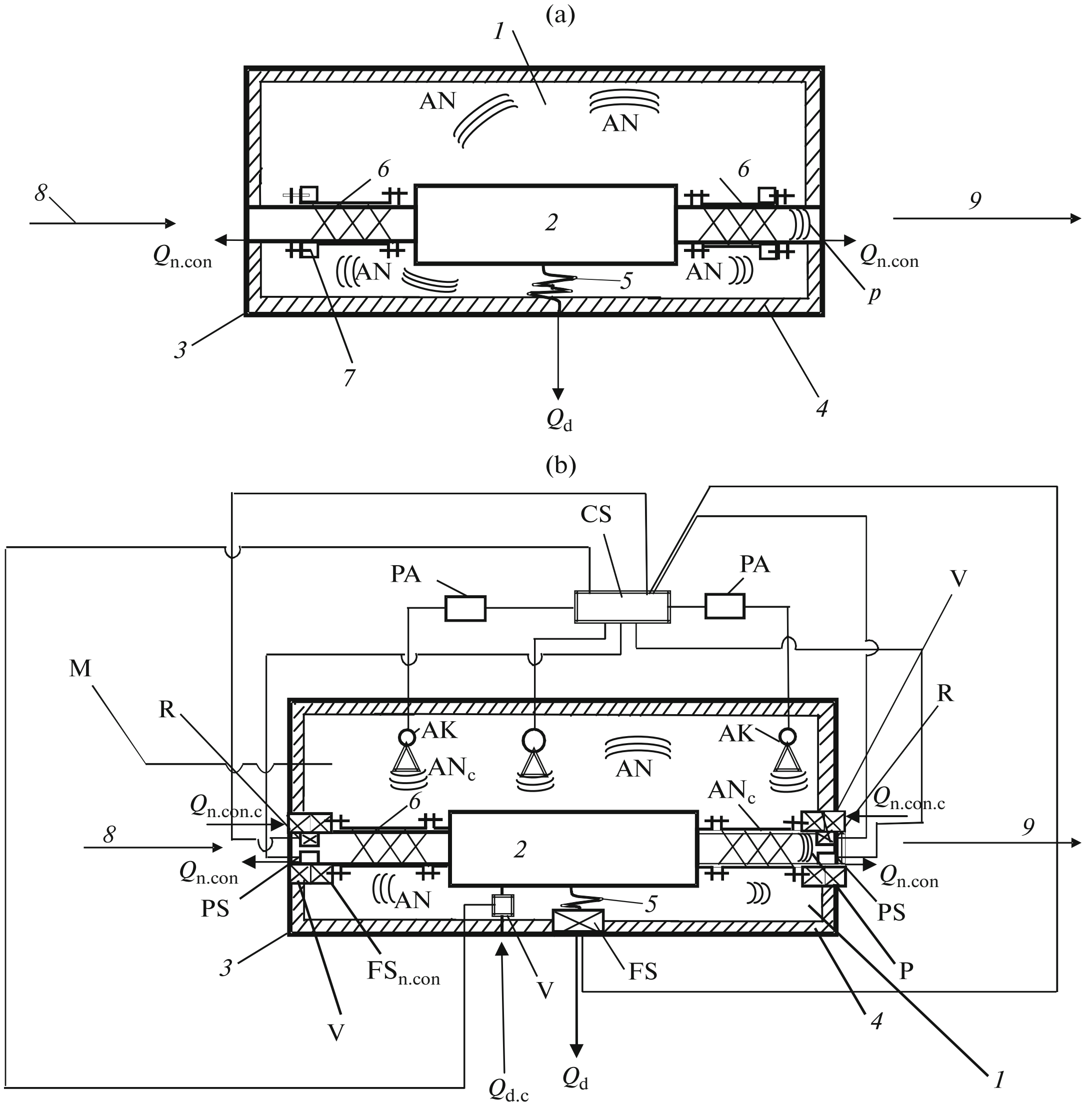

Fig. 1. Vibroprotection of a power plant. (a) Scheme of transfer of vibration and noise from a power plant and of the means of its passive vibration isolation; (b) layout of elements of full-size active system of vibroprotection of the plant; (*1*) plant premises; (*2*) power plant; (*3*) seating; (*4*) sound insulation and sound absorption; (*5*) support vibration isolator; (*6*) vibration-isolating insert (expansion joint); (*7*) pressure pulsation damper; (*8*) input of working medium into environment; (*9*) output of working medium from the environment; Q_d are dynamic forces from the support shock absorption; $Q_{n.con}$ are forces from nonsupport connections; $Q_{d.c.}$ are compensating dynamic forces; $Q_{n.con.c}$ are compensating forces of nonsupport connections.

vibration isolators (or shock dampers) bearing the weight load of this plant. Furthermore, the vibration of seating is caused by dynamic forces Q_d, which are transferred through support vibration isolation. It should be noted that, at present, the support vibration isolation could be used for sufficiently powerful power plants [2].

The vibration transfer also occurs along the nonsupport connections (pipelines with working medium), by the walls in the form of vibrations with dynamic forces $Q_{n.con}$, and by pipeline working medium in the form of pressure pulsations *p* or hydrodynamic noise (HDN). In addition, the vibration energy from the plant is propagated by means of air noise (AN). The passive vibration isolation means (vibroprotection) include the support vibration isolation (shock-absorbing) and the means of vibration isolation by nonsupport connections: expansion joints of pipelines (vibration-isolating inserts) and pressure pulsation dampers of the working medium, vibration-isolating couplings of the shaft lines, sound insulation, and sound absorption.

The possibilities for reducing the vibration transfer from the plants into the environment by traditional passive methods is largely exhausted. In the last few decades, the active techniques of noise and vibration damping in the source and the ways of their propagation have been quickly developed.

It is accepted to call the methods of artificially creating appropriate vibration influences acting in the antiphase with initial vibration influence (propagating from the plant) by the active vibration damping methods (pressure pulsations, air noise, dynamic forces). The system of active damping of vibration signals propagating from the plant creates antiphased signals (with the phase turned by 180° relative to the signal to be suppressed) on each of the frequencies in the desired frequency range. It should be noted that the active vibration suppression—in addition to power engineering (transport, shipbuilding, gas and oil engineering)—could be applied in other fields of technology too. This could be a chip grown for modern electronics (stabilization of equipment), ensuring the silence zone of the operator of any equipment (aircraft pilot, personnel, transport driver), reducing the noise transfer from wheels at railway transport to passengers and driver, etc., i.e., anywhere where it is necessary to reduce the vibration levels. Figure 1b shows a schematic diagram of the active vibration damping by all the above-listed ways of its transfer from a power plant (source).

The executive elements of such systems can be vibrators *V* of different types, radiators *R* of pressure pulsations of the working medium of pipelines' environment and air noise (speakers AK). The initial vibration magnitudes are measured with the relevant receivers (microphones M, sensors of force FS, sensors of dynamic force of nonsupport connections $FS_{n.con}$, sensors of pressure pulsations PS) and are processed by the control system. The generated signals through power amplifiers are fed to the executive elements and, combining with the initial signals in antiphase, reduce the vibration transfer over the appropriate channel. The compensating forces generated by vibrators reduce the forces transferred through the support vibration isolation and over the structure of nonsupport connections. The compensating pressure pulsations reduce the initial pressure pulsations in pipelines p. The compensating sound AN_c emitted by speakers reduces the initial level of air noise. From the energy point of view, the dissipation of vibration energy in the executive elements of an active system with heat release occurs. In contrast with the passive systems where damping noticeably influences only at resonances, the active damping turns out to be effective in the wide frequency band too.

Complete suppression of the vibration is not succeeded by the terms of system stability. The system of active vibration damping is a control system by the deviation from the specific parameter (vibration, dynamic force, pressure pulsations). Furthermore, the stability of such a system is determined by several parameters: the lowest frequency, the number of control channels and connections between them, damping, and the type of signal to be suppressed. In particular, if the system tries to reduce the adjustable parameter from a random component to zero, it will inevitably lose stability, because the reference control signal will be absent. Various algorithms of active damping are described in [3–8], but full vibration suppressing was not achieved.

One can believe that reducing the vibration amplitude or pressure pulsations by 2–3 times already is a good result. The suppression of vibration in a narrow frequency band (or at discrete frequency) turns out to be more efficient than in a wide frequency band. The active vibration damping (pressure pulsations, air noise) in the source is possible when the executive elements directly influence the vibration source. However, this way is more power-intensive and the control system is complicated, since the control channels turn out to be connected through an object (the plant). During active vibration damping behind the vibration isolation elements, the active system does not influence the vibration in the source, which can be used as a reference signal for each of channels, which increases system stability.

A detailed analysis of the problem state by the active vibroprotection methods made in [3–7] shows that the methods of active damping of vibration, transferred over the structure of vibroprotection elements, are sufficiently developed at present. The works on active damping of the medium pulsations in the pipelines and active damping of the air noise in air pipes are conducted. We should note the more widespread distribution of active methods abroad and the lack of their introduction in domestic developments and products, although such research is conducted in Russia [8].

For reducing the power-intensity and overall dimensions of active vibroprotection systems, their joint application with the corresponding passive means of vibration and noise damping is appropriate. For example, the active damping of the dynamic (vibration) forces affecting the plant seating should be carried out behind the elements of its vibration-isolating fastening (behind the shock absorption). These forces at the frequencies lying above natural frequencies of shock absorption are significantly (by multiple times and tens of times) less than vibration forces in the source owing to the effect of vibration isolation widely described in the literature. The effect of vibration isolation is in a considerable attenuation of vibration (dynamic) forces transferring into the seating through vibration isolation (sometimes it is called the support shock absorption) of equipment at the frequencies lying above the nature (resonant) vibration frequencies of this equipment on the shock absorption. The higher the disturbance frequency, the greater the effect of vibration isolation. The physical processes of attenuation of vibration transfer using the vibration isolation are considered in [9] in detail.

In addition, during impact on the seating, the active system slightly influences on the initial vibration of an object, which makes it possible to maximally uncouple the control channels and increase the stability and efficiency of the system.

It is appropriate to carry out active damping of pulsations in the pipeline together with using the high-performance passive pressure pulsation dampers. Figure 2 shows the test section of a broadband passive pressure pulsation damper for the pipeline based on the special pneumatic accumulators with rubber membrane and perforation and the dependence of pulsation level L on frequency f. The effectiveness of such damper reaches 30 dB in the frequency band up to 800 Hz in the pipeline with the diameter of 750 mm. The passive pressure pulsation dampers of different types are described in [10]. It is advisable to apply active damping of air noises together with the means of passive sound insulation and sound absorption.

Figure 3 shows the results of experienced works that we performed in order to reduce the air noise of a diesel generator with 65 kW capacity installed indoors, using the sound insulation of a new type. The effectiveness of the new low-frequency sound insulating enclosure of "heavy rubber" types is 10–20 dB in the wide frequency band beginning from 20 Hz.

Figure 4 presents the results of active damping of air noise of the same diesel generator. In addition to the sound insulating enclosure, the effectiveness of active damping of air noise in the lab at the discrete frequency of 32.5 Hz defined by the diesel generator operation amounts more than 20 dB.

Figure 5a shows the appearance and the scheme of the bench with an active system of vibration suppression (dynamic forces) transferred through support vibration isolation into the seating as well as the diagrams showing the system effectiveness. Influence Q_v is transferred into the mechanism simulator by vibrators V, on which the broadband random signal of sound signal generator is fed through power amplifiers PA. The measurement of dynamic forces Q_a behind the shock absorbers is performed with the force sensors FS_c of the control system. The signal from the sensors through preliminary amplifiers *PrA* is fed into the control system CS and then through the power amplifiers into the active vibrators V_c of the control systems, which generate a compensating influence Q_c on the seating. The control of total residual dynamic influence Q_Σ on the seating is carried out with the force sensors SF_Σ. The broadband spatial active damping makes it possible to reduce the dynamic forces behind the shock absorbers by 15–20 dB in the frequency band from 15 Hz to 150–200 Hz.

Figure 6 shows the results of active pulsation damping in the pipeline with passive pulsation dampers and expansion joints with thin-layer rubber-metal elements (TLRME). In a broad frequency band from 70–100 Hz to 500–600 Hz, the active pressure pulsa-

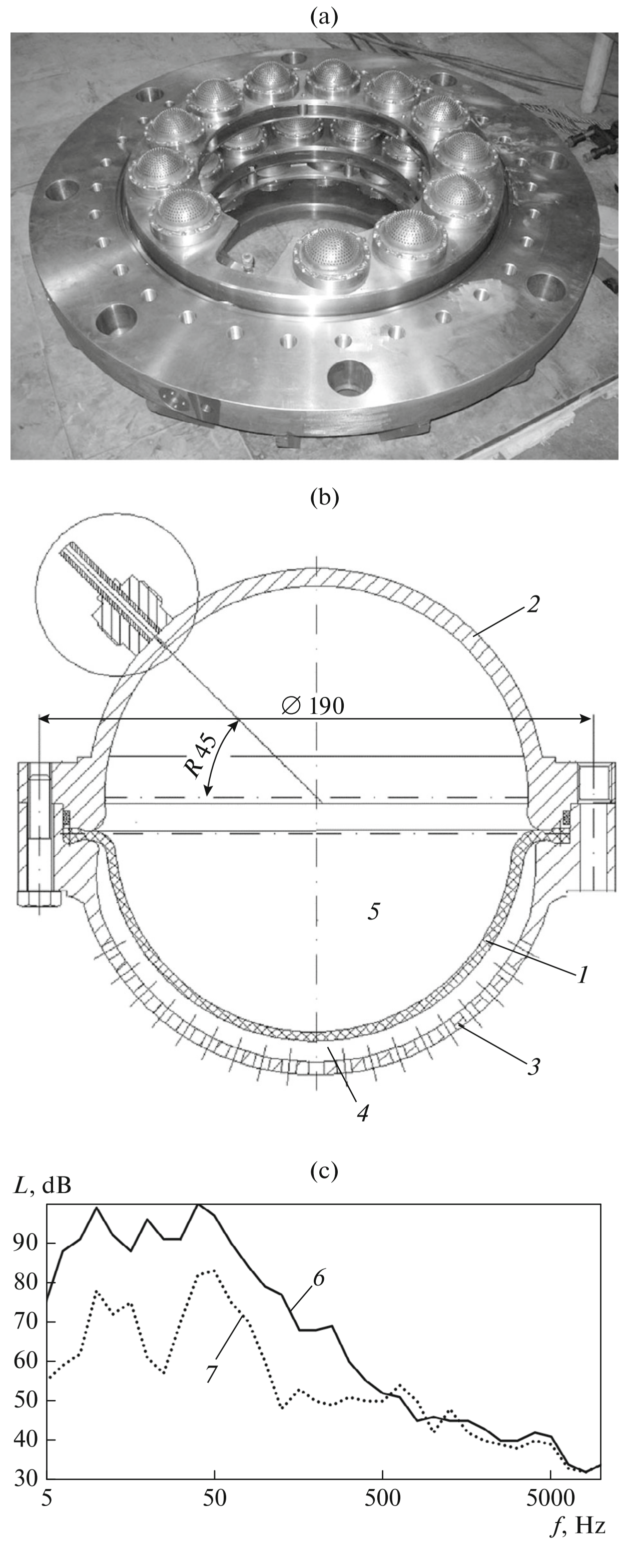

Fig. 2. The influence of the broadband pressure pulsation damper. (a) Damper at assembling; (b) elastic element of damper; (c) dependence of pulsation pressure level in the pipeline with the diameter of 750 mm when flow speed is 1.5 m/s; (*1*) rubber membrane; (*2*) casing; (*3*) holes; (*4*) water; (*5*) air; (*6*) damper is off (without air); (*7*) damper is on.

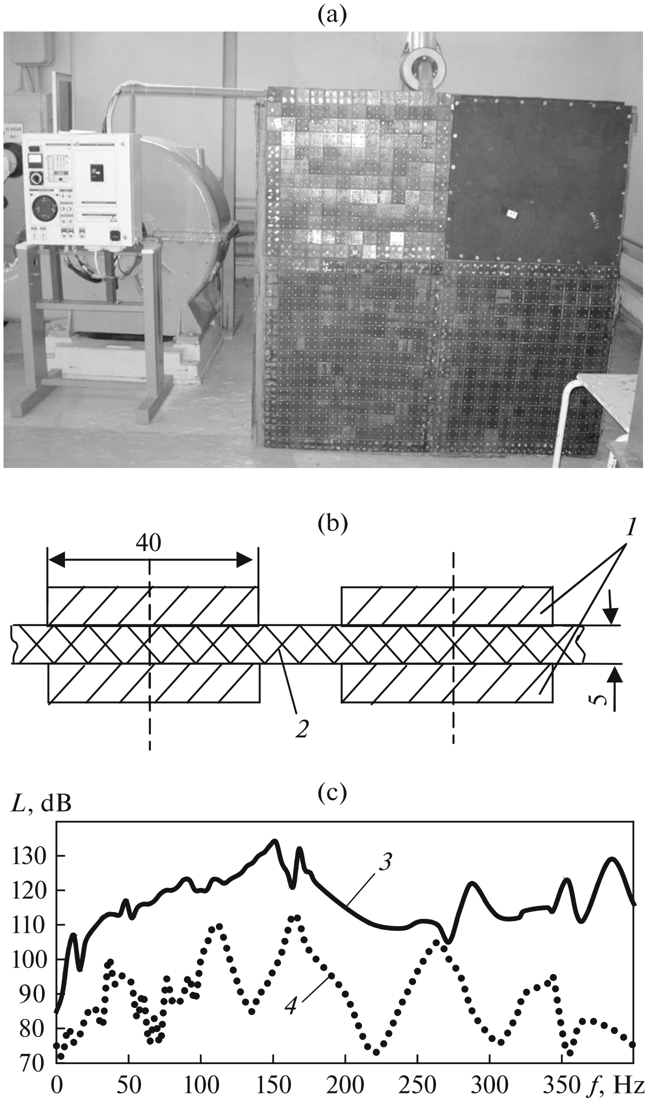

Fig. 3. Reducing air noise indoors by the new low-frequency sound-insulating enclosure of "heavy rubber" types. (a) Sound-insulating enclosure on the diesel-generator; (b) enclosure cross section; (c) effectiveness of sound-insulating enclosure; (*1*) metal plate; (*2*) rubber sheet; (*3*) air noise level inside sound-insulation: (*4*) air noise level outside sound-insulation.

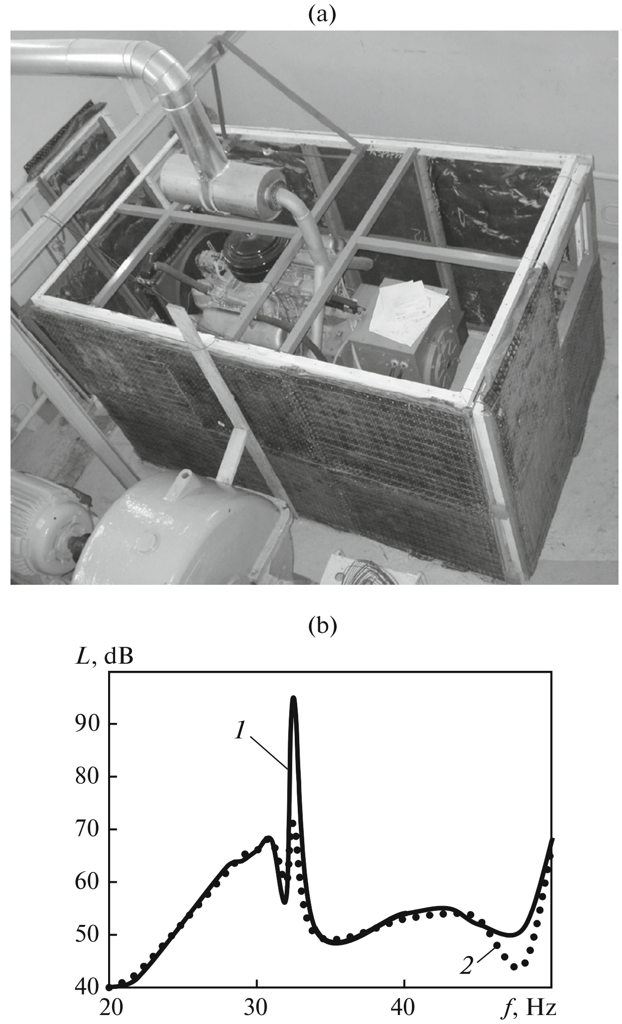

Fig. 4. System of active damping of air noise of diesel-generator indoors. (a) Diesel-generator inside sound-isolating casing (top is removed); (b) effectiveness of air noise suppression; (*1*) active system is turned off; (*2*) active system is turned on.

tion damping amounts to 20 dB or more [8] when the speed of working medium is up to 1.5 m/s.

In the described experiments on active damping, the available laboratory equipment of domestic and foreign manufacture was used: the vibrators of 20JE20D type and the power amplifiers of A438S type produced by Prodera Company (France), the three-component force sensors produced by Kistler Company (Switzerland), the accelerometers and the charge amplifiers of different types of Bruel & Kjaer Company (Denmark) for damping of vibration and dynamic forces. For pressure pulsation damping, the piezoelectric exciters of pressure pulsations of our own production based on the piezoelectric ceramics of Elpa Research Institute (Zelenograd, Moscow, Russia) with the power amplifiers of 2713 type produced by Bruel & Kjaer were used. For active damping of air noise, the speakers with the power amplifiers of domestic production and manufactured by Philips Company were used. When choosing the components of the active system, it is necessary to take into account the frequency range, the maximum amplitudes of dynamic influences, the characteristics of linearity, power consumption and overall dimensions, ease of assembling on an object, and the characteristics of reliability.

There are practically no the works on active damping of dynamic forces, vibrations and pressure pulsations transferred through the expansion joints of pipelines [3–7]. The research, including this article, shows that the transfer of structural vibration and pulsations of working medium through pipelines and their expansion joints in some cases can be essential, for

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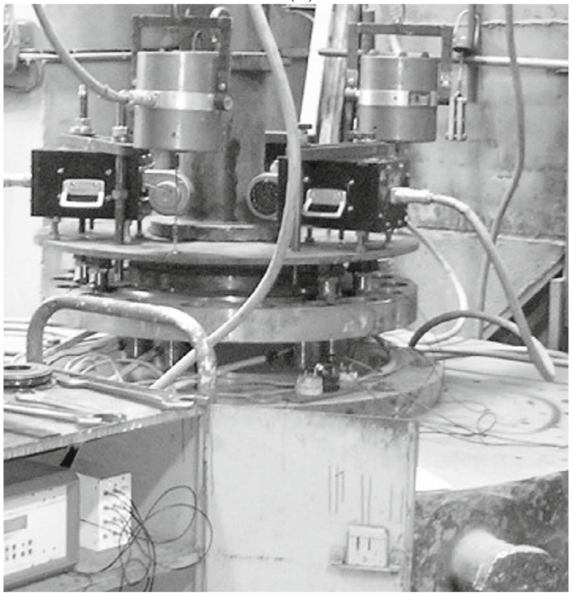

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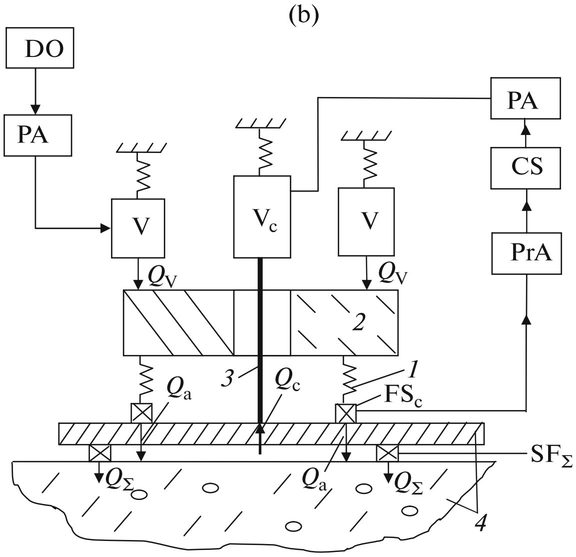

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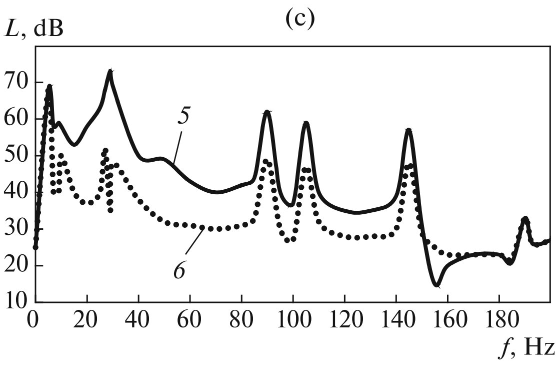
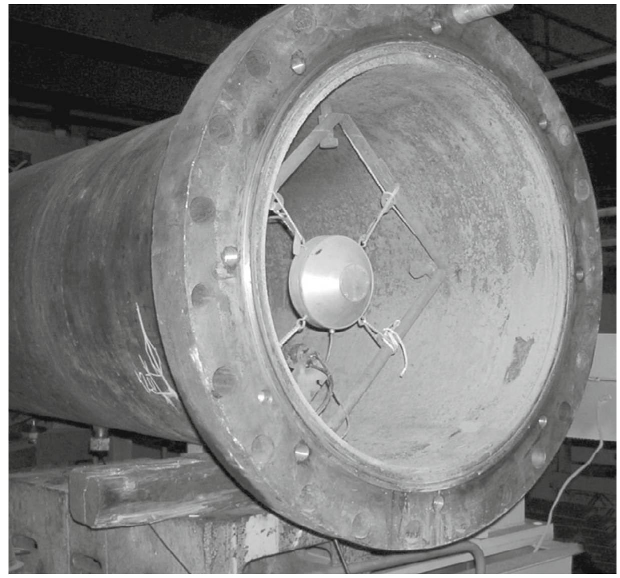


Fig. 5. Active damping of the forces behind the shock absorption. (a) Model of active vibration insulating system (AVIS) of suppression of dynamic forces on the test bench; (b) AVIS circuit; (c) AVIS effectiveness; (*1*) shock absorber; (*2*) simulator of mechanism; (*3*) rod of vibrator; (*4*) seating; (*5*) active system is turned off; (*6*) active system is turned on.

(a)



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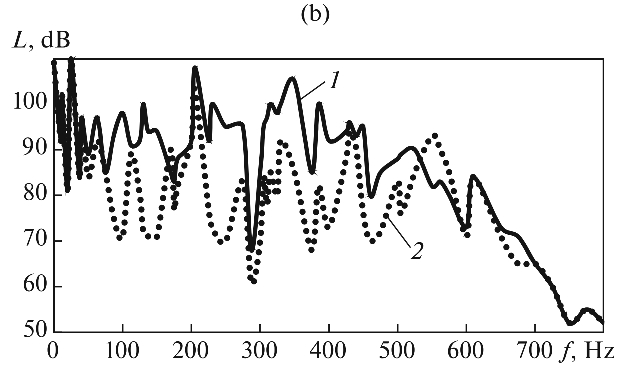

Fig. 6. Active damping of pressure pulsations in the pipeline. (a) Assembling of active radiators of active system of pressure pulsation damping on test bench pipeline; (b) effectiveness of active pulsation damping; (*1*) active system is turned off; (*2*) active system is turned on.

example during oil and gas transportation in the energy and transport engineering industry [10, 11]. It is shown in [11] that the dynamic forces transferred by pipelines can exceed the dynamic forces transferred through the support vibration isolation by two to three orders of magnitude. Furthermore, the expansion joint itself with its vibration deformation can be a source of pressure pulsations of the working medium.

Almost all tested expansion joints of pipelines with liquid have essentially frequency-dependent transfer function of vibration and pressure pulsations through them in a wide frequency range caused by the resonances of expansion joint structure and liquid and interaction of liquid and the structure with their vibrations. This complicates using the active methods. In accordance with the data of [9], the complex matrix describing the transfer of vibration and pulsations through the expansion joint, in general form, has the dimension of 13 × 13.

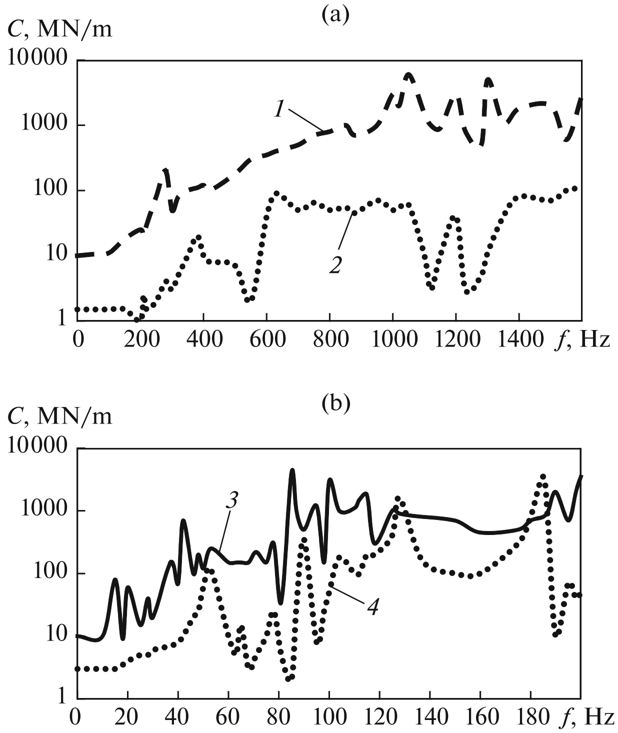

Fig. 7. (a) Dependence of vibration stiffness C of expansion joints D_a = 80 mm with (*1*) hose type and (*2*) expansion joint with TLRME with water pressure of 10 MPa and (b) expansion joints D_a = 750 mm with (*3*) RCC and (*4*) TLRME on the frequency.

The authors' studies showed that, for the most existing expansion joints with the rise of vibration frequency, there is a sharp increase in the vibration stiffness with respect to the static stiffness (at zero frequency). The vibration stiffness is defined as the ratio of dynamic force at the expansion joint output to the amplitude of vibration displacement on its input [9] at the specific disturbance frequency. It describes the vibration-isolating properties of the expansion joint: the smaller the vibration stiffness, the better the vibration isolation.

Increasing the vibration stiffness can amount to ten times or more in a wide frequency range. Figure 7 shows as an example the curves of vibration stiffness for high-pressure expansion joints of the hose type with thin-layer rubber-metal and rubber-cord casings (RCC).

The studies show that increasing the vibration stiffness of expansion joints is caused by emergence of pressure pulsations inside of the expansion joint during its vibration deformation and structural resonances of elastic elements of the expansion joint. New expansion joints, which we developed based on the TLRME, having a minimum connection of pressure pulsations and vibrations due to the special design, reduce the vibration transfer over the structure by an order of magnitude or more in a wide bandwidth as compared with the expansion joints with rubber-cord casings [11]. One of such expansion joints with TLRME on the test bench is shown in Fig. 8. Owing

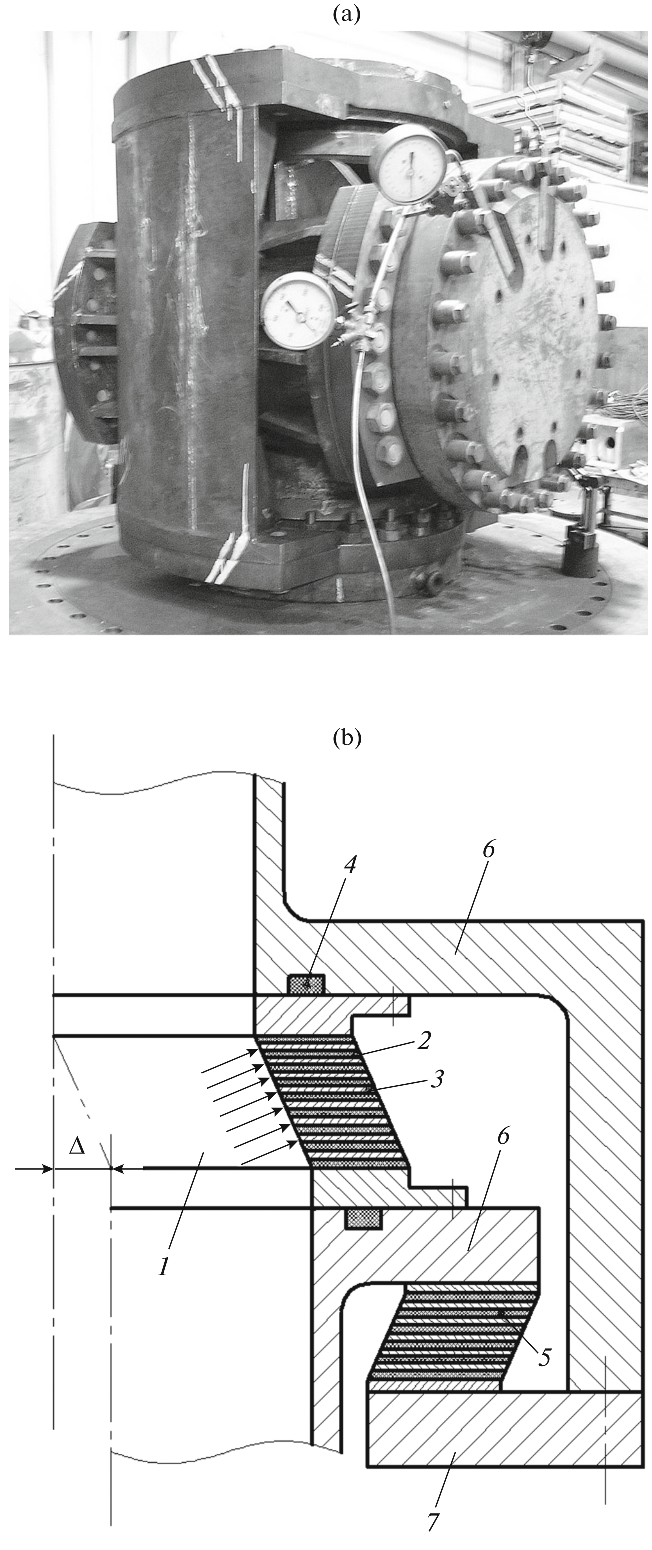

Fig. 8. (a) Expansion joint based on TLRME on the test bench for determining the transitional vibration stiffness and (b) scheme of deformation of TLRME consisting of a pipeline. (*1*) Ring TLRME; (*2*) rubber layer; (*3*) metal plates; (*4*) ring seal; (*5*) pressing plane TLRME; (*6*) flanges; (*7*) removable ring.

to the large anisotropy of elastic properties, the ring TLRME works only on shift, minimizing the pressure pulsations occurring in the pipeline, since its volume is not changed under shift. This will make it possible in the future to take the expansion joints with TLRME as the basis when developing the systems of active damping of vibration and hydrodynamic noise in the expansion joints of high-pressure, large diameter pipelines.

CONCLUSIONS

(1) The results of experimental works conducted by the authors showed the possibility of active broadband suppression of vibration and dynamic forces behind the shock-absorption up to 15 dB in the frequency band up to 150 Hz as well as the pressure pulsations of water in the pipeline up to 20 dB in the frequency band up to 600 Hz and spatial low-frequency air noise indoors from a diesel generator on discrete frequency up to 20 dB.

(2) A reduction of vibration transfer through vibration-isolating junctions (expansion joints) of pipelines with liquid, including the active methods, is the most complicated and was hardly developed thus far. At the same time, this problem is essential for the vibration isolation of power equipment from the seating and environment through pipelines with working medium in power and transport engineering, shipbuilding, and in oil and gas pipelines in pumping stations.

(3) For most existing expansion joints of pipelines, during increase in the vibration frequency, there is a sharp increase of vibration stiffness with respect to the static stiffness, which can amount to tens and hundreds times in the wide frequency range. Increasing the vibration stiffness of expansion joints with frequency increase is caused by the emergence of pressure pulsations inside of the expansion joint with its vibration deformation and structural resonances of elastic elements of the expansion joint.

(4) Using the new, specially designed by the authors expansion joints, which have a minimum connection of pressure pulsations and vibration owing to the special design, reduces the vibration transfer over their structure by an order of magnitude or more in a wide band of frequencies as compared with the existing designs. This makes it possible to take them as a basis when developing the systems of active damping of vibration and hydrodynamic noise in the expansion joints of high-pressure large diameter pipelines. It is necessary to continue the research on the study of mechanisms of pressure pulsation emergence in the expansion joints of pipelines of different types.

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REFERENCES

1. A. G. Kostyuk and O. A. Volokhovskaya, "Vibration activity evaluation of double-span rotor at rundown caused by its initial curvature and residual unbalance," Therm. Eng. **64**, 37–45 (2017). doi 10.1134/S0040601517010049
2. M. E. Skarednov, "Effective vibration protection of turbine unit foundations at power plants," Gazoturbinnye Tekhnol., No. 8, 32–36 (2016).
3. A. V. Kiryukhin, V. A. Tikhonov, A. G. Chistyakov, and V. V. Yablonskii, "Active vibration protection — Purpose, Principles, State. 1, Purpose and principles of the development," Probl. Mashinostr. Avtom., No. 2, 108–111 (2011).
4. A. V. Kiryukhin, V. A. Tikhonov, A. G. Chistyakov, and V. V. Yablonskii, "Active vibration protection — Purpose, Principles, State. 2. History of the development and the state," Probl. Mashinostr. Avtom., No. 3, 63–69 (2011).
5. A. V. Kiryukhin, V. A. Tikhonov, A. G. Chistyakov, and V. V. Yablonskii, "Active vibration protection — Purpose, Principles, State. 3. Active vibration insulation in cars," Probl. Mashinostr. Avtom., No. 2, 56–59 (2012).
6. A. V. Kiryukhin, V. A. Tikhonov, A. G. Chistyakov, and V. V. Yablonskii, "Active vibration protection — Purpose, Principles, State. 4. Active vibration and noise insulation of pipelines. Theoretical and experimental studies," Probl. Mashinostr. Avtom., No. 1, 72–80 (2013).
7. A. V. Kiryukhin, V. A. Tikhonov, A. G. Chistyakov, and V. V. Yablonskii, "Active vibration protection — Purpose, Principles, State. 5. Active vibration and noise insulation of pipelines. Patent studies," Probl. Mashinostr. Avtom., No. 3, 125–131 (2013).
8. G. N. Kuznetsov, A. V. Kiryukhin, V. A. Fedorov, E. S. Belogubtsev, S. G. Mikhailov, A. A. Pudovkin, and D. A. Smagin, "Problems and preliminary results of testing of active low-frequency signal damping systems in water and air media," Fundam. Prikl. Gidrofiz. **4**, 93–107 (2011) [in Russian].
9. V. I. Popkov and S. V. Popkov, *Oscillations of Mechanisms and Constructions* (Sudarynya, St. Petersburg, 2009) [in Russian].
10. R. F. Ganiev, *Non-Linear Resonances and Catastrophes. Reliabillity, Safety, Noiselessness* (Dynamika, Moscow, 2013) [in Russian].
11. A. Kiryukhin, O. Milman, and A. Ptakhin, "A search for the physical principles of improving the power unit pipeline expansion joint with fluid vibro-isolating properties," Int. J. Appl. Eng. Res. **11**, 11176–11183 (2016). https://www.ripublication.com/ijaer16/ijaerv11n23_12.pdf.

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