Theoretical and Experimental Researches of Spring Damping Flexural Oscillations for Beam Structures

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Abstract. One of the main problems in the design of rigid bus in open switchgears is the problem of structural stabilization under the action of various dynamic loads. The implementation of rigid bus with large spans of tube showed a real risk of wind resonance for cylindrical tube structures (aeroelastic self-oscillations). This phenomenon is dangerous due to the fact that at low speeds of the wind flow, intensive oscillations of the tubes occur in the vertical plane, adding to the static loads an essential dynamic component. Its level is comparable with loads from its own weight of structures and can, in combination with other loads, will be the cause of stresses close to the limiting ones by the 1st group of limiting states. The problem of reducing the level of structural oscillations in many cases is associated with the need to increase rigidity and reduce the material consumption of structures, however, it is important to fulfil the technological requirements of the operating conditions and protect people from the harmful effects of oscillations. The article highlights some of the main ways to damp oscillations of rigid bus structures. Is given a mathematical model of the simplest single-mass spring inertial dynamic oscillation damper. Experimental studies have been carried out on the joint work of the rigid bus structure with the spring damper. Has been confirmed the effectiveness of spring dampers using, both outside and inside the tube-bus.

1. Introduction
Rigid bus structure [1,17,18] is intended for transmission and distribution of electrical energy between high-voltage devices as part of open (OSG) and closed switchgear (CSG) fast-mounted complete transformer substations. One of the main problems in the design of rigid bus in open switchgears is the problem of structural stabilization under the action of various dynamic loads. Aeroelastic self-oscillations of large spans rigid bus structures arise quite easily at low wind speeds (1-3 m/s) [2-8,19].

2. Analysis of recent research and publications
National and foreign standards [9, 10, 11] claim that stable (stabilized) wind resonant oscillations of the buses do not occur if the maximal deflection of the buses with a periodic breakdown of air vortices with a resonant frequency \( f_{p,\text{res}} \) does not reach acceptable values \( f_{p,\text{don}} \), i.e.
If condition (1) does not perform, then it is necessary to use special devices [1, 12–15] for damping the resonant oscillations of the tube-bus. Further methods for the selection of such devices and their parameters are absent.

The research goal is to create a new (figure 1) rational damping device for damping oscillations of rigid bus structures.

3. Main material
Spring inertial single-mass dynamic damper is in the form of a solid (figure 1), attachable elastically to the damped object at the point, the oscillations of which are required to be extinguished. The dissipative losses in the damper have a significant effect on the resulting characteristics of the motion of an object with a damper.

Let's consider the forced oscillations of two loads attached to the springs, under the action of the harmonic perturbing force $P(t) = P\sin\theta t$, where one of the masses is the oscillation damper (figure 1) disregarding dissipative forces. This force is applied to one of the masses, for example, to the mass $m_1$, and get the system of equations:

$$\begin{cases}
y_{1}' + ay_{1} - by_{2} = q \sin \theta t;
y_{2}' - cy_{1} + cy_{2} = 0,
\end{cases}$$

where $a = \frac{k_1 + k_2}{m_1}$, $b = \frac{k_2}{m_1}$, $c = \frac{k_2}{m_2}$, $q = \frac{P}{m_1}$.

Also note that in the absence of the damper - the spring with stiffness $k_2$ and mass $m_2$ ratio $\frac{k_2}{m_1}$ is the square of the natural frequency of a system with one degree of freedom:

$$\omega_0^2 = \frac{k_2}{m_1} = a - b.$$  

Because the resistance to movement is not taken into account, there will be no phase shift between the perturbing force and the movement caused by it and the particular solution of system (2) corresponding to the forced oscillations can be taken as:

$$y_1 = A_1 \sin \theta t;$$
$$y_2 = A_2 \sin \theta t.$$
Substituting this solution into system (2), obtain the amplitudes:

\[ A_1 = \frac{q(c - \theta^2)}{(a - \theta^2)(c - \theta^2) - bc}; \]
\[ A_2 = \frac{qc}{(a - \theta^2)(c - \theta^2) - bc}. \]  

At very low perturbation frequencies \( \theta \to 0 \), can be neglected of value \( \theta \) at (5), and the amplitudes of both masses will be the same and equal to the displacement of masses from a statically applied load \( P \):

\[ A_1 = A_2 = \frac{q}{a - b} = \frac{q}{\omega_0^2} = \frac{P}{k_1} = y_{cm}. \]  

Then, expressing the oscillation amplitudes (5) in terms of the static displacement parameter, we get:

\[ A_1 = \frac{(c - \theta^2) \cdot \omega_0^2}{(a - \theta^2)(c - \theta^2) - bc} y_{cm} = \alpha \cdot y_{cm}; \]
\[ A_2 = \frac{co_0^2}{(a - \theta^2)(c - \theta^2) - bc} y_{cm} = \beta \cdot y_{cm}. \]  

Dependence of the coefficients \( \alpha \) and \( \beta \) from frequencies ratio \( \theta/\omega_0 \) for the case when \( k_1 = k_2; m_1 = 4m_2 \) presented in figure 2.

**Figure 2.** Graph of coefficients of increase oscillations \( \alpha \) and \( \beta \) in dependence of frequencies ratio \( \theta/\omega_0 \)

For the verifying the theoretical prerequisites a full-scale experimental installation was assembled (figure 3, a) in the form of a steel double-support beam with a span of 13.5 m. The beam cross-section – round tube 159х5,5 mm. Fastening at the both ends is the hinge. The spring damper (figure 3, b, c) was selected from a spring of appropriate stiffness and mass so that the frequency coincided with the natural frequency of the flexural oscillations of the tube-bas. A spring damper was installed both outside and inside the tube [16].

The determination of the amplitude of oscillations was made in the following order:

1. A vibratory machine is installed on the tested structure (tube-bus).
2. Are being made the flexural oscillations of the first-tone for rigid bus structure, smoothly transforming into resonance.
3. The amplitude swing of the forced oscillations of the structure is fixed with the help of a geodetic rail.
4. The vibratory machine is turned off and the process of own damped oscillations of the structure is recorded.
5. Processing of results.

The process of tube oscillation is a certain total movement, which includes many different forms, connected simultaneously with both different oscillations of the element (flexural, longitudinal, etc.), and oscillations of individual parts of the element (rattling, etc.). The practical problem of the vibration method is the selection from all this variety of forms of the calculated vibration shape based on the existing mathematical apparatus of oscillation processing. The process of “recognition” of the calculated oscillation pattern is reduced to the analysis of the amplitude-frequency characteristic (AFC) of the recorded signal. Theoretically is determined the range of possible values of the natural frequency of the calculated form. Typically, this frequency range does not exceed 10-15 Hz. In the resulting frequency range, the frequency response is examined AFC for frequencies with a maximum amplitude (often, with an element's flexibility $\lambda > 80$, this amplitude is maximum at the entire frequency response). The resulting frequency determines the calculated form of oscillations.

Since the instrumental systematic error of the registration of oscillations is small and the main measurement error is random, for each method of damping were made several records of free oscillations of the tube.

The initial stage of the vibration signal processing is the separation of the vibrogram into individual oscillations (oscillation processes caused by a single disturbance) and discarding the time interval associated with the direct action of the vibratory machine by each oscillation.

![Figure 3. Experimental installation of double-support beam: a) general view; b), c) spring damper](image)
Further analysis of the vibrational program of natural oscillations can be made by presenting the process of oscillations in the form of a Fourier series (harmonic analysis). This approach allows analysis with any frequency step (spectral bandwidth).

The effectiveness of the installed spring damper was recorded by a decrease in the amplitude of the forced oscillations for the tube-bus (a decrease in the amplitude almost to zero in the resonant mode) [16].

4. Conclusions
1. The practical importance is that the mass $m_1$ doesn’t move. This means that with a certain selection of characteristics of the added system - springs and weights, the initial mass $m_1$ remains at rest, despite the fact that the disturbing load is applied to the mass $m_1$.
2. For the first time it is theoretically justified and experimentally confirmed that the spring damper reduces the amplitude of oscillations of the structure by 12 times, however, it requires fine tuning. These dampers rationally positioned inside the tube (2-3 pieces) for uniform perception of tube oscillations with settings close to the resonant frequency.

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