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INVESTIGATION OF VIBRATION BEHAVIOR OF FR4 AND ALUMINUM PCB MATERIALS USING THE MAC METHOD

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Abstract: In this study, plates made of two different materials used in PCB (Printed Circuit Board), FR4 and Aluminum, have been investigated by experimental modal analysis (EMA) and finite element method (FEM) modal analysis. Firstly, dynamic structural characteristics of plates, consisting of modal shapes, natural frequencies have been calculated by the help of ANSYS® R23 software. Afterwards, to determine the vibrational characteristics of the plates by EMA, roving hammer, roving accelerometer, and modal shaker tests have been performed by the help of DEWESOFT® data acquisition hardware and its corresponding software. Frequency response function (FRF) is measured by EMA software and FRF were exported to be analyzed by MEScope® modal analysis software. In MEScope® software, FRF's between excitation points and measurement points were matched with corresponding points of the structure geometry. Modal assurance criteria (MAC) were calculated between the EMA and FEA modal shapes. Results of impact hammer test method obtained to closer to the results of numerical analysis than other methods. It has also been observed that the dimensions of the variable plates affect the MAC results. Additionally, importance of the gravity and mass balance on the structure were analyzed specially for thin structures, indicated actions to cope with mass balance issue.

Keywords: Experimental modal analysis, Finite element modal analysis, MAC, Vibration

FR4 ve Aluminyum PCB Malzemelerinin MAC Metodu Kullanılarak Titreşim Davranışlarının İncelenmesi

Öz: Bu çalışmada, PCB'de (Baskılı Devre Kartı) kullanılan iki farklı malzeme olan FR4 ve Alüminyumdan yapılmış plakalar, deneysel modal analiz (EMA) ve sonlu elemanlar yöntemi (FEM) modal analizi ile incelenmiştir. İlk olarak, plakaların modal şekilleri, sönümleme oranları, doğal frekanslarından oluşan dinamik yapısal özellikleri ANSYS® R23 yazılımı yardımıyla hesaplanmıştır. Daha sonra, plakaların titreşim özelliklerini EMA ile belirlemek için DEWESOFT® veri toplama donanımı ve yazılımı ile birlikte darbe çekici gezdirme, ivmeölçer gezdirme ve modal sarsıcı testleri gerçekleştirilmiştir. Frekans tepki fonksiyonu (FRF), EMA yazılımı ile ölçülmüş ve FRF'den elde edilen veriler numerik analiz ile karşılaştırılmak için MEScope® yazılıma aktarılmıştır. MEScope® yazılımında, tetikleyici noktalar ile ölçüm noktaları arasındaki FRF'ler yapı geometrisinin ilgili noktaları ile eşleştirilmiştir. EMA ve FEA modal şekilleri arasında modal güvence kriterleri (MGM) hesaplanmıştır. Darbe çekici test yönteminin sonuçlarının diğer modal test yöntem sonuçlarına göre, numerik analizlere daha yakın olduğu görülmüştür ve mod şekillerinin benzerliği açıklanmıştır. Değişken plakaların boyutlarının MGM sonuçlarını etkilediği

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de gözlemlenmiştir. Ayrıca, yerçekimi ve kütle dengesinin yapı üzerindeki önemi özellikle ince yapılar için analiz edilmiş ve kütle dengesinden kaynaklanan sorunlar için aksiyonlar belirtilmiştir.

Anahtar Kelimeler: Deneysel modal analiz, Sonlu elemanlar modal analizi, MGM, Titreşim

1. INTRODUCTION

The dynamic behavior of structures is gaining significant attention and becoming increasingly popular day by day. During the product design phase, it is essential to determine the dynamic characteristics of the products, which include natural frequencies, damping ratios, and modal shapes. These determinations help in understanding how the products behave under dynamic loading, and they play a crucial role in ensuring the reliability of the designed products.

Vibration durability, in particular, poses challenges for various industries. Manufacturers, engineers, and technicians may encounter unexpected issues during laboratory testing due to vibrations exciting resonances within the structures, leading to unexpected failures. Consequently, modal analysis has gained substantial attention. At this point, modal analysis provides insights into the dynamic behavior of the structures, revealing resonance frequencies and modal shapes. This information helps us anticipate how the structures will respond to external dynamic loads. To prevent problems arising from external dynamic loads, engineers routinely perform modal analysis to identify and scrutinize the dynamic structural behavior of the structures. This type of analysis has become a standard practice in the product development stage. To validate FEA, EMA becomes a necessity for the engineers. The primary reasons for employing experimental methods are as follows:

- Ensuring the correctness of theoretical analyses and results by employing experimental tests to uncover the behavior of structures.
- Identifying vibration characteristics experimentally, which are challenging to determine through theoretical analysis.

Voorhees et al. (1984) investigated the techniques of modal testing on spacecraft structure. Modal tests were conducted using multiple shakers with sine vibration signals, single shakers with random signals, and multiple shakers with random signals. Subsequently, the results of these modal tests were obtained for each testing method and compared with each other. Furthermore, the operational times for each testing method were revealed to streamline the testing process. Hunt et al. (1983), in their study, demonstrated that modal tests performed using multiple shakers yield more reliable and accurate results in terms of coherence compared to other testing methods. Additionally, it was apparent that the use of multiple shakers significantly reduces testing time. To validate the modal vectors, the Modal Assurance Criterion (MAC) was applied, and this matrix method showed that energy is adequately distributed in a wideband by utilizing multiple shakers. Ren et al. (2015) conducted modal analysis in Ansys software by employing free boundary conditions in the finite element model to determine the natural frequencies and vibration modes of the circuit board. Ren also compared PCBs with different thickness plates and stated that the error rate between the analysis and test models did not exceed 10%. Between the experimental test methods, only impact hammer test was carried out in order to compare with numerical analysis. The modal behavior of CEM-1 single layer PCB plate investigated, one of the PCB materials, by experimental modal test methods thus resonance frequencies of the structes were identifed by the impact hammer but no numerical results were obtained to verify the test results (Anuar et al., 2013). The finite element model utilized to investigate the dynamic properties of PCBs with electronic components and eliminate the need for certain physical tests to save time. Natural frequency modes and FRF of PCBs were obtained and verified with physical test unlike modal test progress which is random vibration (Somashekar, V. N et al., 2016). Similarly, Zhang et al. (2016) conducted a study to validate the dynamic properties of a printed circuit board. A modal analysis of the circuit board was performed in a virtual environment to obtain the shape and values of the first four modes by impact hammer. Respect to modal test results, An optimization was done on circuit board with whole system to enhance the durability and suggested proper materials in order to provide high reliability. In this study, only impact hammer method was carried out so the other methods did not used but optimization was done on the materials according to impact hammer test's results.

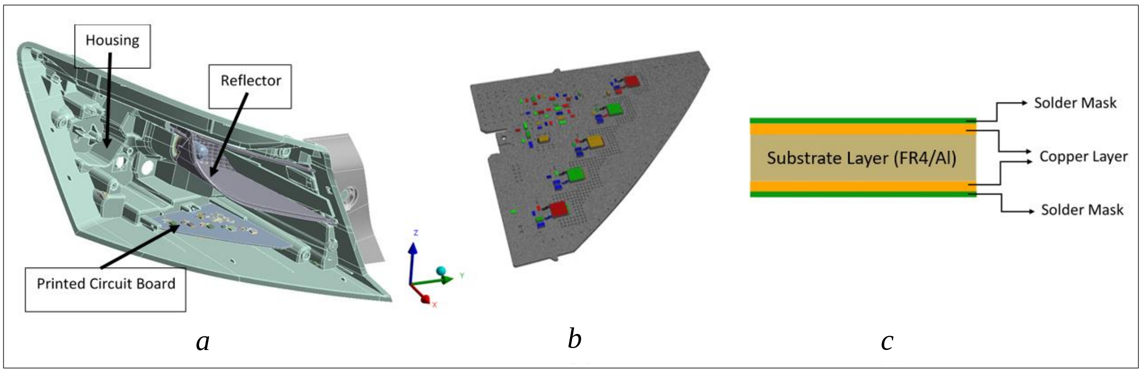


Figure 1:
*The PCB used in a lighting part **a.** Internal structure of rear lamp, **b.** printed circuit board, **c.** PCB layers (Fahri Berk et al., 2023)*

The primary objective of this investigation is to analyze the vibration characteristics of PCB plates, aiming to ascertain the most reliable EMA method that aligns consistently with the modal FEA outcomes. FR4 is the material used in the LED driver electronic card of an automotive lighting product and is a flame retardant and reinforced epoxy resin composite material. In Figure 1a shows the components used in automotive lighting. The LED driver used in the rear lamp is shown in Figure 1b. The PCB layers are formed by sandwiching the insulating FR4 material which is used so common or Al between copper layers with conductive paths on the surface, and these pathways provide the conductivity on this surface. In addition, it is covered with a solder mask in order to protect the copper layer from external factors such as dust, corrosion, and oxidation. The sandwich structure of a circuit board consisting of conductive and insulating layers is shown in Figure 1c (Fahri Berk et al., 2023). The Aluminum plate used in the vibration test, is a fixture of the electronic card also used in PCB as a substrate and preferred due to its durability, low temperature resistance, and non-magnetic properties. Various EMA test methods, including roving accelerometer, roving hammer, and modal shaker tests, were performed to obtain EMA results. The results of different EMA tests were compared with the results of the FEA analysis. These studies helped determine which EMA test methods are applicable to specific types of structures and provided solutions to challenges encountered during EMA tests.

Unlike mentioned previous studies that either focused solely on experimental modal tests or numerical analyses, this study employs both experimental modal tests and finite element analysis. This comprehensive approach provides a more thorough understanding of the vibration characteristics of FR4 PCB plates. The comparative analysis helps in understanding the strengths and limitations of different EMA techniques in evaluating the vibration characteristics of PCB plates. The study also adds an additional practical aspect to literature by, considering the aluminum plate's durability. The primary aim of our research is to demonstrate the importance of using different experimental methods for different materials, allowing us to evaluate and understand material behavior across varied plate structures more effectively. Overall, by focusing on PCBs used in automotive lighting products, the study addresses a specific application area where reliability and durability are critical factors.

2. MATERIAL AND METHODS

In all systems exposed to vibration, it is observed that the external factors causing vibration gradually to dissipate over time, leading to a gradual weakening and eventual disappearance of the vibrations. This phenomenon is attributed to the vibration dampers within the systems that absorb mechanical energy. These effects are referred to as damping and act counter to the motion of the system, generating damping forces. General formulation of damped free vibration;

$$\ddot{x} + 2\zeta\omega_d\dot{x} + \omega_n^2 x = 0 \tag{1}$$

Here w_n is the natural frequency, ζ is the damping factor and is illustrated as in equation below.

$$\zeta = \frac{c}{c_c} = \frac{c}{2\sqrt{km}} \tag{2}$$

Under the condition of $\zeta < 1$ (the condition for vibration movements to occur), the solution of the differential equation is as follows;

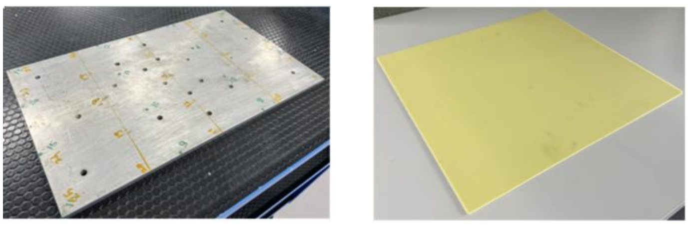
$$x(t) = e^{-\zeta w_n^t}(A_1 \sin w_d t + A_2 \cos w_d t) \tag{3}$$

The solution of the differential equation under the condition of $\zeta < 1$ leads the emergence of vibration movements. Here w_d defined as in the equation indicates that the frequency of damped vibration (Yokoyama, 2023)

$$\omega_d = \omega_n\sqrt{1-\zeta^2} \tag{4}$$

2.1. Materials used and Numerical Model configuration

The modal tests utilized the plates depicted in Figure 2. Figure 2a represents the Aluminum plate measuring 480x280x10 mm. In Figure 2b, the FR4 plate of the PCB is seen, which measures 160x400x1.5 mm.



a *b*

Figure 2:
*The plates used in the modal tests **a.** Aluminum plate **b.** FR4 plate*

In Figure 3, the workflow of this study is presented, and phase of modal analysis are showed. First of all, FEA was performed to determine the resonance frequencies and modal shapes of the structure. Afterwards, suitable EMA method was chosen. At the end MAC has been performed for comparison of results.

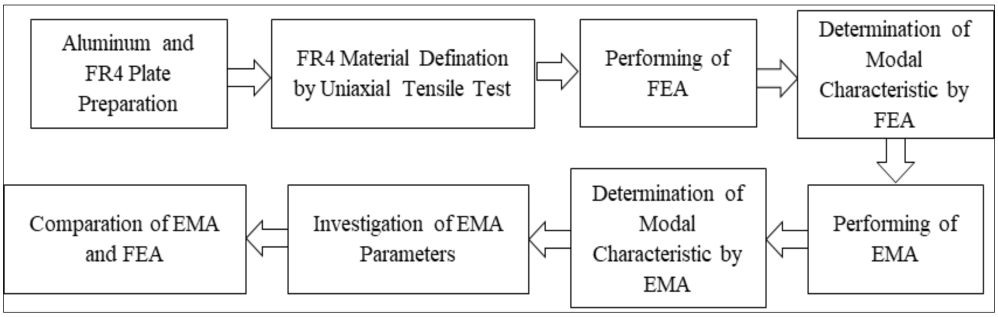
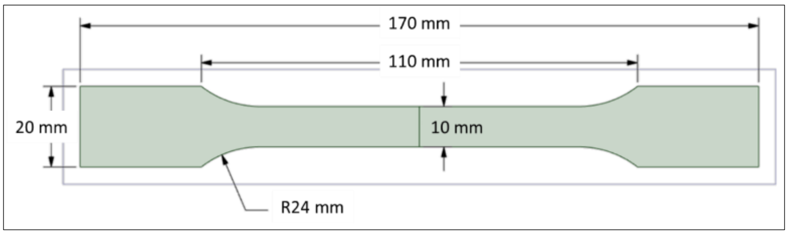


Figure 3:
Workflow for determinig the modal characteristics of Aluminum and FR4 plate

In the study, tensile test samples were prepared to determine the Elastic modulus of the FR4 material in two directions, allowing us to ascertain the orthotropic properties. The thickness of the samples was 2 mm with the dimensions given in Figure 4a in accordance with ISO 527 tensile test standard (ISO 527-4, 2021). Figure 4b depicts the sample after preparation in accordance with the dimensions specified in the ISO 527 tensile test standard.



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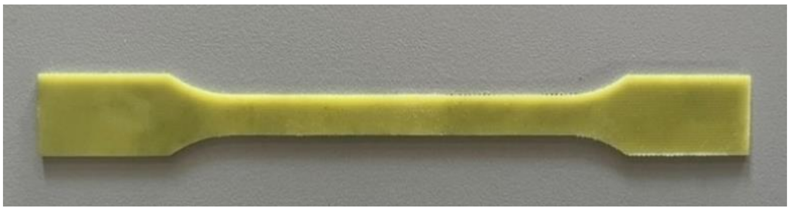


b

Figure 4:
ISO 527-4 tensile test sample
a. *dimensions* ***b.*** *untested sample*

Finite elements analysis (FEA) was performed on related plates under free free body conditions at room temperature. For the Aluminum plate material properties provided by Ansys material library was used. For the FR4 plate, ISO 527 tensile test samples were initially produced in two orthogonal in plane directions (x, y) and tested to determine the Young's modulus values. Additionally, a value from the literature was referenced for the thickness direction (z) (Amalu et al., 2012). These material properties are given in Table 1. They were used in ANSYS 2023 R2 version for modal analysis. During the finite element analysis studies secondary order solid brick elements (C3D20) were utilized.

Table 1. Material Properties of FR4 and Aluminum

Aluminum	Young's modulus (GPa)	71
	Density g/cm^3	2,77
FR4	Young's modulus (GPa) Ex/Ey/Ez	25,7 / 23,9 / 22
	Density g/cm^3	1,9

2.2. Experimental Modal Test (EMA) setup

After performing FEA, the modal shapes and deformations obtained at resonance frequencies were used to identify and label the locations for excitation and response on the test plates. Additionally, FRF geometry was constituted, which will be utilized by the EMA software. Once the excitation points were marked on the structures, the plates were suspended from a test bench using elastic ropes to ensure free free boundary conditions. During this phase, several experimental modal tests were conducted on the structures, primarily using the roving impact hammer test and roving accelerometer method for each plate. Devices used during EMA with their pictures are listed below in Figure 5.

Figure 5:
Modal test equipment

In Table 2, the name of devices shown in Figure 5 and their features are depicted with sensitivities and channel information.

Table 2. Used devices during experimental modal tests

1	DEWESoft® with 16 channel DAQ
2	NI USB-44331 with 5 channel DAQ
3	Impact hammer which is force sensitivity of 25 Mv/N
4	MB Dynamics MODAL 50A Shaker
5	Force sensor of modal shaker which is sensitivity of 11.2 Mv/N
6	IEPE accelerometer which is sensitivity of 100 mV/g
7	IEPE accelerometer which is sensitivity of 10.65 mV/g
8	DEWESoft® and MEScope® Softwares

2.2.1. Modal Test Setup of the Aluminum Plate

First, Aluminum plate was suspended using elastic ropes, and the modal test method with a roving hammer was applied on the 25 points. During the test, a leakage problem was detected. The plate was ringing due to a low damping ratio of the Aluminum. To address the leakage issue, the Aluminum plate was placed on a sponge pad, and the test was repeated. Subsequently, a roving accelerometer test was conducted on the sponge pad after performing the roving hammer test to validate the test method. Figure 6 shows the Aluminum test sample marked at response and excitation points, suspended using elastic ropes from the designated test bench to capture flexible mode shapes.

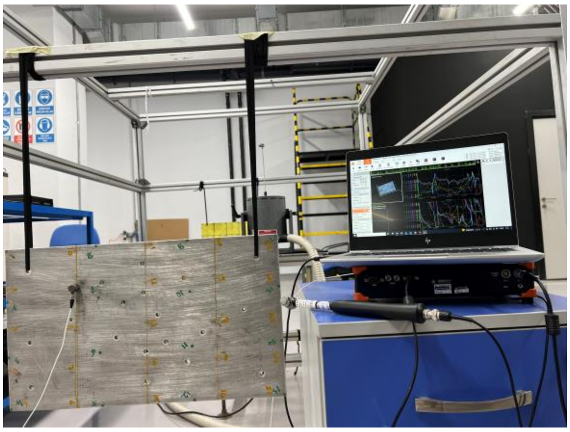


Figure 6:
Aluminum plate test with elastic ropes

In Figure 7, the Aluminum plate test sample, identified with response and excitation points, is placed on a sponge pad to mitigate leakage issues and facilitate damped vibration.

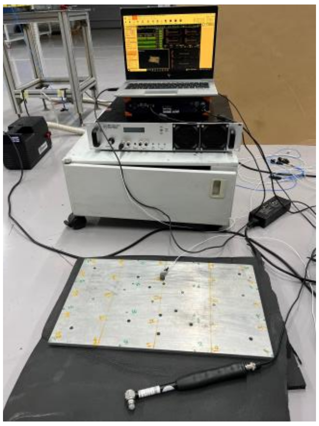


Figure 7:
Aluminum plate test with damping mat

Lastly, A modal test was conducted on the Aluminum plate using a modal shaker, as depicted in Figure 8. Upon analyzing the modal test outcomes, it became evident that the modal shaker was better suited for heavier components characterized by higher inertia. Additionally, it was observed that the force transducer used with the modal shaker was not suitable for testing lighter components like plates, due to its measurement range.

As a result, the modal shaker test was excluded for the FR4 plate. Instead, alternative modal testing methodologies were utilized for the FR4 plate.

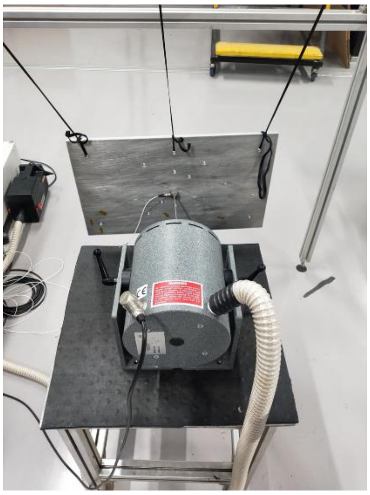
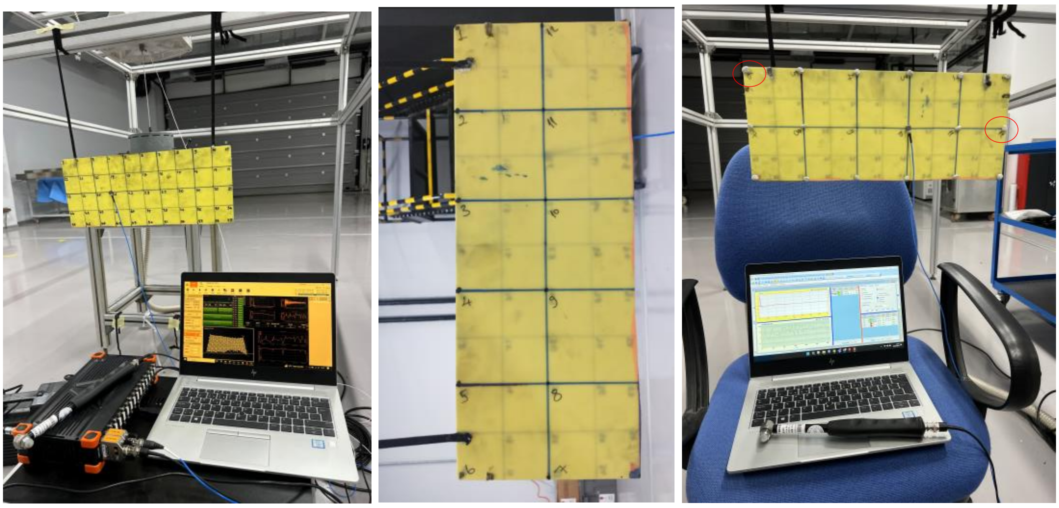


Figure 8:
Aluminum plate test with modal shaker

2.2.2. Modal Test Setup of the FR4 Plate

Testing the FR4 plate by suspending it with elastic ropes is adequate, considering the material composition and weight of the plate, both of which inherently contribute to sufficient damping. The test sample of the FR4 PCB plate is depicted in Figure 9. As seen in the picture, the plate was discretized into 55 and 18 equidistant points, and the hammer was sequentially positioned at each of these points for impact excitation and positioned one points for roving accelerometer. Signals were then recorded by a fixed accelerometer, employing data acquisition tools, and subsequently transferred to a computerized environment. Subsequently, the acquired data was subjected to an analysis that encompassed the evaluation of FRF, data consistency, and modal verification. After all tests were performed and evaluated, the FRF results, the resonance points and damping ratios of each plate were obtained. Thus, the differences between the 55 point structure and the 18 point structure were evaluated for the hammer applied in each application. According to results, the effect of increasing DOF points for the results were analyzed. Specifically, after employing EMA methods on the FR4 plate, roving accelerometer tests were performed once more with rubbers of weights equal to those of the accelerometers at each point, in order to reinforce the results and to prevent mass imbalances and maintaining the center of gravity. This aspect of modal results influenced by accelerometer weight was highlighted in a previous study. (Ay et al., 2019)

It is necessary to compute certain matrix methods by identifying them from FRF values obtained through analyses and tests in order to determine the modal shapes of the plates.



a *b* *c*

Figure 9:
*FR4 plate test with elastic ropes **a.** 55 points plate **b.** 18 points plate **c.** 18 points plate with rubbers positioned on each point*

2.3. Modal Assurance Criterion (MAC)

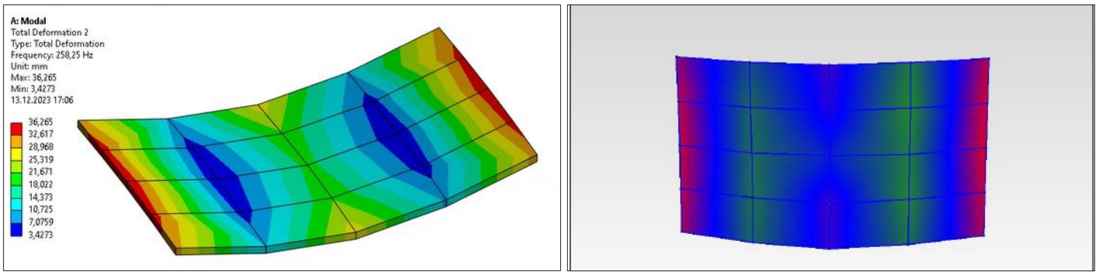
The Modal Assurance Criteria (MAC) is an important parameter in modal analysis, serving as a fundamental metric for comparing modal shapes and assessing the degree of similarity between them. MAC is quantified according to Equation 5, which leverages the square of the correlation between two modal vectors, denoted as φ_r and φ_s. The MAC value is bounded within the range of 0 to 1. A MAC value approaching zero signifies substantial dissimilarity between mode shapes, while a value approaching 1 indicates a high degree of similarity, rendering the mode shapes suitable for comparison. This can also be evaluated as the square of the correlation between two modal vectors φ_r and φ_s, where $*$ and t represent the complex conjugate and transpose of the vector, respectively. (Maia et al., 1998)

$$MAC(\{\varphi_r\},\{\varphi_s\}) = \frac{[\{\varphi_r\}^{*t}\{\varphi_s\}]^2}{(\{\varphi_r\}^{*t}\{\varphi_s\})(\{\varphi_s\}^{*t}\{\varphi_s\})} \tag{6}$$

MAC calculations can be performed between FEA-EMA, FEA-FEA, EMA-EMA in order to validate and compare. It should be noted that MAC calculations are based on comparison of modal shapes, not the resonance frequencies.
In this study, FEA-EMA and EMA-EMA, MAC calculations have been performed and the FRF results of FR4, and Aluminum plates were initially imported into MEScope® to compare EMA and FEA modal shapes. These results were then matched with excitation and response points of the EMA geometry defined in MEScope®. Following this, modal shapes were obtained for each plate, and animations illustrating these modal shapes were inspected. Furthermore, point matching between the EMA and FEA geometries was performed, and the MAC matrix was calculated as part of the analysis.

3. RESULTS

Initially, Ansys Mechanical 2023 R2 was employed to solve for the first two flexible body mode shapes of the Aluminum plate, depicted and subsequently, a similar approach was used for the FR4 plate in Ansys Mechanical 2023 R2, revealing its first and second flexible body mode shapes showcased in Figure 10. It is observed that the numerically and experimentally obrtained mode shapes are highly consistent with each other.



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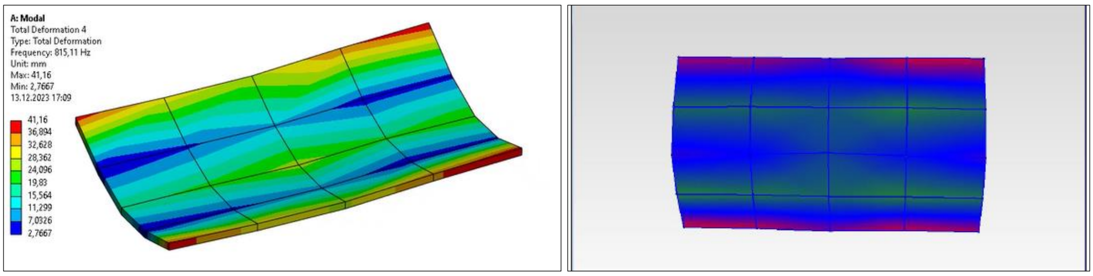

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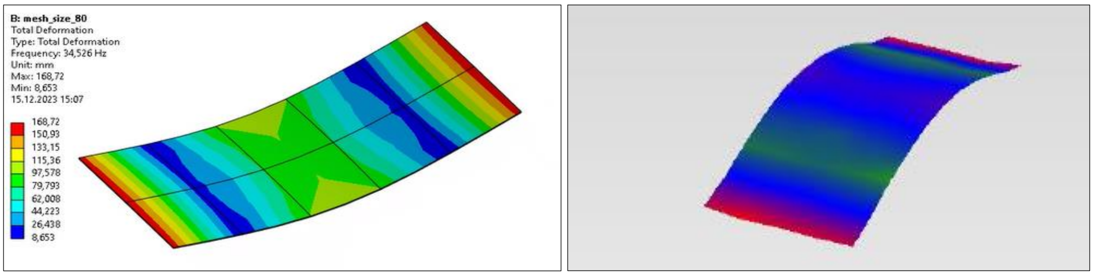

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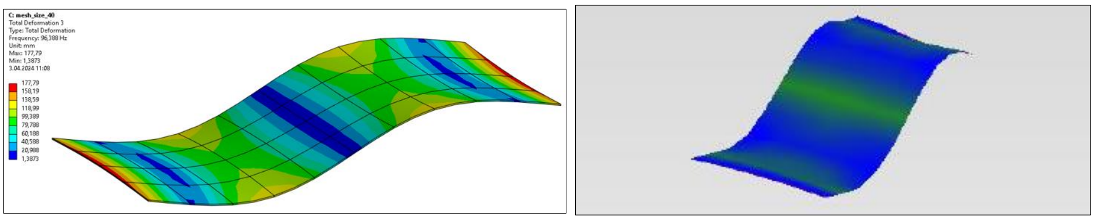


d

Figure 10:
Numerical modal analysis and experimental modal analysis of the plates
***a. Left** / 1st mode shape of Al plate FEA – 258,25 Hz **right** / 1st mode shape of Al plate EMA– 250 Hz*
***b. Left** / 2nd Mode shape of Al plate FEA – 815,11 Hz **right** / 2nd Mode shape of Al plate EMA – 724 Hz*
***e. Left** / 1st Mode shape of FR4 plate FEA – 34,526 Hz **right** / 1st Mode shape of FR4 plate EMA– 35,5 Hz*
***f. Left** / 2nd Mode shape of FR4 plate FEA – 96,4 Hz **right** / 2nd Mode shape of FR4 plate EMA – 98,3 Hz*

3.1. FRF Results of Experimental Modal Test (EMA) of Aluminum Plates

In all figures presenting FRF results of both aluminum and FR4 plate, the two most significant modes are selected and represented as two distinct curves. In Figure 11, illustrates the FRF results of the roving impact hammer test conducted on the Aluminum plate. Within this graphic, the presence of a leakage problem between the resonance points is observed, as previously mentioned. Excitation point (E.point) and response point (R.point) were given in Figure 11.

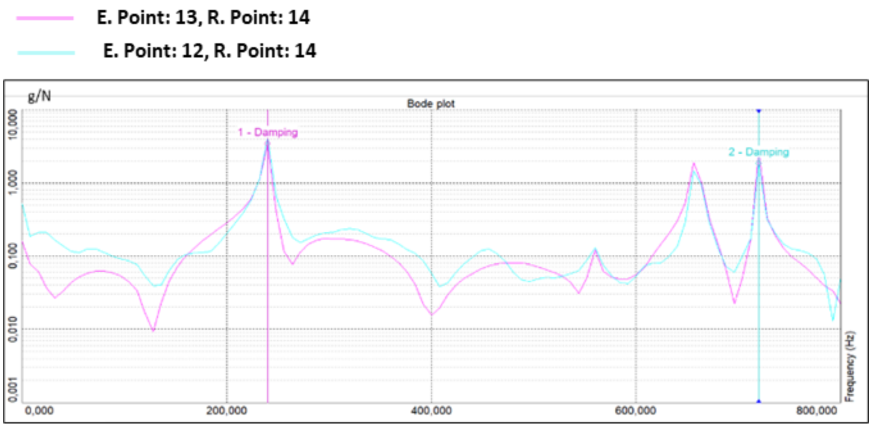


Figure 11:
FRF results of Al plate (roving impact hammer test with elastic ropes)

Moreover, in Figure 12, the Frequency Response Function (FRF) outcomes for the Aluminum plate are presented. This graph was generated through the same roving impact hammer test; however, it involved the use of a damping mat to address issues with the elastic ropes. Excitation point (E.point) and response point (R.point) were given in Figure 12.

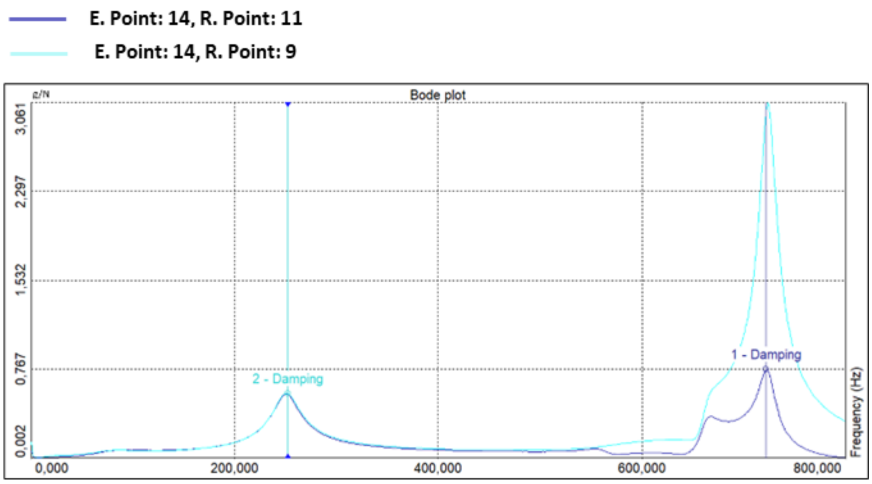


Figure 12:
FRF results of Al plate (roving accelerometer test with damping mat)

In Figure 13, the FRF results of the roving accelerometer test conducted on the Aluminum plate are depicted. Excitation point (E.point) and response point (R.point) were given in the Figure 13.

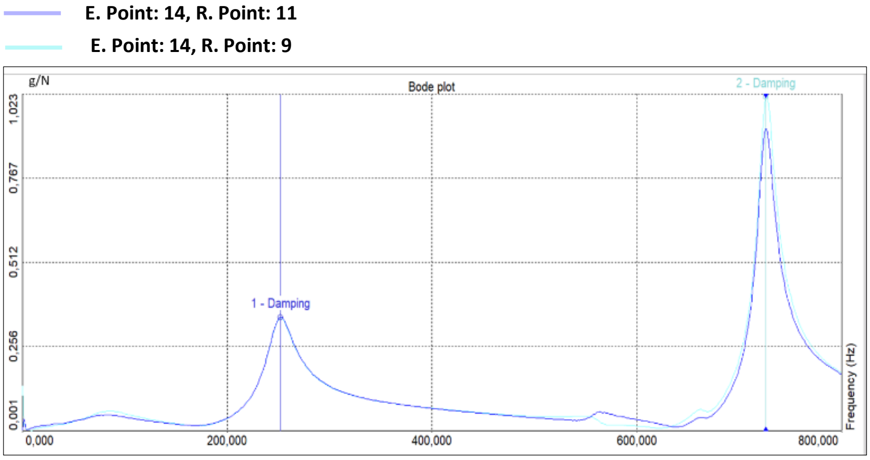


Figure 13:
FRF results of Al plate (roving impact hammer test with damping mat)

Additionally, Figure 14 illustrates the FRF results of modal shaker test on the aluminum plate. By also using this method, various approaches were applied to the plates to implement the available testing techniques and obtain accurate results. The FRF amplitudes may vary depending on the experimental method, likely due to factors specific to the chosen technique. For example, impact force can differ based on the hammer tip material, strike force, and the test operator. In a linear system, such as in modal analysis, these factors are expected to create proportional changes in both the response and excitation, meaning the FRF ratio should remain the same. Therefore, the differences in peak amplitudes observed in various testing configurations are likely due to the damping mat's effect. Excitation point (E.point) and response point (R.point) were given in Figure 14.

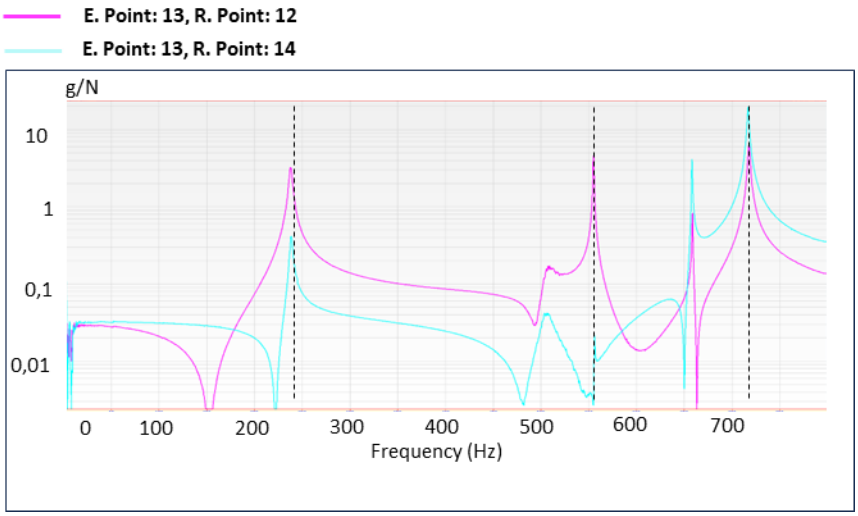


Figure 14:
FRF results of Al plate (modal shaker test)

The damping ratios were calculated by using half power method from the peaks in the Figures regarding Al plate and the values are depicted in Table 3. Since the damping ratios are close to each other, the overall damping values of 25 points calculated for mode 1 are taken as basis. Half power method is employed to identify the damping ratios and Q factors of the structures. This method employs the 3dB calculations which involve determining the difference between the upper frequency where the response is 3dB below from maximum and the lower frequency where the response is 3dB down. To estimate damping, the half power points, f_2 and f_1 are initially located on each side of the identified peak. The damping factor can then be estimated from the width of the resonance peak as:

$$\zeta = \frac{f_2 - f_1}{2f_0} \tag{7}$$

Figure 15 illustrates in terms of finding the half power points (FU et al., 2001)

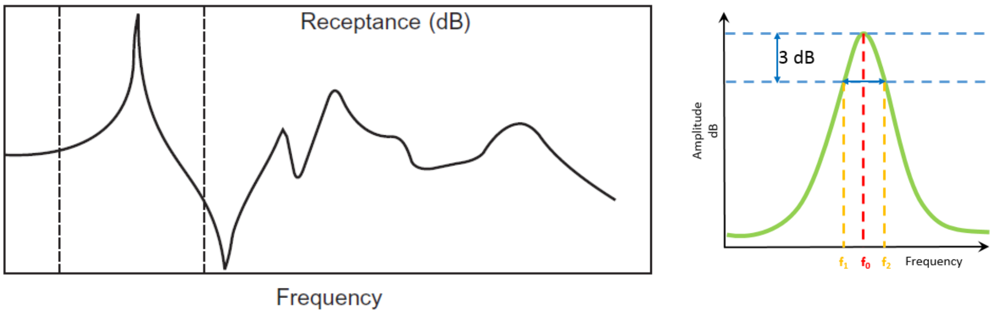


Figure 15:
Half power method (He et all., 2001)

Table 3. Comparison of Al plate frequencies of EMA and FEA results

	EMA Methods	Modes	EMA Frequencies of Al	FEA Frequencies of Al	Difference %	Damping
Aluminum Plate	Roving İmpact Hammer	1. Mode	250 Hz	258 Hz	3,1	0,053
		2. Mode	724 Hz	815 Hz	11,16	0,011
	Roving Accelerometer	1. Mode	248 Hz	258 Hz	3,8	0,052
		2. Mode	722 Hz	815 Hz	11,41	0,010
	Modal Shaker	1. Mode	237 Hz	258 Hz	8,8	0,050
		2. Mode	550 Hz	815 Hz	32,5	0,007

In the preceding sections, it was noted that the modal shaker test was exclusively conducted on the aluminum fixture plate. The acquired FRF curves exhibited imbalances, wherein certain resonance points did not precisely align or exhibited shifts when comparing different FRF curves. Upon thorough examination of this discrepancy, it was determined that several factors, namely the structural weight, modal shaker specifications, and the force transducer, significantly influenced the obtained results. Subsequently, it was deduced that conducting a modal shaker test on the FR4 plate would yield inaccurate FRFs due to these identified influencing variables. Drawing insights from the modal tests conducted on the aluminum plate, appropriate methodologies for plate modal testing were discerned. Consequently, to facilitate a comparative analysis between the two testing methodologies, modal tests employing a roving hammer and roving accelerometer were performed on the FR4 plate. Additionally, regarding tests on FR4 plate are performed by using elastic ropes as hung because during this test it was not encountered any leakage and low damping issue such as Al plate.

3.2. FRF Results of Experimental Modal Test (EMA) of FR4 Plate

In Figure 16, the frequency response function (FRF) outcomes from the roving impact hammer test conducted on the FR4 plate are depicted. Excitation point (E.point) and response point (R.point) were given in the Figure 16.

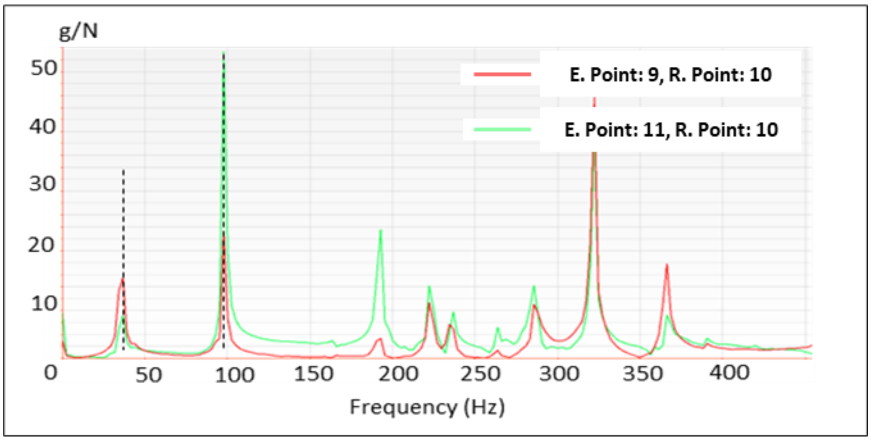
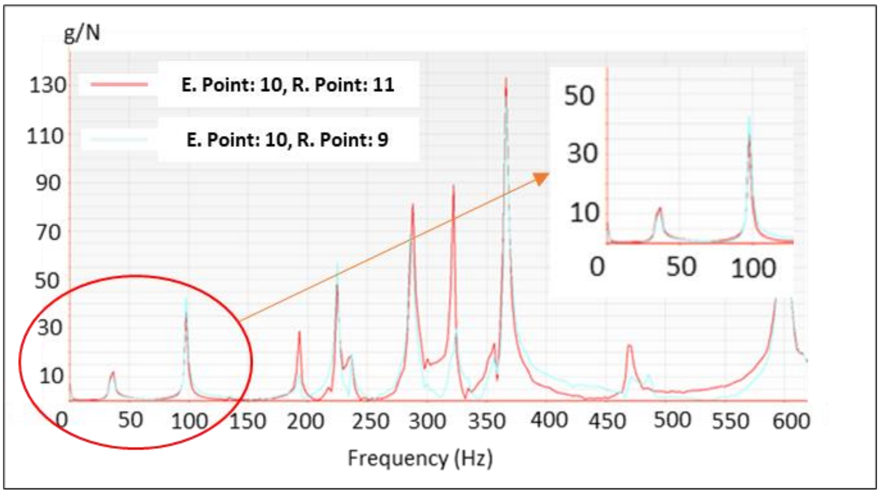
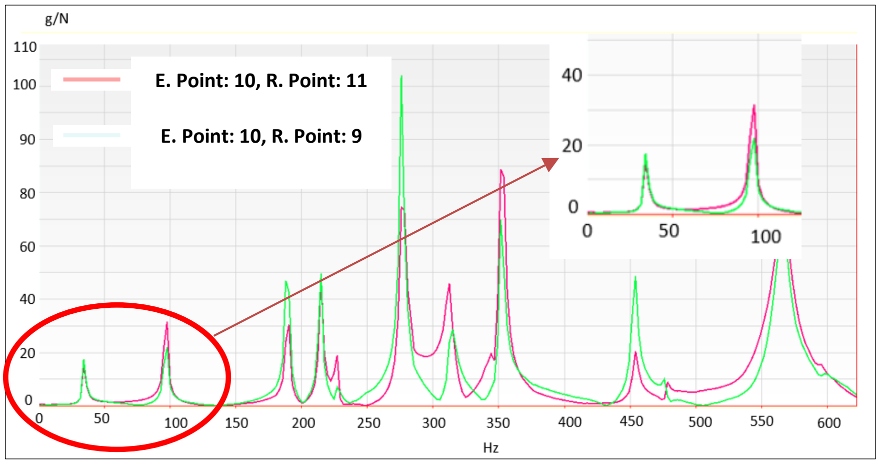


Figure 16:
FRF results of FR4 plate (roving impact hammer test)

Conversely, Figure 17 showcases the FRF results derived from the roving accelerometer test performed on the FR4 plate. After the roving impact hammer test was completed on 18 point and 55 point FR4 plates, both results were analyzed. According to the results, it has not been observed major differences in the effect of the number of points positioned on the plates. The results of the 18 point plates were analyzed to be very close to the results of the 55 point plates. Therefore, in order to save time, the roving accelerometer test was performed on the 18 point plates. Excitation point (E.point) and response point (R.point) were given in the Figure 17.



a



b

Figure 17:
FRF results of FR4 plate ***a.*** *Roving accelerometer test* ***b.*** *roving accelerometer test with rubbers positioned on each point*

The damping ratios were calculated by using half power method from the peaks in the Figures regarding FR4 plate and the values are depicted in Table 4.

Table 4. Comparison of FR4 plate frequencies of EMA and FEA results

	EMA Methods	Modes	EMA Frequencies of FR4	FEA Frequencies of FR4	Difference %	Damping
FR4 Plate	Roving İmpact Hammer	1. Mode	35,5	34,5	2,81	0,039
		2. Mode	98,4	96,4	2,03	0,018
	Roving Accelerometer	1. Mode	35,4	34,5	2,54	0,035
		2. Mode	97,6	96,4	1,22	0,025
	Roving Accelerometer with Rubbers	1. Mode	34,7	34,5	0,57	0,048
		2. Mode	96,1	96,4	0,31	0,020

3.3. MAC results

The MAC matrices of both Aluminum and FR4 plate for different test configurations are determined. These matrices give a visual representation of the degree of similarity between the experimental mode shapes obtained through testing and the numerical mode shapes acquired from FEA. The MAC values of the first 2 modes obtained by MEScope® software are given in Table 5.

Table 5. Similarity of EMA and FEA mode shapes

	EMA Methods	Modes	Similarity of Between Mode Shapes of EMA and FEA
Aluminum Plate	Roving İmpact Hammer	1. Mode	0,97
		2. Mode	0,97
	Roving Accelerometer	1. Mode	0,97
		2. Mode	0,97
	Modal Shaker	1. Mode	0,95
		2. Mode	0,61
FR4 Plate	Roving İmpact Hammer	1. Mode	0,99
		2. Mode	0,96
	Roving Accelerometer	1. Mode	0,87
		2. Mode	0,70
	Roving Accelerometer with Rubbers postinoed	1. Mode	0,96
		2. Mode	0,75

These similarity ratios of mode shapes given in Table 5 are represented as matrices in Figure 18 with more than 2 modes. It was observed that the similarity degree decreasesas as the frequency gets higher.

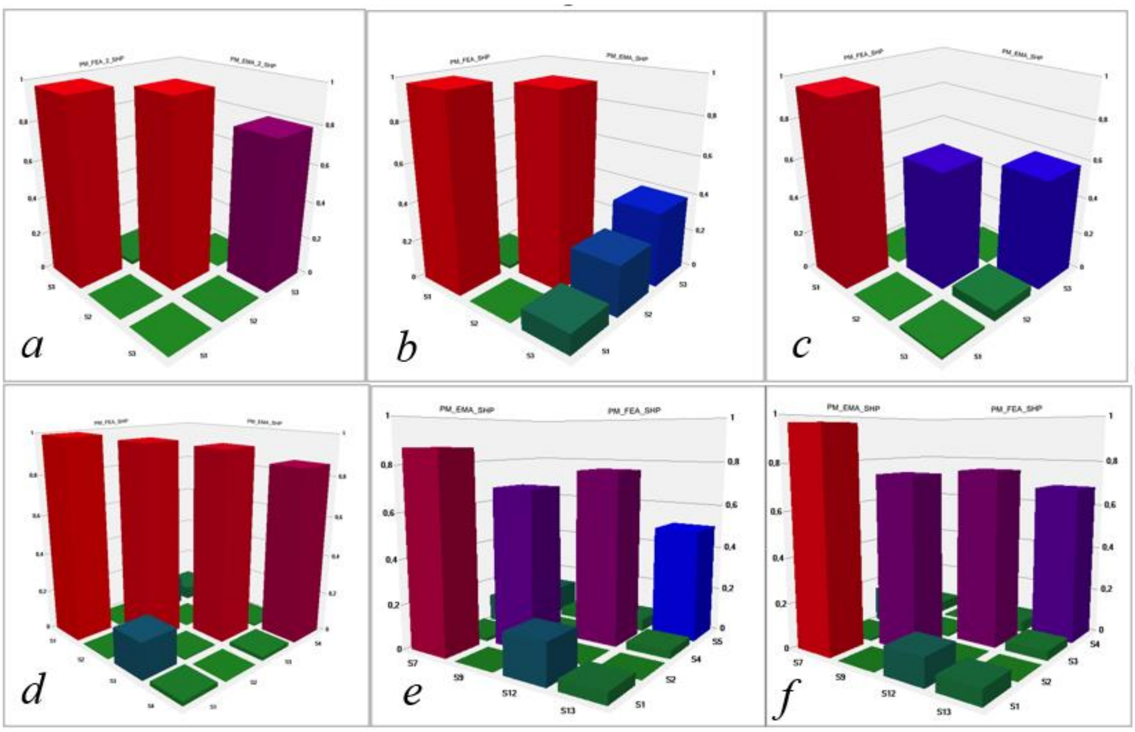


Figure 18:
MAC results in visual between EMA and FEA of plates
***a.** roving impact hammer test of Al **b.** roving accelerometer test of Al **c.** modal shaker test of Al **d.** roving impact hammer test of FR4 **e.** roving accelerometer test of FR4 **f.** Roving accelerometer test of FR4 with rubbers positioned*

As seen the figures, generally the most similarity results between EMA and FEA obtained on first modes and second modes. Once the frequency values of plates increased, ratio of similarities of EMA and FEA decreased so the most effective modes shapes investigated on the first two modes as correctly. These ratios can be taken into account for optimization parameters in order to prevent some structure issues. Also as evaluate the figures, the ratio of similarity can be influenced by several modal tests. The most similarity mode shape is detected with roving impact hammer method and the most similarity frequencies are detected with roving accelerometer, so the difference of EMA and FEA obtained %0,31 by roving accelerometer with rubbers. The less similarity is detected with modal shaker test method of Al and frequency of differences obtained %32,5.

The occurrence of a leakage problem associated with the Aluminum plate was addressed through the implementation of a sponge mat, resolving the issue. Additionally, several preventative measures have been identified to mitigate and prevent similar occurrences in the future. These measures encompass the following strategies:

• Utilizing accelerometers to measure the response and firmly affixing them to the structure for stability.

• Employing appropriate accelerometer mounting techniques such as bonding or magnetic attachment to ensure reliable measurements.

• Implementing damping methods to counteract leakage effects, including the application of damping materials to mitigate undesired vibrations and reflections at measurement points.

• Employing precise impact techniques with the impact hammer, ensuring perpendicular impact angles at defined points with controlled and consistent impact forces. Avoidance of excessive forces is crucial to prevent unintended vibrations and leakage.

• By adhering to these precautionary measures, the occurrence of leakage problems during modal tests can be circumvented, thus ensuring the accurate determination of modal parameters

Strategically selecting measurement points to minimize leakage effects. This involves avoiding points that may excite multiple modes simultaneously, leading to complex motion and increased leakage. Optimal points should emphasize the dominance of the mode under examination for clear and distinct analysis.

4. CONCLUSION

In this study, plates made of FR4 and Aluminum material were tested by different EMA methods and EMA results were compared with FEA results. Roving hammer, roving accelerometer and modal shaker tests were performed on the plates. During these tests, some requirements and issues were detected and solved accordingly. In conclusion, proper EMA test methods for different structures were determined.

- A difference of 8.8% was observed between the EMA resonance frequencies of the Al plate obtained via modal shaker and the FEA resonance frequencies for the first mode. The difference was 3.8% for the roving accelerometer test and 3.1% for the roving hammer test performed with a sponge mat.
- For the second mode frequencies, a 32.5% difference was found between the modal shaker EMA and FEA resonance frequencies of the Al plate. This difference decreased to 11.41% for the roving accelerometer test and 11.16% for the roving hammer test performed with a sponge mat.
- In the FR4 plate tests, the roving accelerometer EMA resonance frequencies differed by 2.54% from the FEA resonance frequencies. When using rubber supports at each test point, the difference between the roving accelerometer EMA and FEA frequencies was reduced to 0.57%, while the difference for the roving impact hammer test was 2.81% in the first mode.

- For the second mode frequencies of the FR4 plate, a 1.22% difference was observed between the roving accelerometer EMA and FEA resonance frequencies. This difference decreased to 0.31% with rubber supports at each point, and was 2.03% for the roving impact hammer test.

- Evaluating the FR4 plate’s mode shapes and results, the roving accelerometer test caused a mass imbalance in the structure, especially in light and thin configurations, affecting the center of gravity and reducing the accuracy of higher-frequency mode shapes. To address this, rubber supports were added at each point to balance the structure, and the test was repeated. Comparing results with and without rubber supports showed a clear improvement in balance. Replacing rubber supports with equivalent weight cables at each point could further enhance accuracy.

- Based on these tests, it is essential to evaluate the geometry and dimensions of parts before conducting modal shaker tests, select an appropriate force transducer, and ensure the structure is suitably prepared for the modal shaker.

- It was also observed that the assembly conditions on the test bench, such as using elastic ropes and sponge mats, can influence test results depending on the plate properties.

In conclusion, the roving impact hammer test is a practical option for many plates to obtain accurate mode shapes quickly. The roving accelerometer test, however, provides detailed results that closely match the EMA frequencies. When applying a modal shaker, careful consideration of the structure's characteristics is critical for reliable results.

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CONFLICT OF INTEREST STATEMENT

The authors declare that there is no conflict of interest in the study.

AUTHORS CONTRIBUTION

Enes Çal: Conceptualization, Investigation, Methodology, Software, Validation, Writing – original draft, Barış Ediz: Conceptualization, Investigation, Methodology, Writing-review & editing, Software, Fahri Berk Bilbay: Investigation, Software, Writing–review & editing Betül Gülçimen Çakan: Supervision, Writing – review & editing.

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