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RESEARCH PAPER

EXPERIMENTAL ANALYSIS OF DAMPING PROPERTIES OF VISCOELASTIC MATERIALS

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ABSTRACT

This study presents results of an experimental investigations of the materials used in passive damping vibrations. The main purpose of this paper was to examine the damping properties of selected viscoelastic materials (VEM), using the modal analysis. In presented analysis three configurations of specimens were considered. At first, the separated steel beam was analyzed. As results of this analysis, the frequencies and amplitudes of the beam during resonance were obtained. In next part of the work the modified specimen was investigated. In this modification the bitumen-based material (as a damper) was fixed to the surface of the beam. This method is known as free layer damping (FLD) treatment. In last configuration, the butyl rubber layer was connected to the steel beam. Using the Unholtz-Dickie UDCO-TA250 electrodynamic vibration system, the natural frequencies and amplitudes of free vibrations for all examined specimens were obtained. The vibration amplitude of the beam was measured using piezoelectric acceleration sensors. In order to define the damping capabilities of both the bitumen based material and the butyl rubber, the relative amplitude of specimens and the loss factor using half-power bandwidth method were calculated.

Keywords: passive vibration damping, experimental modal analysis, viscoelastic material, free layer damping, loss factor.

INTRODUCTION

There is many sources of sounds and vibrations in the aircraft structures and the cars. The main reason of vibrations is an unbalanced engine. The next sources of vibrations are related to: aerodynamics (i.e. fluctuations of pressure in the boundary layer of the stream flowing around the structure) and mechanics (i.e. driving wheeled vehicles on uneven surfaces). The large amplitudes of vibrations can be reason for decrease of the fatigue life of structures. Moreover the high level of noise in the pilots or drivers cabin has an indirect influence on the safety of the air and car transport.

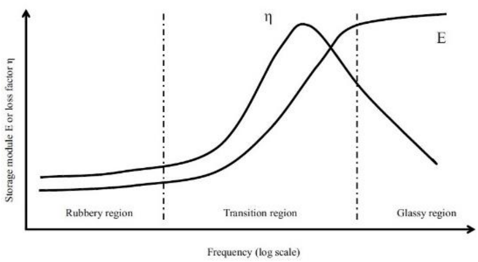

Fig. 1 Loss factor and storage modulus as a function of frequency [1, 6, 14]

From mentioned above reasons there is need to decrease a noise in the cabin, mainly in aviation and the automotive industry. The first way to decrease the vibration of aircraft structures and cars is minimization of vibrations of engine. Unfortunately each turbine engine (and especially the piston engine) cannot be perfectly balanced.

Vibrations are transmitted in various ways, generating unpleasant noise and directly affecting the human body and the vehicle structure. For example, in the aircraft during flight, the compressor blades are subjected to vibration. The resonant vibrations decrease the fatigue life of the blades [5, 15-17]. The most common way of suppressing vibration is attaching to the face of an element the layer made of damping material. Viscoelastic polymers are commonly used in automotive and aviation industry as passive damping [1, 18]. Such materials have both viscous and elastic properties. Because viscoelastic polymers have such features, after removing the load from them, some of the energy is recovered and some is dissipated in the form of thermal energy. Their damping capabilities strongly vary with the temperature and frequency (**Figs 1, 2**). The best damping effect (**Fig. 1**) is achieved in transition region, where the loss factor has the highest value. The storage modulus E' has the highest value in the glassy region.

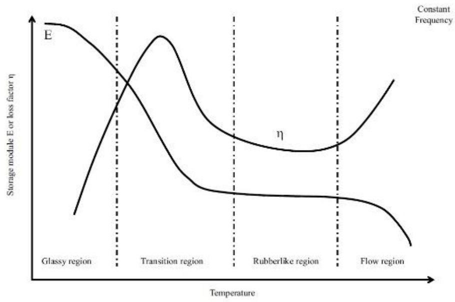

Fig. 2 Loss factor and storage modulus as a function of temperature [1, 6, 14]

The most effective way for using viscoelastic materials are free layer damping (FLD, **Fig. 3**) and constrained layer damping (CLD [18,19, 20]).

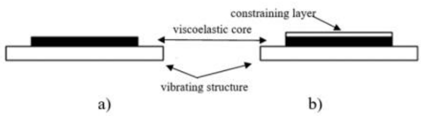

Fig. 3 Viscoelastic materials damping: FLD treatment (a) and CLD treatment (b)

The first configuration dissipates energy caused by vibration, through extension, the second one mostly due to shear. FLD treatment is basically a viscoelastic layer connected with the structure. In CLD treatment a thin constraining layer is added, providing the shear deformation [2]. The aim of those treatments is to minimize vibration amplitude of the structure. In consequence the decrease of both the noise and stress is observed. It also causes extension of the fatigue life. The most common viscoelastic materials used in automotive and aviation industry are butyl rubber and bitumen-based material. The advantage of this approach is no need of external source of power like in active vibration isolation. Viscoelastic materials work best in high range of frequency. In some situations, such passive damping cannot be used at high temperature [7].

Kerwin [8, 9] was one of the first who observed that a stiff constraining layer placed on top of the viscoelastic layer can increase the damping capabilities. In his work, he also examined the influence of temperature on the loss factor. Di Taranto [10] developed an analytical model of a freely vibrating beam with free boundary conditions, which enables the determination of the damping coefficient. Mead and Markus [11] derived mathematical formula determining the transverse displacement of a three-layer beam with a viscoelastic core. Rao [12] presented the formula for the natural frequency and the loss factor for a multi-layer beam under different boundary conditions. Hujare and Sahasrabudhe [13] conducted a study of CLD treatment with various damping materials. The damping properties of seven kinds of viscoelastic materials were investigated. The considered beam was symmetrical in the form of a sandwich structure.

EXPERIMENTAL ANALYSIS

In order to determine the damping capabilities of investigated viscoelastic materials, the experimental modal analysis was performed. During the experiment the Unholtz-Dickie UDCO TA-250 electrodynamic vibration system (equipped with the shaker, controller and the amplifier) was used. In **Fig. 4** the investigated beam covered by bitumen based material was shown. The beam was fixed to the movable head of the shaker.

During the experiment the Oberst beam method was utilized [2]. As a base beam a steel cantilever was used. It is typical material for such analysis, because it has very low loss factor and its storage modulus does not depend on frequency [1].

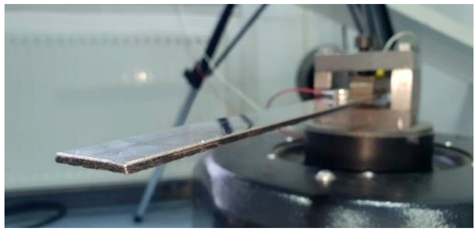

Fig. 4 View of investigated beam covered by bitumen based material. The beam was fixed to the movable head of the shaker using the special grip

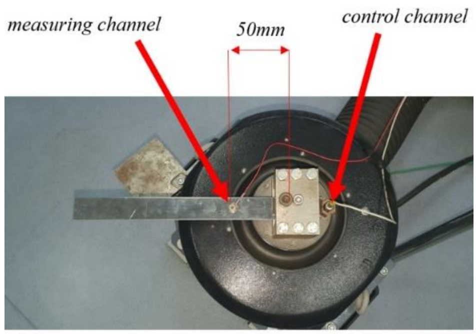

Fig. 5 Top view of the experimental setup. Location of the acceleration sensors used in modal analysis (related to the measure and control channel)

The dimensions of investigated beam were as follows: length 250 mm, width 20 mm, thickness 1 mm. The thickness of damped material was 2 mm. In performed experiment three configurations were studied. In first configuration the steel beam (specimen 1, without damping material) was examined. In second configuration the beam was covered by bitumen-based material (specimen 2). The last examined specimen (no. 3) was the steel beam with layer made of the butyl rubber. The specimens were constructed based on the ASTM Standard E756(05) [3] and were investigated at room temperature (20 °C) in frequency range from 20 Hz to 4 kHz. The modal analysis was performed at constant intensity of excitation 1g (where 1g = 9,81 m/s^2). The intensity of excitation (vibration) was measured using the piezoelectric vibration sensor on the movable head of shaker (**Fig. 5**). Signal from this sensor was assigned to the control channel. The second vibration sensor was located on the beam, 50 mm from the restraint (**Fig. 5**). Signal from this sensor was assigned to the measure channel and was used to creation of frequency-amplitude characteristics, for all investigated specimens.

RESULTS AND DISCUSSION

The results of performed modal analysis are presented as Frequency Response Function (FRF). Based on the ASTM

Standard E756(05) [3] the measure was started from the second mode of resonant vibration of the beam.

Obtained results of experimental modal analysis showed that the best damping of beam is observed for frequencies above 1000 Hz. For first considered modes (from 2nd to 4th) the moderate damping effect is observed. From fifth investigated mode, the strong damping effect of the beam is visible. The quantitative results of performed modal analysis were presented in **Table 1**. In order to compare the damping effect of investigated materials, the relative amplitudes (A2/A1 and A3/A1 were computed for each resonance. In the relative amplitude calculation, the amplitude A2 of damped beam (covered by bitumen material) was divided by amplitude A1 of separated beam (without any damping material).

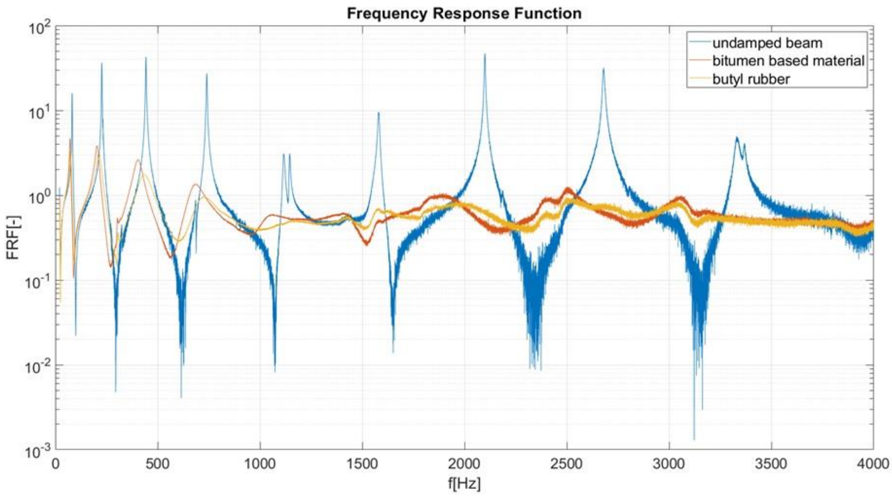

Fig. 6 Frequency response curves for considered specimens

Table 1 Frequency of resonant vibration and amplitude of acceleration of specimens

Specimen 1 (beam without damping)			**Specimen 2 (beam and bitumen-based material)**			**Specimen 3 (beam and butyl rubber)**		
No. of mode	Freq. of resonant vibration [Hz]	Ampl. of accel. A_1 [g]	Freq. of resonant vibration [Hz]	Ampl. of accel. A_2 [g]	Relative amplitude A_2/A_1 [-]	Freq. of resonant vibration [Hz]	Ampl. of accel. A_3 [g]	Relative amplitude A_3/A_1 [-]
2	81.3	15.89	71.9	4.674	0.294	74.0	3.502	0.220
3	226.8	36.27	203.7	3.852	0.106	212.1	3.128	0.086
4	444.5	42.36	403.2	2.6	0.061	434.4	1.763	0.042
5	741.5	27.2	686.2	1.351	0.050	730.5	0.9606	0.035
6	1115.8	3.098	1059	0.59	0.190	1169	0.5082	0.164
7	1146.4	3.097	1408	0.6289	0.203	1435	0.5425	0.175
8	1584.3	9.591	1581	0.521	0.054	1581	0.7061	0.074
9	2100	47.21	1879	1.073	0.023	1982	0.8722	0.018
10	2680	31.79	2503	1.266	0.040	2501	0.9557	0.030
11	3331	4.957	3059	0.9891	0.200	3057	0.8526	0.172
12	3368	4.084	3185	0.7136	0.175	3185	0.5941	0.145

Obtained results of investigations (**Fig. 6**, **Table 1**) showed that after adding the damping mats to the beam an amplitude of acceleration (during resonances) significantly decreased. The results presented in **Table 1** shows that the butyl rubber provides (for most modes) a bit more effective damping than the bitumen material. Both the butyl rubber and the bitumen material achieve the best results in damping at the following modes of vibration: 4, 5, 8-10. For these modes the amplitude of acceleration was decreased more than 10 times.

For better estimation of the damping factor from frequency domain, a half-power bandwidth method was used. In this method the amplitude of FRF was used. In first part of the procedure the amplitude A_{max} and frequency f_o for each resonance were extracted (**Fig. 7**). In next step the value of A_{max} was divided by √2. The vertical line (which has coordinate $A_{max}/\sqrt{2}$) intersects the curve at 2 points. The first coordinate of intersection points are: f_1 and f_2. These frequencies are used in half power bandwidth method (**Fig. 7**).

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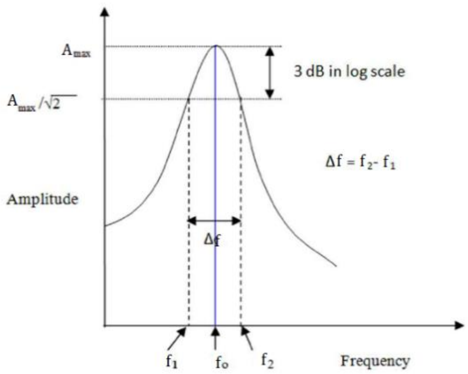

Fig. 7 Visualization of half power bandwidth method [4]

Structural loss factor η is equal to twice the viscous damping factor ζ [1]. Using equation shown below, the loss factors have been computed for every investigated mode of resonant vibration.

$$\eta = 2\zeta = \frac{\Delta f}{f_0} = \frac{f_2 - f_1}{f_0} \quad (1.)$$

where: η [-] – loss factor

ζ [-] – damping factor

f_0 [Hz] – natural frequency

f_2, f_1 [Hz] –frequencies used in half power bandwidth method (**Fig. 7**)

The frequencies f_1 and f_2 were next used for calculation of the damping capabilities of examined materials. The structural loss factor and the damping factor for considered beams are presented in **Table 2**.

Obtained results of investigations (**Fig. 6**, **Table 2**) showed that after adding the damping mats to the beam, the loss factor and the damping factor significantly increased. The beam without damping (specimen 1) has very low loss factor for all modes. The small value of loss factor is typical for the steel, which dissipates very much of energy during vibration. Performed experiment showed that both the butyl rubber and the bitumen material increased the structural damping significantly. For modes in medium frequency range the loss factor achieves the highest values (**Table 2**).

In **Fig. 8** the relative damping factor of specimen no. 2 and 3 was shown. From this figure is visible that for specimen 3 (beam with butyl rubber) the highest values of relative damping were observed. It means that the butyl rubber has better damping properties than the bitumen material. The highest values of relative damping factor were observed for modes 4-7 and 9.

Table 2 Loss factor, damping factor and relative damping factor of specimens

Specimen 1 (beam without damping)			**Specimen 2 (beam and bitumen-based material)**			**Specimen 3 (beam and butyl rubber)**		
No. of mode	Loss factor (η_1) [-]	Damping factor (ζ_1) [-]	Loss factor (η_2) [-]	Damping factor (ζ_2) [-]	Relative damping factor ζ_2/ζ_1 [-]	Loss factor (η_3) [-]	Damping factor (ζ_3) [-]	Relative damping factor ζ_3/ζ_1 [-]
2	0.032	0.016	0.067	0.033	2.1	0.095	0.047	3.0
3	0.011	0.006	0.083	0.042	7.6	0.118	0.059	10.7
4	0.006	0.003	0.104	0.052	18.5	0.147	0.074	26.2
5	0.005	0.003	0.112	0.056	20.8	0.159	0.079	29.4
6	0.008	0.004	0.154	0.077	19.1	0.256	0.128	31.7
7	0.005	0.003	0.131	0.065	25.0	0.132	0.066	25.3
8	0.005	0.003	0.021	0.010	4.1	0.025	0.012	4.9
9	0.002	0.001	0.103	0.052	48.2	0.151	0.075	70.4
10	0.003	0.002	0.033	0.016	9.8	0.053	0.026	15.7
11	0.007	0.003	0.046	0.023	7.0	0.050	0.025	7.5
12	0.004	0.002	0.026	0.013	5.9	0.038	0.019	8.6

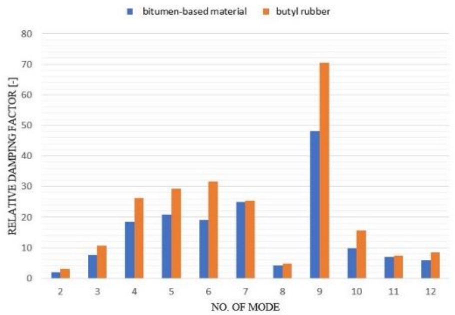

Fig. 8 Relative damping factor of specimen no. 2 and 3

CONCLUSIONS

In this study the experimental modal analysis of beams covered by the viscoelastic material was performed. Viscoelastic materials, especially the bitumen and the butyl rubber are often used in automotive and aviation industry in order to damping the vibrations of thin-walled structures. In this work three specimens were examined (steel beam without damping material, the beam with bitumen material and the beam covered by the butyl rubber layer). During experimental investigation both the natural frequencies and also the vibration amplitude of examined specimens were obtained. In order to define the damping capabilities the relative amplitude and also the loss factor using half-power bandwidth method were computed. Obtained results of investigations (**Fig. 6**, **Fig. 8**, **Table 1**, **Table 2**) showed that after adding the damping mats to the beam an amplitude of acceleration (during resonances) significantly decreased. The results presented in **Tables 1** and **2** shows that the butyl rubber provides (for most modes) a bit more effective damping than the bitumen material. The highest loss factor is achieved in medium frequency range. Obtained results are important from both the research and also the practical point of view because the passive damping of vibration is used in many branches of industry (automotive, aviation, railway technology). Reduction of vibration amplitude causes increase of the fatigue life of thin-walled structures. Moreover, the passive damping of vibration causes reduction of noise in cabins of cars and aircraft what indirectly affects the safety of transport.

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