Reduction of vibrations of mechanical and power plant pipelines using vibration insulators with design power characteristics based on guides of a special shape

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Abstract. The article deals with the problem of reduction of the level of pipeline vibrations caused by resonance with disturbing frequencies of pumping units. Each pipeline vibration has to be analyzed since there are no universal methods for reducing pipeline vibrations. For vibration isolation, it is proposed to use vibration isolators on the basis of negative rigidity. The pipeline acts as a stabilizer for a vibration isolator becomes flat which reduces its rigidity and natural frequencies. Analytical calculations were performed to confirm the effectiveness of the solution. A passive vibration absorber based on the elastic element moving along the guides perpendicular to their axis creates power characteristics which are required to protect against vibrations.

1. Introduction
One of the most common elements of pumping stations is pipelines directly connected to high-pressure units and equipment operating in non-stationary modes which causes vibrations (transmitted from pumping equipment). Pipeline vibrations can exceed permissible values and cause damage to the pumping equipment. A frequent cause of failures is depressurization of connections of the pump nozzles and pipelines due to their fatigue damage during vibrations [1, 2].

Pipeline oscillations are transmitted to the pumping unit which creates large stresses on its body and increases vibrations. It is important when these frequencies coincide. All this can lead to failures of the pumping equipment in case of damage to its parts and assemblies, as well as to unforeseen expenses on repair works [2].

There are no universal methods for reducing vibrations of technological pipelines. Each type of vibrations requires a careful and qualified analysis, as methods of damping are not efficient.

2. Methods and materials
Considering the pump unit as a source of a large number of forced vibrations with different frequencies, it should be noted that in some cases it is necessary to reduce natural frequencies of the pipeline. Analysis of a number of piping vibrations [1-6] revealed that the main causes are the resonance of natural frequencies with disturbing frequencies of the pump unit.
According to the results of vibrodiagnostics of horizontal pumping units NMP 5000-90 (Figure 1) with a speed of 1000 rpm, elevated values of vibration speed were observed both on bearings in the vertical, horizontal and axial directions, and on binder pipelines [4].

The main frequencies of vibrations of the pump unit and natural frequencies of vibrations of the piping are presented in Table 1.
Table 1. The main frequencies of vibration of the NMP 5000-90 pump.

| Forcing frequency of the pump unit, Hz | Natural frequency of the strapping, Hz |
|---------------------------------------|---------------------------------------|
| 16,67 – main rotary frequency          | 13,03                                 |
| 33,33 – second harmonic of the rotor frequency | 26,06                                 |
| 116,69 – impeller blade frequency      | 34,5                                  |
| 233,38 – second harmonic of the blade frequency | 69                                    |

The detuning of frequencies of natural oscillations $f_0$ from frequencies of forcing loads $f_p$ is the main method for achieving pipeline vibration strength. The following condition has to be met:

$$\frac{f_0}{f_p} \leq 0.75 \text{ and } \frac{f_0}{f_p} \geq 1.3$$

(1)

3. Results and discussion

Let us consider the ratio of forcing frequencies and natural frequencies of the binding vibrations (Table 2).

Table 2. The ratio of the forcing frequencies and the natural frequencies of the binding vibration.

| Natural frequencies | Forcing frequencies | Ratio   |
|---------------------|---------------------|---------|
| 13,03               | 16,67               | 0,78    |
| 26,06               | 33,33               | 0,78    |
| 34,5                | 33,33               | 1,04    |
| 69                  | 33,33               | 2,07    |

The increased vibration of the system “pump–binder” system is caused by high vibrations of the suction pipe in the horizontal section.

The most characteristic resonant frequency is $f_{v3} = 34.5$ Hz at an excitation frequency $f_{p2} = 33.3$ Hz. The ratio of natural and disturbing frequencies is $\frac{f_{v3}}{f_{p2}} = 1.04 \approx 1$ which causes resonant oscillations.

Under these conditions, the “detuning” from the resonance between driving forces and perturbing frequencies of the MNA is required to ensure acceptable operational parameters.

To protect against vibrations of pipelines, systems with negative rigidity have the highest efficiency [5-7], allowing to reduce vibration frequencies of the “pipeline–vibration isolator” system.

Considering the pipeline as an elastic system (simulating its mass on a spring), we obtain the missing elastic element that acts as a “stabilizer” for a vibration isolator with negative rigidity (Figure 2).

Combining these two elements, the power characteristic of the “pipeline–vibration absorber” system becomes flat [6, 8-12], thereby reducing its rigidity and natural frequencies determined by formula:

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

(2)

where $c$ is the rigidity coefficient determined by the pipeline, N/m; $m$ is the mass of the protected object, kg.
Negative rigidity can be obtained on the downward section of the power characteristics of the vibration protective system with negative rigidity (Figure 2). The working range of the vibration-proof system is set so that its power characteristic decreases with an increasing amplitude of pipeline oscillations. In combination with the linearly growing power characteristic of the pipeline which acts as a “stabilizer” with positive rigidity, we obtain a gentle section with an almost constant restoring force with increasing values of vibration movements. When working in this area (with a preliminary preload), rigidity of the whole “pipeline-vibration isolator” system decreases.

The system with negative rigidity can be built on the basis of guides that have a specially designed shape (Figure 3).

The elastic element of the system is compression spring 3 which moves perpendicularly to the axis of guides of a specially designed shape 1 by means of roller supports 2. The compression spring is connected through rigid connection 5 to pipeline 6.

For the case under consideration, the value of negative rigidity should be calculated in such a way as to reduce the frequency of vibrations of the “pipeline-vibration isolator” system to a level that does not coincide with forcing frequencies of the pump unit taking into account condition (1).

The shape of guides must be calculated for each case separately. The design scheme is presented in Figure 4.
**Figure 3.** Vibration-proof system with negative rigidity based on specially shaped guides: 1 - special profile guides; 2 - roller support; 3 - compression spring; 4 - guide glass; 5 - hard link; 6 - pipeline; 7 - clamps; $F_0$ - driving force; $F^*$ - restoring force.

**Figure 4.** The design scheme of a vibration-proof system with negative rigidity based on specially shaped guides.

The amplitude of restoring force $F(x)$ is determined by formula:

$$F(x) = -\frac{\partial P}{\partial x},$$

where $P$ is the potential energy of the elastic element; $c_1$ is the spring rigidity coefficient; $(\frac{l_0}{z} - y)$ - spring compression amplitude.

The potential energy of the compressed spring is

$$P = \frac{c_1}{2} (l_0 - 2y)^2,$$
where $y$ is the function that determines the shape of the guides; $l_0$ is the length of the unstressed spring. In this case, the length of the unstressed spring $l_0$ is greater than the length of the spring $L_0$ in the zero position ($x = 0$). Since function $y$ depends only on coordinate $x$, it is possible to reduce the partial differential equation to an ordinary differential equation:

$$F(x) = -2c_1(2y - l_0) \frac{dy}{dx}. \quad (5)$$

The initial condition for this differential equation is: at $x_0=0$ $y_0=L_0/2$. The amplitude of restoring force $F(x)$ can be written as

$$F(x) = b - c_* \cdot x, \quad (6)$$

where $b = mg; c_*$ – required negative rigidity.

Solving the differential equation (5), you can get the shape of the guides determined by function $y$ for force $F(x)$. It is required to calculate the restoring force to determine negative rigidity. Differential equation (5) can be rewritten in as

$$-(b - c_* \cdot x) = -2c_1(2y - l_0) \frac{dy}{dx}, \quad (7)$$

The initial condition at $x_0=0$ $y_0=L_0/2$.

Solving differential equation (7), we have a function to determine the profile of the guides:

$$y = \frac{c_1l_0 \pm \sqrt{-2c_1D_0+c_1^2l_0^2}}{2c_1}, \quad (8)$$

where $D_0 = \frac{c_* x^2}{2} - bx + c_1l_0L_0 - \frac{c_1L_0^2}{2}$.

With the minus sign, we get the profile of the guides for the compression spring (Figure 5).

Figure 5. The profile of guides for the vibroprotection system with negative rigidity

The profile of the guides was calculated to create negative rigidity $c_* = 19558000$ N/m under which the rigidity of the pipeline-vibration isolator system is $c_c = 1800000$ N/m. The natural frequency of vibrations decreases up to $f_{01} = 9.5$ Hz. The ratio of the natural and disturbing frequencies is $f_{01}/f_p = 0.285$, condition (1) is fulfilled.

4. Conclusion

The proposed design of the vibration isolator allows for the use of a section with negative rigidity to reduce the vibration of the binder operating in resonance conditions by reducing natural frequencies of oscillations of the "pipeline-vibration absorber" system. A passive vibration absorber based on the elastic element moving along the guides of perpendicular to their axis creates e power characteristics required for vibration protection.

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