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РАЦІОНАЛЬНІ ПАРАМЕТРИ «ГАСНИКА НА НИТЦІ» ДЛЯ ГАСІННЯ ЗГИНАЛЬНИХ КОЛИВАНЬ КОНСТРУКЦІЙ ЖОРСТКОЇ ОШИНОВКИ

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Анотація. У статті розглянуто принцип динамічного гасіння коливань конструкцій балкового типу жорсткої ошиновки відкритих розподільчих пристроїв. Для гасіння коливань використано спеціальний «гасник на нитці», який розташовується всередині труби трубчатого перерізу і «включається в роботу» при виникненні поперечних коливань шини. Ефект гасіння досягається за рахунок передачі енергії коливань шини гаснику і розсіюванню енергії за рахунок дисипативних властивостей матеріалів гасника. У статті наведено числові дослідження гасника при різних його параметрах. Визначені раціональні параметри «гасника на нитці». Також у статті наведені дані експериментальних досліджень гасіння згинальних коливань консольної балки трубчатого перерізу із застосуванням «гасника на нитці».

Ключові слова: гасіння коливань, гасник на нитці, динамічний гасник коливань, демпфірування, жорстка ошиновка.

РАЦИОНАЛЬНЫЕ ПАРАМЕТРЫ «ГАСИТЕЛЯ НА НИТИ» ДЛЯ ГАШЕНИЯ ИЗГИБНЫХ КОЛЕБАНИЙ КОНСТРУКЦИЙ ЖЕСТКОЙ ОШИНОВКИ

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Аннотация. В статье рассмотрен принцип динамического гашения колебаний конструкций балочного типа жесткой ошиновки открытых распределительных устройств. Для гашения колебаний использован специальный «гаситель на нити», который помещается внутрь шины трубчатого сечения и «включается в работу» при возникновении поперечных колебаний шины. Эффект гашения достигается за счет передачи энергии колебаний шины гасителю и рассеиванию энергии за счет диссипативных свойств материалов гасителя. В статье приведены численные исследования эффективности гасителя при различных его параметрах. Определены рациональные параметры «гасителя на нити». Также в статье приведены данные экспериментальных исследований гашения изгибных колебаний консольной балки трубчатого сечения с применением «гасителя на нити».

Ключевые слова: гашение колебаний, гаситель на нити, динамический гаситель колебаний, демпфирование, жесткая ошиновка.

RATIONAL PARAMETERS OF A «DAMPER ON THE THREAD» FOR DAMPING BENDING OSCILLATIONS OF RIGID BUS STRUCTURES

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Abstract. The paper deals with the principle of dynamic damping of girder rigid bus structures of the outdoor switchgears. The special «damper on the thread» has been used to suppress of oscillations, which is placed inside the tubular bus conductor and «turns on» when the occurrence of cross sectional oscillations of the bus conductors. The effect of damping is achieved due to the transfer of vibrational energy bus to damper and the dissipation of energy due to dissipative material properties of the damper. The article presents numerical researches of the efficiency of the damper in its various parameters. The rational parameters of a «damper on the thread» are set. The experimental tests data of damping bending oscillations of cantilever tubular section beams with the application of a «damper on the thread» are given in the article.

Keywords: suppress of oscillations, damper on the thread, dynamic oscillation damper, damping, rigid bus.

Introduction

Rigid buses constructions in outdoor switchgears are becoming increasingly popular in the modern power supply network engineering [4, 12–18]. Rigid conductive buses made usually from aluminum alloy pipes in such constructions and is supported by insulators, which mounted on a special frame [4, 12–18] (Fig. 1).

In connection with large spans buses, these structures are susceptible to the phenomenon of vortex excitation of oscillations (wind resonance) [1, 2, 4–12], which in turn actualizes the problem of damping the oscillations of these structures. The problem of buses oscillations damping complicated electrical requirements – the buses operating mode up to 750 KV. High voltage makes it impossible to use most of the known methods of damping the oscillations of the beam type structures [4, 12–18]. For example, the damping of the oscillations by joining a dynamic oscillation dampers [10–18] leads to the appearance of the corona, which is not allowed.

Modern trends in the development of methods for oscillation damping of this type of constructions are more focused on the transfer of the damping element or damper inside the conductive tube. For example, the well-known work in this direction «Electroengineering, Diagnostics and Service Scientific and Technical Center» Ltd, Moscow, having extensive experience in the operation, design, maintenance and diagnostics of electrical equipment. «Electroengineering, Diagnostics and Service Scientific and Technical Center» embedded the damping device type «butterfly» in the production now (the patent for useful model № 100859). The damping device (Fig. 2) is a rope with a weight at the end, which is fixed to the bottom of bus, into the bus or mounted out of the bus-bar. Experimentally obtained value of the logarithmic decrement of the oscillations for damper type «butterfly» was 0.346.

One relatively new, but not explored in enough ways of bend oscillation damping of rigid bus

Figure 1. Rigid buses constructions of «Kievskaya» electric substation.

constructions is a string damper or a «damper on the thread» (Fig. 3) [3, 12, 19].

The damping nozzle of little rigidity string with gaps to the tube surface is set up into the rigid bus tube. The string is strained with a certain force. On the tube faces the string is fixed with elastic-damping pads. The damper doesn't operate when the small oscillations of a tube. The tube strikes against «theoretically stable» damping nozzle and begins the process of oscillations of the two subsystems (tube and damper), which are interact through multiple impacts in the process of oscillations. The tube oscillations damped by energy transfer to the damper and the dissipation of energy due to dissipative properties of a nozzle and a string material. Unexplored today is the issue of selection of rational parameters of a «damper on the thread», including:

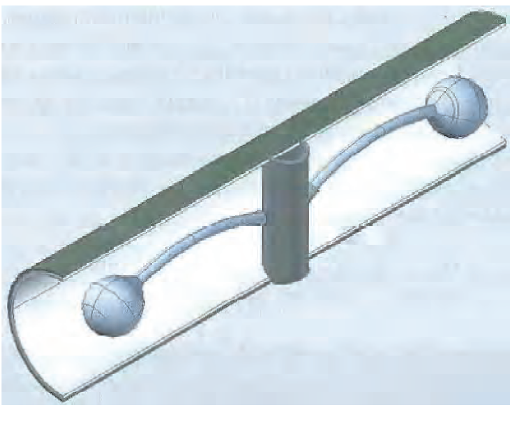

Figure 2. The damping device type «butterfly» with the installation into the bus.

- the mass of the damping nozzle;
- elastic-dissipative properties of the damper nozzle;
- value of a preliminary gap between a string and a tube-bus construction;
- force of string tension and its rigidity.

To study these characteristics were conducted theoretical, numerical and experimental researchers, the results of which are introduced in this article.

Theoretical researches of the behavior of a «damper on the thread»

It was considered a model in the form of oscillations of the two subsystems – a tube-bus and a «damper on the thread» to examine the behavior of a damper on the thread when the bus is oscillate (Fig. 4).

The first subsystem is a weighty tubular cross-section bar with continuous mass m and with a localized mass M, the compressed force T (Fig. 5). A localized mass M takes into account the weight of the possible equipment that may be place on the bus (in the absence of this parameter is equal to zero). The compression force is equal the force of the thread tension of the second subsystem – damper.

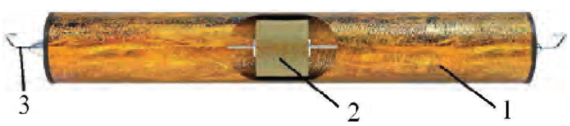

Figure 3. «Damper on the thread» construction for rigid bus constructions: 1 – tube-bus; 2 – the damping nozzle; 3 – string.

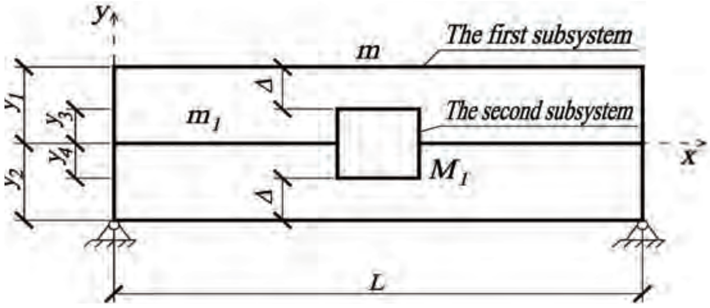

Figure 4. Scheme of location of a «damper on the thread» into the tube: y_1, y_2 – the coordinates of the movements of the upper and lower points of the tube; y_3, y_4 – the coordinates of the movements of the upper and lower points of the damper; m – linear weight of a tube; m_1 – linear weight of a thread; M_1 – mass of a damper; Δ – value of a preliminary gap between a string and a tube.

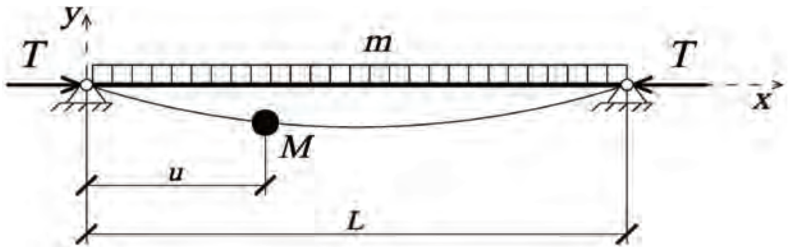

Figure 5. Dynamic model of the first subsystem – tube-bus.

Differential equation of natural cross-section oscillations of the first subsystem can be written as:

$$EI\frac{\partial^4 y}{\partial x^4}+m\frac{\partial^2 y}{\partial t^2}+2\beta m\frac{\partial y}{\partial t}+T\frac{\partial^2 y}{\partial x^2}+\frac{M}{L}\delta(x-u)\frac{\partial^2 y}{\partial t^2}=0,$$

where $\delta(x-u)$ – the Dirac delta- function.

The equation of movement of the middle section of the bar in the initial parameters is:

$$y(x,t)=e^{-\beta\cdot t}\left(y_0 A_{kx}+\frac{y_0^{/}}{k_0}B_{kx}+\frac{M_0}{k_0^2 EI}C_{kx}+\frac{P_0}{k_0^3 EI}D_{kx}+\frac{P}{k_0^3 EI}D_{k(x-u)}\varepsilon(x-u)\right)\sin(\omega t+\mu).$$

The second subsystem is a weighty thread with continuous mass m_1 and with a localized mass M_1, the tension force T (Fig. 6).

Differential equation of cross-section oscillations of the thread with a localized mass is:

$$m_1\frac{\partial^2 y}{\partial t^2}-T\frac{\partial^2 y}{\partial x^2}+\frac{M_1}{L}\delta(x-u_1)\frac{\partial^2 y}{\partial t^2}+2\beta_1 m\frac{\partial y}{\partial t}=0.$$

The equation of movement of the middle section of the thread in the initial parameters is:

$$y_1(x,t)=e^{-\beta_1\cdot t}\left(y_0\cos kx-\frac{P_0}{kT}\sin kx-\frac{P_1}{k_n T}\sin k(x-u_1)e(x-u_1)\right)\sin(\omega_1 t+\mu_1).$$

There is a collision of damping nozzle of the damper with the wall of the tube (Fig. 7) at certain points in time, when the oscillations occur, there are conditions:

$$\begin{cases} y_1\left(\frac{L}{2},t\right)\le y_3\left(\frac{L}{2},t\right) & \text{– is the collision at the top of the tube;} \\ y_2\left(\frac{L}{2},t\right)\ge y_4\left(\frac{L}{2},t\right) & \text{– is the collision at the bottom of the tube.} \end{cases}$$

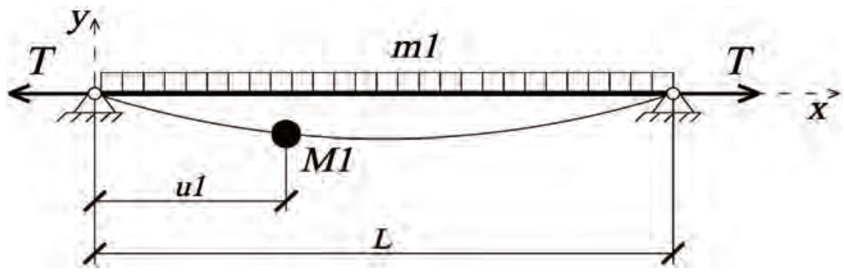

Figure 6. Dynamic model of a «damper on the thread» oscillation.

The equations of movement must consider with the time step is not more than 0.1 from the smallest period of two subsystem oscillations. Collision of the subsystems leads to a redistribution of the kinetic energy of the moving mass of the subsystems, as well as to the irreparable loss of energy. To account for the redistribution of energy subsystem are given by known methods [11] to systems with the same mass (and one degree of freedom), which located at the point of impact. The velocity of a given mass of subsystems after the collision:

$$V_{1i} = \frac{(M_1 - k \cdot M_2)\cdot V_1 + (1+k)\cdot M_2 \cdot V_2}{M_1 + M_2},$$

$$V_{2i} = \frac{(M_2 - k \cdot M_1)\cdot V_2 + (1+k)\cdot M_1 \cdot V_1}{M_1 + M_2},$$

where M_1 – the reduced mass of the bar (first subsystem) at the point of collision;

M_2 – the reduced mass of a «damper on the thread» (second subsystem) at the point of collision;

V_1, V_2 – point velocity to bring the masses up to the collision;

k – the coefficient of restitution, taking into account the irreversible loss of energy at collision.

The obtained point reduced masses velocity after the subsystems collision determine the initial conditions of natural oscillations of each subsystem until the next collision. So, the present oscillations model is the nonlinear system and the influence of the parameters of a «damper on the thread» on the maximum amplitude, the loss of dissipation oscillations energy of the first subsystem (bus) depends on a number of factors. As a result of factor analysis the most significant factors of a damper highlighted, significantly affecting the above mentioned parameters of the bus oscillations with the external perturbation:

1 – clearances Δ (Fig. 4) between the wall of the tube-bus and oscillation damper [limit values of the design considerations Δ=(0,05 ÷ 0,35) D_{tube}];
2 – the ratio of the oscillation frequency of the damper and a tube [limit values frequency of the damper of the design features and conditions of critical stresses in the tube from the thread tension f_{damper} = (0,5 ÷ 2,2) f_{tube}];
3 – damping nozzle mass of the oscillation damper [limit values of the design considerations M_{dampe} = (1–5 %)M_{tube}].

The damping nozzle material should provide the large irreversible loss of energy at impact. This parameter can be studied experimentally, the impact were accepted absolutely elastic in the theoretical researches.

As a parameter observations in the analysis, we used a notional logarithmic decrement of the oscillations δ, which is defined as the exponent is approximated by an exponential envelope of the points of

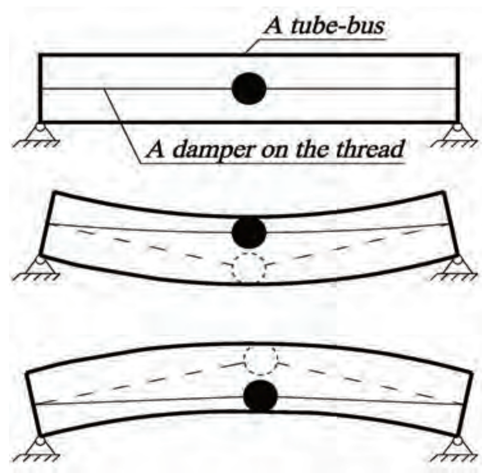

Figure 7. Scheme of the collaborative system «tube-damper».

maximum displacement in time of the mid-span of the tube (Fig. 8) multiplied by the period of oscillation T:

$$\delta = \beta T .$$

The received oscillations vibrational record of the middle section of the bus with a «damper on the thread» was compared with the similar vibrational record in the absence of a damper. The efficiency of application of a «damper on the thread» k was the ratio of the notional logarithmic decrement is (the approximated curve (Fig. 8) for a system with a damper to the logarithmic decrement δ_0 for a system without damper (Fig. 9):

$$k = \frac{\delta}{\delta_0} .$$

So, for a particular case is shown in Fig. 9, the logarithmic decrement by adding of a «damper on the thread» has increased in $k = \frac{\delta}{\delta_0} = \frac{0.23}{0.80} = 2.88$ times.

Of the three mentioned above of the factors studied, two were taken as fixed, and the third was varied in the range of the limit values. The change of the index k efficiency of the application the

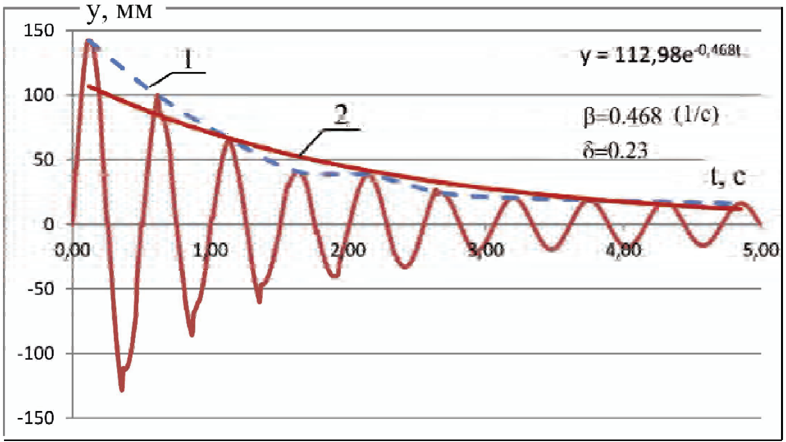

Figure 8. Vibrational record of mid-span of the tube-bus oscillations: 1 – envelope of the points of maximum displacement in time; 2 – approximation of the envelope exponentially.

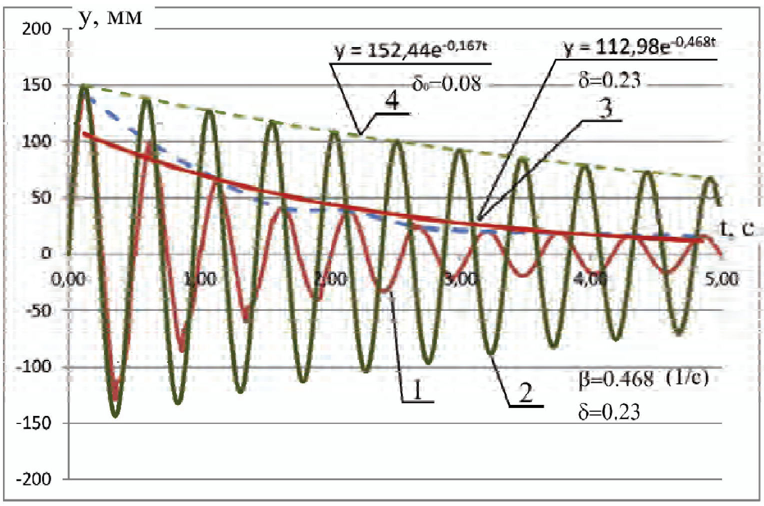

Figure 9. Vibrational records of the mid-span of the tube-bus oscillations: 1 – vibrational records of the bus with a damper oscillations; 2 – vibrational records of oscillations without damper; 3 – envelope approximation exponentially for vibrational records with the damper (δ=0.23); 4 – the envelope of the points of maximum displacement in time of the bus without damper (δ_0=0.08). (In this case the natural frequency of the bus f_{tube} = 1.883 Hz, a damper – f_{damper} = 3.5 Hz).

«damper on the thread» with changing the influencing factors are presented in Fig. 10.

The numerical researches have shown (Fig. 10 c) that the value of a gap between the tube wall and the damping nozzle is almost no influence on the damper efficiency. This parameter can be assigned based on the design considerations associated with avoidance of large amplitude oscillations of the tube, because the development of large displacements of the bus oscillations in the wind flow leads to sustained oscillations and the effect of the wind resonance. The values of the gaps between the damper

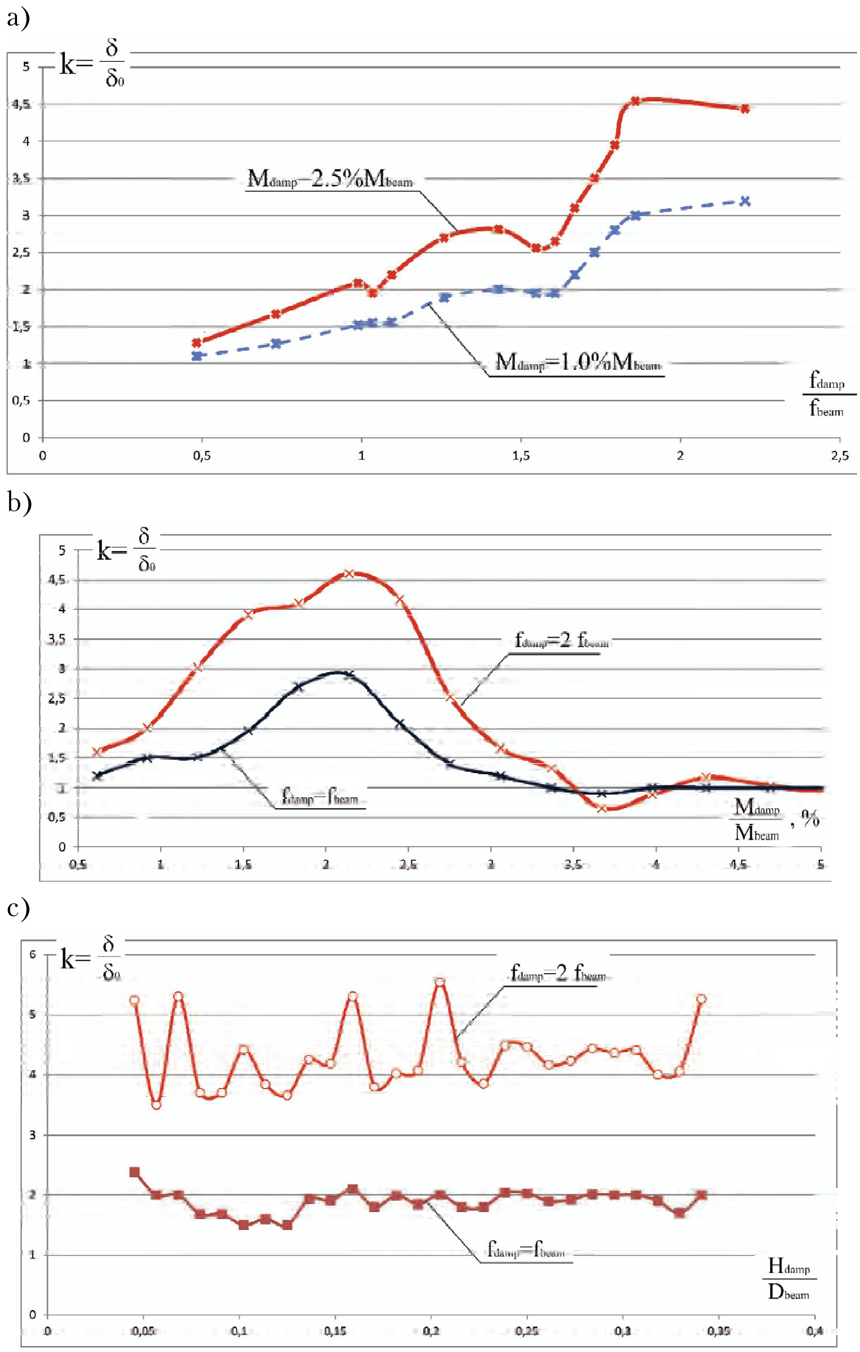

Figure 10. The change of the index *k* efficiency of the application the «damper on the thread» with changing the influencing factors: a) when changing the natural frequencies of the damper; b) when changing the damper mass; c) when changing the damper dimensions.

and the bus wall are recommended to take in the range of 0.15–0.20 of its diameter.

The frequency of damper natural oscillations is an increase with the increase in the tension force of a damper string. According to the obtained results (Fig. 10 a) with the increase the natural frequency of the damper is the increase in the velocities of collisions of the tube and the damper, thereby increasing the damping effect. The upper boundary of the frequency difference between the damper and the bus caused by the occurrence of significant stresses in the tube, in which the possible loss of stability due to the large flexibility of the tube. The rational parameters of the damper frequency are (1.75–2.20) by frequency of a bus.

The damping nozzle mass has also influence on the effect of the application of a «damper on the thread». The low mass (less than 1 % by weight of the bus) does not allow to accumulate the damper kinetic energy. The large mass on the contrary leads to the process of pumping energy bus – damper – bus with minimal losses. Rational parameters of the damper mass are (1–3) % by bus weight.

Experimental researches

It was collected an experimental plant (Fig. 11) of a steel cantilever beam length 3 m for experimental researches of the efficiency of a «damper on the thread». The cross-section of beam is a square tube with the dimensions of 100×3 mm. The damper is a steel wire string with a diameter of 1 mm, stretched over supports, with a damping nozzle with adjustable weight. The natural frequency of the damper is changed using the thread tension, the weight using the internal filling of the damping nozzle.

To test the operation of the damper was used resonance method, for which was installed with a special vibration machine on the structure (Fig. 12). The frequency of vibration machine was chosen equal to the natural frequency of the structure. The selection of resonant frequency were made based on the conditions of occurrence of the maximum displacements of the cantilever edge. It was compared to the amplitude of the steady-state oscillations of beam with damper and without it in the resonant mode.

a) b)

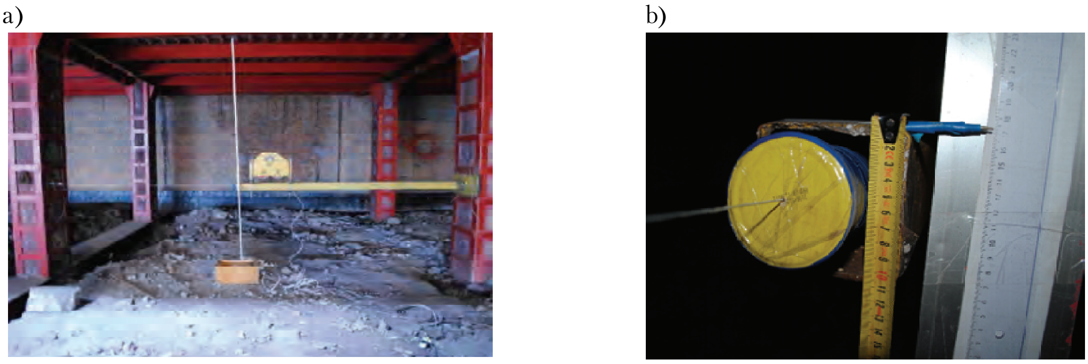

Figure 11. The experimental plant of a cantilever beam with a «damper on a thread»: a) general view of the experimental plant; b) a damping nozzle and a sounding stick for measuring the amplitude.

a) b)

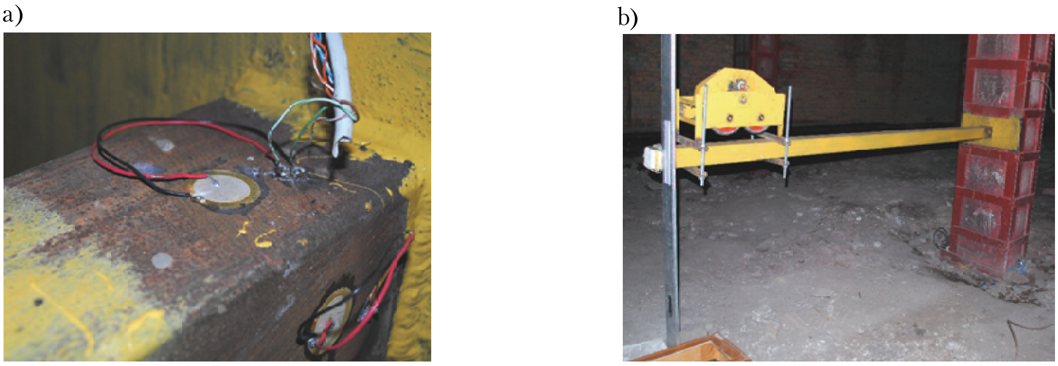

Figure 12. The scheme of registration of stimulated oscillations of beam: a) piezoelectric pick-up; b) vibration machine, mounted on the beam.

The control of amplitudes was carried out by the sounding stick (Fig. 11). It used the method of fixation video to increase the accuracy of the information received (up to 1 mm). It was produced record the vibrational records of natural oscillations at «running-out» also for comparison of the rate of energy loss of oscillations of a system with damper – when you turn off the vibration machine for beams with damper and without it (Fig. 13). The registration of oscillations was produced by the piezoelectric pick-up (Fig. 12) with subsequent conversion data and save it to personal computer (Fig. 13).

The mass of a damper nozzle and the frequency of natural oscillations of the dynamic damper are ranged due to the thread tension in experimental researches. The natural frequencies, the amplitude of the steady-state oscillations in the resonant mode, and the value of the logarithmic decrements of the oscillations have been determined experimentally. The results have been given in Table.

The test results have been showed the close agreement with the theoretical background. The application of a «damper on the thread» with the rational parameters has been reduce the amplitude of the oscillations in the resonant mode in 1.5 times and has been increase the logarithmic decrement of the oscillations in 2.0 times.

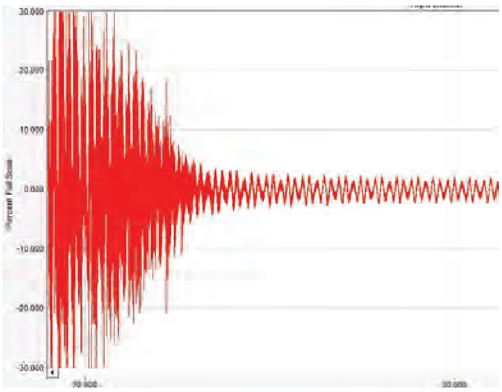

Figure 13. Vibrational records of natural damped oscillations of the tube with a damper at «running-out».

Conclusions

1. The studies found that the mass of the damping nozzle of a «damper on the thread» is significantly influence on the oscillation of the tube-bus. The rational parameters of the damper mass are (1–3) % by bus weight when a single-span support.
2. The studies found that the value of a gap between the tube wall and the damping nozzle is almost no influence on the damping of the tube-bus oscillations. The values of the gaps between the damper and the bus wall are recommended to take in the range of 0.15–0.20 of its diameter to prevent sustained bus oscillations before the damper «turns on».
3. The studies found that the frequency of damper oscillation is significantly influence on the

Table. The results of experimental researches

Damping nozzle weight, kg	Cantilever beam with an installed the vibration machine without damper (oscillations in resonant mode)			Cantilever beam with an installed the vibration machine and with a «damper on the thread»			Damper efficiency $k = \frac{\delta}{\delta_0}$
	Frequency f_0, Hz	Amplitude A_0, cm	Logarithmic decrement δ_0	Damper frequency f, Hz	Amplitude A, cm	Logarithmic decrement δ	
0,27	3,1±0,1	6,0±0,1	0,012± ±0,005	1,7±0,1	4,0±0,1	0,0168±0,005	1,4
				3,8±0,1	3,8±0,1	0,0216±0,005	1,8
				5,0±0,1	3,7±0,1	0,024±0,005	2,0
0,55				1,2±0,1	3,5±0,1	0,0168±0,005	1,4
				2,8±0,1	3,4±0,1	0,0192±0,005	1,6
				4,8±0,1	3,2±0,1	0,024±0,005	2,0
0,82				1,0±0,1	3,8±0,1	0,0192±0,005	1,6
				2,2±0,1	3,8±0,1	0,024±0,005	2,0
				4,2±0,1	3,7±0,1	0,024±0,005	2,0
1,0				0,9±0,1	4,2±0,1	0,0192±0,005	1,6
				2,0±0,1	4,1±0,1	0,024±0,005	2,0
				4,0±0,1	4,0±0,1	0,0288±0,005	2,4

oscillation of the tube-bus. The rational parameters of the damper frequency are (1.75–2.20) by frequency of a bus.
4. The test results have been showed the close agreement with the theoretical background. The application of a «damper on the thread» has been reduce the amplitude of the oscillations in the resonant mode in 1.5 times and has been increase the logarithmic decrement of the oscillations in 2.0 times.

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