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VIBRATION ATTENUATION IN COMPOSITE LAMINATES BY VISCOELASTIC MATERIALS

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Abstract. *Due to the uncertainty of its origin, which can originate from a number of different sources, the problem of damping in materials has not yet been fully resolved. Also, composite materials are increasingly becoming a part of everyday systems displacing metals as a construction material due to their exceptional characteristics of strength, flexibility and lightness. The uses span the automotive industry to aerospace, where they are employed to construct items like pressure containers, fairing sections, wind turbine blades, and aircraft wings. When loads and vibration modes are in resonance, problems might occur in composite material constructions. With proper damping control, this problem can be somewhat mitigated. In this article, it was conducted a campaign of experimental tests to quantify damping in composite materials and its connection to layers of viscoelastic material. In the end, an increase of about 168% was possible to be obtained using self-adhesive aluminum tape, which represents a good figure for practical applications when compared to passive attenuator devices.*

Keywords: *composite material, damping identification, viscoelastic material, mechanical vibrations, Hilbert transform.*

1. INTRODUCTION

Composite materials have received a lot of attention and use in recent years, due to their characteristics of high strength and very low weight when compared to the usual metallic materials used by the industry for the manufacturing of parts and equipment. Its employment is vast, ranging from the aerospace, aeronautical, nautical and automotive sectors. Fatigue problems have been reported when using composite materials in apparently usual situations, which leads to imagine resonance situations being the cause of these failures. Solutions for attenuation and vibration exist in the most diverse forms, depending on whether the interest is to mitigate the transmission of vibration (in this case by isolation), directly eliminate vibration in the causative sources or even just attenuate the vibration in active, passive or semi-active form, with the use of devices coupled to parts and equipment. The most direct solution to attenuation is by means of stiffening, if the excitation is of a lower frequency than the resonance of the part, or by damping, if this excitation is greater than that of resonance. Investigations in the literature have shown that the damping in parts of composite materials is, in general, low, due to its high stiffness and low weight, however still higher than in metallic materials such as steel or aluminum. It is possible, within the feasible limits, to control this vibration relatively cheaply, by controlling the damping.

The damping ratio is a critical parameter in the analysis and design of composite structures, as it directly influences their dynamic behavior and response to external loads. Damping in composites can be attributed to various mechanisms, including viscoelasticity, interfacial friction, and energy dissipation within the constituent materials. Understanding and optimizing damping characteristics are essential for applications such as aerospace and civil engineering, where structural integrity and vibration control are paramount.

This work aims to study the structural behavior regarding the damping of composite materials, applying to them viscoelastic materials, pursuing the attenuation of vibration. By an experimental study, with the use of damping identification techniques, different viscoelastic materials applied to the composite laminate in different positions and situations.

2. LITERATURE REVIEW

The status on composite materials damping nowadays may be summarized by the pursuit of the several sources of energy dissipation in such materials. On source is the visco-elasticity of matrix and/or fiber materials that primarily contribute to composite damping, but high-damping fibers like carbon and Kevlar also play a role. Another investigated source is the interphase damping, Gibson et al. (1991), the region near the fiber surface, that has unique properties that affect the mechanical properties and damping behavior of composite materials. The nature of the interphase, whether weak, ideal, or strong, has implications for damping. Also the damping caused by damage, Nelson and Hancok (1978),

by the means of frictional damping due to slipping between fiber-matrix interfaces, delamination, and energy dissipation in regions with matrix cracks or damaged fibers. Moreover, the inherent damping by viscoplastic behavior at high vibration amplitudes or stress levels, Kenny and Marchetti (1995), that thermoplastic composite materials exhibit, may cause non-linear damping due to stress and strain concentration in local regions. And finally the thermoelastic damping according to Hashin (1970), generated by cyclic heating from the compressive stress region to the tensile stress region, particularly in thermoplastic composites under cyclic loads. The extent of temperature increase depends on various factors like load, sample thickness, and stress amplitude of cycles.

In 2001, Tita et al. (2001) investigated a procedure to evaluate the damped behavior of beams reinforced with fiberglass in vibration by bending. Damping ratios are evaluated for experimental samples of fiberglass and epoxy resin $[(\pm 45)_4 \, (0/90)]_s$. Modal tests with impact hammer, accelerometer and spectrum analyzer are used for this. Damping ratios (ζ (ω)) of the order of 0.04, 0.034 and 0.026 were obtained for the first three bending modes of free-set beams (0.4 m x 0.025 m) of thickness 1.8 mm and mass 0.028 kg. The results obtained report that it would be possible, from the reformulation of the lamination sequence, to change the damping characteristics of the compound.

In a similar work (for fiber angles), Haung and Tsai (2015) evaluate the effect of the addition of silica nanoparticles on the damping properties of fiber/epoxy compounds, concluding that despite the reduction in stiffness, there is an improvement in the damping properties. They report that the use of a micromechanical analysis, together with the modal analysis, presents a good agreement with the experimental data evaluated from the loss factor, (loss factor $\eta_{(\omega)} = 2\zeta_{(\omega)}\sqrt{1 - \zeta^2_{(\omega)}}$), but, in general, the reported damping ratio values (ζ) are around 0.15% to 0.18%.

In their work, Hameed et al.(2015) state that the damping of composite materials is usually a function of several parameters such as, for example, the volumetric fraction of the fibers, their diameter, orientation relative to the loading axis, constituent matrix, lamination form among others. They carry out an investigation of the damping on plates of composite materials of fiberglass and carbon with epoxy matrix by means of modal tests. They investigate two boundary conditions (fixed-free, free-free) with various orientations ($0^0, 45^0, 90^0$) in laminated composites made by manual process using an accelerometer, a 4-channel FFT analyzer and a modal impact hammer to obtain the FRFs (Frequency Response Functions) and, subsequently, the damping. They conclude that 0^0 lamination orientations have greater damping than 45^0 or 90^0. They report that the thickness had little influence on the damping, unlike the support condition. The steps, in a detailed way of how to evaluate the damping factor in structures by modal tests, can be seen in Carvalho (2002).

A first attempt to model the behavior of damping variation analytically/numerically is made in the work of Berthold et al (2008), where viscoelastic layers are assumed between the layers of composite materials. The results are compared with experimental data obtained with modal tests. Comparisons between experimental results of FRFs and those corresponding by finite element are reported and the results are very similar. Zheng and Liang (2020) present a proposal for modification of damping and sound insulation properties based on the inclusion of viscoelastic layers in composites to modify damping. They describe the procedure by modal analysis with accelerometers and analysis in the time domain by means of logarithmic decrement to obtain the damping ratio in composite materials. With the procedure, they report an increase of more than 100% in the damping ratio (reaching a damping ratio of 1.5%). More recently, Zhang et al. (2021) presented a study of the influence of the volumetric fraction of fiberglass and epoxy and the fiber orientation on the strength and damping of composite laminates. The studies were performed under vacuum to remove the effect of aerodynamic damping, reporting that air has an important effect on small damping values. Again, modal tests with the use of logarithmic decrement were used to quantify the damping ratios (ζ) in these materials with same dimensions, reaching values ranging from 2.42%, in ambient condition (air), to 1.15% in vacuum. Unlike the surveyed works, this work proposes the control and modification of damping through the external inclusion of layers of viscoelastic materials to the composite laminate.

3. THEORETICAL BASIS

3.1 Damping Ratio by Logarithmic Decrement

For the damping ratio evaluation in composite materials, assuming that it comes mostly from viscous damping, one can use the logarithmic decrement formulation of free vibration signals with certain boundary conditions. In Zheng (2020) it is said that it is possible to make a least squares adjustment to the law of decay of free vibrations and obtain a good estimate of this damping property. The system can be characterized by its period (or frequency) and damping ratio. The relationship between the natural undamped and dampened frequency, under these conditions, is given by.

$$\omega_d = \omega_n\sqrt{1 - \zeta^2}, \tag{1}$$

where ω_d is the damped frequency (rad/s) and ω_n the natural undamped frequency and ζ the so-called damping ratio which, in general, for composite laminated material systems, lies between $0 < \zeta < 1$, representing an underdamped condition. Thus, the answer of the system can be given by:

$$x(t) = e^{-\zeta \omega_d t} X\, sen(\omega_d\, t + \varphi), \tag{2}$$

where $x(t)$ is the response of the system in time, X is its initial amplitude, t is the instants of time, and φ a phase angle. For initial conditions x_0, $\dot{x}_0$, $X = \sqrt{x_0^2 + [(\dot{x}_0 + \zeta \omega_n x_0)/\omega_d]}$ and $\varphi = tan^{-1}[\omega_d\, x_0/(\dot{x}_0 + \zeta \omega_n x_0)]$. The envelope of equation (2) can be represented as:

$$z(t) = me^{-n(t-l)}, \tag{3}$$

with constants m, n, and l to be determined, for example, by means of a function adjustment with the envelope of the decaying signal, obtained experimentally (Figure 1). Therefore, logarithmizing equation (3), results in:

$$ln\,[z(t)] = ln\,(m) - nt + nl, \tag{4}$$

By defining $f = ln[z(t)]$ and $h = ln\,ln\,(m) + nl$, the equation is linearized ($f = -nt + h$) and can be used to be adjusted by least squares to a series of k data of the envelope, obtained experimentally, and subsequently obtained the damping ratio (minimizing the function ϕ)

$$\phi = \sum_{i=1}^{k}[f_i - (nt_i + h)]^2, \tag{5}$$

Applying the necessary condition for an optimum, $\partial\phi/\partial n = 0$ and $\partial\phi/\partial h = 0$, and solving the system, one arrives at:

$$n = \left[\sum_{i=1}^{k} t_i f_i - \frac{1}{k}\left(\sum_{i=1}^{k} t_i \sum_{i=1}^{k} f_i\right)\right] / \left[\frac{1}{k}\left(\sum_{i=1}^{k} t_i\right)^2 - \sum_{i=1}^{k} t_i^2\right], \tag{6}$$

$$h = \frac{1}{k}\sum_{i=1}^{k} f_i + \frac{n}{k}\sum_{i=1}^{k} t_i, \tag{7}$$

If one takes the ratio between two measurements of displacements separated from a dampened period τ_d, this results in

$$\frac{x(t_1)}{x(t_2)} = \frac{e^{-\zeta \omega_d t_1} Xsen(\omega_d\, t + \varphi)}{e^{-\zeta \omega_d t_2} Xsen(\omega_d\, t + \varphi)} = e^{\zeta \omega_d \tau_d}, \tag{8}$$

and the logarithmic decrement, calculated for a set of n separate measures taken $n\tau_d$, is: $\underline{\delta} = (1/n)\,ln\left(x(t_1)\backslash x(t_n)\right) = \zeta \omega_d \tau_d = 2\pi\zeta/\sqrt{1-\zeta^2}$) and which, resolved for ζ, results in

$$\zeta = \underline{\delta}/\sqrt{2\pi^2 + \underline{\delta}^2}, \tag{9}$$

The relation between n, ζ and ω_n can be placed as $n = \zeta \omega_n$. The so-called loss factor is defined η as $\eta = 2\zeta$. Figure 1 indicates a generic free vibration curve, indicating the main parameters required for the calculation of the damping ratio (or loss factor) in the logarithmic decrement method.
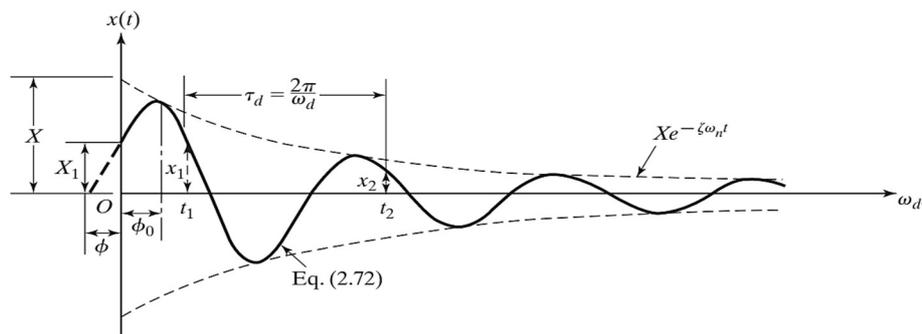

Figure 1. Free vibration curve with viscous damping to define logarithmic decrement.

4. METHODOLOGY

The way in which the experimental analyses were developed is divided into three steps: (i)the first being the definition of the experimental tests; (ii) building the experimental bench and performing the tests, and (iii) the post processing of the data to obtain the damping ratios using the filters, Hilbert transform and the decrement logarithmic technique. The

test samples consisted of 16 equal samples of composite laminates, 2x2 twill weave, 3k wire yarn, with cross-ply orientation $(0/90)_{10}$, in the dimensions of 50×390×2 mm, with total mass of 61g.

For the tests performed, 5 different viscoelastic materials were used in the format of adhesive tape, width 50 mm, similar to the width of the tested specimens. A brief description of such materials can be seen in Table 1. Figure 2 shows these materials attached to the composite laminates.

Table 1. Brief description and mass of the viscoelastic materials used in each of the samples.

Viscoelastic Material	Description	Mass (g/m)
1	Adhesive demarcation tape	9.62
2	Self-adhesive asphalt blanket	67.7
3	Adhesive tape with transverse and longitudinal fiberglass reinforcement	11.96
4	Adhesive tape with longitudinal fiberglass reinforcement	7.31
5	Self-adhesive aluminum tape	5.53

Figure 2. Viscoelastic materials according to Table 1 attached to the composite laminate in the situation of one-side and 50% covering treatment.

4.1 Design of Experiments

For the design of the experiments, it was considered to submit each of the beams to simple damped free vibration in fixed-free situation, applying each of the treatments with available viscoelastic materials. For each test condition, five valid measurements were performed. There are 5 identical specimens in total that will receive 5 different damping treatment besides the control test specimen. Tests were performed with a control sample, without any viscoelastic treatment, and for each of the five treatment, three different adhesion conditions were investigated. The first one consisting of attaching the viscoelastic material on the two surfaces of the sample, the second only on one of the faces, but still covering its entirety. The third condition, in turn, consists of keeping on only one side the material adhered from the setting to half the length of the sample. In accordance with this, the final design of experiments is presented in Table 2.

Table 2. Design of Experiments and number of realizations for each configuration.

Treatment	Design of Experiments	Viscoelastic configuration		
		100% Covering 2 faces	100% Covering 1 face	50% Covering 1 face
0	Control sample (testimonial sample)	5	-	-
1	Adhesive demarcation tape	5	5	5
2	Self-adhesive asphalt blanket	5	5	5
3	Adhesive tape with transverse and longitudinal fiberglass reinforcement	5	5	5
4	Adhesive tape with longitudinal fiberglass reinforcement	5	5	5
5	Self-adhesive aluminum tape	5	5	5

A representation on how the adhesion of viscoelastic material is made in in half a face can be observed in Figure 3(b).
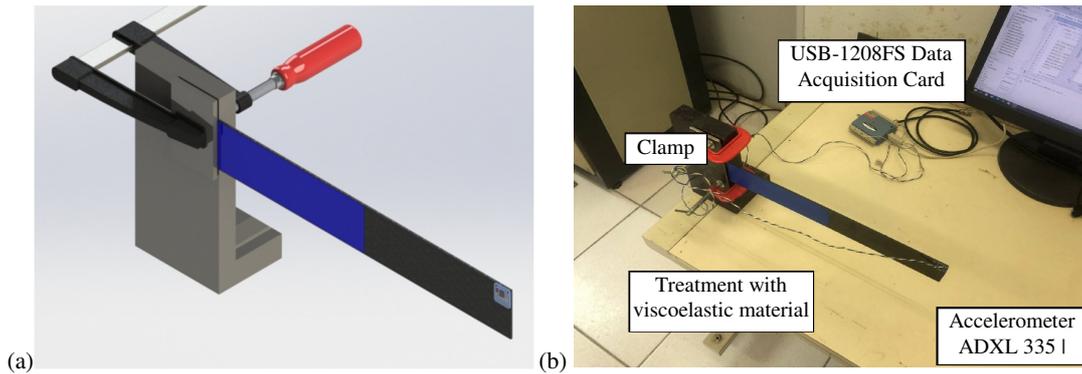
(a)　　　　　　　　　　　　　　　　　　　(b)

Figure 3. Arrangement of the viscoelastic material in the sample under the conditions of (a) 1 face 50% covering, (b) Sample 3, 50% covering in 1 face, fixed to the test bench.

4.2 Experimental setup

To perform the measurements, the samples were fixed with the aid of clamps, restricting their rotation and translation in relation to the bench. The bench, as well as the sample number 3 properly fixed, can be seen in Figure 3.

The samples were fixed on the bench in the set-free condition, using 40 mm of one end for fixation and obtaining a useful length of 350 mm. To assure the same stiffness in the fixation, the same number of tightening turns was used for all samples and the overhang length was confirmed in each test by a caliper rule. An accelerometer was fixed at the end of the sample in order to measure the acceleration values at this point. The form of fixation was with beeswax, since the acceleration levels were relatively low to the point that the wax was sufficient to keep the accelerometer fixed at the tip of the sample.

Initially, tests were performed using an impact hammer with tip in rubberized material, model Endevco ISOTRON 4416-1, with gain of 10X and sensitivity 1.14 mV/N. Tests were performed previously using a stopper in order to excite the sample only by applying a prescribed displacement (equal for all samples) at the end of the sample to assure only the first frequency mode was significantly excited. The accelerations were measured with an accelerometer connected to the end of the beam, model ADXL 335 by Analog Devices. The sensitivity is 970 mV/g, and a maximum acceleration of ±3.0 g, with a frequency range from 0 up to 550 Hz. The acceleration signals were acquired with a data acquisition board model USB-1208-FS of 12 bits. Data were acquired with a sampling rate of f_s=1000 Hz.

4.3 Data processing

The data obtained through the experiments were stored and treated using the Matlab software (2012). Initially, some treatments were applied in order to obtain frequency response functions, taking into account the feedback obtained through the instrumented hammer, but two unwanted effects were observed making the measurements in such a way. The first is the excitation of the second and third modes of vibration being more significant in relation to the first mode. The second, in turn, is the difficulty of finding a stable damping ratio throughout the duration of the measurement, since it is necessary to adjust a series of parameters to obtain this value from the FRF, making the procedure non-repetitive and quite dependent on the parameters used in each evaluation.

For the follow-up of the experiments, a new method was adopted, which consists of applying a prescribed displacement at the end of the beam as a form of excitation. The analysis of the graph of the accelerations allows identifying a wave with sine format, without indications of noise or overlapping of frequencies. An example of an acquired acceleration signal at the tip of the sample over time is presented in Figure 5.
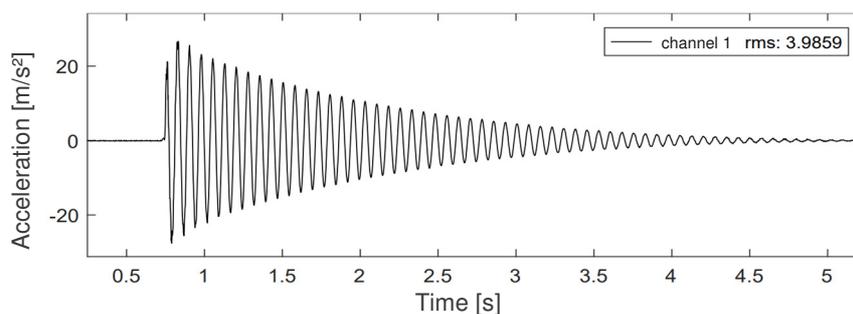

Figure 4. Measurement #48, free-vibration wave with damping.

After such acquisition, a Fast Fourier Transform FFT was performed for this set of values, aiming to find the relevant oscillation frequencies within the range in which the accelerometer is able to read, which can be observed in Figure 6.
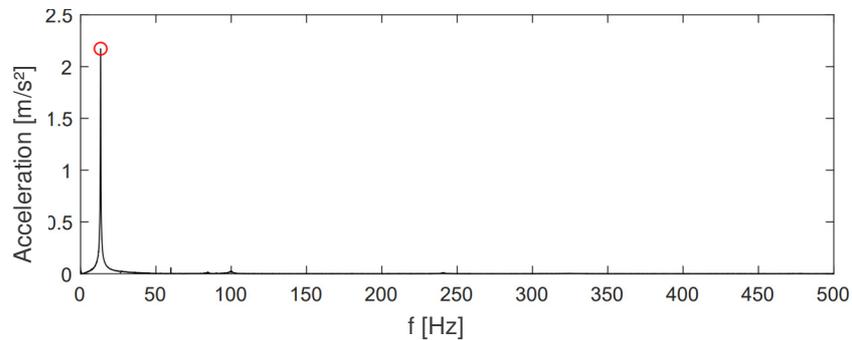

Figure 5. FFT of measurement #48 indicating the first mode frequency. Measured value: 13.32 Hz.

Notably, the value is quite clear and noise-free, containing only the first vibration frequency referring to the first bending vibration mode of the measured system. After obtaining the frequency value of the first mode, a filter was applied to the measured values in order to obtain only the accelerations at specific frequency, and, using the coordinates of the upper peaks of the sine wave (envelope) and making the adjustment by least squares, the value of ζ was found. The wave formed by the filtered data, as well as the approximate exponential curve is presented in Figure 7. Repeating this process for all measurements present in the design of experiments, it is obtained the values for first modal frequency and damping ratio for all desired configurations of treatment with the viscoelastic material.
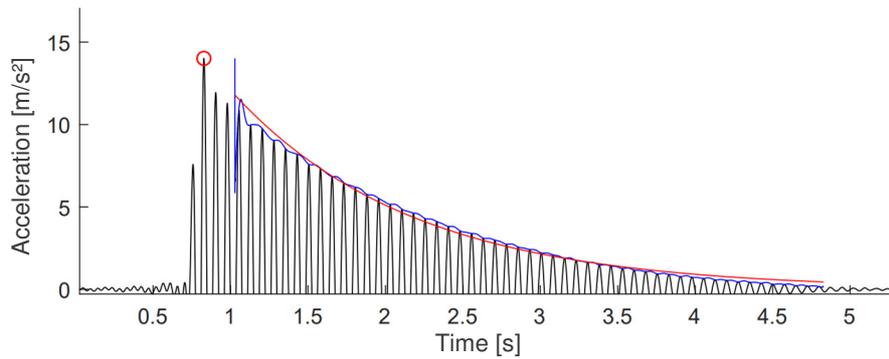

Figure 6. Measurement #48, envelop by Hilbert Transform and fitted exponential curve (in red).

5. RESULTS

A summary of the data, with obtained mean values for each test condition of all valid replicated measurement obtained by the prescribed displacement method is presented in Table 3. By combining the density data of the samples and viscoelastic materials with the values represented in Table 3, it is possible to perform an analysis of how much mass is being added to the system with the treatment and verify, for each test condition, what is the ratio between the increase in mass given to the system and the increase in the damping ratio for the first vibration mode. The data for this analysis are presented in Table 4.

Table 3. Mean/std.dev. for frequency and damping ratio values for each test condition.

Sample	Situation	Mean value f μ_f (Hz)	Std. Dev. ζ σ_f (Hz)	Mean value of ζ μ_ζ	Std. Dev. ζ σ_ζ
Control	–	14.5513	0.0000	0.0080	0.0003
1	2 faces	13.4845	0.0000	0.0082	0.0002
1	1 face	13.6317	0.0190	0.0071	0.0004
1	½ face	13.9107	0.0147	0.0101	0.0007
2	2 faces	11.0308	0.0183	0.0120	0.0001
2	1 face	12.1579	0.0155	0.0136	0.0007
2	½ face	14.2248	0.0155	0.0161	0.0007
3	2 faces	12.9511	0.0000	0.0113	0.0010
3	1 face	13.3191	0.0000	0.0103	0.0001
3	½ face	13.7255	0.0143	0.0100	0.0009
4	2 faces	14.3251	0.0277	0.0150	0.0007
4	1 face	14.3923	0.0143	0.0132	0.0007
4	½ face	14.6590	0.0143	0.0136	0.0004
5	2 faces	13.4247	0.0155	0.0146	0.0006
5	1 face	13.3922	0.0143	0.0141	0.0009
5	½ face	13.2781	0.0143	0.0214	0.0005

Table 4. Mean values for Total mass, mass increase due treatment, damping ratio, and ratio

Sample	Situation	Total sample's mass (g)	Added mass by treatment (%)	Increase in ζ (%)	Ratio
Control	–	61.00	–	–	–
1	2 faces	67.73	11.04	2.26	0.20
1	1 face	64.37	5.52	-10.69	-1.94
1	½ face	62.68	2.76	26.92	9.75
2	2 faces	108.39	77.69	49.96	0.64
2	1 face	84.70	38.84	70.30	1.81
2	½ face	72.85	19.42	101.44	5.22
3	2 faces	69.37	13.72	42.11	3.07
3	1 face	65.19	6.86	29.03	4.23
3	½ face	63.09	3.43	24.83	7.24
4	2 faces	66.12	8.39	88.13	10.51
4	1 face	63.56	4.19	64.81	15.45
4	½ face	62.28	2.10	69.98	33.37
5	2 faces	64.87	6.35	82.86	13.06
5	1 face	62.94	3.17	76.77	24.19
5	½ face	61.97	1.59	168.09	105.95

Another analysis that can be made from the obtained data, and that brings a better view on the behavior of the samples as the superficial treatments change is presented in Figures 8 and 9 by means of line graphs.
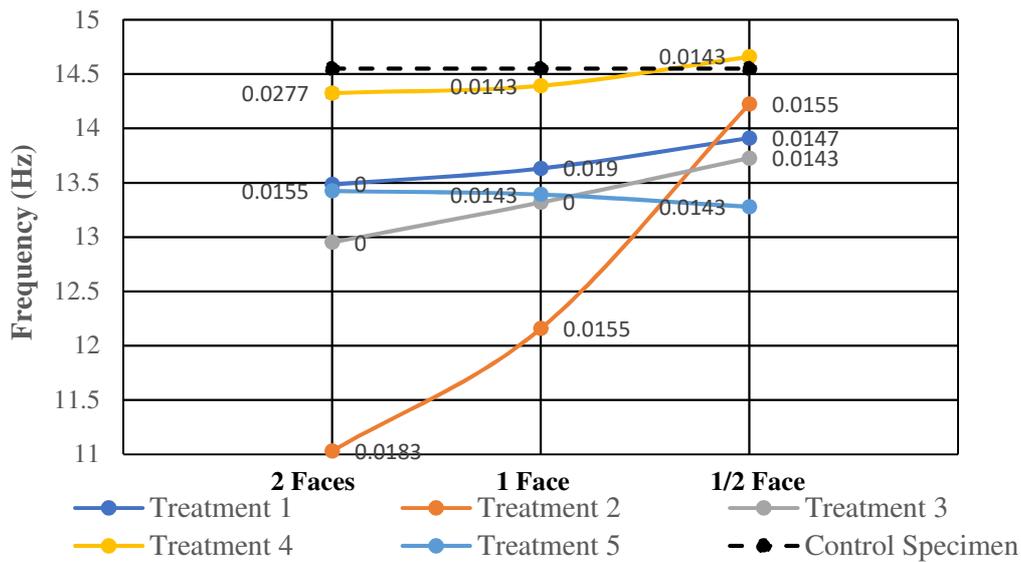
Figure 7. Graphs for Frequency × Treatment condition (numbers are the standard deviations).

Figure 8 shows that in general, the natural frequency has a tendency to reduce as the severity of the treatment increases, with sample 5 being the exception in this case. Another detail that draws attention in this graph is the large frequency change for sample 2, bringing a reduction of approximately 25% in frequency compared to the Control sample, while the second largest effect was caused by sample 3, with a reduction of approximately 11%. Analyzing Figure 9, in turn, it is observed that for samples 3 and 4, both composed of fibrous materials, their damping ratio tends to approach the damping ratio of the Control sample with the reduction of the severity of the treatment, while the other samples present a reverse behavior for the case with half-face treatment.
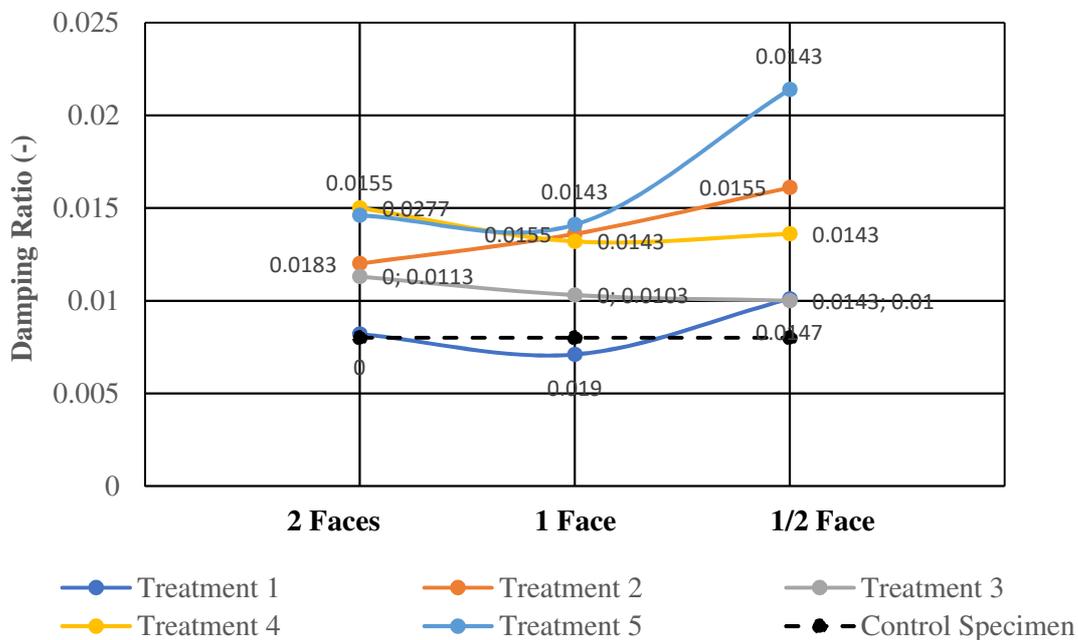

Figure 8. Graph for Damping Ratio × Treatment condition (numbers are the standard deviations).

6. CONCLUSIONS

By analyzing the values in Table 3, it is possible to observe that both the frequencies and the calculated damping ratios, the standard deviations of the measurements present values of at least one order of magnitude smaller in relation to the measured values themselves, which is an indication of the good repeatability of the experiment.

Treatment 1 (self-adhesive demarcation tape) showed the smallest change in the damping ratio in relation to the other treatments, and even so, it was not the one that showed the smallest change in mass. Treatment 2 (Self-adhesive asphalt blanket), in turn, showed more significant changes in the damping ratio, but caused a significant increase in mass to the system. This increase in mass, along with the geometric changes caused by the adhesion of the viscoelastic material, justify the large change in natural frequency when compared to the control sample.

Treatment 3 (Adhesive tape with transverse and longitudinal glass fiber reinforcement) and 4 (Adhesive tape with longitudinal glass fiber reinforcement) showed similar behavior both in relation to the damping ratio and natural frequency, and treatment 4, among these two, was the one that achieved the greatest increase in the damping ratio. The behavior of the damping ratio, being different from the other samples and reducing for the case of half-face treatment, can be justified by the similarity of the viscoelastic material to the composite material, both containing fibers as a structural element in its composition.

In highlight, it is observed treatment 5, whose material used was the self-adhesive aluminum tape. This not only obtained the highest values of damping ratio, but also was the case in which the least amount of mass was added to the system, and therefore, it is clear to justify the change in damping caused only by the treatment of the viscoelastic material. Unlike the other treatments, it was the one that presented a decreasing behavior for the progression of natural frequencies with the surface treatments. A possible justification for this is that the change in frequency caused by the change in damping ratio, according to Equation 1, is more significant than the change in frequency caused by the difference in geometry and mass when compared to the control sample. Finally, the causes for the increase in damping ratio with the half-face treatment in samples 1, 2 and 5 were not revealed.

7. ACKNOWLEDGEMENTS

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