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# DESIGN OF A COMPACT STRUCTURE FOR VIBRATION EXPERIMENTS OF RECTANGULAR PLATES INSIDE ENVIRONMENTAL TEST CHAMBER

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***Abstract.*** *In the context of the study of vibrations and noise suppression, the Mechanical Structures Laboratory (LMest) located at the Federal University of Uberlândia is developing a prototype of a piezoelectric damper to be used with plates in the aeroacoustics environment. To this purpose, an experimental bench is being designed and tested in order to replicate the conditions during flight situations using an environmental test chamber and that the vibrations of the structure don't interfere with vibrations from the plate in the frequency band of 30 – 400 Hz. The problem was used as a case study and this paper describes the design of such experimental apparatus. The design approach was conducted at first through a series of operational constraints, including size and weight. Secondly a FEM analysis was executed in order to verify the bench's stiffness and frequency response, the result was used in the dimensioning of the bench. Following to the manufacturing of the proposed structure, a modal analysis comparison was made to verify the coherence with the finite element models.*

***Keywords:*** *Structural design, Plate, Vibration and noise suppression, Piezoelectric transducers, FEM analysis*

## 1. INTRODUCTION

Vibration and noise suppression on mechanical structures like industrial equipment and air vehicles have been a major research topic over the last decades (Qureshi et al, 2014; Yan et al, 2017). A variety of techniques have been developed intended for tackling problems of operators and passengers discomfort caused by operational noise and also the wear and fatigue of mechanical components.

According to Gripp and Rade (2018) the emergence of smart materials has unveiled new routes for innovative solutions and among them piezoelectric materials are the most mature and those with the most widespread applications.

In the context of LMest researches, a new strategy is being developed through Self-Tuning Multimodal Piezoeletric Shunt Damping integrated with flexible electronics whose prototype is already in the testing phase and needs to be validated experimentally. To this purpose, this paper is a case study and presents the design and simulation of a test bench intended to evaluate the performance of the previously mentioned device.

It's worth mentioning that the assessed problem has inherent temperature and humidity requirements, which will be addressed by a dedicated test chamber, the aircraft structural and acoustic particularities that will be reproduced by a thin panel clamped around its boundaries by continuous metal frames, also the bench dimensioning is an important requirement so that it's influence in the vibration in the frequency band of interest would be minimized.

There are several published articles with experimental apparatus (Figure 1) for tests in literature. In a recent work Aridogan and Basdogan (2015) review current state-of-the-art of active vibration and noise suppression systems for plate and plate-like structures with various kinds of boundary conditions.

About the methodology, most of the studies have used finite element models to predict forced vibration response e.g. Strassberger and Waller (2000) work that FEM supplies numerical description of the noise radiating structure and the effects of the attached piezo actuators, but also it can allow one to choose where the piezoelectric patches and sensors are to be placed as stated in Halim and Moheimani (2003) and Ducarne et al (2012) articles.

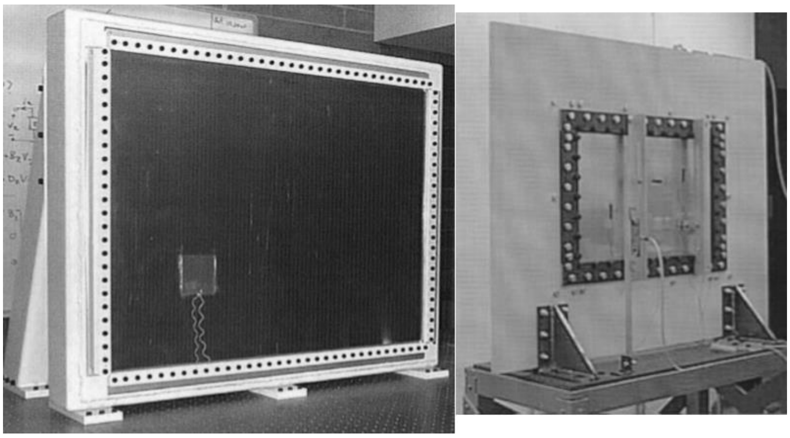


Figure 1. Experimental apparatus – left: Halim and Moheimani (2003), right: Strassberger and Waller (2000).

## 2. MATERIALS

The experimental bench was develop based on the materials available in the laboratory and presented as follows. Furthermore, the aluminum plates used in the experiments were provided by EMBRAER with dimensions of 500 x 500 x 1.7 mm thick.

### 2.1 Environmental test chamber

In order to control the testing environment i.e. temperature and humidity, a climatic chamber, ESPEC model EPL-4H (Figure 2), will be used. Such chamber has inner dimensions of 1000 x 900 x 1000 mm (length, width and height) and maximum payload capacity of 900 N distributed over the inner floor of the equipment.

In regards to the operation of this equipment it is expected some vibration noise due to its inner components, thus the structure will be place inside of the chamber by using vibrational absorbers (*vibrostops*) at its base.

### 2.2 Shaker

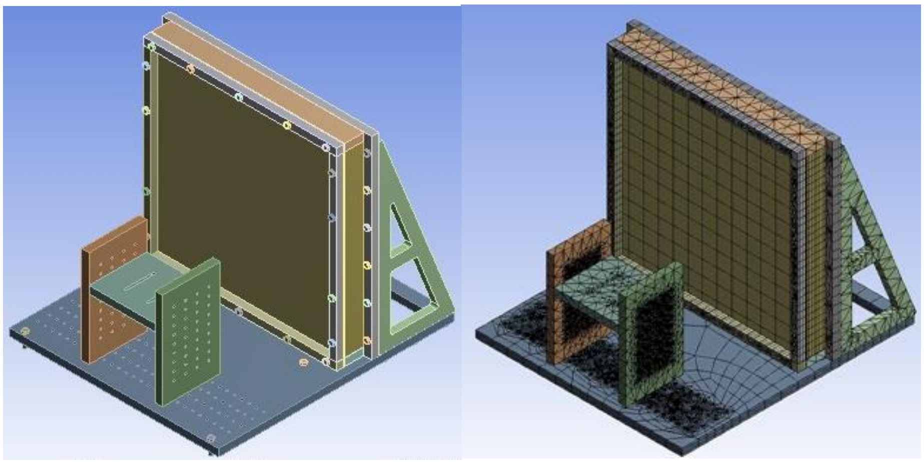
The system will be excited by the TMS K2007E01 Smartshaker (Figure 2). The electrodynamic exciter is a small, portable permanent magnet shaker with a compact precision power amplifier integrated in its base. The exciter provides up to 31N sin peak and 67N impact force.

Figure 2. Used equipment – left: Environmental test chamber EPL-4H, right: Shaker TMS K2007E01.

## 3. DESIGN AND FINITE ELEMENT ANALYSIS

Based on the study review presented in the introduction section and also the available materials, the design approach was conducted as to fulfill the constraints and follow the standards of the study field. The main objective of design was to verify the bench's stiffness and to predict the structure response in a single frequency band, in this case, lower-frequency modes 30Hz - 400 Hz – the main contributor to vibration, with higher vibration amplitudes than the higher-frequency modes (Shen and Homaifar, 2001).

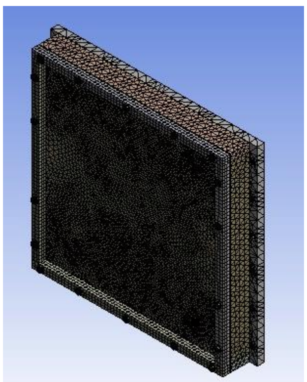
In order to specify the dimensions and components of the experimental bench, a series of FEM simulations were made with different bench geometries, aiming to minimize the displacement of the structure's components under test conditions. The resulting geometry, finite element model and the modal analysis are presented in Figure 3. The experimental bench, due to the constraints imposed by the testing equipment, was made out of ¾" aluminum plates. It was designed to be manufactured with water jet machining and assembled entirely by use of M8 bolts. It's also worth mentioning that for the modal analysis at hand, the frequencies and modes observed had displacement components mainly in the test plate, the structure's displacement were orders of magnitude above those found on the plate.



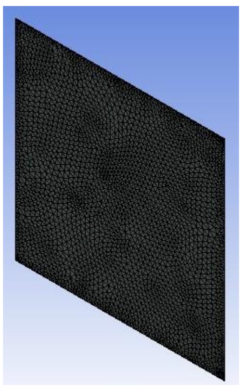
| Mode | Frequency [Hz] |
|---|---|
| 1 | 72.89 |
| 2 | 147.35 |
| 3 | 149.39 |
| 4 | 217.77 |
| 5 | 264.79 |
| 6 | 268.40 |
| 7 | 331.44 |
| 8 | 333.12 |
| 9 | 413.84 |
| 10 | 428.28 |
| 11 | 431.03 |
| 12 | 445.12 |
| 13 | 487.69 |

Figure 3. Experimental bench, Finite Element Model and Modal Analysis.

After the completion of the design phase, the focus of the finite element analysis was changed to the assess the plate's response. In order to evaluate the influence of the experimental bench on the results, several simplified models were made and assessed using the same methodology. Two models were created, the first one assumed that the plate's fixture was clamped on the structure of the bench, the second one considered only the plate itself and the clamped boundary as shown in Figure 4.



| Mode | Frequency [Hz] |
|---|---|
| 1 | 72.90 |
| 2 | 147.40 |
| 3 | 149.42 |
| 4 | 217.79 |
| 5 | 265.08 |
| 6 | 268.55 |
| 7 | 331.59 |
| 8 | 333.17 |
| 9 | 422.89 |
| 10 | 428.80 |
| 11 | 444.35 |
| 12 | 488.05 |
| 13 | 489.38 |



| Mode | Frequency [Hz] |
|---|---|
| 1 | 72.21 |
| 2 | 147.38 |
| 3 | 147.61 |
| 4 | 217.68 |
| 5 | 265.00 |
| 6 | 266.31 |
| 7 | 332.44 |
| 8 | 332.72 |
| 9 | 424.28 |
| 10 | 426.00 |
| 11 | 444.10 |
| 12 | 489.80 |
| 13 | 490.98 |

Figure 4. Simplified models and Modal Analysis - left: Fixture model, right: Plate model.

With the finite element model it was obtained the FRF in the center of the plate and the vibration modes (Figure 5). Analising the results and comparing it can be concluded that the use of the full experimental bench in the FEA has little influence, as designed, into the final result of the modal analysis. The resulting frequencies had small changes within the frequency band of interest (30 – 400 Hz) but it was possible to observe some divergences due to the presence of the various components in the experimental bench. The main differences found were on the modes at 413.84 Hz and 431.03 Hz where the strucutre displacements are higher but still lower than in the plate. It also important to note that the solution to the simplified models were obtained at a fraction of the time spent on the full model, reasserting the advantages of the simplified models.

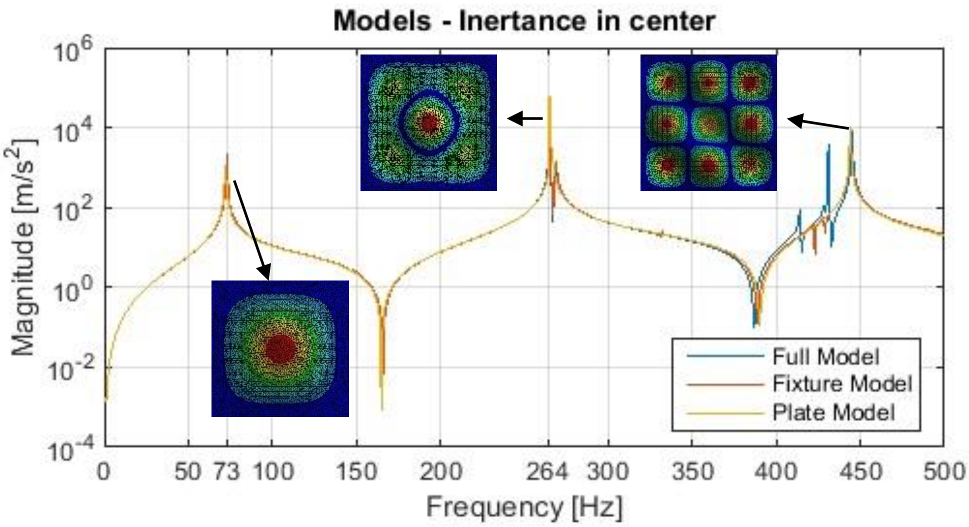



Figure 5. Inertance FRF and some vibration modes in the center of the plate.

According to the results it is observed that the experimental bench itself has minor influence on the plate’s frequency response since the complete model, the fixture model and the plate model had similar results. On the other hand, once included a more realistic type of fixture (i. e. bolts) it was possible to observed a slight change on the frequencies due to the more complex interaction of the various parts. All results will be later used to compare with the expermental data but it’s important to note that the boundary conditions on a panel can never be perfectly clamped as discussed in Shen and Homaifar (2001) work. It’s also important to emphasize that all results obtained will be relevant when considering the electro-mechanical coupling of the piezo-dampening devices.

The assessment made with the plate’s center serves as a start point to compare the different models, however because of the symmetry of such position it fails to observe certain mode shapes. Therefore, a second analysis was made with a point located at 115mm directly above the plate’s center. The results are shown in Figure 6.

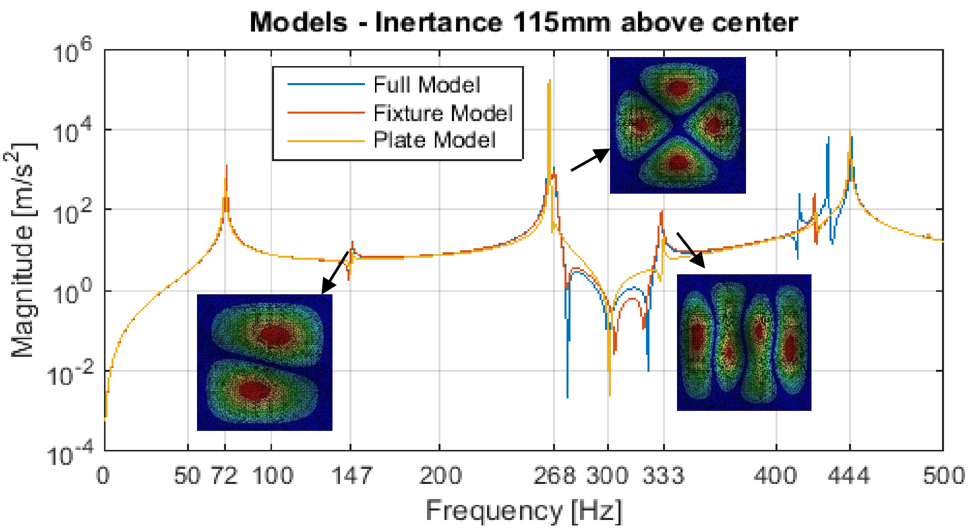
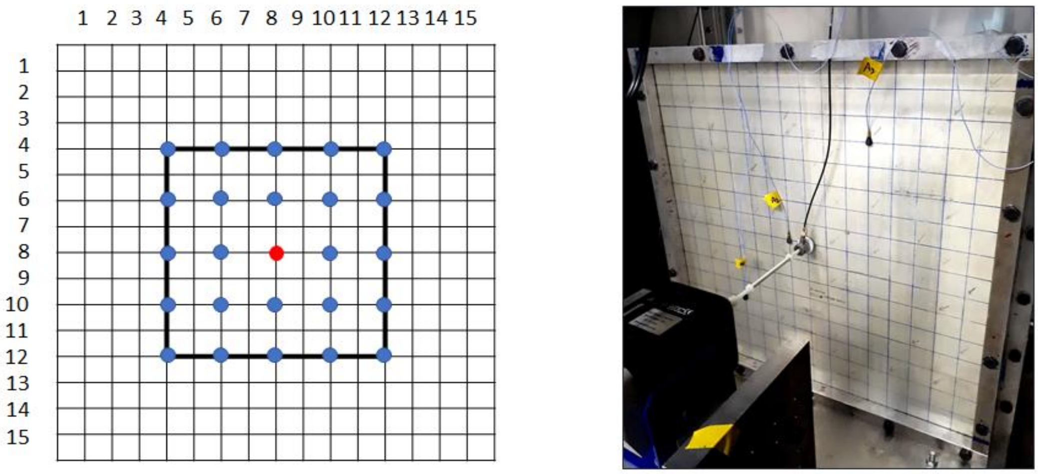



Figure 6. Inertance FRF and vibration modes at 115 mm above center.

According to the results obtained for the plate's center, the more complex the model the more issues can be observed. Although there were some differences inside the interest range, the goal of such assessment is to determine the frequencies and positions for the accelerometer's placement, which was not affect by the presence of the structure's antiresonance.

## 4. RESULTS

As preparation for the experimental phase, the test plate was subdivided into a 15x15 grid (Figure 7) of acelerometers' postions and with the shaker placed at the center (Figure 7.b). Experimental tests were conducted to evaluate the behavior of the inner 25 points as shown in Figure 7.a . The goal of such phase was to recreate the mode shapes as observed in the FEM analisys and compare the FRF of both experimental and numerical phases.




a) b)

Figure 7. a) 15 x 15 grid and test points b) Assembled experimental bench with grid positions.

In each test the acelerometer and load cell data was collected by a dynamic signal analyzer that also sent the signal to the shaker for a low level burst chirp excitation.

Figure 8 shows the FRF obtained for both the plate's center and the point 115 mm above center. As expected, given the singular particularities of the center point, it was possible to observe only the few mode shapes and natural frequencies in which the displacement distribution involved the central portion of the plate. On the other hand, through the displaced position it was possible to point out a few intermediate mode shapes since its not entirely simetric with the plate's boundaries.

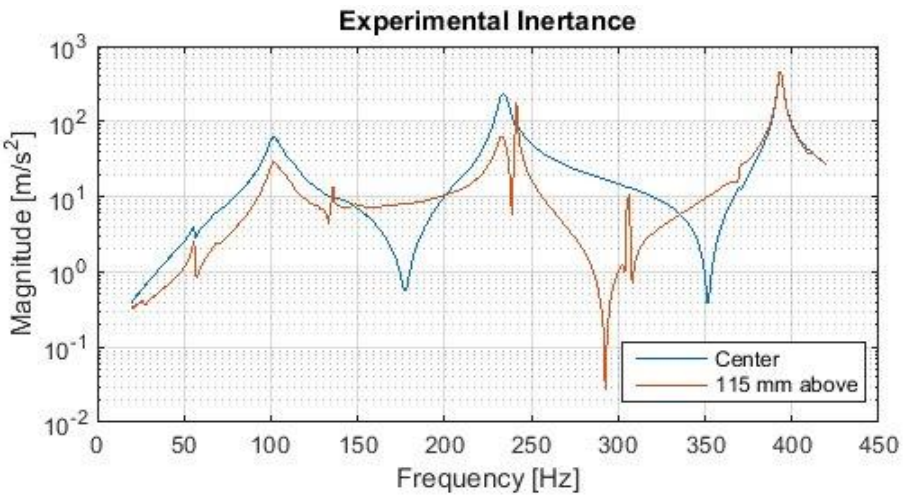



Figure 8. Experimental Inertance FRF.

In order to validate the gathered data, the theorical and experimental results were compared (Figure 9). For the experimental system at hand was observed a considerabily large difference between both experiment and numerical. It’s worth mentioning that the previous comparison made was before any parameter corretions like damping factor, rigidity and boundary conditions.

For that purpose, it was necessary a modal parameter extraction and an experimental modal analysis using a complex exponential method (Ewins, 2000). With the feedback obtained some attemps were made in order to correctly represent the fenomenom inside the Finite Element enviroment.

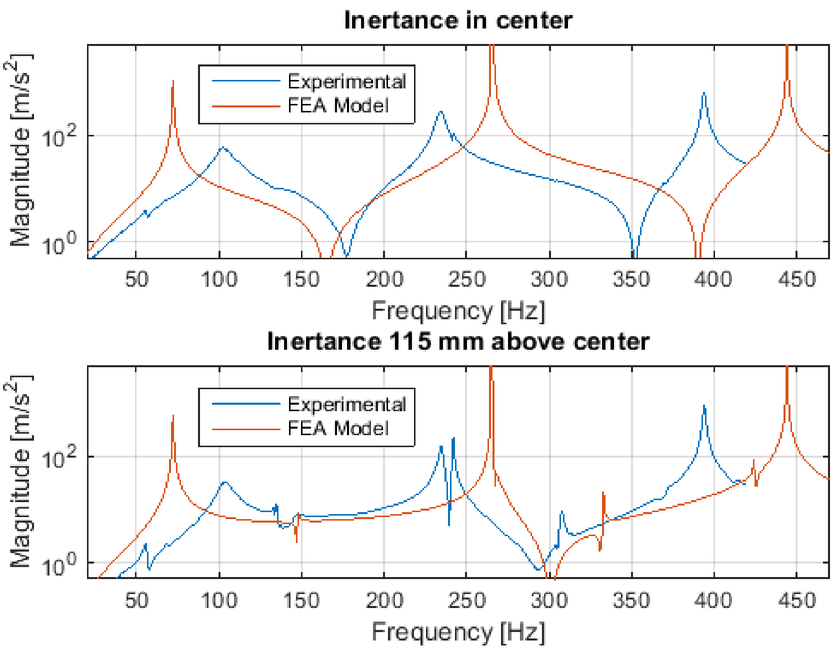



Figure 9. Comparative analysis between Experimental and finite element Plate Model.

The adjustments were made primarly on geometry parameters (mechanical imperfections of the experimental aparatus), interface between components (boundary conditions imperfections) and damping factor. Figure 10 shows the deformed plate (concave) and the FRF for the deformed plate model.

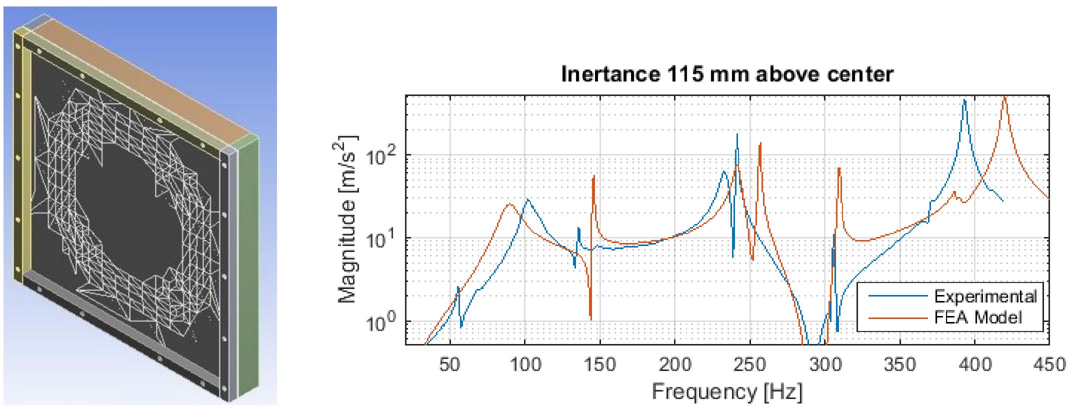



Figure 10. Deformed Plate – left: FE Model representation, right: Inertance FRF at 115 mm above center

The deformed plate model was derived of the fact that the plates used in the study had a small deflection on its center and as observed in the FRF (Figure 10) such model was able to correct a few of the deviations observed in the original flat plate model. At the current state of the analisys several factors were already taken into account. The plate mass and width were precisily measured to fit the FEM model, also the fixtures of the plate were conducted using standard practices as to avoid torsion, compression and tension, which were validated by using strain gages adhered to the plate. The bolt’s pretesion were also calculated through the torque used when tightening the bolts. The biggest influencing factor observed was that the plates were slight deformed in a concave manner. This deformation could not be precisily quantified but at some level were able to aproach the desired results.

With the corrections on the parameters a Modal Assurance Criterion (MAC) analysis was used to determine the similarity of modes of the finite element deformed plate model and the experimental model (Figure 11.a).

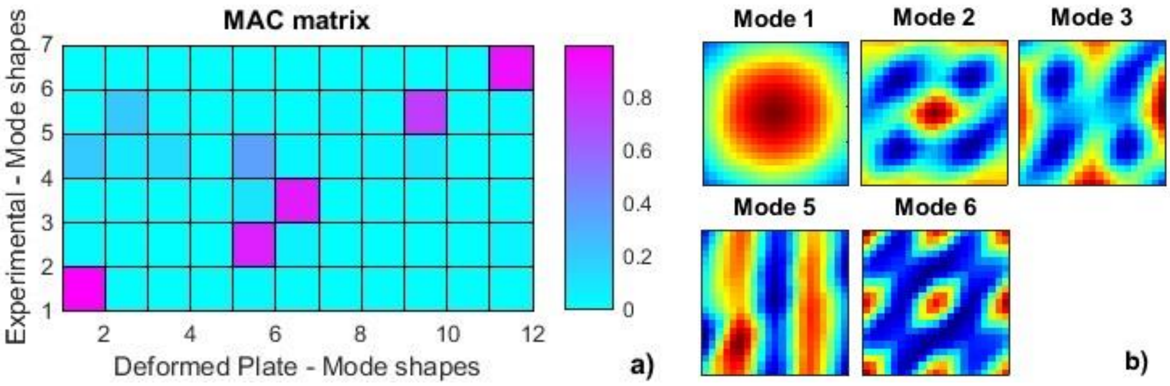



Figure 11. a) Modal Assurance Criterion matrix b) Experimental vibration modes

As observed in the MAC analysis, although the frequencies don't match perfectly five of the modes are similar (higher MAC values) and represent the structure response. Fig. 11.b shows the similar vibration modes that can be compared to the ones presented in Figures 5 and 6 for the inner 25 points (Figure 7.a). The mode around 140 Hz could not be well represented by this model and will be adressed in future developments.

Since the objective of this experimental bench is to test the prototype of a piezoelectric damper, the location of the piezoelectric is important factors to be considered. For that purpose, Ducarne et al (2012) work in beams could be useful as it demonstrate that the coupling factor is directly related to the mode shape slope difference at the piezoelectric patch ends. Such information can be used to determine the optimum location for the piezoelectric transducers and accelerometers but it's important to note that some shift of natural modes due to geometrical variabilities is expected (Gripp and Rade, 2018).

## 5. CONCLUSIONS

This paper describes the design phase of an experimental bench to assist in the developing of a prototype of a piezoelectric damper to be used with plates in the aeroacoustics environment. To replicate the conditions during flight situation, the structure should enable tests in a 500 mm x 500 mm aluminum plate and be placed inside an environmental chamber.

The main objectives were to determine the influence of the structure on the test plate vibration and to obtain a finite element model that could successful represent the fenomenom inside the Finite Element enviroment.

Following to the manufacturing of the proposed structure the theorical and experimental results were compared and was observed a considerabily large differences. Some attemps were made in order to minimize such erros and a deformed (concave) plate model was proposed that have similar results with the experimental model based on the Modal Assurance Criterion (MAC).

Several other factors (plate dimensions, density, deformation, boundary conditions, pre loading, bolt tensions) were also impactfull on the frequency response (with smaller contributions), and although those adjustments may approach the experiments, without a systematic parameter optimization, it wasn't possible to faitfully translate the plate's behaviour with the numerical approach. The optimatization of said parameters will be adressed in future developments.

At last it was discussed the usage of the mode shapes information for determine an optimum location for the piezoeletric patches that will be used in future work.

## 6. ACKNOWLEDGEMENTS

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