Designing a Metal Structure with Allowance for Damping Properties

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Abstract. The use of low-modulus foam capable of intensive absorption of vibrational energy consists in reducing their number and amplitude under dynamic external influences. Computational experiments in the SolidWorks environment and field tests with the construction of vibrograms of damping flexural vibrations of a thin-walled cantilever beam have shown that volumetric damping with foam can be an effective means of damping free vibrations. The paper investigates the effect of damping on the resonant amplitude of forced beam vibrations. In the estimated design calculation, it is proposed to use an analytical expression to determine the logarithmic decrement of the foamed beam and the calculated data on natural frequencies obtained in a computer complex. A significant decrease in the resonant amplitude of the steady-state forced oscillations of the foam beam under bending is predicted.

INTRODUCTION

The problems of calculating the fatigue strength of metal structures belong to non-classical problems of mechanics. Traditionally, their solution is based on data from experimental studies of a specific class of machines [1]. One of the most effective ways to increase the durability of load-bearing metal structures of transport vehicles is to reduce dynamic loads during operation, in particular, by increasing the damping properties [2–8]. As a rule, the accurate determination of dissipative properties of materials and structures is associated with costly experimental studies [1]. Estimates based on numerical methods and analytical solutions are useful for design calculations.

In [8], an expression was given for determining the logarithmic decrement δ_2 of free bending vibrations of a beam foamed with a low-modulus energy-intensive material,

$$\delta_2 = \frac{1}{\omega_2}\sqrt{(\delta_1\omega_1)^2 + \omega_1^2 - \omega_2^2}\ , \tag{1}$$

where ω_1 and ω_2 are eigenfrequencies of empty and foamed beams defined in standard computing systems – a logarithmic decrement of an empty beam, which is characterized as the ratio of neighboring amplitudes of the *i*-th and (*i*+1)-th oscillations.

The values of this decrement are usually taken from design tests. There are some recommendations for the values of the logarithmic decrement of oscillations (Table 1).

Expression (1) was tested experimentally [10] in laboratory tests of the cantilever beam. Despite the widespread use of load cells [11] and optical oscilloscopes [12], a high-speed recording camera was used. The value of the foam beam decrement obtained during the processing of the vibrogram (the value marked "*" in Fig. 1) was close to the calculated value (red curve in Fig. 1) and one and a half times the decrement δ_1 of the empty beam [10].

Mechanics, Resource and Diagnostics of Materials and Structures (MRDMS-2020)
AIP Conf. Proc. 2315, 030010-1–030010-4; https://doi.org/10.1063/5.0036853
Published by AIP Publishing. 978-0-7354-4057-9/$30.00

TABLE 1. Values of the logarithmic decrement of oscillations for various structures [9]

Type of metal structure of different types of cranes	Values of the decrement
Box spans of bridge cranes	0.05÷0.12
Supporting structures of gantry cranes	0.10÷0.25
Supporting structures of portal cranes	0.30÷0.40

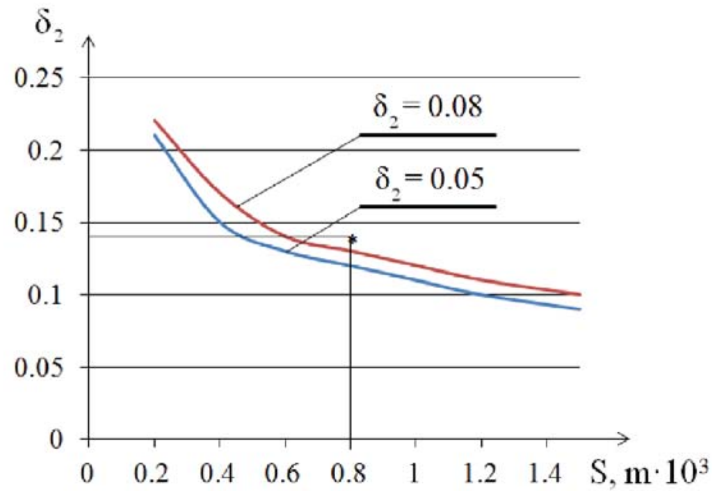

FIGURE 1. Dependence of the logarithmic decrement δ_2 on the wall thickness *s* of the filled beam [5]

This experimental fact provides a basis for using expression (1) at the design stage. Besides, it is shown that volume damping can be an effective means of damping *free* vibrations of a metal structure under bending. In cyclic machines, reducing the amplitude and number of free vibrations increases the durability of the structure and the accuracy of the positioning of the working link [8].

The next stage of research on the effect of volume damping concerns *forced* bending vibrations of a thin-walled metal structure at resonant frequencies. The effect of damping on the amplitude of beam vibrations in the resonant mode is estimated by the example of a single-mass scheme with a periodically changing set external force. *The purpose of the work* is to determine the change in the dynamic coefficient of the structure when using light mounting foam as a filler for structural damping.

FORCED OSCILLATIONS IN A SINGLE-MASS DESIGN SCHEME WITH HYSTERESIS

In oscillation problems, spatial metal structures are often reduced to a simple one-mass scheme. The motion of such a mass under the action of an external periodic force $F = F_0 \cdot \sin \omega t$ and an elastic restoring force is described by the differential equation [13]

$$m\ddot{x} + 2\Psi\dot{x} + cx = F \, , \tag{2}$$

where *m* is the reduced mass of the structure obtained from the equality of the kinetic energy of the original and replacement system, *C* is the equivalent stiffness, Ψ is the attenuation coefficient reflecting the dissipative properties of the structure, ω is the frequency of the driving force.

There is no strict way to describe the attenuation coefficient for a structure. In the case of small oscillations, $\Psi = 2\delta$ is taken, that is, all the kinetic energy of the structure is absorbed by the material. It is proposed to use expression (1) in the estimation calculation to determine the logarithmic decrement of the foamed beam. The numerical values of the empty beam decrement are given in the reference literature for different product groups or determined experimentally [9]. Equation (2) is reduced to the canonical form

$$\ddot{x} + 2n\dot{x} + px = F / m \,, \quad (3)$$

where $n = \delta / m$, $p = \sqrt{c / m}$ is the circular oscillation frequency.

The solution of equation (3) gives an expression for the amplitude of steady-state oscillations [13]

$$A = \frac{F_0}{c\sqrt{(1 - \frac{\omega^2}{p^2}) + \frac{4n^2\omega^2}{p^4}}} \,.$$

The ratio of the amplitude *A* to the static deflection of the beam $y_{cm} = F_0 / c$ termed the dynamic coefficient is equal to

$$\mu = \frac{1}{\sqrt{(1 - \frac{\omega^2}{p^2}) + \frac{4n^2\omega^2}{p^4}}} \,. \quad (4)$$

The dependence of the dynamic coefficient on the frequency ratio ω / p is usually given for different *n*/*p* values that characterize the damping effect of hysteresis losses. As an example of using expressions (1) and (4) in the analysis of the near-resonant behavior of the structure, we present the calculation of the cantilever beam used earlier in the experiment [10].

CALCULATION OF RESONANT AMPLITUDES OF EMPTY AND FOAMED BEAMS

A beam made of a rectangular tube with a box section was selected for the study (Fig. 1), made of the st3sp steel, with a width of *b* = 25 mm, a height of *h* = 50 mm, a wall thickness of *s* = 0.8 mm, and a length of *l* = 600 mm. The lowest oscillation frequencies of an empty and foamed beam were determined in the SolidWorks environment [14, 15] and then by a study in the SolidWorks Simulation Supplement. The characteristics requested by the program for the filler (foam) were selected based on the technical sheets of the professional Kudo Trend line of mounting foams.

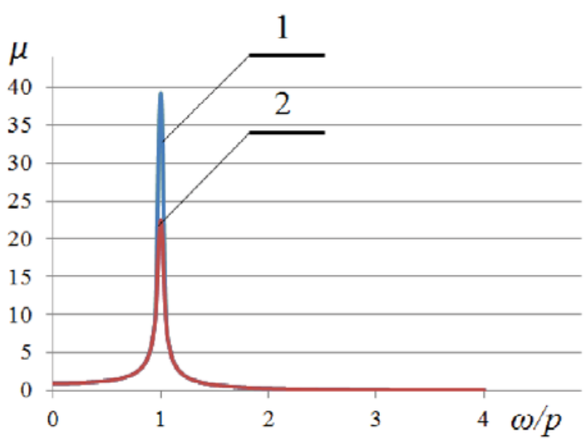

FIGURE 2. Dependence of the dynamic coefficient on the frequency ratio ω / p : 1 – resonant dynamic coefficient of the empty beam, 2 – that of the foamed one

The eigenfrequencies of the empty and filled beam were ω_1 = 141.63 s^{-1} and ω_2 = 140.87 s^{-1}. The experimental value of the logarithmic decrement of an empty beam δ_1 = 0.08. Calculated using expression (1), the value of the foam beam decrement δ_2 = 0.13 differs slightly from the experimental value 0.14. Figure 2 shows the dependence of the dynamic coefficient of the beam (4) on the ratio of the frequency of the disturbing force to the circular frequency of free oscillations.

The graph in Fig. 2 shows that the calculated resonant values of the dynamic coefficient for the empty and foamed beams differ by more than a factor of 1.5. This conclusion requires direct experimental verification.

CONCLUSION

The obtained result makes it possible to predict a significant decrease in the resonant amplitude of the steady-state forced oscillations. This indicates the effectiveness of volumetric damping of a hollow beam with foam not only for free, but also for forced bending vibrations. In the estimated design calculation, it is proposed to use expression (1) and the calculated data on natural frequencies obtained in a computer complex.

REFERENCES

1. V. N. Chelomey, *Vibrations in Engineering: Handbook in 6 Volumes, V. 6. Vibration and Shock Protection,* ed. by K. V. Frolov (Mashinostroenie, Moscow, 1981).
2. D. K. Grishin and V. M. Matveev, Science and Education: Scientific Publication of BMSTU **11**, 69–78 (2013).
3. V. N. Meshcheryakov, V. V. Kolmykov, and D. V. Migunov, Russian Electromechanics **3**, 48–54 (2016).
4. N. N. Evseviev, Bulletin of Science and Creativity **11**(23), 17–22 (2017).
5. V. S. Shcherbakov, M. S. Korytov, and E. O. Shershneva, Mekhatronika, Avtomatizatsiya, Upravlenie **6**, 368–374, (2016).
6. N. N. Panasenko, V. V. Rabey, and L. S. Sinelshchikova, Vestnik of Astrakhan State Technical University **2**, 41–49 (2013).
7. V. I. Mironov, O. A. Lukashuk, and D. A. Ogorelkov, J. MATEC Web of Conferences **129**, 01024 (2017).
8. V. I. Mironov, O. A. Lukashuk, and D. A. Ogorelkov, J. MATEC Web of Conferences **253**, 01004 (2019).
9. S. A. Sokolov, *Structural Mechanics and Metal Structures of Machines* (Politekhnika, St. Petersburg, 2011).
10. V. I. Mironov, D. A. Ogorelkov, and O. A. Lukashuk, J. MATEC Web of Conferences **298**, 00043 (2019).
11. E. V. Khromov, O. V. Khromov, and I. V. Khromov, Fundamental and Applied Problems of Technics and technology **5**, 24–28 (2015).
12. I. S. Kamantsev, A. P. Vladimirov, and E. M. Borodin, Vestnik Tambovskogo Universiteta **18**, 1881–1882 (2013).
13. Ya. G. Panovko, *Basics of Applied Theory of Elastic Vibrations* (Mashinostroenie, Moscow, 1967).
14. A. A. Alyamovsky, A. A. Sobachkin, E. V. Odintsova, etc., *SolidWorks. Computer-Aided Design for Engineering Purposes* (BKhV-Peterburg, St. Petersburg, 2005).
15. K. A. Basov, *ANSYS: User Manual* (DMK Press, Moscow, 2005).