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Newly Designed Compensators to Reduce Vibration Transmission through the Pipes

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Abstract *- Reducing oscillation transmission through the pipe compensators is important for vibration transmission reduction through the pipes of numerical engines. The latter becomes especially relevant in the conditions of permafrost and the observed global warming when vibration transferred by pipelines can lead to soil subsidence and accidents. The design of vibration-insulating compensators based on thin-layer rubber-metal elements (TRME) is considered to improve vibration isolation of power equipment along pipelines with liquid for the frequencies from 10 to 1,600 Hz by order of magnitude or more in comparison with existing expansion joint designs. The model shows experimentally the possibility to reduce further vibration transfer by means of a series connection with a 90 ° turn of two expansion joints with TRME. This provides a reduction in the vibration transfer through the compensators by 20 - 40 dB or more at frequencies from 10 to 400 Hz due to a significant decrease in the torque (rotational) rigidity of two compensators with TRME in comparison with a single one.*

Keywords - *pipe, compensator, vibration transmission, frequency, vibration isolation.*

I. INTRODUCTION

The vibration transmission reduction from numerical equipment is very advisable today. It is limited by the technical possibilities and natural factors for the equipment itself. Vibration protection is the effective alternative way to reduce vibration transmission, including transmission through the pipes. In [1], [2] are presented the results of the study of effective vibration protection means for the working environment. For the vibrations transfer reducing through the pipe, including the operating fluid, various numerical compensators (expansion joints), pressure pulsation dampers, and active methods of vibration suppression are used [1], [2].

The vibration force *Fd(f),* transferred through the pipe, determines the vibration level of the foundation [1]. In practice, the foundation rigidity is usually higher than the pipe compensator rigidity. Then the amplitude of the dynamic (vibration) force *Fd(f),* transferred by the *compensator* to the foundation, calculated as

$$F_d(f) = A(f)\, C(f) \quad (1)$$

An (f) is the vibration amplitude of the compensator behind the installation, *C (f)* is the vibration rigidity of the compensator, Figs. 1, 2. Accordingly, *C (f)* is defined as the ratio of the vibration force *Fd (f)* transferred to the foundation at the fixed outlet of the expansion joint to vibration at its inlet at frequency *f* and is a complex quantity [3]. Methods for determining the frequency-dependent transient resistance and rigidity of vibration isolation elements are given in [3]. The author's studies [1], [2], [4], [5] had shown that interaction of the operating fluid pulsations and vibration of structure in compensator cause *C (f)* rapidly increase when frequency f grew. Static pressure-balanced compensators have been considered, in which spacer forces are absent.

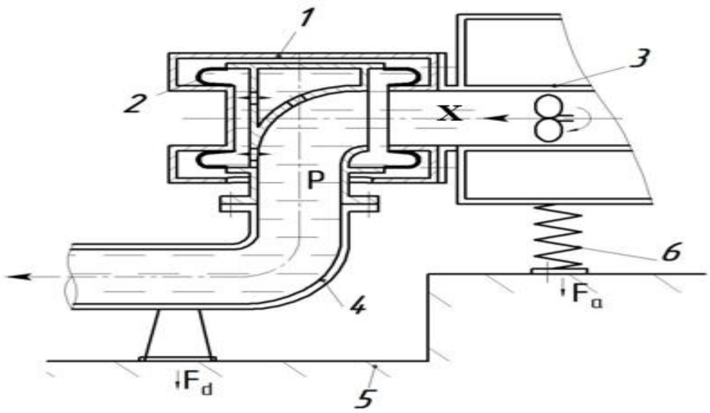


Fig. 1 Angular pressure-balanced compensator 1 with the elastic elements 2, 3- the pump, 4 – the pipe, 5 – foundation, 6 - amortization

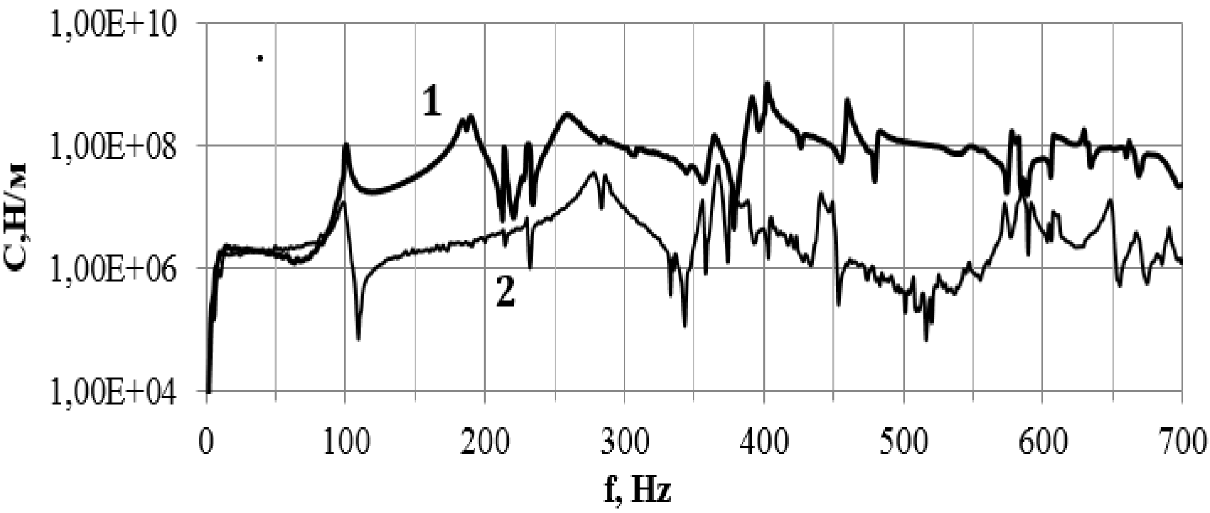


Fig. 2 Vibration rigidity *C (f)*. of an angular pressure-balanced compensator, shown in Fig. 1, 1 - with operating fluid (water), 2 - with air inside

The quality of the expansion joint vibration isolation is determined by the rigidity *C (f)*. Testing [1], [4]–[6] numerous pipeline compensators of various types and sizes [1], [2] with diaphragm rubber – cord casings (RCC) (Fig. 1 pos. 1) and compensators based on sleeve RCC and bellows as well as experimental designs of compensators showed a significant dependence of their transient vibration rigidity *C (f)* on the frequency *f* and the existence of fluid inside the compensator. For the expansion joint in Fig. 1, the *C (f)* value had grown by a hundred times or more with the increasing frequency compared to static rigidity (see Fig. 2). The expansion joint rigidity turns out to be significantly higher than the amortization 6 under the installation, Fig. 1, which has the order of 10^6 N/m in the entire frequency range.

Experiments [1], [2], [4], [5] and Fig. 2 also show that the fluid operating presence increases the compensator rigidity *C (f)*. Models of interaction between operating fluid (water)and structure in compensators have been researched by the authors in works [4], [5].

II. DESIGN AND CHARACTERISTICS OF EXPANSION JOINTS WITH TRME

A flat annular TRME with a diameter of 80 mm is shown in Fig. 4c. Flat alternating annular layers of rubber metal with a thickness of about 1 mm each are glued together and to the flanges. The width of the layers is more than ten times their thickness; therefore, the main rubber mass operates under conditions of volumetric compression, which ensures high axial rigidity (across the layers) of the TRME. In this case, the TRME shear rigidity is determined by the shear modulus of the rubber. The TRME deformation scheme in the transverse direction as part of the pipeline is shown in Fig. 3. The compensators of a new design based on thin-layer rubber-metal elements (TRME) are shown in Fig. 4b. Compared with a diaphragm-type compensator based on RCC in Fig. 4a, it has a significantly lower (10-100 times) transient vibration rigidity *C (f)* in a wide frequency range, Fig. 5. Similarly, experimental expansion joints DN 80 mm at a pressure of 10 MPa based on TRME have in the entire frequency range up to 1600 Hz 10-100 times lower rigidity than sleeve expansion joints DN 80 mm based on RCC, Fig. 6. A feature of the elastic TRME elements, due to which a significant decrease in the rigidity of expansion joints with TRME was obtained in a wide frequency range in comparison with standard designs, there is a strong anisotropy of the TRME rigidity characteristics. The rigidity across flat thin annular rubber-metal layers exceeds the shear rigidity along with the layers (across the TRME axis) by three orders of magnitude [1], [2]. Therefore, during deformation in the joint expansion composition, TRMEs work only in shear, as shown in Fig. 3, which minimizes water flow during the expansion joint vibration deformation between its internal cavities [4], [5], and reduces its rigidity in a wide frequency range. Calculation of the required vibration and strength characteristics of TRME is a complex engineering problem [6], [7].

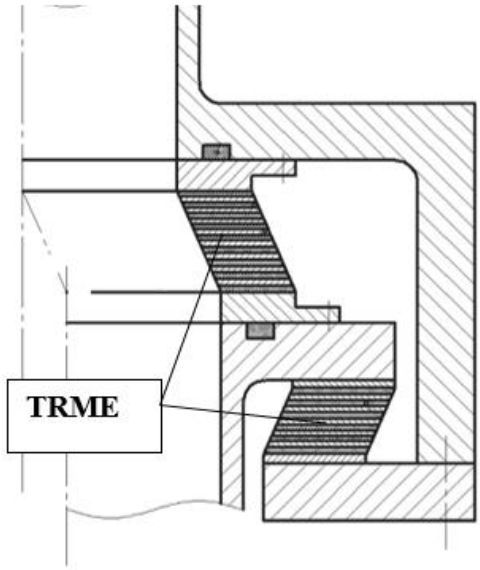


Fig. 3 TRME deformation scheme as a part of a pipeline

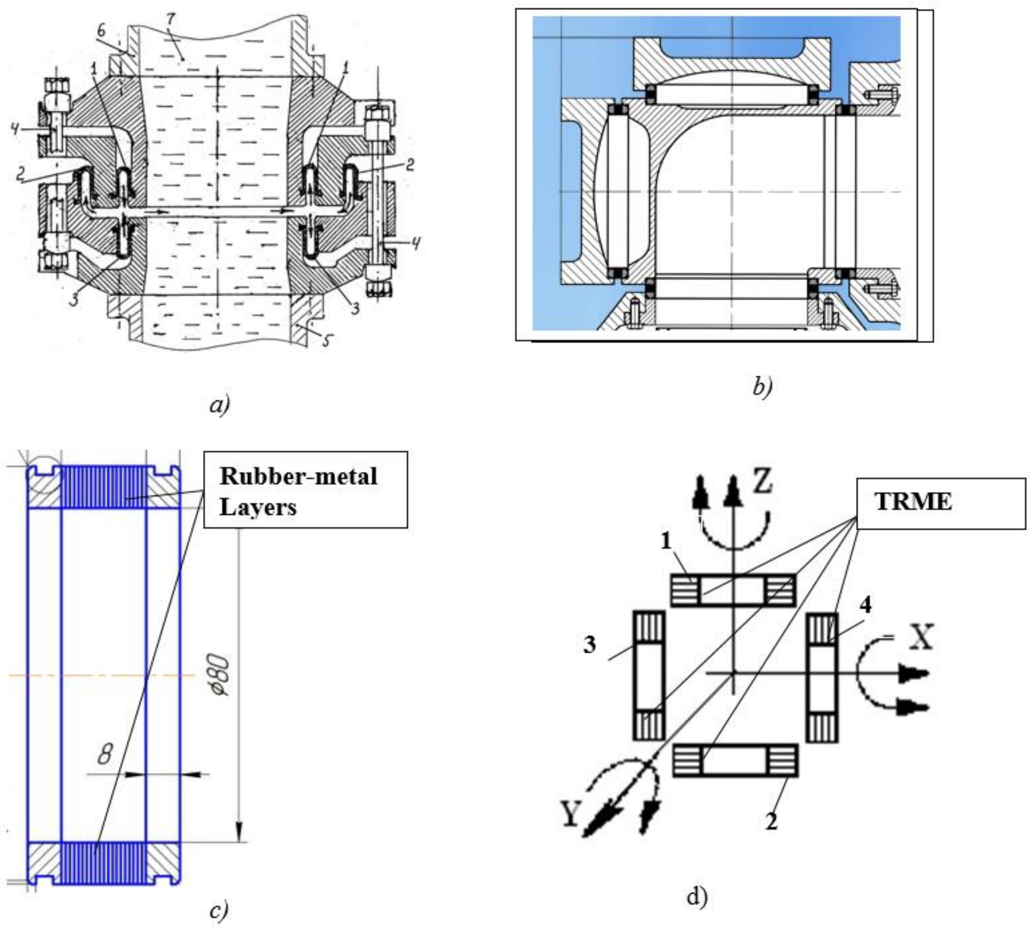


Fig. 4 pressure-balanced direct-flow expansion joint with RCC a) angular expansion joint with TRME b) TRME with an 80 mm diameter c), the layout of TRME in the expansion joint and the coordinate axis d). X, Y, Z - coordinate axes, arrows show rotations around the axes

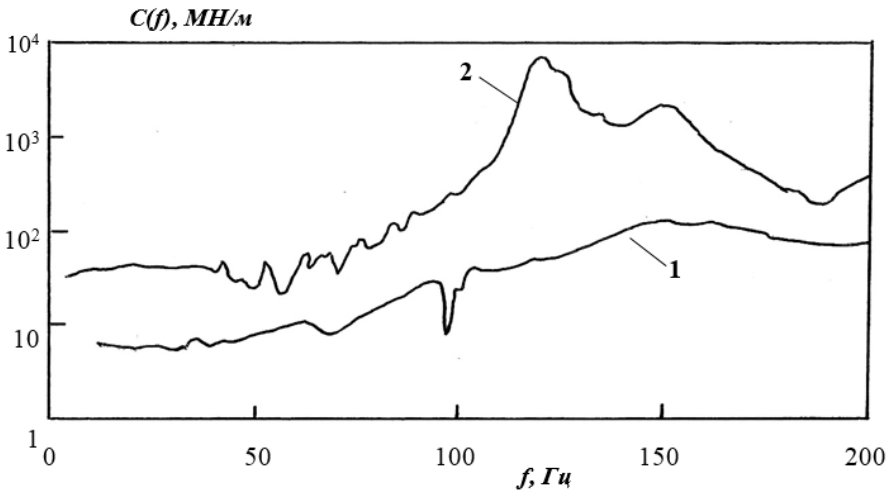


Fig. 5 Comparison of the experimentally measured transient vibration rigidities *C(f)* of the expansion joint Fig. 3b with TRME (1) and Fig. 3a with RCC (2) with water

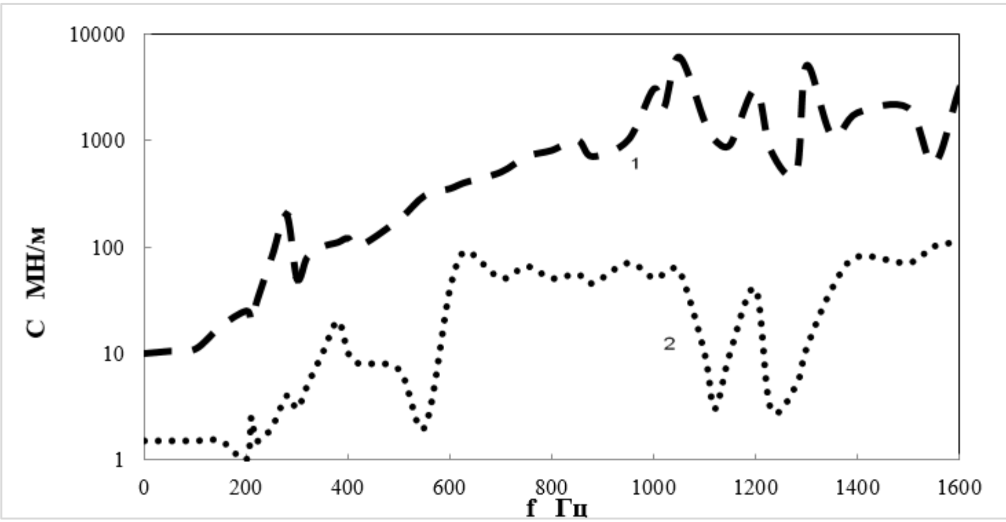
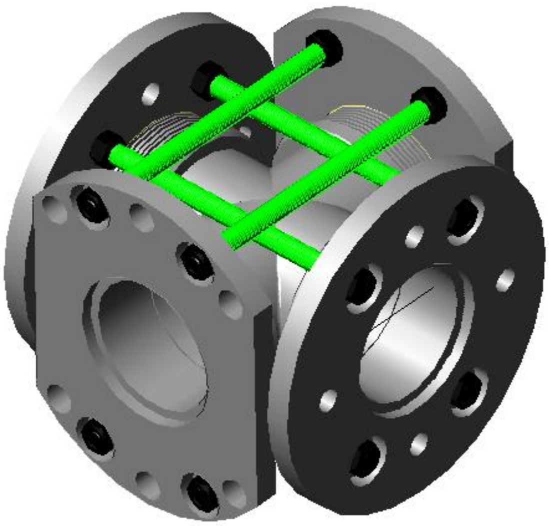


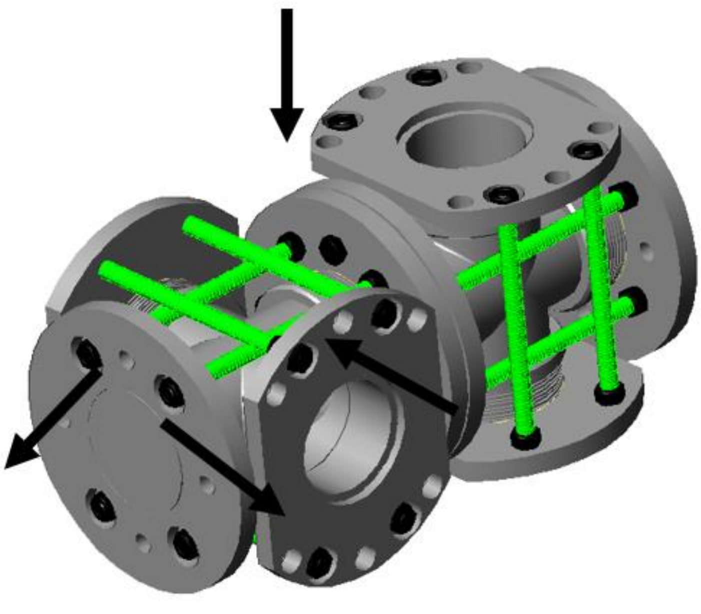
Fig. 6 Comparison of vibration rigidity: 1 - sleeve-type compensator DN 80 mm, 2 – compensator, based on TRME, operating fluid-pressure 10 MPa

III. POSSIBILITIES FOR ADDITIONAL REDUCTION IN THE EXPANSION JOINT RIGIDITY WITH TRME

When operating as part of a pipeline, the expansion joint must compensate not only linear (along the X, Y, Z axes, Fig. 4d) mutual movements of the flanges but also rotary (bending) mutual movements of the inlet and outlet pipelines, i.e., turns around these axes (shown by arrows in Fig. 4d). The corresponding rigidities will be designated as rotary or angular. In the Z direction (Figs. 4b and d), pairs 1 and 2 work in the shear; both pairs 1 and 2, 3 and 4 work in the shear in the Y direction; in the X direction, a pair of TRMEs 3 and 4 work in the shear. When turning around the X, Z axes, they work, respectively, pairs 1 and 2, 3 and 4. When turning around the Y-axis, the rotational rigidity *Cny(f)* is significantly greater since both pairs of TRMEs work across thin rubber layers. Therefore, bending vibrations of the pipeline in the plane of the X, Z axes can be effectively transferred by such an expansion joint in the form of a bending moment. Let us consider the possibility of a significant reduction in the rotary (angular) transient vibration rigidity *Cny(f)*. For this, it is proposed to use a double expansion joint with TRME. In this design, one of the flanges of a single expansion joint(Fig. 7a) is connected to another expansion joint of the same type but rotated 90 ° around the axis. Such a design, made up of two model expansion joints, is shown in Fig.7b.



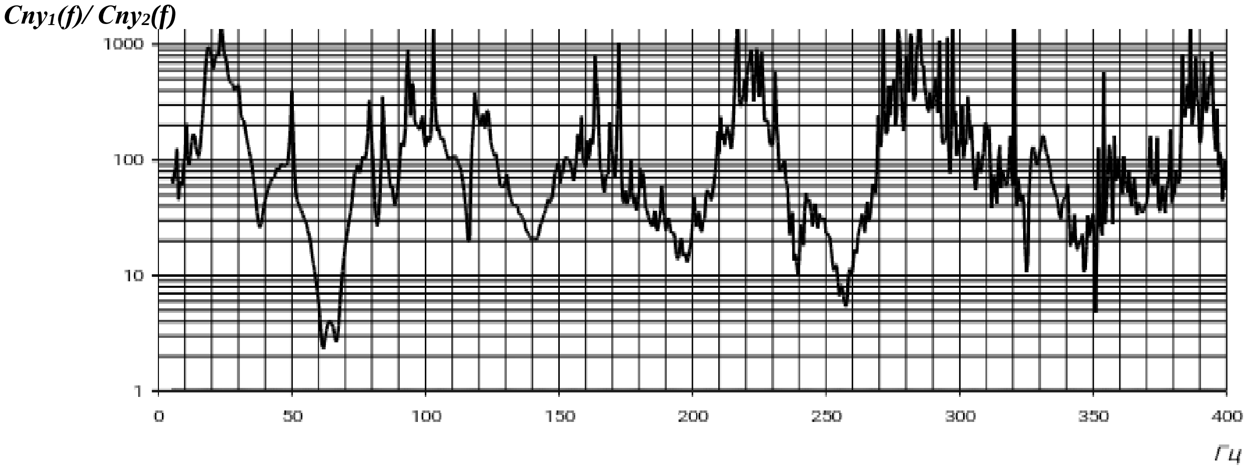
a)



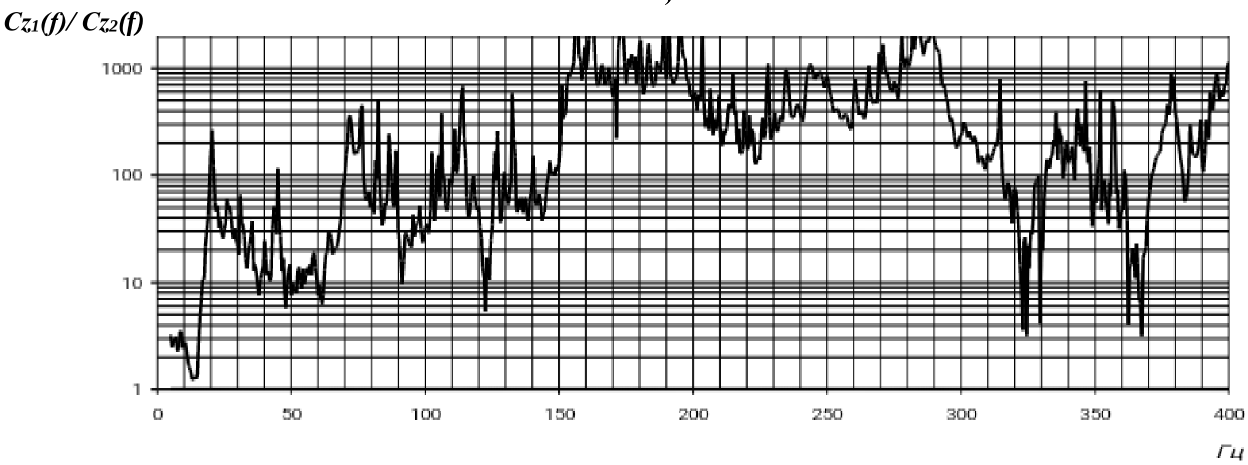
b)

Fig. 7 Single experimental a) and double expansion joints with TRME. The arrows show the movement of the working fluid

Fig. 8a shows a comparison of the torsional (with excitation of rotary vibrations and measurement of the output torque) dynamic rigidities of the single model expansion joint ***Cny1(f)*** 1 and the double expansion joint ***Cny2(f)*** 2. The ratio of the rigidities ***Cny1(f)/ Cny2(f)*** when turning is compared around the Y-axis with the original maximum rotational rigidity (see above). The maximum rotary rigidity decreased by two orders of magnitude in the frequency range 0-400 Hz. Fig. 8b shows the ratio of linear rigidities of single and paired expansion joints ***Cz1(f)/ Cz2(f)*** experimentally measured on experimental expansion joints with displacement along the Z-axis. The linear rigidity also decreased by two orders of magnitude or more.



a)



b)

Fig. 8 a) - comparison of the transient dynamic rigidity of a standard single 1 and double 2 expansion joint with TRME when turning around the Y-axis, *$Cny_1(f)/Cny_2(f)$*; b) - with displacement along the Z-axis, *$Cz_1(f)/Cz_2(f)$*

The reduction in torsional and linear transient rigidities of a double expansion joint in comparison with a single expansion joint is one to two orders of magnitude. For torsional rigidity around the Y-axis, which is the most unfavorable for a single expansion joint, the reduction is from 100 to 1000 times in the frequency range from 0 to 400 Hz. The increase in the hydraulic resistance of the double expansion joint can be reduced by 30 - 40% due to the installation of straightening grids in their flow path. This is confirmed by model tests.

IV. CONCLUSIONS

1. Calculated and experimental studies have shown that the vibration transfer through pipelines of power plants can significantly exceed the transfer through support structures. Existing expansion joint designs can significantly enhance the vibration transfer due to the dynamic interaction of the working fluid and the structure of the expansion joints during pipeline vibrations. It has been shown experimentally that expansion joints based on thin-layer rubber-metal elements (TRME) have a transient vibration rigidity ten to one hundred times less than serial expansion joints of various designs in a wide frequency range from 10 to 400 and 1600 Hz.

2. Serial connection of two expansion joints with TPME, rotated 90 ° relative to each other, allows reducing the vibration rigidity of a double expansion joint by one or two orders of magnitude in comparison with a single expansion joint due to a significant reduction in rotational (angular) rigidity.

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