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On the vibration performance assessment of a plate with damaged constrained layer damping patches

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Abstract. The purpose of this study is to investigate the metrics to be used for the vibration performance assessment of a plate with damaged CLD patches in operating conditions. In particular, Experimental Modal Analysis is used to determine the effects of the CLD patches on the Frequency Response Function (FRF). EMA can be performed during maintenance operations or in a laboratory. However, during operating conditions the excitation cannot be measured. Alternative output-only strategies, e.g. Frequency Domain Decomposition (FDD), can be implemented to extract information about some of the structural properties of the system. Three damaging conditions of the CLD are investigated. It is shown that the deterioration of the CLD can actually improve the vibrational performance of the plate in the desired frequency range. Two metrics are therefore suggested to assess the vibration performance: acceleration peak and frequency shift.

1. Introduction

Constrained-Layer Damping (CLD) is a damping treatment used for passively reducing structural vibrations and noise of lightweight structural components. It consists of a viscoelastic polymer layer sandwiched between a structural element, whose vibration need to be reduced, and a relative stiffer layer [1]. When the out-of-plane vibrations of the structural element occur, the vibrational energy is dissipated through shear deformations of the viscoelastic layer [1]. These treatments, due to their inherent advantages (low weight, no structural modifications) and high efficiency, are widely used in the aeronautical and aerospace industry. CLD can be applied uniformly to the surface of a structural element or in patches. While the uniform configuration can have an effect on a broad frequency range, patches can be used to target specific modes or smaller frequency bands. The performance of a CLD can be assessed by quantifying the drop in amplitude of the Frequency Response Function (FRF) and/or by considering the shift in resonant peaks [1-3].

Experimental Modal Analysis (EMA) [4-5] is a popular approach for the evaluation of the FRF. This approach requires measuring both excitation and response of the structure, and the implementation of signal processing strategies [4-5]. EMA can be performed during maintenance operations, or in a laboratory. However, measurement of the excitation is often impractical. Alternative output-only strategies, e.g. Frequency Domain Decomposition (FDD) [6-7], can be implemented to extract some modal information, such as modal damping, natural frequencies and mode shapes, but not the FRF peak amplitude. The application of this two approaches for the the vibration performance assessment of a plate with damaged CLD patches is investigated in this paper.



An aluminium plate with and without CLD patches is investigated at room temperature. EMA is used to determine the effects of the CLD patches on the FRF. The FDD approach is then applied to yield modal damping and natural frequencies showing a very good agreement with the results yielded by the FRF. Subsequently, the viscoelastic layer and the aluminium layer of the CLD patches are damaged. In particular, three different damage configurations are investigated. For each configuration, EMA and FDD are applied, showing again a very good agreement.

Interestingly the results of this experimental campaign shows that the deterioration of the CLD can actually improve the vibrational performance of the plate in the desired frequency range. Because of these results, it is suggested to assess the vibration performance of a plate with damaged constrained layer damping patches by monitoring the acceleration peak (in the time domain) and the frequency shift. The paper is structured as follow: the test setup is described in Section 2; the CLD is described in section 3 and the FRF obtained for the bare plate and for the plate with CLD are compared; in section 4 the vibration performance of damaged CLD is experimentally investigated. FDD is introduced in section 5 and applied to output only measurements to yield the modal parameters.

2. Test setup

The test specimen is an aluminium plate with dimensions 200 mm x 300 mm x 2 mm, density 2800 kg/m³, Young modulus 72 GPa, Poisson ratio 0.3. The plate was suspended with thin lightweight nylon wires attached to a frame to approximate free boundary conditions, as shown in Figure 1. The plate was excited with an instrumented hammer (PCB 086C03 with a nylon tip) and the velocity response was measured with a single pointer Laser Doppler Vibrometer (Polytec NLV-2500). The sampling rate was 10 kHz and the signal was recorded for 30 seconds. Three averages were taken for each measurement point to evaluate an average FRF.

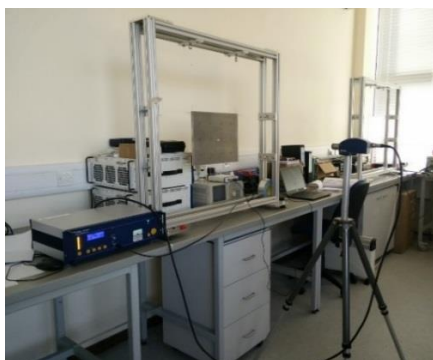


Figure 1. Test setup

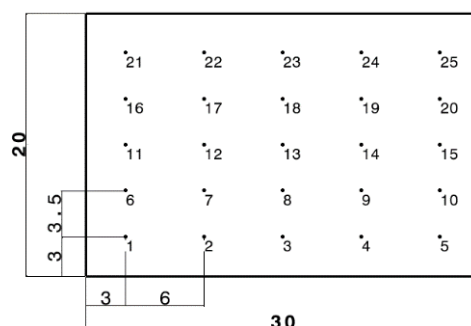


Figure 2. Grid of points

A grid of 25 points was drawn on the plate, as schematically shown in Figure 2. The Laser Doppler Vibrometer was used to measure the velocity response at point 7, while the hammer was used to excite the plate in the 25 grid points, to yield 25 averaged FRFs. These FRFs were used to characterise the mode shapes of the plate, and evaluating the damping ratio (using the circle fitting method) at each resonance. In what follows, only the FRF measured at point 7 is considered.

3. Constrained Layer Damping

The damping layer used in this test campaign is the 3M Vibration Damping Tape 435 [8], shown in Figure 3. This damping layer is characterised by a top aluminium layer of thickness 0.20 mm, and a damping polymer with thickness of 0.14 mm [8], and the tape has a width of 50 mm. The dynamic mechanical properties of the damping polymer vary with frequency and temperature, and will also vary with time. 3M provides the shear storage modulus and the loss factor of a viscoelastic adhesive in the form of a nomograph to account for the frequency and temperature dependence [8]. The room temperature was monitored with a temperature probe, and it was varying between 18 °C and 22.5 °C. In this temperature range there was no substantial variation of the CLD properties. The CLD was used to

partially cover the plate along the edges, as shown in Figure 4, with the aim of affecting mostly the first two modes (at 106 Hz and 115 Hz), which are a twisting and a bending modes.



Figure 3. The 3M Vibration Damping Tape 435 [8]



Figure 4. Plate with CLD

The Frequency Response Functions (in terms of acceleration divided by force) obtained for the bare plate and for the plate with the CLD partial treatment are shown in Figure 5. It can be observed that the peaks at the first two resonance frequencies have dropped of more than 10 dB. The shifts in resonant frequencies and the changes in damping ratios are detailed in Tables 1 and 2.

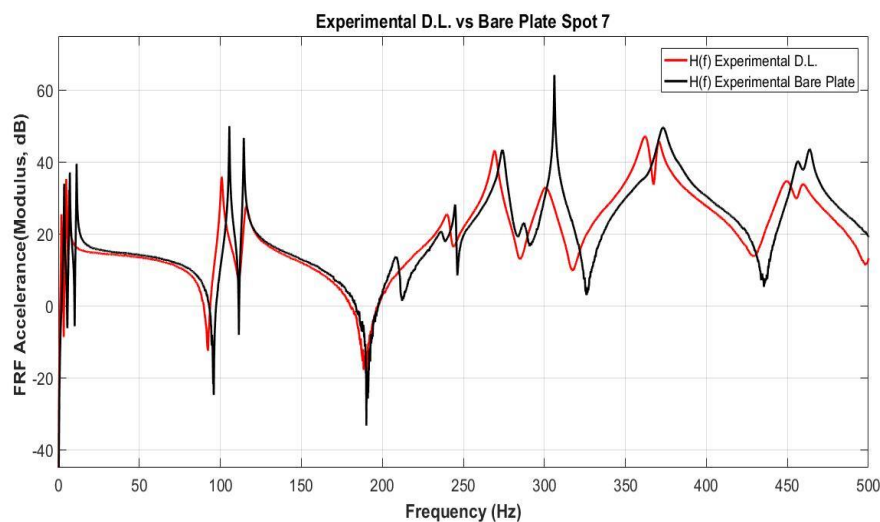


Figure 5. Comparison between the FRF obtained for the Bare Plate (black curve) and for the Plate with CLD treatment (red curve)

Table 1. Resonances obtained for the Bare Plate and for the Plate with CLD treatment.

	f_1 (Hz)	f_2 (Hz)	f_3 (Hz)	f_4 (Hz)	f_5 (Hz)
Bare Plate	105.58	114.51	244.93	274.13	306.2
Treated plate	100.89	116.11	239.84	268.28	300.7

Table 2. Damping ratio obtained for the Bare Plate and for the Plate with CLD treatment.

	ζ_1	ζ_2	ζ_3	ζ_4	ζ_5
Bare Plate	0.0005	0.00077	0.0023	0.005	0.003
Treated plate	0.004	0.012	0.0107	0.0054	0.0114

4. Vibration performance of damaged Constrained Layer Damping

In this section the effect of a partially damaged CLD on the average FRF is investigated. In particular, three damage cases were considered, indicated A, B and C, as shown in Figure 6. While A is a light damage condition mostly at the edges of the CLD, case B is characterised by a more diffused damage of the viscoelastic and aluminium layer. Case C represents the most severe type of damage: part of the damping layer is removed, and it can be observed that in many areas the CLD is not attached to the aluminium plate.



Figure 6. Damaged CLD: case A (left), case B (centre) and case C (right)

The averaged FRFs obtained at point 7 (where both the input and output are measured) in the frequency range 85 Hz – 130 Hz (focusing on the twisting and bending mode) are shown in Figure 7. The effect of the damage on the CLD is clearly visible in terms FRF resonant peaks drop and resonant frequencies shift. Variations in the resonant frequencies and in the damping ratio are detailed in Tables 3 and 4.

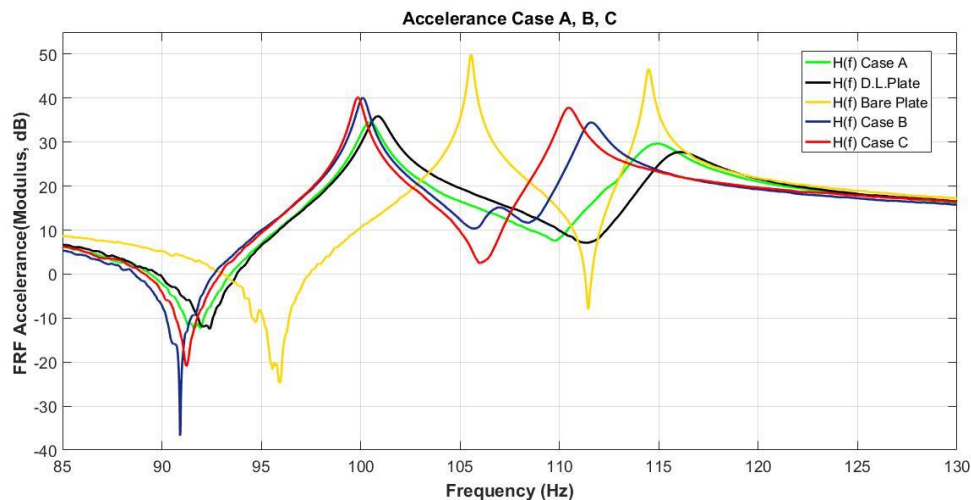


Figure 7. Comparison between the FRF obtained for the Bare Plate (yellow curve) and for the Plate with CLD treatment (black curve, indicated as DL Plate), and the Damaged CLD cases A (green), B (blue) and C (red).

Table 3. First two resonances obtained for the Bare plate, the plate with CLD treatment (CLD), and the plate with damaged CLD (cases A, B and C).

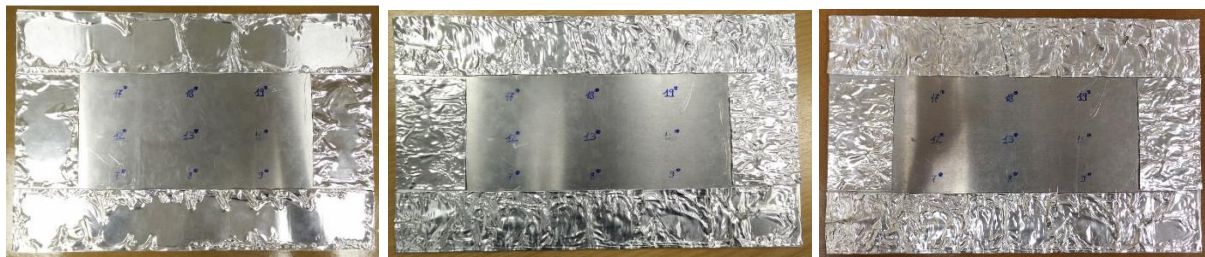
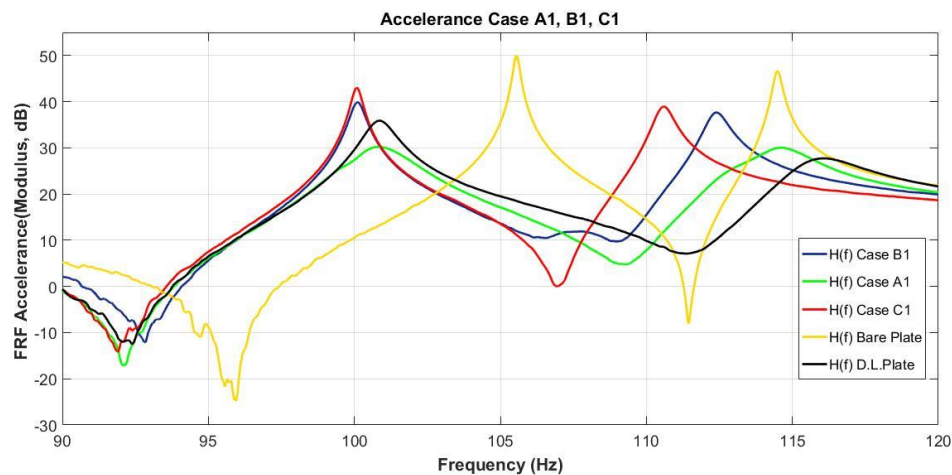
	CLD	A	B	C	Bare
f_1 (Hz)	100.89	100.49	100.13	99.88	105.58
f_2 (Hz)	116.11	114.93	111.62	110.48	114.51

Table 4. Damping ratio for the first two resonances obtained for the Bare plate, the plate with CLD treatment (CLD), and the plate with damaged CLD (cases A, B and C).

	CLD	A	B	C	Bare
ζ_1	0.0045	0.0043	0.0022	0.002	0.0005
ζ_2	0.0120	0.0102	0.0047	0.0037	0.0008

From these results, it can be observed that the damping ratio decreases as damage increases. Damaging the CLD leads to less rigidity of the CLD assembly, which in turn will produce less shear deformations and therefore less energy dissipation. However, other energy dissipation mechanisms can occur, such as friction. An additional effect of damage to the CLD is that of a small reduction of the resonance frequencies. This can be attributed to the CLD performing as an added mass to the system due to loss of rigidity.

A second test campaign was performed on the same plate. Damage of the CLD was produced by detaching and reattaching the CLD on the plate. The three damage conditions (A1, B1, and C1) are shown in Figure 8, and the corresponding averaged FRFs are shown in Figure 9.

**Figure 8.** Damaged CLD study 2: case A1 (left), case B1 (centre) and case C1 (right)**Figure 9.** Comparison between the FRF obtained for the Bare Plate (yellow curve) and for the Plate with CLD treatment (black curve), and the Damaged CLD cases A1 (green), B1 (blue) and C1 (red). Variations in the resonant frequencies and in the damping ratio are detailed in Tables 5 and 6.

These results show that the damping ratio does not decrease for case A1, although they do for the other damage cases. A reduction of the resonance frequencies is observed also in this case as damage is increased. It can be also observed that the first resonance peak is not increasing as the damage increases.

In particular, Case A1 shows a bigger peak drop than the undamaged CLD case. It might be concluded that specific forms of low-level damage may be beneficial.

Table 5. First two resonances obtained for the Bare plate, the plate with CLD treatment (CLD), and the plate with damaged CLD (cases A1, B1 and C1).

	CLD	A1	B1	C1	Bare
f_1 (Hz)	100.89	100.86	100.13	100.11	105.58
f_2 (Hz)	116.11	114.64	112.42	110.62	114.51

Table 6. Damping ratio for the first two resonances obtained for the Bare plate, the plate with CLD treatment (CLD), and the plate with damaged CLD (cases A1, B1 and C1).

	CLD	A1	B1	C1	Bare
ζ_1	0.0045	0.0091	0.002	0.0013	0.0005
ζ_2	0.0120	0.0098	0.0031	0.0025	0.0008

Overall it can be concluded that: (i) modal damping may not be an optimal metric to determine the performance of the plate with CLD treatment; (ii) damaging the CLD patches may in some case of low level damage produce a bigger peak drop compared to the undamaged CLD; (iii) damaging the CLD patches can induce a reduction of the resonance frequencies. Therefore, the deterioration of the CLD could potentially improve the vibrational performance of the plate in the desired frequency range. Because of these results, it is suggested to assess the vibration performance of a plate with damaged constrained layer damping patches by acquiring data with an accelerometer to monitor the acceleration peak in the time domain, and using output-only strategies to assess the frequency content, and therefore monitoring resonance shifts. In the next section, the use of the FDD to this type of problems is investigated. The objective of the proposed analysis is to improve the structural performance, not to identify or quantify the damping layer damage.

5. Frequency Domain Decomposition

Frequency Domain Decomposition (FDD) [6-7] is a technique for output-only modal identification. The spectral density matrix is decomposed into a set of single degree of freedom system, and the individual SDOF auto-spectral density functions are transformed back to time domain to identify damping and frequency [6-7]. Given a standard input ($x(t)$)-output ($y(t)$) relationship in the frequency domain:

$$Y(\omega) = H(\omega)X(\omega) \quad (1)$$

Being $H(\omega)$ the Frequency Response Function. This can be rearranged in terms of Power Spectral Densities (PSD):

$$G_{yy}(\omega) = H^*(\omega)G_{xx}(\omega)H^T(\omega) \quad (2)$$

Where the apex * indicates the complex, and T indicates the transpose operator. The FDD approach assumes that the output PSD $G_{xx}(\omega)$ is constant. This assumption is valid for broadband random inputs. The first step of the FDD is to estimate the power spectral density matrix $\mathbf{G}_{yy}(j\omega)$ at discrete frequencies $\omega = \omega_i$. This can be done using the Matlab function `cspd(x,x)`. $\mathbf{G}_{yy}(j\omega)$ is then decomposed by using the Single Value Decomposition (SVD):

$$\mathbf{G}_{yy}(j\omega) = \mathbf{U}_i \mathbf{S}_i \mathbf{U}_i^H \quad (3)$$

where the matrix $\mathbf{U}_i = [u_{i1}, u_{i2}, \dots, u_{im}]$ is a unitary matrix with entries corresponding to the singular vectors u_{ij} , \mathbf{S}_i is a diagonal matrix with entries corresponding to the scalar singular values s_{ij} , and the apex H indicates the Hermitian. At certain frequencies, only a limited number of modes will contribute significantly (assuming low modal overlap). Thus, according to the FDD theory [6-7], the first singular vector u_{i1} is an estimate of that mode shape:

$$\hat{\phi} = u_{i1} \quad (4)$$

and the corresponding singular value defines the auto-PSD of the corresponding SDOF system.

This PSD function is identified around a peak by comparing the mode shape estimate $\hat{\phi}$ with singular vectors for the frequency lines around the peak [6-7], and using the modal assurance criterion (MAC). The resonant frequency and damping ratios are obtained from the SVDs around the isolated peak singular value [6-7].

FDD makes the assumption of a white noise input excitation or a free vibration response. Hence in hammer tests, the part of the response of the plate after the impact is over can be used, where indeed the data correspond to free vibration response.

The modal parameters identified from EMA (using input and output data) and FDD (using output only data) are compared in Tables 7 and 8.

Table 7. Modal parameters for the first resonance peak – second and first test campaign

<u>1st mode</u>	Frequency (Hz)	Frequency FDD (Hz)	Damping Ratio	Damping Ratio FDD
Damped Plate	100.88	100.85	0.0045	0.0048
Case A1	100.86	100.85	0.0091	0.0093
Case B1	100.13	100.09	0.002	0.0022
Case C1	100.11	100.09	0.0013	0.0016
Bare Plate	105.57	105.58	0.0005	0.0007

<u>1st mode</u>	Frequency (Hz)	Frequency FDD (Hz)	Damping Ratio	Damping Ratio FDD
Damped Plate	100.88	100.85	0.0045	0.0048
Case A	100.48	100.40	0.0043	0.0044
Case B	100.13	100.09	0.0022	0.0029
Case C	99.88	99.79	0.002	0.0027
Bare Plate	105.57	105.58	0.0005	0.0007

Table 8. Modal parameters for the second resonance peak – second and first test campaign

<u>2nd mode</u>	Frequency (Hz)	Frequency FDD (Hz)	Damping Ratio	Damping Ratio FDD
Damped Plate	116.11	116.11	0.0119	0.0122
Case A1	114.64	114.59	0.0098	0.0113
Case B1	112.42	112.45	0.0031	0.0029
Case C1	110.62	110.62	0.0025	0.0011
Bare Plate	114.51	114.43	0.0008	0.0007

<u>2nd mode</u>	Frequency (Hz)	Frequency FDD (Hz)	Damping Ratio	Damping Ratio FDD
Damped Plate	116.11	116.11	0.0119	0.0122
Case A	114.93	114.89	0.0102	0.0108
Case B	111.62	111.53	0.0047	0.0034
Case C	110.48	110.47	0.0037	0.0021
Bare Plate	114.51	114.43	0.0008	0.0007

It can be observed that in both investigation cases, the FDD method provides natural frequencies and damping ratios values very close to that obtained by EMA. The FDD can be used as a strategy for real-time vibration performance assessment of a plate with damaged CLD patches.

6. Conclusion

This paper has addressed the vibration performance assessment of an aluminum plate with damaged constrained layer damping patches. In particular, two test campaigns with three different damage configurations were investigated. For each configuration, EMA and FDD were applied, showing a very good agreement. The results of this experimental campaign have shown that some level of the deterioration of the CLD can improve the vibrational performance of the plate in a desired frequency range. Because of these results, it is suggested to assess the vibration performance of a plate with damaged constrained layer damping patches by acquiring data with an accelerometer to monitor the acceleration peak in the time domain, and using FDD to assess the frequency content, and therefore monitoring resonance shifts.

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