Research Article:

# Composite Sandwich Panels for Vibration and Noise Reduction in Aircraft: Experimental Validation

Marta Sofia Neves Freitas dos Santos

AlfAl / EngAer / 137736-B

**Abstract:**

The current study presents a solution to a problem introduced by the Portuguese Air Force (FAP). While making its latest UAV (Unmanned Aerial Vehicle), the vibration and noise caused by the engine, led to the need to protect the aircraft avionics systems by creating two insulation composite panels. As such, the aim of this study was to present the best configurations for the two composite sandwich panels using only the materials available in the Air Force's Centre for Research, Development and Innovation (CIDIFA).

The experimental study was divided into two parts: vibration and noise. The vibration tests, performed in Instituto Superior Técnico (IST), had the purpose of gathering the damping coefficients in each panel samples (plates), using two different processing methods. The noise tests, performed at Faculdade de Engenharia da Universidade do Porto (FEUP), where able to quantify, through simple methods, the amount of noise that is absorbed by each plate. The results presented in this study take into account different scenarios and limitations that may be required by the FAP.

***Key-Words:*** *FAP, UAV, Vibration, Noise, Composite Panels, Sandwich, Experimental Tests*

## 1. Introduction:

The great success and evolution of the latest generation of UAVs developed for information gathering, surveillance and reconnaissance missions has led to a significant advance in small-scale avionics systems [1]. While researching, and developing these systems, it was determined that their percentage costs were considerably higher than the total costs of the remaining aircraft. In some military aircraft, targeting and monitoring systems are so evolved and expensive that their value reaches 80% of the total cost of the aircraft. With such high costs, it is only natural that the avionics' installation zone should be one of the most well protected areas of the whole structure [2]. Engines are the main cause for the vibration problems to which the components and structure are subjected. As such, due to their sensitive nature, avionics are normally installed in attenuating platforms that guarantee a passive isolation of vibration [3].

Through the PITVANT project and more recently the PERSEUS project, the Portuguese Air Force has developed several UAVs with the purpose of using them in an operational environment and integrate them into the European maritime surveillance system [4]. Thus, in the development of its most recent UAV, the need arose to create two sandwich panels: one that protected avionics of the vibrations propagated to the structure by the engine; and another capable of reducing the amount of noise (from the engine) that reaches the avionics. The panels had to be constructed with materials already available for the construction of the aircraft.

Composite sandwich structures were chosen so their dynamic behaviour could be characterized experimentally, and so that the introduction of a core (in this case foam) could provide information on how it affects the vibration and noise absorption capability of the panel. To this end, several composite sandwich plates where developed with different configurations and materials, namely glass fibre and carbon fibre. In addition, two plates with dual foam cores (and glass fibre on the inside) were made, so they could be compared with traditional plates of single core sandwich.

## 2. Experimental Component: Production of the Test Plates

Experimental tests of vibration and noise where aimed to assess the absorption capability of composite sandwich structures with different internal configurations. For this purpose, it was necessary to fabricate smaller samples of the composite sandwich panels (plates) with different compositions.

The choice of the **Hand Lay-Up** method was the most suitable to produce the test plates, because of the available infrastructure, its lower costs, and its simplicity in both procedure, necessary material and tools.

The materials chosen from CIDIFA were: Glass Fibre (GF); Carbon Fibre (CF); Foam AIREX C70.75, with 3 different thicknesses 2, 5 and 10mm (F2, F5 and F10); Resin Epoxy SR 1500 and Hardener SD 2505. In both fibres, unidirectional strips were chosen for manufacture, so that the author could select the orientation of the layers without restrains and because it was cheaper than the bidirectional strips.

One of the first aspects to be defined was, necessarily, the geometry of the test plates to be manufactured.

This parameter was considered analogous in all fabricated plates, to guarantee a constant of comparison between them. Due to the limitations of raw material, the plates needed to have dimensions that were big enough to guarantee good results experimentally, but not so big that implied a great expense of material. In addition, it was also intended to develop a panel that was easy to manufacture, handle manually in experimental tests and that presented a good aspect ratio (length vs height), as well as to simplify the analysis of properties and its computational modelling. The aspect ratio (AR) of a plate influences various characteristics of its behaviour. For example, consecutive modes of a plate tend to growth if the AR increases (AR up to 2.5 are considered good) [5]. Therefore, the selected AR was 3/2, with a geometry of 300mm x 200mm (Length x Height), which fulfilled all the requirements, like reducing the consecutive modes, for example.

In account to the material used and since the entire process of plate development, testing and analysis of results was an iterative process, the author chose to start with relatively simple plates, of a single material (GF and CF), that served as reference for the remaining plates. Subsequently, were introduced sandwich plates (with a foam core) with 2 and 3 materials that allowed to draw conclusions regarding the best internal structure for the stipulated goal. The orientation of the fibres was defined in stacks of [0,90,90,0] or [0,90], according to the desired thickness, in order to guarantee an increase in the stiffness and resistance of the material in different load directions (x and y). The choice between 0° and 90° orientation, exclusively, was to simplify the manufacturing process, since it was manual. Orientations of ±45° would create a non-uniform distribution of the fibres, which would result in an uneven thickness of the plate. The characteristics of the plates manufactured throughout the work can be found at Table 1.

Table 1 - Characteristics of the plates developed throughout the study:

| Manufacturing Characteristics of Plates | | | | | | |
|---|---|---|---|---|---|---|
| **Plate No** | **Plate Constituition** | **Layers** | | **Stacking** | **Thickness [mm]** | **Layer Thickness hi [mm]** |
| **1** | [GF] | 8 | | [0, 90, 90, 0]s | 4 | 0,5 |
| **2** | [CF] | 8 | | [0, 90, 90, 0]s | 2 | 0,25 |
| **3** | [GF] | 4 | | [0, 90]s | 2 | 0,5 |
| **4** | [GF] + [F5] + [GF] | 4 + 1 + 4 | 9 | [0, 90, 90, 0] + [F] + [0, 90, 90, 0] | 8,5 | [0,5x4] + [ 5] + [0,5x4] |
| **5** | [GF] + [CF] + [F5] + [CF] + [GF] | 2 + 4 + 1 + 4 + 2 | 13 | [0,90] + [0, 90, 90, 0] + [F] + [0, 90, 90, 0] + [90, 0] | 9 | [0,5x2] + [0,25x4] + [5] + [0,25x4] + [0,5x2] |
| **6** | [CF] + [GF] + [F5] + [GF] + [CF] | 4 + 2 + 1 + 2 + 4 | 13 | [0, 90, 90, 0] + [90, 0] + [F] + [0, 90] + [0, 90, 90, 0] | 9 | [0,25x4] + [0,5x2] + [5] + [0,5x2] + [0,25x4] |
| **7** | [CF] + [F2] + [GF] + [F2] + [CF] | 4 + 1 + 4 + 1 + 4 | 14 | [0, 90, 90, 0] + [F] + [0, 90, 90, 0] + [F] + [0, 90, 90, 0] | 8 | [0,25x4] + [2] + [0,5x2] + [2] + [0,25x4] |
| **8** | [CF] + [F5] + [GF] + [F5] + [CF] | 4 + 1 + 4 + 1 + 4 | 14 | [0, 90, 90, 0] + [F] + [0, 90, 90, 0] + [F] + [0, 90, 90, 0] | 14 | [0,25x4] + [5] + [0,5x2] + [5] + [0,25x4] |
| **9** | [CF] + [GF] + [F10] + [GF] + [CF] | 4 + 2 + 1 + 2 + 4 | 13 | [0, 90, 90, 0] + [90, 0] + [F] + [0, 90] + [0, 90, 90, 0] | 14 | [0,25x4] + [0,5x2] + [10] + [0,5x2] + [0,25x4] |

## 3. Experimental Component: Vibration Testing

The experimental tests were designed with the purpose of quantifying, through a parameter, the reduction of vibration and noise that the plates induce. In the vibration tests, it was established that this parameter would be the **damping coefficient** (**ζ**). The calculation of this parameter was attained through the **Frequency Response Function** (**FRF**) of each plate, by two different methods. As a result, the experimental trial was designed to obtain this function. Experimental modal parameters are obtained by artificially exciting the plates and measuring the amplitude of the response at predefined points. The excitation and response signals are then sent to a signal analyser which calculates its response functions (FRF). FRF are functions that characterize each plate, in the sense that they graphically show their amplitude of response, when subjected to an excitation, in a given frequency or time interval. In this study, the FRF are presented as a function of frequency. Thus, knowing the FRF, it is then possible to apply the methods that allow the identification of the modal parameters and to reach the objective of calculating the damping coefficient of each plate [6], [7].[1]

The test was conducted in the Laboratório de Vibrações of IST using the following measurement devices:

**1) Signal Generator** - Hammer (Bruel&Kjaer Type 8203 - Serial No 1887696; Sensitivity 3,58 pC/N);

**2) Transducer** - Laser (Polytec Laser Classe II:  Model OFV 518 - Serial No 1980028 & Model OFV 2802i - Serial No 1980029);

**3) Processing Unit** - Spectral Analyzer (Bruel&Kjaer Type 3560D - Serial No 2343574);

Using the hammer and laser as signal generator and transducer, respectively, in a transient impact test, was the most logical choice since it allowed the author to perform a test without any type of mass addition, substantially reducing the experimental error. This experiment consisted in a Multi-Input/Single-Output (MISO) process. The <u>hammer</u> (MI), by its very nature, implies the introduction of more than one excitation signal, in different locations, to ensure that all its natural frequencies and modes are obtained. The <u>laser</u> (SI), in contrast, has only one response registration point, located on the left corner, so as to record all of the modes in each plate [7].

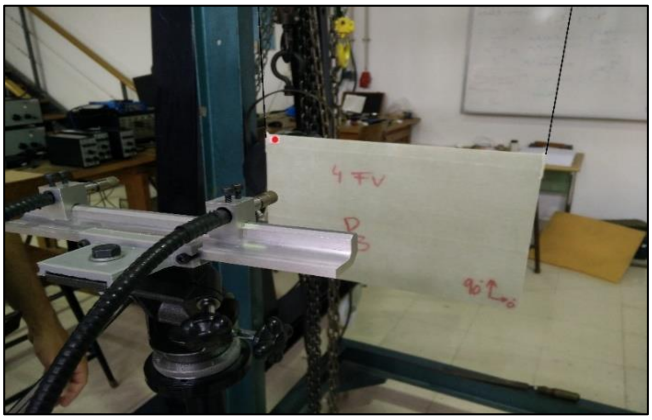

Figure 1 - Example of the experimental vibration test, using the hammer and laser.

[1] From the experimental vibration tests, performed within the thesis, only the trials with the Hammer and Laser shall be described. Also, only the RFP approximation to the damping coefficient shall be presented in this article.

During the experiments, one of the most significant aspects to be defined was how to support the plates, since it could condition the quality of the collected data. Ideally, the plate should be restricted as little as possible, so that the system can remain unchanged. If the mass of the system is altered, it can modify its corresponding natural frequencies. Thus, in the present study, all the plates were suspended by wires, which guaranteed a full range of movements and elastic conditions so smooth, that the effect of the supports was neglected [7]. Figure 1 shows one of the plates ready to be tested, while suspended and with the laser pointed.

There are several methods that allow the calculation of the modal parameters through the response functions. The most popular ones use a technique called <u>Curve Fitting</u>, which consists of adjusting the curve of a theoretical expression to the experimentally obtained FRF. Thus, finding the theoretical damping coefficient ($\zeta_M$) that is closest to the measured data. Logically, the better the adjustment of the theoretical curve to the real one, the better is the damping coefficient obtained. The zone that is adjusted is, of course, in the resonance modes (NF) and its neighbourhoods, so that the modal parameters obtained are relative to the most relevant zone of the curve, where damping is more important [7], [8].

In this work, the curve fitting method used was the <u>Local MDOF</u> (Multiple Degree of Freedom), using the **Rational Fraction Polynomials** (RFP) algorithm, from reference [9], which is applied directly to an FRF measurement in the frequency domain and uses the model of viscous damping in its analysis [7], [10]. The algorithm's input is the number of modes existing within the frequency interval in analysis, and its output is the estimated natural frequency (NF) and damping coefficient ($\zeta_M$). The analysis is performed by entering different values of m until the curve fit is satisfactory. In an acceptable curve fitting, it is typical for the number of modes entered to be much higher than the real case [10], [11]. Figure 2 exemplifies the adjustment sequence of the theoretical curve to the experimental curve for Plate 1, frequency 364 Hz.

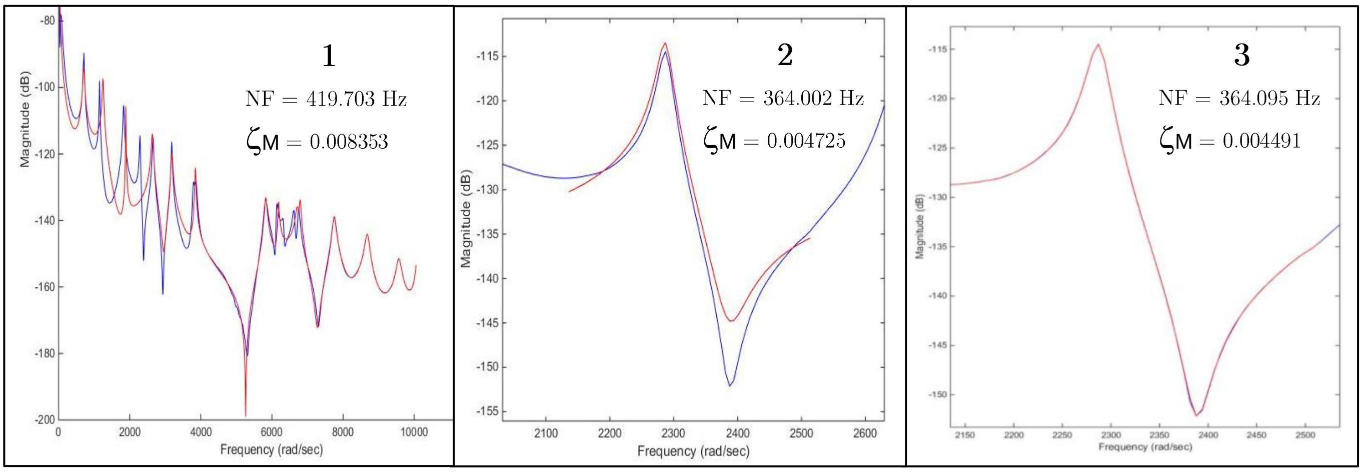


Figure 2 - Typical sequence of a curve fit (Plate 1: NF=364 Hz): **1)** Overall estimate over the entire frequency range using m=18; **2)** Analysis in the frequency range [340; 400] Hz using m=2; **3)** Analysis in the frequency range [340; 400] Hz using m=7;

Each theoretical damping coefficient pertains to its corresponding natural frequency and mode of vibration. As such, to effectively compare different plates using their damping coefficient, it is necessary to relate the damping coefficient to the same mode of vibration for each plate. Therefore, it is necessary to know exactly the modes of vibration corresponding to each natural frequency. So, by using an inverse identification technique, from references [12]–[15], it was then possible to identify the properties of the material, and to identify the three-dimensional modes of vibration by computational simulations, Figure 3.

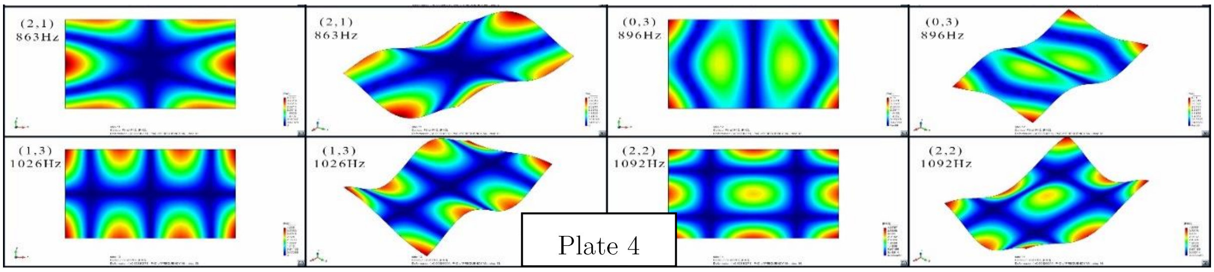


Figure 3 - Three-dimensional modes of vibration of Plate 4.

Since the excitation from the impact test is of the transient type (a temporary impulse applied to the structure), it is necessary, while processing the response signal, to add two windows (Force and Exponential) to reduce the noise and distortion present in the measurement. The exponential window adds artificial damping to all modes of the structure, so that the structure response is completely contained in the sampling window. This artificial damping is subsequently corrected by subtracting the damping added to the damping coefficient obtained ($\zeta_M$), in each plate [6], [7], [16], [17]:

$$\zeta_r = \zeta_M - \frac{1}{2\,\pi\, f_N\, \tau} \quad (1)$$

With $\zeta_r$ being the real damping coefficient, $f_N$ being the corresponding natural frequency and $\tau$ being the time constant.

## 4. Experimental Component: Noise Testing

In the noise test the established parameter used to quantify the noise reduction, caused by the plates, was the insertion loss (IL). The experimental trial was based on the work developed by PCB and presented by Doctor Andrew Barnard in the reference video [18], it was then complemented with the help of Professor Hernâni Lopes, from Instituto Superior de Engenharia do Porto (ISEP).

The experimental test aimed to quantify the amount of noise that is lost from the sound source when plates are introduced, so as to simulate, as closely as possible, the actual sound insulating panel between the engine (sound source) and the avionics. The IL is calculated by making the ratio between the pressure levels of the sound source without enclosure and with enclosure (plates) [19]:

$$IL\ [\mathrm{dB}] = 20\ \log\left(\frac{Pressure\ without\ enclosure\ (2')}{Pressure\ with\ enclosure\ (2)}\right) = L_{P2}{'} - L_{P2} \qquad (2)$$

The test was conducted in the Laboratório de Ótica e Mecânica ExperimentaL (LOME) of FEUP using the following measurement instruments:

1) **Signal Generator** - 4 Tectonic Loudspeakers - model TEBM36S12-4/A;
2) **Transducer** - Acoustic Probe G.R.A.S. 46AE - ½" CCP Free-field Microphone;
3) **Signal conditioner** - Acoustic Control TRX 500;
4) **Amplifier** - Power Amplifier PA-700- Brimaquinas - 70 Watts;
5) **Processing Unit** - Multi-Spectral Analyzer - Oros OR35;

Since the trial was performed on an open room, the probe was placed at a relatively close distance from the sound source, so as to reduce noise dissipation. The probe was therefore placed at a 22.8cm distance from the centre of the plate and at 14.8cm of height, Figure 4. For a more rigorous analysis, the signal introduced in the loudspeakers was white noise, which is a signal of random amplitude over a frequency range (in this case [0; 1600] Hz). The test consisted in first measuring the pressure of the white noise in the sound source (loudspeakers), and stipulating that signal as the reference measurement. Then, under the same conditions as the reference test, each plate was introduced between the loudspeakers and the microphone and the sound pressure recorded, Figure 4.

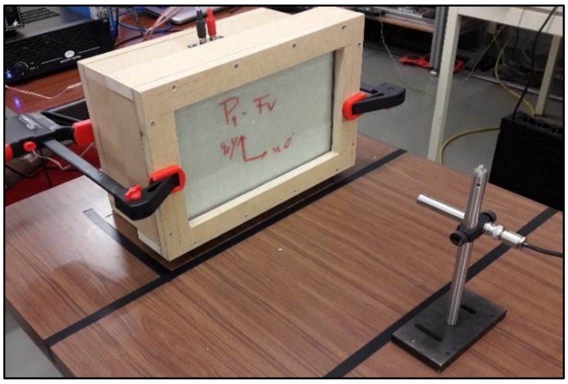

Figure 4 - Experimental noise test.

## 5. Analysis of Results:

### i) Vibration (RFP Method) - Damping Coefficient ($\zeta_r$):

Table 2 - Results for damping coefficient $\zeta r$ (corrected) and presented in percentage ($\zeta r$ x 100), for all plates:

| Corrected Damping Coefficients ($\zeta_r$) using the RFP Method | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Modes | | Plate 1 - GF (4mm) | | Plate 2 - CF (2mm) | | Plate 3 - GF (2mm) | | Plate 4 - GF + F5 (8,5mm) | | Plate 5 - GF + CF + F5 (9mm) | |
| | | FN [Hz]: | $\zeta r$ | FN [Hz]: | $\zeta r$ | FN [Hz]: | $\zeta r$ | FN [Hz]: | $\zeta r$ | FN [Hz]: | $\zeta r$ |
| 1 | (1 , 1) | 113 | 0,686% | 65 | 0,773% | 58 | 0,726% | 273 | 0,729% | 277 | 0,627% |
| 2 | (0 , 2) | 183 | 0,211% | 157 | 0,219% | 105 | 0,117% | 425 | 0,402% | 491 | 0,373% |
| 3 | (1 , 2) | 291 | 0,592% | 200 | 0,337% | 156 | 0,491% | 612 | 0,742% | 651 | 0,648% |
| 4 | (2 , 0) | 364 | 0,315% | 298 | 0,141% | 154 | 0,287% | 795 | 0,642% | 868 | 0,659% |
| 5 | (2 , 1) | 422 | 0,470% | 320 | 0,302% | 190 | 0,574% | 863 | 0,631% | 934 | 0,652% |
| 6 | (0 , 3) | 506 | 0,261% | 428 | 0,096% | 291 | 0,261% | 896 | 0,700% | 970 | 0,674% |
| 7 | (1 , 3) | 604 | 0,503% | 464 | 0,265% | 334 | 0,326% | 1026 | 0,792% | 1091 | 0,735% |
| 8 | (2 , 2) | 614 | 0,645% | 418 | 0,379% | 297 | 0,648% | 1092 | 0,822% | 1153 | 0,794% |
| Medium Value | | 0,460% | | 0,314% | | 0,429% | | 0,683% | | 0,645% | |

| Modes | | Plate 6 - CF + GF + F5 (9mm) | | Plate 7 - CF + F2 + GF (8mm) | | Plate 8 - CF + F5 + GF (14mm) | | Plate 9 - CF + GF + F10 (14mm) | |
|---|---|---|---|---|---|---|---|---|---|
| | | FN [Hz]: | $\zeta r$ | FN [Hz]: | $\zeta r$ | FN [Hz]: | $\zeta r$ | FN [Hz]: | $\zeta r$ |
| 1 | (1 , 1) | 279 | 0,711% | 186 | 0,635% | 346,0 | 0,750% | 428 | 0,842% |
| 2 | (0 , 2) | 539 | 0,418% | 375 | 0,241% | 690,0 | 0,482% | 732 | 0,710% |
| 3 | (1 , 2) | 687 | 0,675% | 468 | 0,446% | 855,0 | 0,708% | 920 | 0,910% |
| 4 | (2 , 0) | 998 | 0,659% | 633 | 0,345% | 1210,0 | 0,755% | 1319 | 1,003% |
| 5 | (2 , 1) | 1049 | 0,674% | 671 | 0,364% | 1285,0 | 0,850% | 1414 | 0,977% |
| 6 | (0 , 3) | 1172 | 0,777% | 701 | 0,310% | 1403,0 | 0,898% | 1239 | 1,102% |
| 7 | (1 , 3) | 1252 | 0,795% | 783 | 0,472% | 1520,0 | 0,873% | 1541 | 1,027% |
| 8 | (2 , 2) | 1593 | 1,037% | 816 | 0,419% | | | | |
| Medium Value | | 0,718% | | 0,404% | | 0,759% | | 0,939% | |

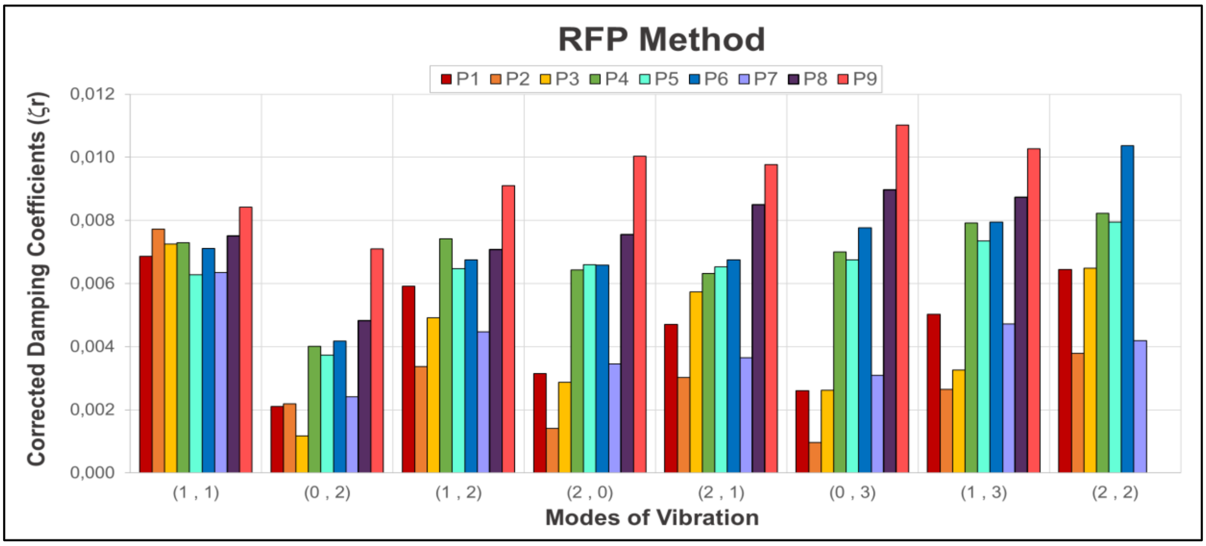


Figure 5 - Graphical representation of the results for damping coefficient ζr (corrected), obtained by the RFP method.

## ii) Noise - Insertion Loss (IL):

Table 3 - Maximum and average values of IL and pressure for Plates and Sound Source (Reference):

| Pressure and Insertion Loss (IL) of Plates and Sound Source (Reference) | | | | | | | |
|---|---|---|---|---|---|---|---|
| Interval | [0;1600] Hz | | | | | | |
| | Maximum Values | | | Medium Values | | | |
| Plates | IL Max | Max Pressure [Pa] | Max Pressure [dB] | Med Pressure [Pa] | Med Pressure [dB] | IL [dB] | Noise Reduction [%] |
| Plate 1 | 50,69 | 0,0687 | 70,72 | 0,0048 | 47,62 | 12,24 | 20% |
| Plate 2 | 38,02 | 0,1246 | 75,89 | 0,0092 | 53,23 | 6,63 | 11% |
| Plate 3 | 41,19 | 0,0593 | 69,44 | 0,0068 | 50,61 | 9,25 | 15% |
| Plate 4 | 49,98 | 0,0266 | 62,49 | 0,0027 | 42,50 | 17,36 | 29% |
| Plate 5 | 50,34 | 0,0238 | 61,50 | 0,0026 | 42,37 | 17,49 | 29% |
| Plate 6 | 47,70 | 0,0215 | 60,61 | 0,0023 | 41,37 | 18,49 | 31% |
| Plate 7 | 50,70 | 0,0426 | 66,58 | 0,0035 | 44,87 | 14,99 | 25% |
| Plate 8 | 49,96 | 0,0164 | 58,30 | 0,0021 | 40,38 | 19,48 | 33% |
| Plate 9 | 50,67 | 0,0126 | 56,00 | 0,0021 | 40,27 | 19,59 | 33% |
| Reference | | 0,0551 | 68,80 | 0,0197 | 59,86 | Nr [%] = IL / Pmed (dB) | |

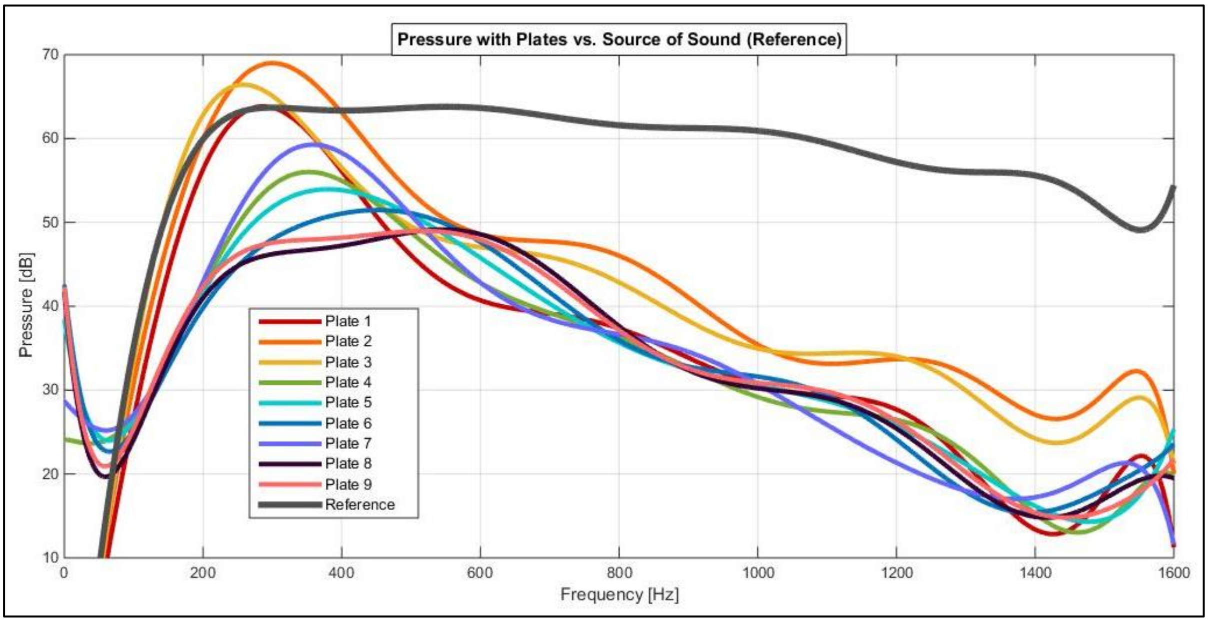


Figure 6 - Pressure curves approximated by a 10th degree polynomial, for all plates and reference.

As can be seen in the results from both experimental tests (tables 2 and 3), the plate with the worst performance (in red) was Plate 2 with a damping coefficient of only ≈0,314%, on average, and an insertion loss of only 6,63 dB, which means a decrease of about 11% when compared to the sound source (reference). The best results (in green) came from Plate 9, almost 3 times better than Plate 2, with a damping coefficient of ≈0,939%, on average, and an insertion loss of 19,59 dB, a decrease of about 33% when compared to the sound source (reference).

From the two experimental tests (vibration and noise) the following conclusions were drawn:

1) **Glass Fibre vs. Carbon Fibre - Which material is best for reducing vibration and noise?** In both trials the conclusion was the same, glass fibre is a better absorber than carbon fibre. In the vibration trial glass fibre presented an increase in the damping coefficient of ≈0,12%, at a minimum (P3 vs P2). In the noise trial the decrease of noise, compared to carbon fibre, was 3 dB (≈4%), at a minimum (P3 vs P2).

2) **Effect of Foam Introduction - what is the effect of introducing or increasing foam?** From both trials the same conclusion can be reached: the introduction of foam or the increase of its thickness can significantly improve the absorbing capability of the composite. In the case of vibration, the results are more significant, having in some situations doubled the damping coefficient. In its best instance, increasing the thickness of the foam can improve the damping coefficient in ≈0,36% (P8 vs P7). In the noise test, introducing foam in a plate can lower the noise levels in about 5 dB (≈9%) (P4 vs P1).

3) **2 vs. 3 Materials - how does the introduction of another material, as reinforcement, alters the performance of the plate? And what is the best configuration for the exterior and interior of the sandwich plate**? The conclusions drawn from both tests were: adding another material as a reinforcement is only favourable if the configuration sequence is the same as Plate 6, with carbon fibre on the outside and glass fibre on the inside. In the vibration tests the growth of the damping coefficient is ≈0,04% and in the noise test the reduction is only 1 dB (≈2%). Even though Plate 6's configuration exhibits better results than Plate 4 (with only 2 materials), the difference is so small that it does not compensate the increase in manufacturing costs (more than double). One probable reason why the structure of P5 isn't as favourable as P6 is that, the absorption capability of a composite sandwich is related to the malleability of its core (foam) and the stiffness of its exterior. These characteristics ensures an increased resistance to shearing and bending stresses. Since CF is more rigid than GF, if GF is in the exterior and CF is in the interior, the advantages of a composite sandwich are deteriorated.

4) **Single vs. Double Core - what is the best setting for vibration and noise reduction? A plate with a single foam core or two (separated on the inside by glass fibre)?** The comparison between plates

6, 7, 8 and 9, in both trials, confirmed that double core plates are worse at reducing vibration and noise than single core plates with the same thickness. In addition, the increase in damping of a thicker double core plate (P8) versus a thinner single core plate (P6) is substantially lower (≈0,04%), than, for example, the growth shown by increasing the same amount of thickness in a single core plate (P9) (≈0,22%). In the noise tests P6's noise reduction is approximately 4 dB (≈6%) higher than P7 (with double core), and only 1 dB (≈2%) lower than P8. Which means that it's almost the same having a thinner single core plate than a thicker double core plate for vibration and noise reduction.

In short, it can be stated that the best plates for vibration and noise reduction, in their different categories (when compared to other plates of same thickness or configuration), are **P1**, **P4**, **P6**, and **P9**, in order.

## 6. Conclusions:

There are several solutions that can be implemented to reduce vibration and noise inside the UAV, the best one depends on the type of restrictions required by the Portuguese Air Force. The author presents the two most common cases and the best resolution for each:

**a)** **No Constraints of Cost, Weight or Thickness** - For this case the author suggests **Plate 9** as the solution for both problems, since it is the plate that consistently guarantees a better performance. It costs about 51,61€, weighs around 430g, is 14mm thick and has a decent damping coefficient (≈0,939%) and a great insertion loss (19,59 dB (≈33%)).

**b)** **Constraints of Cost, Weight or Thickness** - In the worst case scenario, if the limitations are very restrictive then the author's choice would be **Plate 3**, since it is cheap (15,17€), lightweight (226g), thin (2mm) and guarantees a moderate damping coefficient (≈0,429%) and insertion loss (9,25 dB (≈15%)). If the constrains are more flexible, then the best possible solution would be **Plate 4**, it is twice the price of P3 (30,52€), weighs around 447g and as an average thickness (8,5mm) but guarantees a much better damping coefficient (≈0,683%) and insertion loss (17,36 dB (≈29%)), almost at the same level of P9.

In conclusion, the author discovered that the simpler solutions are usually better suited to solve the presented problems, namely plates made only of glass fibre (P1 or P3) and with a single foam core (P4 or P9). The author presents several solutions to different problems, with the notion that for each, there is always some level of commitment, whether it be in price, weight, thickness, or damping. The best solution to a problem always depends on the type of limitations present.

## 7. Acknowledgements:

The financial support of FCT, Fundação para a Ciência e Tecnologia, Portugal, through IDMEC, under LAETA, project UID/EMS/50022/2013, and interinstitutional projects from LAETA, is acknowledged.

## 8. References:

[1] J.R.Wilson, "The evolution of UAV avionics," *Military & Aerospace Electronics*, 2005. [Online]. Available: http://www.militaryaerospace.com/articles/print/volume-16/issue-9/features/special-report/the-evolution-of-uav-avionics.html. [Accessed: 01-Jan-2016].

[2] Federal Aviation Administration, "Chapter 11: Communication and Navigation," in *Aviation Maintenance Technician Handbook–Airframe Vol. 2*, United States Department of Transportation, 2012.

[3] Joseph Ellison; Goodarz Ahmadi ; Mike Kehoe, "Passive Vibration Control of Airborne Equipment Using a Circular Steel Ring," *NASA Dryden Flight Res. Facil. Edwards, CA*, 1997.

[4] Academia da Força Aérea, "PERSEUS." [Online]. Available: http://www.academiafa.edu.pt/subPagina-10D00-019.005.003.005-perseus.

[5] J. N. Reddy, *Mechanics of Laminated Composite Plates and Shells: Theory and Analysis*, 2nd ed. 2004.

[6] B. J. Schwarz and M. H. Richardson, "Experimental Modal Analysis," *CSI Reliability Week*. Vibrant Technology, Inc., Jamestown, California, 1999.

[7] D. J. Ewins, *Modal Testing: Theory, Practice and Application*, 2nd ed. Research Studies Press Ltd., 2000.

[8] Peter Avitabile, "Experimental Modal Analysis," *Sound Vib.*, no. January, 2001.

[9] C. A. G. Acuña, "Rational Fraction Polynomial Method," *MathWorks*, 2002. [Online]. Available: https://www.mathworks.com/matlabcentral/fileexchange/3805-rational-fraction-polynomial-method?requestedDomain=www.mathworks.com.

[10] M. H. Richardson and D. L. Formenti, "Parameter Estimation from Frequency Response Measurements Using Rational Fraction Polynomials," in *International Modal Analysis Conference*, 1982.

[11] D. L. Formenti and M. H. Richardson, "Parameter Estimation from Frequency Response Measurements Using Rational Fraction Polynomials (Twenty Years of Progress)," in *International Modal Analysis Conference (IMAC) - XX: Conference & Exposition on Structural Dynamics*, 2002.

[12] C. M. M. Soares, M. Moreira De Freitas, A. L. Araújo, and P. Pedersen, "Identification of material

properties of composite plate specimens," *Compos. Struct.*, vol. 25, no. 1–4, pp. 277–285, 1993.

[13] A. L. Araújo, C. M. Mota Soares, and M. J. Moreira De Freitas, "Characterization of material parameters of composite plate specimens using optimization and experimental vibrational data," *Compos. Part B Eng.*, vol. 27, no. 2 PART B, pp. 185–191, 1996.

[14] A. L. Araújo, C. M. Mota Soares, M. J. Moreira De Freitas, P. Pedersen, and J. Herskovits, "Combined numerical-experimental model for the identification of mechanical properties of laminated structures," *Compos. Struct.*, vol. 50, no. 4, pp. 363–372, 2000.

[15] A. L. Araújo, C. M. Mota Soares, J. Herskovits, and P. Pedersen, "Development of a finite element model for the identification of mechanical and piezoelectric properties through gradient optimisation and experimental vibration data," *Compos. Struct.*, vol. 58, no. 3, pp. 307–318, 2002.

[16] W. Fladung and R. Rost, "Application and Correction of the Exponential Window for Frequency Response Functions," Cincinnati, EUA, 1997.

[17] C. P. Ratcliffe, "Correcting a Significant and Consistent Error in the Modal Damping Obtained using Transient Vibration Data." Division of Engineering and Weapons, United States Naval Academy, 1998.

[18] Dr. Andrew Barnard and PCB PIEZOTRONICS, "Transmission Loss in Acoustic Testing." [Online]. Available: https://www.pcb.com/Resources/Videos/TransmissionLossInAcousticTesting.

[19] M. P. Norton and D. G. Karczub, *Fundamentals of Noise and Vibration Analysis for Engineers*, 2nd Editio. Cambridge University Press, 2003.