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An analytical and experimental study on dampening material effects on the dynamic behavior of free-free aluminum sheets

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This work aims to present an experimentally verified analytical solution to examine damping properties of systems including viscoelastic treatments. Although there are several methods for characterizing the behavior of three-layer damping systems, the RKU method is the most frequently used one. In this paper, this method is modified such a way that to be applied for a five-layer damping system. The achieved analytical relations are then employed to study the effects of a four-layer vibration-absorbing coating on the dynamic behavior of an aluminum sheet with free-free boundary conditions. Since the vibration-damping properties of the coating are unknowns, its loss factor and shear modulus are experimentally extracted based on the ASTM E756-05 standard method. The comparison between the analytical solution and performed modal tests expresses the efficiency of the presented method.

Nomenclature and Abbreviations

- Nomenclature

E_i	Elastic modulus corresponding to *i*th layer
G_2^*	Complex shear modulus of damping material
ν	Poisson's ratio
EI^*	Complex flexural rigidity
ρ	Density
H_i	Thickness of the *i*th layer (as depicted in Fig.1)
η	Loss factor
k	Wave parameter
n (subscript)	Corresponds to the n^{th} mode of vibration of bare steel beams (section 4)
s (subscript)	Corresponds to the s^{th} mode of vibration of sandwich beam (section 4)
v (subscript)	Corresponds to the damping material (section 4)
i (subscript)	Corresponds to the i^{th} mode of vibration of a thin sheet (section 5.1)

- Abbreviations

RKU	Ross, Kerwin and Ungar
CLD	Constrained Layer Damping
CDC	Constrained, Damping and Constrained Layers

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1. Introduction

Achieving high-quality in-flight service and creating conditions for the passengers to travel in comfort, have always been competitive subjects between airlines. The noise phenomenon is a major problem in airplanes that disturbs the human hearing system and so, corresponding standards have set limits for preventing those problems. Therefore, the investigations on the developing methods for reducing the unfavorable outcome of vibration phenomena have been always in progress.

Emitted noise in an airplane depends on the way of transmission which is usually classified as structure-born noise and air-borne noise. In the structural way, one of the main cases that causes noise, is the vibration of body panels. Besides to existing annoying noise inside the cabin due to structural vibration, it causes fatigue in structure and should be considered in the design phase. Vibration absorber coatings are helpful products to prevent the structural way of noise transmission. The purpose of employing vibration absorber coating is to increase the damping of structures to a secure level of reliability. In rubber-like materials, such as plastics and elastomers, the high damping property is accompanied by low and unfavorable stiffness. Thus, in those cases in which both properties are important, a smart merging process of attaching a high damping material to a high stiffness material may form an ideal combination.

More than half a century ago, engineers began to develop the ability to measure the properties of damped systems. Primary models were presented for a simple beam and then non-beam structures such as sheets and panels. As reported in (Jones, 2001), Ross, Ungar, and Kerwin offered the RKU method to analyze viscoelastic material damping effects on the vibrational behavior of structures. The RKU method which has been formulized in terms of complex stiffness of a three-layered beam (i. e. main material, damping layer, and constraining layer) together with assuming pure sinusoidal mode shapes, still is the basis of many analytical works. This method is a tool for quick estimation of effect of different CLD systems parameters (i.e., dampening layer material, constraining layers material, thickness of each layer and etc.) on vibrational response of structurs. In other word, high-speed response along with proper accuracy for preliminary design of damping systems is the main benefit of employing RKU method in comparison with other available methods.

Jones (2001) discussed how Rao experimentally developed a correction factor to predict damping properties of a simple beam with different boundary conditions using the RKU method. Nashif et al. (1985) also expressed a method to extend the RKU relations for a simply supported sheet. They presented a methodology for dealing with systems containing more than one damping layer and/or one constraining layer. Teng et al. (1996) studied the optimal design of a dynamic absorber by using polymer-laminated steel sheets. According to the optimal tuning and damping rate, they obtained the thickness of constraining layers and the interlayer by employing the RKU model. Ren et al. (2011) proposed a variant method for the characterization of MPS beams. The proposed optimization method which is based on simple experimental setup, updates the four parameters of the fractional derivative model by using the RKU equation, the normal mode superposition method, and experimental FRFs. Numerous research have been conducted about the effects of viscoelastic damping materials on noise and vibration reduction of various structures (Lewandowski and Baum, 2015; Park, 2001; Ovaert et al., 2003; Kurano, 2005; Ghayesh and Moradian, 2011; Rossikhin and Shitikova, 2008; Fan et al., 2009; Kumar and Singh, 2010; Saidi et al., 2011). Zhou et al. (2016) presented a comprehensive review of the various research methods and theory to study the vibration characteristics of viscoelastic damping material. Many authors have recently investigated different problems in the field of sandwich panels with viscoelastic core (Sheng et al., 2018; Biswal and Mohanty, 2019; Claude et al., 2019; Kordkheili and Khorasani, 2019; Zhang et al., 2020). In particular, there is a huge number of works dealing with constrained viscoelastic layer (Rouleau et al., 2018; Biswal and Mohanty, 2018; Zarraga et al., 2020; Gupta et al., 2020; D'Ottavio et al., 2020).

This work presents a modified analytical method to investigate dampening material effects on the dynamic behavior of free-free aluminum sheets based on the RKU method. The novelty of this work comes from the limitation of the RKU method, in which, although it is compatible with long and pinned-pinned three layered beams, it can't be used for other situations directly. Also, the formulation is modified for a five layer damping system by introducing an equivalent CLD system. The motivation of this work originates from the reduction of the noise level in an airplane employing an unknown viscoelastic coating which is pasted on aluminum sheets. To extract vibration-damping properties of this available four-layer viscoelastic coating, some experiments are performed based on ASTM E756-05 standard method (Standard, 2010). This viscoelastic coating effects on the dynamic behavior of a free-free aluminum sheet is investigated analytically and the results are verified experimentally.

2. Problem description

An aluminum sheet with 23 *cm* × 30 *cm* × 0.3 *cm* dimensions and free-free boundary conditions (see section 5.2 for test setup details) together with a constrained layer damping system is considered. The CLD system contains two aluminum foils and two viscoelastic layers with 0.1 *mm* and 0.4 *mm* thicknesses for each layer, respectively (Fig. 1a). The vibration-damping properties of this CLD are unknown.

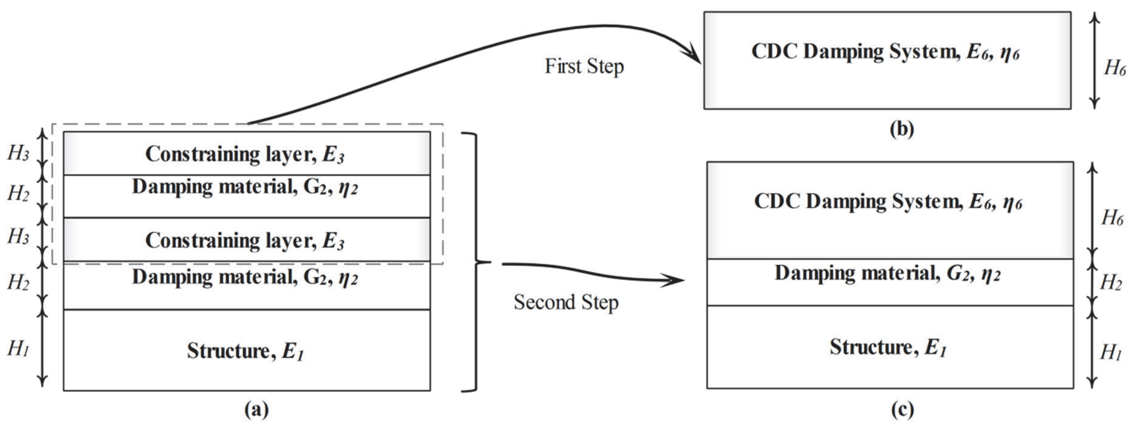


Fig. 1. a) Five layer damping system, b) Replacing the three upper layers with a CDC system, c) Equivalent CLD with three layers

3. Modified analytical method

There are several approaches to describe the behavior of a system with a damping treatment. The most common method in the vibrational application is the RKU method (Nashif et al., 1985). The complete set of RKU method equation for a CLD system is written in a form as bellow

$$\begin{aligned} EI^* = {} & E_1\frac{H_1^{\,3}}{12} + E_2\frac{H_2^{\,3}}{12} + E_3\frac{H_3^{\,3}}{12} - E_2\frac{H_2^{\,2}}{12}\left(\frac{H_{31}-D}{1+g}\right) + E_1H_1D^2 + E_2H_2(H_{21}-D)^2 \\ & + E_3H_3(H_{31}-D)^2 - \left[\frac{E_2H_2}{2}(H_{21}-D) + E_3H_3(H_{31}-D)\right]\left(\frac{H_{31}-D}{1+g^*}\right) \end{aligned} \tag{1}$$

where subscripts 1, 2 and 3 are related to the base structure, damping material and constrained layer, respectively (also see Fig. 1 for the definition of the parameters). Other unknown parameters are defined as follows

$$D = \frac{E_2H_2\left(H_{21} - {}^{H_{31}}\!/_{2}\right) + g^*(E_2H_2H_{21} + E_3H_3H_{31})}{E_1H_1 + {}^{E_2H_2}\!/_{2} + g^*(E_1H_1 + E_2H_2 + E_3H_3)} \tag{2}$$

$$H_{21} = \frac{H_1 + H_2}{2}, \qquad H_{31} = \frac{H_1 + H_3}{2} + H_2, \qquad g^* = \frac{G_2^*}{E_3 H_3 H_2 k^2}$$

also G_2^* is the shear modulus of damping material and k is the wave parameter in which depends on the system boundary conditions. The output from RKU relation is the flexural rigidity, *EI**, which is in general a complex expression. Thus bending stiffness and equivalent loss factor of the system are calculated by

$$EI = Re(EI^*), \qquad \eta = \frac{Im(EI^*)}{Re(EI^*)} = \frac{Im(EI^*)}{EI} \tag{3}$$

The fundamental limitation of this method is that the RKU equations are just applicable for a three-layer system with pinned-pinned boundary conditions.

In this section, the idea to solve the problem of extracting damping properties for free-free boundary conditions and for the case of five-layer damping treatment systems is discussed. Two approaches may be employed for this purpose; 1) knowing from experiments, the first damping layer resists most shear deformations however other layers generally act to increase the total stiffness of the system (Jones, 2001). Therefore, as the first approach one may simply replace the five layers damping system with three layers one by directly neglecting other damping layers. The thickness of the constraining layer is assumed to be the sum of layers 3 and 5 thicknesses. 2) In the second approach, which is developed here, to achieve a modified RKU equation, the three outer layers of the damping system are firstly replaced by an equivalent damping system (Fig. 1). Then this CDC damping system along with the base structure and the other damping layer make a new three-layered damping system (equivalent CLD) in which employ to extract damping properties.

Based on the second aforementioned approach, the flexural rigidity of the CDC system can be calculated and presented as follows

$$(EI^*)_6 = \frac{1}{6} E_3 H_3 (H_3^2 + 6A_6 B_6^2 + 12 A_6^2 B_6^2) \tag{4}$$

where

$$A_6 = \frac{g_6^*}{1 + g_6^*}, \qquad B_6 = H_2 + H_3, \qquad g_6^* = \frac{1}{k^2} \frac{G_2^*}{E_3 H_2 H_3} \tag{5}$$

From Eq. 4, the equivalent Young modulus and loss factor of this new layer (CDC system) are extracted as follows

$$\begin{aligned} E_6 &= \frac{1}{I_6}(EI)_6 = 2E_3 H_3 \frac{12 A_6^2 B_6^2 + 6 A_6 B_6^2 + H_3^2}{H_6^3} \\ \eta_6 &= \frac{Im(EI^*)_6}{Re(EI^*)_6} \end{aligned} \tag{6}$$

where $H_6 = H_2 + 2H_3$ is the equivalent thickness of the CDC system. Now, using Eq. (1) and Eq. (4) the total flexural rigidity of the new three-layered damping system consisting of the base structure, damping layer, and the CDC damping system can be calculated as follows

$$(EI^*)_{eq} = E_1 H_1 \left(\frac{1}{12} H_1^2 + (AB)^2\right) + \Lambda \left(\frac{1}{12}(H_2 + 2H_3)^2 + B^2(1 - A^2)\left(\frac{g^*}{1 + g^*}\right)\right) \tag{7}$$

where

$$\begin{aligned} \Lambda &= 2E_3 H_3 \frac{12\left(\frac{g_6^*}{1+g_6^*}\right)^2 (H_2 + H_3)^2 + 6\left(\frac{g_6^*}{1+g_6^*}\right)(H_2 + H_3)^2 + H_3^2}{(H_2 + 2H_3)^2} \\ A &= \frac{g^* \Lambda}{E_1 H_1 (1 + g^*) + g^* \Lambda}, \qquad B = \frac{H_1 + 3H_2 + 2H_3}{2} \\ g^* &= \frac{g_6^*}{\Lambda} E_3 H_3, \qquad g_6^* = \frac{1}{k^2} \frac{G_2^*}{E_3 H_2 H_3} \end{aligned} \tag{8}$$

Also, η_2 which is viscoelastic material damping could be achieved from the experiment (see section 4). As Young modulus of the viscoelastic material is much less than the aluminum one ($E_2 \ll E_1$), therefore it has vanished during formulation development.

4. Extracting vibration-damping properties of the viscoelastic material

The vibration-damping properties of the considered viscoelastic coating are extracted using ASTM E756-05 standard test method. For this purpose a sandwich specimen is constructed with two 250 *mm* × 30.5 *mm* × 1.4 *mm* steel beams and a viscoelastic layer with total thickness H_v=1.0 *mm* as depicted in Fig. 2. A rigid fixture is also designed to fix one end of the sandwich specimen properly. Fig. 3 shows the test setup which includes a hammer, a PCB accelerometer, the sandwich specimen, cables, and the fixture.

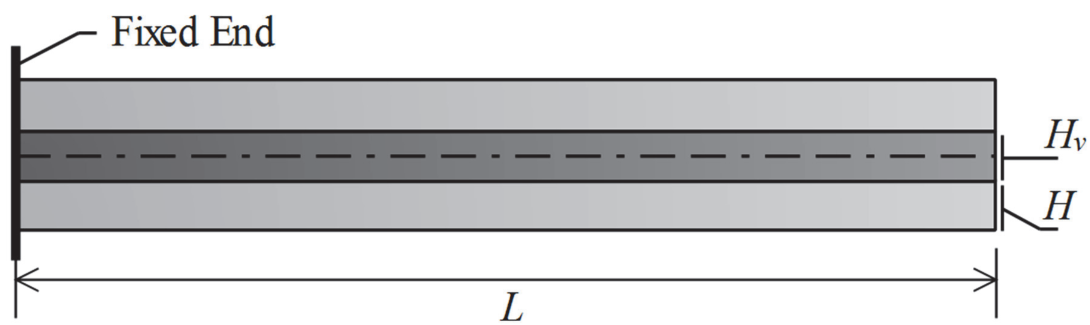


Fig. 2. A sandwich specimen to obtain vibration-damping properties of viscoelastic material

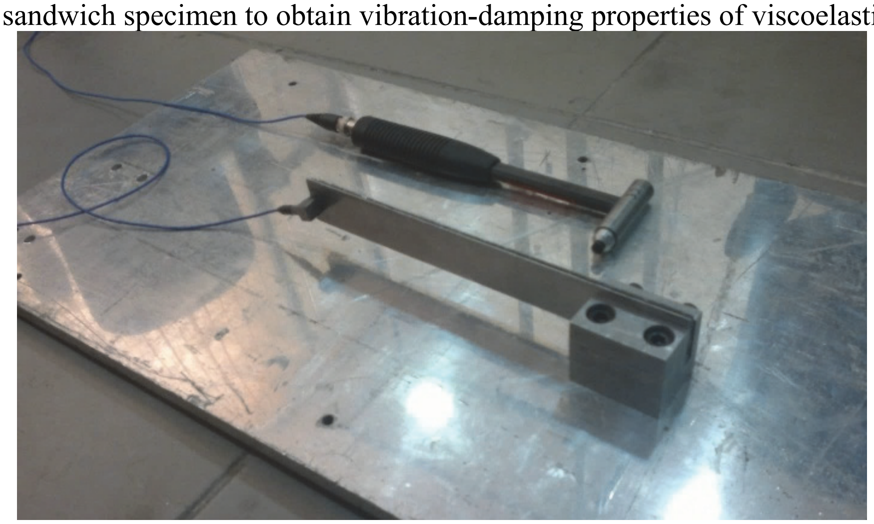


Fig. 3. A setup assembly to examine vibration-damping properties of the viscoelastic material

To determine the damping properties of the viscoelastic material, at first, the natural frequencies of the two bare steel beams are obtained via performing modal tests. These tests results are then employed to compute the elastic modulus and loss factors of the beams using relations $E_n = 12\rho L^4 f_n^2 / H^2 C_n^2$ and $\eta_n = \Delta f_n / f_n$. In these relations n refers to the n^{th} mode of the beams, η_n is found using the Half-power bandwidth method and $C_1 = 0.55959$, $C_2 = 3.5069$ and $C_n = \left(\frac{\pi}{2}\right)(n - 0.5)^2$ for $n \geq 3$. Similarly, the constructed sandwich specimen is tested to find its natural frequencies and the corresponding elastic modulus and loss factors. Now, the required vibration-damping properties of the viscoelastic material are extracted using the following relations

$$G_v = [a - b - 2(a-b)^2 - 2(a\eta_s)^2]\,\frac{2\pi C_n E H H_v}{L^2((1-2a+2b)^2 + 4(a\eta_s)^2)}$$
$$\eta_v = \frac{a\eta_s}{a - b - 2(a-b)^2 - 2(a\eta_s)^2} \tag{9}$$

where subscript s refers to the s^{th} mode of vibration of the sandwich specimen and other parameters are defined as follows

$$a = \left(\frac{f_s}{f_n}\right)^2 \frac{(2+\alpha T)b}{2}, \qquad b = \frac{1}{6(1+T)^2}, \qquad \alpha = \frac{\rho_v}{\rho}, \qquad T = \frac{H_v}{H} \tag{10}$$

Fig. 4 depicts the shear modulus and loss factor of the considered viscoelastic material versus frequency. From this figure, one may observe that by increasing in frequency, the shear modulus of the viscoelastic material increases but its loss factor decreases, both nonlinearly. It should be noted that these results are achieved in the laboratory temperature.

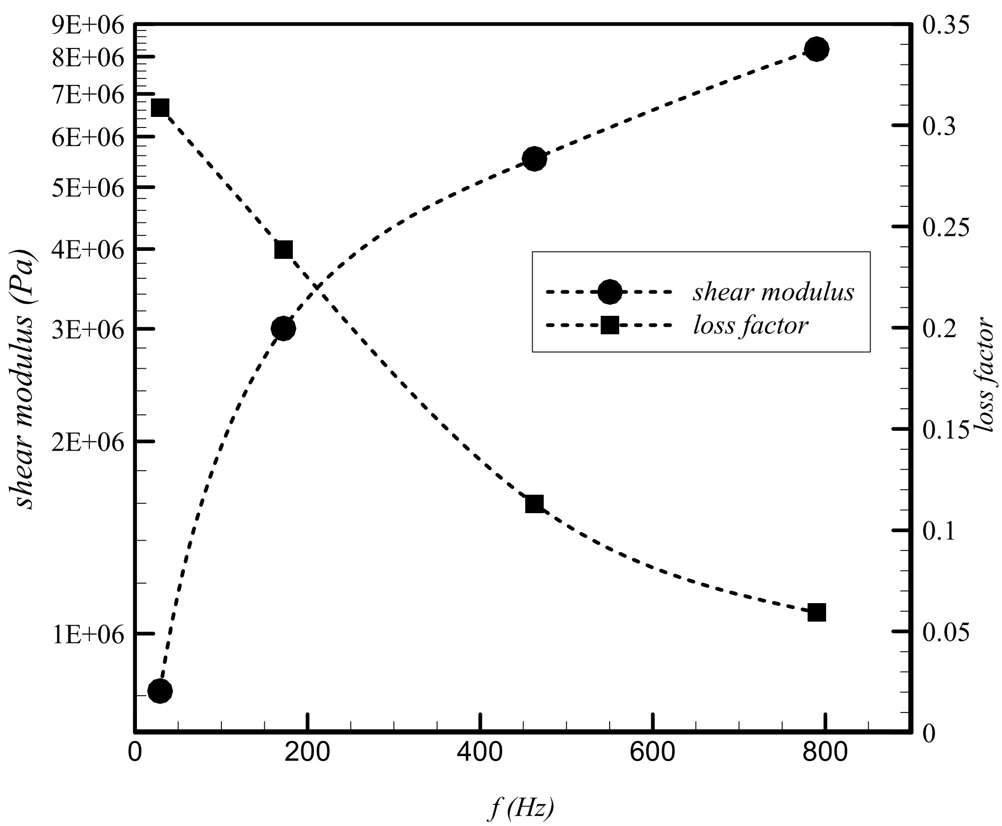


Fig. 4. Shear modulus and loss factor of the viscoelastic material verses frequency.

5. Results and discussions

5.1. Extracting damping properties for the problem; analytically

To extract damping properties of the problem (itemized in section 2) by employing previously derived relations together with extracted damping properties of viscoelastic material, we need to calculate the wave parameter k_i corresponding to i^{th} natural frequency of the sheet. In general, the natural frequency of a thin sheet could be extracted using the following equation

$$f_i = \frac{k_i^2}{2\pi}\sqrt{\frac{EH^2}{12\rho(1-\nu^2)}} \tag{11}$$

Knowing that $EI = EH^3/12(1-\nu^2)$ for thin plats, yields a relation for natural frequency independent from Poisson's ratio as follows

$$f_i = \frac{k_i^2}{2\pi}\sqrt{\frac{EI}{\rho H}} \tag{12}$$

According to this equation, wave parameters may be estimated using natural frequencies of the aluminum sheet. Table 1 listed the first six natural frequencies and their corresponding mode shapes and wave parameters for the free-free sheet which are computed numerically.

Now using these wave parameter values and measured CLD system properties, we can calculate EI^* using Eq. (7) for our five-layer system. The real part of the calculated EI^* is then used to extract EI in relation to each natural frequency. Later on, we employ these data in relation (12) to recalculate new values of the natural frequencies. These data describe the effects of damping material on the natural frequencies of the sheet which are listed in Table 1. The imaginary part of EI^* gives the loss factors in relation to natural frequencies that are also listed in the same table. Regarding to these results, the effect of damping material on higher natural frequencies is more considerable. However the system's loss factor decreases by increasing in its natural frequency.

5.2. Extracting damping properties for the problem; experimentally

Modal testing is an essential test for vibration behavior estimation of structures and sub-structures. Here damping properties of the bare, as well as coated aluminum sheets, are extracted and compared. Fig. 5 shows the test equipment including analyzer, shaker, force transducer, and accelerometer. For excitation of the structure, a burst random signal with frequency span of 0-800 Hz is employed. Burst random excitation is an improved version of pseudo excitation with random amplitude and phase spectrum and contains energy throughout the frequency range (Fu and He, 2001). It is noted that the excitation must be perpendicular to the suspension system, which is considered. Fig. 6 shows the frequency response functions, FRFs, diagrams for the two considered cases of aluminum sheets with and without viscoelastic coating. From this figure, it could be seen clearly that for the same natural frequency, the bandwidth of the coated case increases. This observation indicates that damping of the structure is grown in presence of viscoelastic coating. On the other hand, the diagram for the case of "in presence of coating" is shifted to the left of the bare sheet diagram. It means that, after the coating of the sheet, the effect of added mass is more than the induced stiffness to the system, therefore it leads to lower natural frequencies.

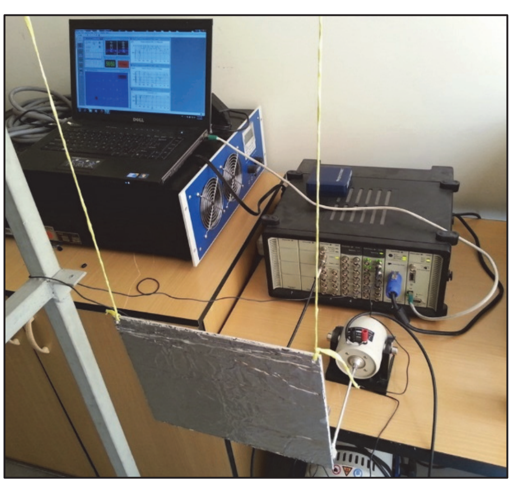


Fig. 5. Modal analysis test setup

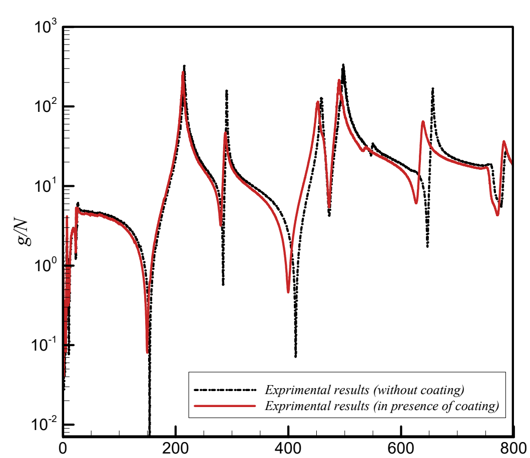
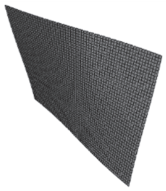
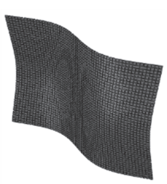


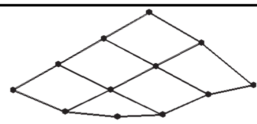
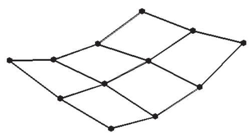
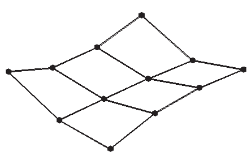
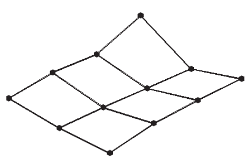
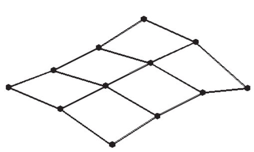
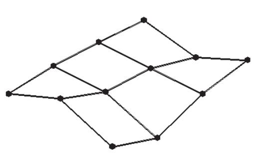
Fig. 6. A comparison between FRF diagrams of Aluminum sheets

Table 1. Damping properties for the problem which are extracted analytically

Mode No.	***Mode Shape***	***Bare sheet natural freq. (Hz)***	***Wave parameter (k)***	***Coated sheet natural freq. (Hz)***	***Decrease in natural freq. (%)***	η
1		142.6	13.89	137.6	3.50	0.01326
2		173.9	15.37	168.2	3.28	0.01285
3		310.0	20.5	298.0	3.87	0.01037
4		339.3	21.5	326.0	3.83	0.00975
5		409.1	23.6	392.0	4.15	0.00826
6		506.3	26.2	483.0	4.55	0.00634

In Table 2, the natural frequencies and corresponding loss factors are compared. Also in Fig. 7, extracted loss factors using analytical and experimental solutions are compared together. In this figure, two sets of analytical results are depicted, i.e. with considering Aluminum sheet structural damping and without considering its damping. It means after calculating Aluminum sheet damping values experimentally, employing these data we extracted the sheet loss factors using analytical relations again. In this regards, η_1 is achieved from modal testing and the 'E_1' parameter in Eq. (6) is replaced by 'E_1 ($1+i\eta_1$). Fig. 7 yields that when we implement the sheet's structural damping during the analytical solution, the trend of the results will be in accordance with those from experiments for the first five modes. The main reasons of difference between analytical and experimental results correspond to the inherent simplification in analytical solution (in base RKU method and the modified one), errors and uncertainty in experimental method parameters (both in equipment and implementation) and number of parameters for describing viscoelastic material behavior, extracted from ASTM E756-05 method.

Table 2. Damping properties for the problem which are extracted experimentally

Mode No.	***Mode Shape***	***Bare sheet***		***Coated sheet***		***Decrease in natural freq. (%)***	***Increase in loss factor (%)***
		natural freq. (Hz)	*Loss factor (η)*	*natural freq. (Hz)*	*Loss factor (η)*		
1		212.7	0.00690	211.8	0.01168	0.41	69.2
2		288.5	0.00326	285.1	0.01368	1.19	319.0
3		456.1	0.00991	453.4	0.01557	0.59	57.1
4		497.6	0.00210	490.6	0.00933	1.41	344.0
5		532.7	0.00258	528.3	0.00389	0.83	51.0
6		646.8	0.00448	633.6	0.01186	2.03	164.6

As can be observed from Fig. 8, another effect of viscoelastic coating on the structures is a loss in the amplitude of vibration which may have a significant effect on mechanical fatigue of the structures. Using these figures acceleration amplitudes for two different cases are obtained and then displacement amplitudes of responses are achieved by dividing the acceleration to the square of the corresponding angular frequency value of the modal test. Table 3 lists these data and compares between maximum displacement amplitude for two considered cases. From these results, it can be noted that loss in displacement amplitude is about 47.7% in presence of coating.

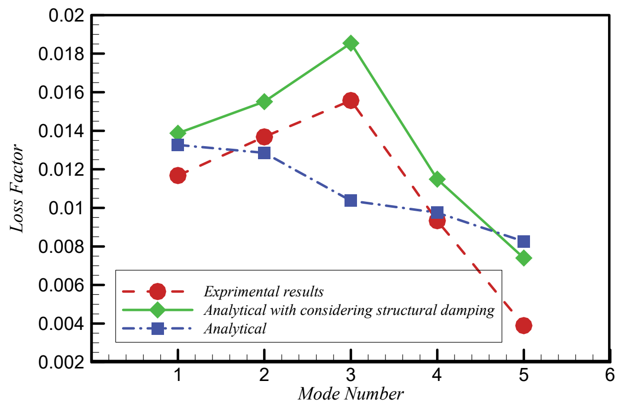
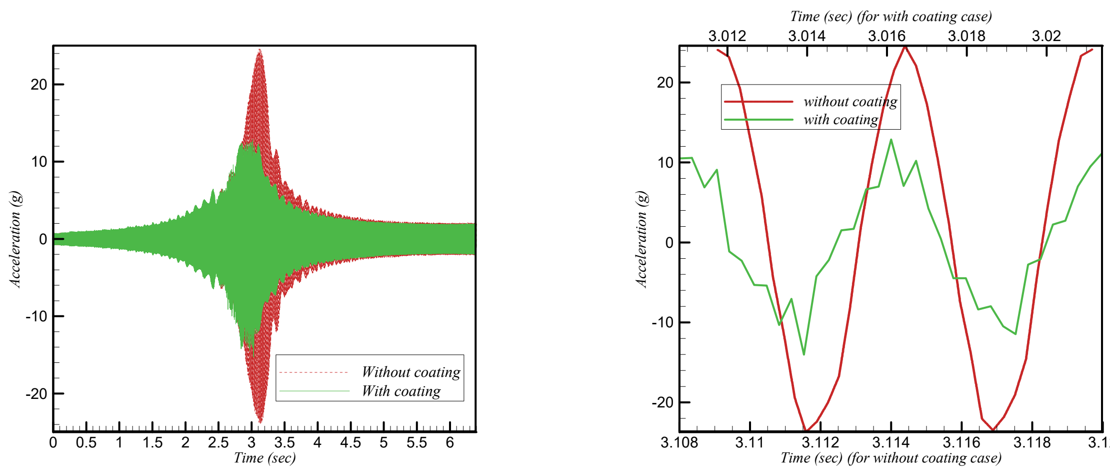


Fig. 7. A comparison between loss factors



(a) (b)

Fig. 8. Sine swept excitation results, a) first natural frequency b) A zoomed view

Table 3. Comparison between maximum displacement amplitude in the first mode

Case	*Acceleration amplitude (g)*	*Displacement amplitude (mm)*
Bare sheet	24.6	**0.1718**
Coated sheet	12.86	**0.0899**
Decrease in displacement amplitude		**47.7 %**

5. Conclusions

In this research, a modified analytical method for a 5–layer damping system with a free-free boundary condition is developed in the framework of RKU relations. The results from the provided relations for an aluminum sheet accompanying by a 4-layer vibration damping coating are confirmed by experiment. Also, the process of extracting frequency-dependent shear modulus and loss factor of the dampening material of the utilized coating is carried out based on the ASTM E756-05 standard. As the presented results express, vibration-damping coatings generally have two major effects on a structure which include increasing the loss factors of the system and decreasing the natural frequencies. Also, from the performed sine swept test, it is obvious that how and how much the CLD coating is helpful in the reduction of vibration amplitude, which is very essential in design considerations from the perspective of fatigue phenomenon. Therefore, it is important to offer reliable relations to predict the effects of employing vibration absorber coating. The results obtained by the presented analytical solution prepare a good vision on how the vibration damping coating will affect the dynamic behaviors of free-free aluminum sheets. It should be emphasized that when the Aluminum sheet structural damping is taken into account in analytical solution, the trend of the results becomes similar to those obtained from experiments.

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