Experimental research of dynamic vibration damping for rigid busbar structures

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Abstract. Introduction: in the case when flexible structures interact with the wind flow, various phenomena of aerodynamic instability may arise. Typical representatives of such phenomena are vortex wind excitation of cylindrical structures, galloping of poorly streamlined structures with a square, rectangular or rhomboid cross-section, etc. The article highlights some basic ways of damping vibrations of rigid busbar structures.

Materials and methods: the method of dynamic vibration damping consists in attaching additional devices to the vibration protection object to change its vibration state. The work of dynamic dampers is based on the formation of force effects transmitted to the object. It differs from another method of vibration reduction, characterized by the imposition of additional kinematic connections on the object such as fixing its particular points.

Results: a mathematical model of the plate dynamic damper operation with a point load is presented. To determine the optimal parameters of dynamic vibration dampers, their calculation was performed, taking into account the joint action of the rigid busbar and the damper. Experimental research of the conjoint work of a rigid busbar with a plate dynamic damper is carried out. Conclusions: the effective application of plate dynamic dampers with a point load has been confirmed both outside and inside the tube busbar. Is proposed the special plate vibration damper. This allows to increase the logarithmic decrement of oscillations by 3-3.5 times and reduce the amplitude of oscillations of rigid busbar structures in the resonant mode by 12 time.

Key words: rigid busbar, outdoor switchgear, vortex wind excitation, plate dynamic damper, point load.

1 Introduction

Nowadays, the problem of structural damping vibrations is relevant not only in construction but also in other areas of technology. A huge number of special damping devices to reduce the amplitudes of forced structural vibrations are used in mobile transport. These are the various dampers (including dynamic and shock) and vibration absorbers, vibro-impact dampers, leaf springs with dry friction, wedge-type shock absorbers, etc. Generally, the use of damping devices in construction is caused by a large dynamic component of internal forces. Equally important is the damping of both lengthy structures such as high-rise

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relatively flexible buildings, towers (solid and lattice, steel and reinforced concrete), tubes, masts, etc. as well as their elements. Vibrations of the entire structure arise under the action of wind, seismic, transport, technological and other dynamic loads. The amplitudes of such vibrations must be reduced.

The use of a rigid busbar [1] in electric networks construction has become widespread in many countries. Outdoor switchgears (OSG) with a voltage of 110-500 kV have been built and are being successfully operated according to standard designs in such countries as Great Britain, Germany, Japan, USA and Canada. 765 kV OSG are always produced with rigid busbars in USA and Canada. In recent years, OSG with rigid busbars with a voltage of 110 kV and more are increasingly being used next to flexible busbars in Russia.

The rigid busbar of the OSG is made of sufficiently strong aluminium alloys with bus spans of 9 m or more (110 kV and more). It is exposed to solar radiation, wind load and ice. As research and operating experience show, the resonant wind speed is low and is no more than 2–3 m / s (aeroelastic self-oscillations) [2-8]. The stresses in the busbar material and the loads on the insulators in this mode are usually significantly less than the permissible values. Wind vibrations of rigid busbar structures have a negative psychological effect on staff, causing discomfort for builders, mounters, power engineers and other specialists of maintenance services. Moreover, such vibration leads to weakening of bolted connections (hence, increases the probability of bus failure), and in some cases causes fatigue damage in structural elements (which is especially unacceptable for elements made of aluminium alloys). Therefore, wind resonances must be constructively eliminated.

The development of the structural dynamic theory is revealed in the works of Soviet and foreign scientists [9] in the middle of the XX century. These contain direct recommendations for dynamic analysis to compile the differential equations of motion for "complex" models, taking into account the set of conditions for the interaction of the constituent elements of the model, the mutual influence of the structure and the external environment.

The principles of dynamic loads on buildings and structures are considered in work [10]. A number of scientists were engaged in experimental and theoretical research in the field of wind flow around buildings and structures [11]. The basic knowledge of the liquid and gas flow was provided in works [12]. The phenomenon of transition from laminar to turbulent flow was investigated by [13].

The research of phenomena in the theory of boundary layer is described in [14]. Discovery and mathematical description of the process of vortex wake appearance behind cylindrical objects at certain Reynolds numbers is considered in [15].

As a result of the vortices breakdown, periodic forces act on the bus across the airflow. If the frequency of vortex breakdown coincides with the frequency of natural vibrations, wind resonance (vortex excitation) may occur. It should be noted that stable resonant oscillations are preceded by random oscillations with a small amplitude. At a resonant wind speed, the amplitude gradually increases. It is well known that synchronization of vortex separation and stabilization of vibrations appears when the amplitude of resonant vibrations is not less than 0.015-0.1 of the bus diameter D. A larger value of amplitude was determined in laboratory conditions when testing circular cylinders, whereas a smaller synchronization amplitude was with vortex excitation of bus with a diameter of 250 mm. The relative amplitude is not less than 0.025-0.03D for bus diameter up to 120 mm.

Rigid bus structures are designed and operated in accordance with the industry standards Eurocode (Europe), RUS Bulletin (USA), SOU (Ukraine) and STO (Russia).

The national and foreign standards [16, 17, 18] recommend the use of special devices [19-26] to damp the resonant oscillations of the tube busbar in case if the greatest deflection
of the bus exceeds the permissible value \( y_{p,\text{add}} \) during the periodic breakdown of air vortices with a resonant frequency \( y_{p,\text{max}} \). Further actions of the selection of such devices and their parameters are not described in the standards.

The research aim is to create a new rational damping device to damp vibrations of rigid busbar structures.

2 Materials and methods

The method of dynamic vibration damping consists in attaching additional devices to the vibration protection object to change its vibration state. The work of dynamic dampers is based on the formation of force effects transmitted to the object. It differs from another method of vibration reduction, characterized by the imposition of additional kinematic connections on the object such as fixing its particular points. Dynamic damping is applicable to all types of vibrations: longitudinal, bending, torsional, etc.

A change in the vibration state of an object with a connected dynamic damper can be carried out both by redistributing vibrational energy from the object to the damper and in the direction of increasing the dissipation of vibration energy. The first is implemented by means of correcting the elastic-inertial properties of the system that change frequencies of the acting vibration disturbances. In this case, the devices connected to the object are called inertial dynamic dampers.

The simplest inertial dynamic damper is made in the form of a solid body, which is elastically connected to the damped object at the point, the vibrations of which must be damped. Dissipative losses in the damper have a significant effect on the resulting characteristics of the motion of an object with a damper.

The damper consists of a plate (or plates) cantilevered outside the bus (or inside the bus) with a concentrated mass on the edge, which can be moved along the plate for fine adjustment. The damper is adjusted by moving the mass in such a way that the damper oscillates in antiphase to the main structure in the resonant vibration mode of the bus. This leads to a decrease in the vibration amplitude of the main structure and dissipates the vibration energy.

The conjoint work of the rigid busbar structure and the vibration damper is presented in the form of a system with two degrees of freedom (Fig. 1).

![Fig. 1. Scheme of a plate vibration damper: 1 – damped object; 2 – damper.](image)

By applying a harmonic disturbing force \( P(t) = P\sin\theta t \) to a rigid busbar (mass \( m_1 \)), a system of equations is obtained:
\[
\begin{align*}
    y''_1 + ay_1 - by_2 &= q \sin \theta t; \\
    y''_2 - cy_1 + cy_2 &= 0,
\end{align*}
\]

(1)

Where \(a = \frac{k_1 + k_2}{m_1}, \quad b = \frac{k_2}{m_1}, \quad c = \frac{k_2}{m_2}, \quad q = \frac{P}{m_1}.
\)

Considering that for \(\theta = \frac{C}{2} = \left(\frac{k}{m_1}\right)^{\frac{1}{2}}\) the mass \(m_1\) does not move, the optimal parameter of the vibration damper is chosen. This parameter is the length (overhang) \(l\) of the console of the plate vibration damper:

\[
l = 3\sqrt{\frac{3EI}{m_2\theta^2}}
\]

(2)

As a result, the rigid busbar will maximally transfer the vibration energy to the damper, despite the fact that the disturbing load is applied to it. The mass of the dynamic vibration damper is within 1-3% of the mass of the rigid busbar structure.

The solution of this issue is obtained by analogy with the spring dynamic vibration damper [27].

To test the theoretical premises, a full-scale experimental structure (Fig. 2) in the form of simply supported steel beam with a span of 13.5 m was assembled. The cross-section of the beam is a round tube 159 x 5.5 mm. The supports at the ends are hinged. The damper installed on the bus (Fig. 3) was subjected to the process of 'fine' adjustment by moving the concentrated mass along the plate. At this time, the amplitudes of oscillations in the resonant mode were recorded for structures with and without a damper. The frequency of the damper coincided with the natural frequency of bending vibrations of the tube busbar. The dynamic damper was installed both outside and inside the tube [28].

Experimental research of structures under the operational load represents the true response of a structure on the real impacts, since the nature and position of load application exactly corresponds to the operating conditions. However, it is necessary to apply artificial methods to excite vibrational influences in most cases of dynamic test [28].

To excite vibrations of steel structures, an electromechanical eccentric vibrator was used (Fig. 4-6). It has two eccentrics synchronously rotating in opposite directions, driven by an electric motor [21]. The machine is fixed to the structure. A smooth change in the angular speed of the shaft's rotation of the vibrator gives an opportunity to study the structural behavior at the different frequencies of the disturbing force and to note the resonance.

The horizontal components of both forces \(Q\cos(\omega t)\) at each moment of time are equal to each other and oppositely directed, and therefore their sum is equal to zero (Fig. 4). When the vertical components are added, a pulsating force \(2Q\sin(\omega t)\) appears, which changes its magnitude and direction according to the sine law. The value of \(Q\) depends on the mass \(M\) of one eccentric weight, the value of its eccentricity \(r\) and the angular velocity \(\omega\).

Technical characteristics of the vibrator: direct current collector motor; rated power is 300 W; rated voltage is 24 V; rated speed is 2000 min-1; rated load moment is 1.4 N\(\cdot\)m; rated current consumption is 20 A; the mass of the vibrator is 25 kg; maximum amplitude value of the disturbing force is 5.0 kN.

To register dynamic deformations (vibrations), the following equipment was applied. A piezoelectric sensor was used as a primary transducer (Fig. 7). The piezoelectric sensor possesses a number of advantages [28] associated with the absence of its own inertial component. Nowadays, it has been widely used in various vibration recording devices.
Considering that for $\theta = (C)$
\[ \frac{1}{2} \left( \frac{k_2}{m_2} \right)^{1/2} \]
the mass $m_1$ does not move, the optimal parameter of the vibration damper is chosen. This parameter is the length (overhang) $l$ of the console of the plate vibration damper:

\[ E_{Il} = \text{constant} \]

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Fig. 2. General view of the experimental structure in the form of a simply supported steel beam.

Fig. 3. Experimental structure of the beam with a dynamic vibration damper: a) location across the tube busbar; b) location along the tube busbar.

Fig. 4. The scheme of the electromechanical eccentric vibrator: 1 - rotors; 2 - eccentrics; 3 - balanced components of centrifugal forces; 4 - unbalanced components of centrifugal forces.
The operation of piezoelectric sensors is based on the piezoelectric effect. This means that under the influence of mechanical deformations of a substance with piezoelectric properties (a special type of quartz, tourmaline, ceramics of lead titanate-zirconate, etc.), its polarization occurs. As a result, electric charges appear on certain elements surfaces. Thus, an electromotive force (EMF) appears, the value of which is directly proportional to the value of the mechanical deformation of the sensor [28].

Piezoelectric sensors are acceleration sensors (accelerometers) that operate without a reference point. Inertia-free operation of the sensor provides high-quality measurements. The orientation of the sensor in the gravity field is irrelevant. Piezoceramics made of lead titanate-zirconate have a Curie point temperature above 3200 C°, which ensures its use in a wide temperature range. Additionally, it should be noted that due to the high natural frequency (over 25 kHz), resonance phenomena are normally excluded in the range 2 - 20,000 Hz. Piezoelectric sensors are usually delivered in a special stainless steel case. The sensors use pre-aged piezoelectric elements, which ensures high long-term stability of the sensor parameters. The sensors can be attached to ferromagnetic elements using a built-in adhering magnet when measuring vibrations with a frequency of no more than 5 kHz and an amplitude of vibration accelerations of no more than 50 m/s2. The passport frequency range is from 2 Hz to 20 kHz and is usually determined by the type of amplifier.

To convert recorded by the primary transducer electrical signals into numerical values (‘digitization’ of the signal), a sound card of a personal computer was used.
Determination of the frequency and amplitude of oscillations was carried out in the following order:

1. A vibrator is set on the experimental structure (tube busbar).
2. The first tone bending vibrations of the structure are induced, smoothly turning into resonance.
3. An amplitude of the forced vibrations is fixed with a geodetic rod.
4. The vibrator is turned off and the process of natural damped vibrations of the structure is recorded.

3 Results and analysis

The process of tube vibrations (Fig. 8) is a certain total movement, which incorporates a number of different modes simultaneously associated with both various vibrations of the element (bending, longitudinal, etc.) and vibrations of individual parts of the element (rattling, etc.). The practical task of the vibration method [28] is among the variety of modes to select the design mode of vibration on the basis of the available mathematical apparatus for vibrations processing (Fig. 9). In fact, "identification" of the design mode of vibration is the analysis of the amplitude-frequency characteristic (AFC) of the recorded signal. The range of possible values for the natural vibration frequency of the design mode is theoretically determined. Generally, this frequency range does not exceed 10-15 Hz. AFC for the frequency with the maximum amplitude is explored in the obtained frequency range (this amplitude is often maximum over the entire AFC when element flexibility \( \lambda > 80 \)). The resulting frequency determines the design mode of vibration.

Since the instrumental systematic error in vibrations recording is low and the main measurement error is random [28], several records of free and forced vibrations of the tube have been done for this method of damping.

The initial stage of vibration signal processing is the division of the vibrogram into separate vibrations (vibration processes caused by a single perturbation) and discarding the time interval of each vibration associated with the direct action of the vibrator.

The vibrogram presented in Figure 9 is a graph of vibration acceleration with some variable amplitude. The indicated amplitude on the given vibrogram is in some relative units without specifying the dimension. Since the vibration amplitude of the tube is important to confirm the operation of the damper, it was determined using a geodetic rod.

Further analysis of the natural vibrations vibrogram can be implemented by presenting the vibration process in the form of a Fourier Series [28] (harmonic analysis). Such an approach allows analysis with any frequency step (spectral bandwidth). Thus, all received vibrograms were analyzed. The spectral bandwidth was taken equal to 0.01 Hz during the signal processing. This is greater than the systematic error of the equipment, but 3-5 times
less than the random error of measurements. Thus, the value of the natural vibration frequency as the mathematical expectation of \( \omega \) frequencies separately calculated for each impact was formed under the same external conditions from the analysis of a series of tests. Moreover, the standard deviation \( \sigma \), the coefficient of variation \( \upsilon \) and the range of the confidence interval of the \( \omega \) values have been determined for the availability of \( \alpha = 0.95 \) in accordance to the Student's t-test.

4 Conclusions and discussions

1) Practical importance is the fact that when \( \theta = (C)^{1/2} = (k_2 / m_2)^{1/2} \) the mass \( m_1 \) does not move. This means that within a certain selection of the characteristics of the added system (plate cross-section, plate span and concentrated mass of the damper), the initial mass \( m_1 \) remains static despite the fact that the disturbing load is applied directly to the mass \( m_1 \).
2) The efficiency of the installed dynamic damper (both along and across the tube busbar) was recorded by a decrease in the amplitude of the forced vibrations of the tube busbar (decrease in the amplitude to almost zero in the resonant mode).
3) A special plate vibration damper is proposed. This allows to increase the logarithmic decrement of oscillations by 3-3.5 times and reduce the amplitude of oscillations of rigid busbar structures in the resonant mode by 12 time, however, it requires precise tuning. It is rational to place such dampers inside the tube (2-3 pcs).

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