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Reduction of Pipeline Natural Frequencies by Negative Stiffness Vibration Isolators

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Abstract. The paper is devoted to the problem of reduction of vibration level of pipelines, caused by resonance with the disturbing frequency of pumping units. The causes of large vibrations of main pump units piping and its influence to the level of the pump unit oscillations are analyzed. The source of increased vibration exposure in the "pump - pipe" system is often not only the pump unit, but pump piping also. Piping of the pumping unit in case of its operating parameters changing can be entered to resonance. It is offered to use disk springs with negative stiffness area for vibration isolation. Pipeline in this case acts as a "stabilizer" for a vibration isolator with negative stiffness. Analytical calculations and experiment have been made to prove the effectiveness of proposed solution.

1. Introduction

Technical progress in the development of pipeline systems places high demands to safety and reliability of pumping stations operation, as well as to the pipelines themselves. The practice of pipeline systems operating shows that their vibration can be caused both by the resonance of the pump piping natural frequencies with disturbing frequencies from the pump unit and by the pressure fluctuations occurring at the inlet and outlet of the pumps and then transmitted to the pipeline. All of the above can cause dynamic impacts to the pipeline, which lead to accidents with equipment [1, 2, 3].

The vibrational state of the "pump-pipeline" system must be considered in aggregate as a complex, interrelated and interrelated polyharmonic process. At the same time, reducing the vibration of the pump piping also reduces the vibration of the pump unit, especially in the case of resonance of the natural frequency with disturbing frequencies from the pump unit [4].

2. Criteria of efficiency of vibration isolators

The main way to ensure the vibration resistance of the pipeline is to detune the natural frequencies f0 from the disturbing frequencies fp. According to the requirements of the state standard specification the ratio of frequencies must be [5]:

$$f_0 / f_p \leq 0{,}75 \text{ and } f_0 / f_p \geq 1{,}3 . \quad (1)$$

Considering the force transmission coefficient K_c as a criterion of effectiveness of vibration isolators (Figure 1), it should be noted that for $K_c < 1$ the vibration protection system is most effective, because the amplitude of the force acting to the foundation decreases. All curves, that illustrates the ratio of the disturbing frequency f_p to the natural frequency f_0, regardless of the relative damping value

β, intersect at the point with the coordinates (√2;1). Therefore, in order to the maximum value of the force transferred to the foundation to be less than the amplitude of the disturbing force (force transmission coefficient $K_c < 1$), the condition must be $f_p / f_0 > \sqrt{2}$ [6, 7].

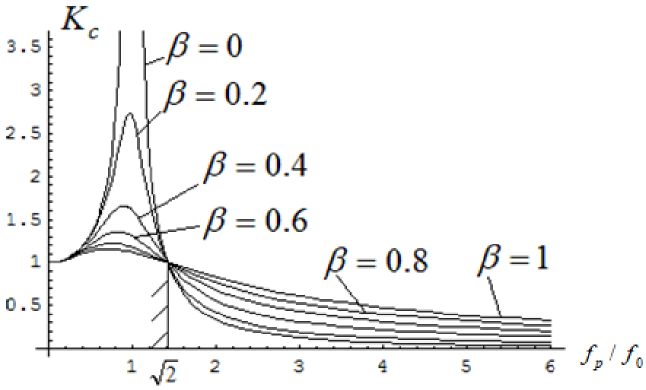

Figure 1. Force transmission coefficient depending on the ratio frequency of the disturbing force to the natural frequency in cases of different relative damping β.

Dynamic coefficient k (Figure 2), which is the ratio of the amplitude of the forced vibrations to the displacement corresponding to the static action of the disturbing force F_0, with increasing stiffness, using classical vibration damping methods cannot be less than one. At values of the disturbing frequency f_p close to f_0 one can observe the resonance state, when the amplitude of the forced vibrations tends to infinity [8].

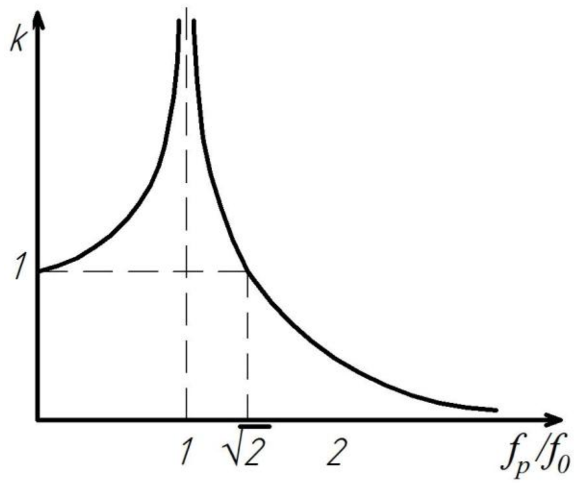

Figure 2. The dependence of the dynamic coefficient k from the frequency ratio.

At large ratios of f_p / f_0, the dynamic coefficient becomes very small, i.e. the high-frequency force does not cause appreciable vibrations in the low-frequency elastic system, the latter, as it were, does not have time to respond to very rapid changes in the disturbing force [8]. Less than one the dynamic coefficient becomes when $f_p / f_0 > \sqrt{2}$. Considering

$$f_0 = \frac{1}{2\pi}\sqrt{\frac{c}{m}}, \tag{2}$$

it becomes obvious that the lower the stiffness coefficient of the elastic element (natural frequency), the wider the frequency range in which the vibration isolator works.

Designing of vibration isolators, which make it possible to increase the ratio f_p / f_0 to a level greater than $\sqrt{2}$, seems very promising from the point of view of effective vibration damping and avoidance of resonance in the case under consideration. This is possible with a decrease of pipelines natural frequencies, which can be achieved by using negative stiffness passive vibration isolator [9, 10].

3. “Pipeline – negative stiffness vibration isolator” system

Considering the pipeline as an elastic system, we get the missing elastic element, which acts as a "stabilizer" for a vibration isolator with negative stiffness. The force characteristic of such system is shown in Figure 3. When two mentioned elements are combined, the force characteristic of the pipeline-vibration isolator system becomes horizontal in a certain section, thereby reducing its stiffness and thus the natural frequencies in accordance with (2).

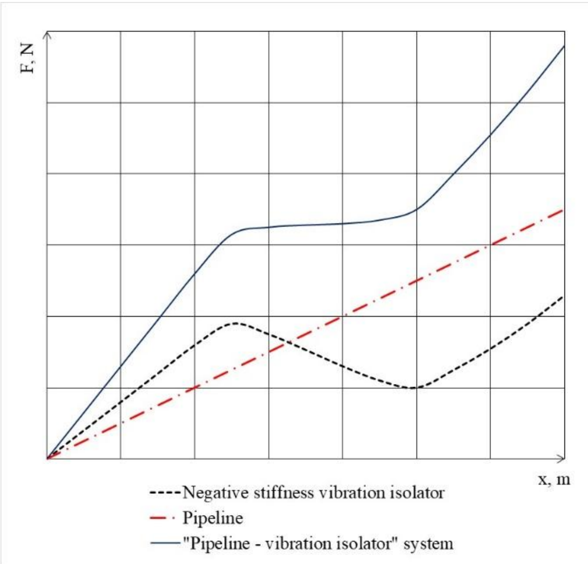

Figure 3. Force characteristic of a vibration protection system with negative stiffness.

Negative stiffness in this system is obtained in the descending section of the power characteristic of the vibration protection system (Figure 3). The working range of the vibration protection system is set in such way that its force characteristics decrease as the amplitude of the pipeline vibration increases. In combination with the linearly increasing force characteristic of the pipeline, which acts as a “stabilizer” with positive stiffness, we obtain a flat section with an almost constant restoring force with increasing values of the vibration displacement. When working in this area (with pre-pressing), the stiffness of the "pipeline-vibration isolator" system is reduced.

3.1. Experimental argumentation

The section of the pipeline that hit the resonance regime is modeled by a mass m located on the spring with stiffness c (Figure 4).

Figure 4. The experimental installation.

Silicone disk springs with a negative stiffness area, used as a vibration isolator with negative stiffness, are pressed to the mass *m* by plates moving along the guides (Figure 4).

The disturbing force F_0 is created by a frequency-controlled electric motor, which makes it possible to achieve resonance with the natural frequency of the model $f_0 = 34\ Hz$, determined in accordance with (2). When carrying out the experiment without plate springs (without a vibration isolator), when the rotor rotates at a frequency of 34 Hz, the value of the vibration velocity is 23.4 mm/s (Figure 5).

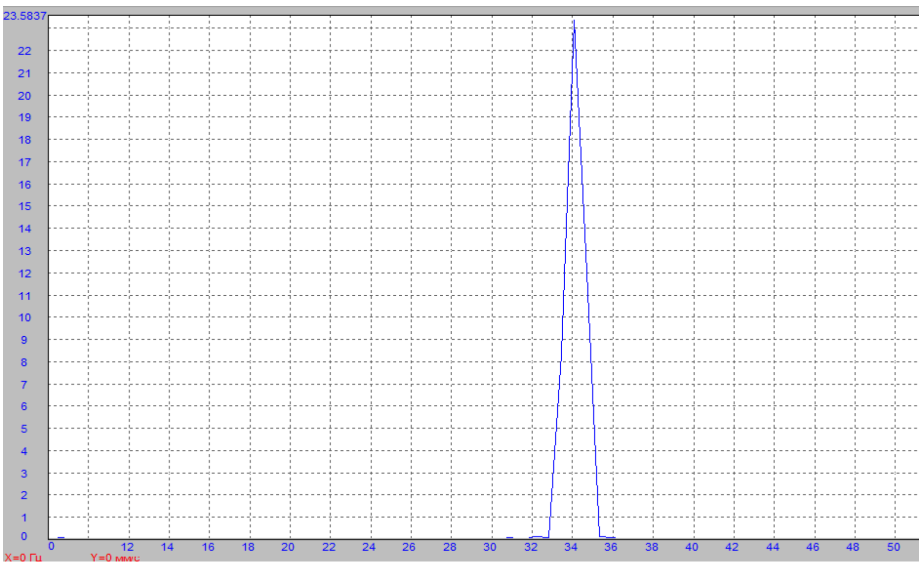

Figure 5. Measurement of vibration without vibration isolators at resonant frequency.

The theoretical force characteristic (the dependence of the restoring force from the displacement) of the disk spring is determined from [11]:

$$F(x)=\frac{8\cdot\pi\cdot E\cdot s\cdot\lambda_1}{(D-d)^2}\cdot\left\{(f-\lambda_1)\cdot(f-\frac{\lambda_1}{2})\cdot\left[\frac{D+d}{(D-d)\cdot 2}-\frac{1}{\ln\left[\frac{D}{d}\right]}\right]+\frac{s^2\cdot\ln\frac{D}{d}}{12}\right\}, \tag{3}$$

where E – the modulus of elasticity of the spring material;

s – thickness of the spring cone;

f – total height of the inner cone;

D – outer diameter of the disk spring;

d – inside diameter of the disk spring;

λ_1 – spring draft.

The theoretical and experimentally measured force characteristic of the spring used during the experiment is shown in Figure 6.

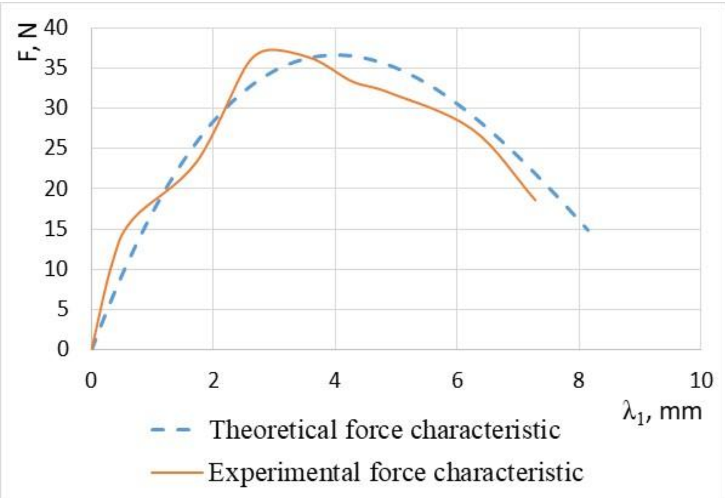

Figure 6. Force characteristics of the disk spring.

When the disc springs are pressed at λ_1 = 6 mm (Fig 6.) to the area of negative stiffness, the resonant frequency decreases to f_{01} = 24.16 Hz (Figure 7). The value of the vibration velocity is 14.92 mm/s.

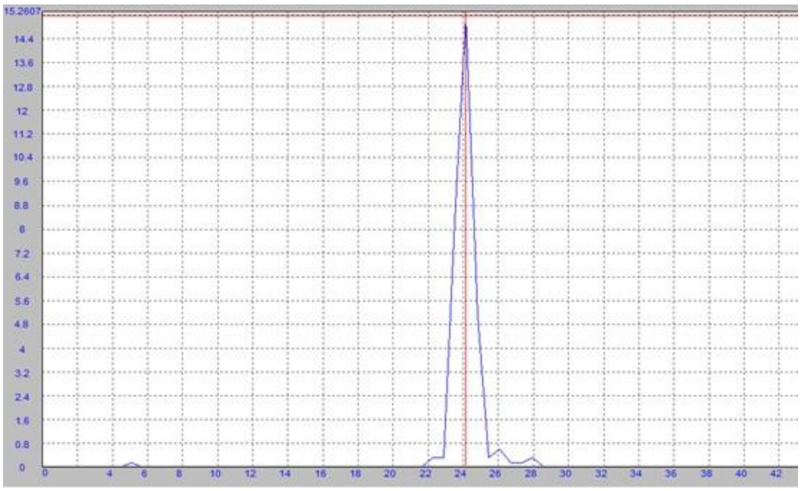

Figure 7. Measurement of vibration with negative stiffness vibration isolator.

The decrease of the resonant frequency indicates a decrease in the natural frequency of the "vibration isolator-protected object" system in comparison with the natural frequency of the protected object itself and, consequently, of the exit from the resonance zone.

4. Summary

The use of negative stiffness vibration isolation systems to reduce the vibration of pipelines will reduce their natural frequencies, which in turn ensures that condition (1) and exit from the resonance zone are met. In addition, it becomes quite easy to obtain the ratio $f_p / f_0 > \sqrt{2}$, resulting in less than 1 becoming both a dynamic coefficient (Figure 2) and a force transmission coefficient K_c (Figure 1), which is the criterion of the effectiveness of using vibration protection systems.

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