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FINITE ELEMENT MODELING, CORRELATION AND MODEL UPDATING OF STIFFENED PLATE

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ABSTRACT

This paper looks into modal parameter extraction of aluminium plate using finite element (FE) analysis and the correlation with experimental modal analysis. The effects of localize stiffening with ribs on modal parameters were considered. Finite element models of an aluminium plate with and without rib were developed and analyzed. Correlation and model updating of the stiffening plate were considered. Results suggest that ribs increase the local stiffness of the plate and they can be added onto a structure during modification.

Keywords: *Finite element, modal analysis, correlation and model updating*

1.0 INTRODUCTION

Finite element (FE) analysis has been widely used in noise, vibration and harshness (NVH) simulation especially in the fast growing automotive industries. Engineers were given the power and ability to predict and assess structure-borne noise in engine, thanks to the current advancement in computer software and hardware. Development of a low noise engine would be an easier task if this knowledge could be harnessed.

Validation of the FE model itself has become automated and more reliable. The FE models are often correlated with experimental modal analysis (EMA) results in order to achieve a high degree of confidence in the FE analysis. The EMA is a process where modal parameters such as natural frequency, mode shapes and damping ratio were extracted from the structures, experimentally. Between late 80's and 90's, the correlation and model updating were done based on the said modal parameters [1]. However, from late 90's onwards there is a shift of trend towards Frequency Response Functions (FRF) based correlation and modal updating [2].

One of the important stages in engine development is the modification of the engine structure, where the dynamic behavior of the structure is modified to enhance the structures fatigue characteristics by raising the frequencies of the critical modes [3]. Usually, the major parameters of the structure such as bore size, stroke length, etc. are not considered. The usual modification is the addition of stiffening ribs, rubber damper, increasing or decreasing the thickness and other acceptable modifications. All these modifications should lead to a low weight but high stiffness structure. A stiffening rib however is the most common method used in structural modification. Other than being simple, it is effective, cheap and does not interfere much with the original design. In engineering terms, a rib is an added structure that increases the localize stiffness of a surface or structure [3]. Figure 1 shows ribs on a crankcase cover of a two-stroke engine.

Before embarking on a complex structure such as the crankcase cover, a basic study is carried out on a simple flat plate with and without stiffener. This paper focuses on the modal analysis of a plate using experimental and FE analysis and the effect of ribs on modal parameters of a plate. Correlation and model updating works are also included.

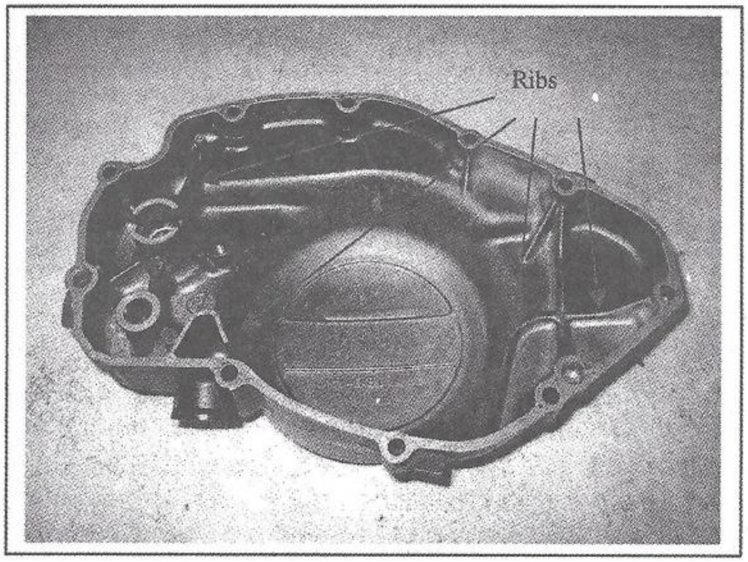

Figure 1 Crankcase cover stiffened with ribs

2.0 FINITE ELEMENT MODELING

Figures 2 and 3 show the dimensions of the flat plate with and without a rib. The rib was located at the centre of the plate. The plate is made of aluminium with the following mechanical properties:

Young's Modulus, E	:	70 GPa
Poison Ratio, ν	:	0.33
Mass Density, ρ	:	2769 kg/ m^3

As for NVH, the use of shell elements has yield the best result compared to solid elements since the later do not have rotational degrees of freedom (d.o.f) [4]. The absence of rotational d.o.f. will give values of natural frequencies much higher than the actual values since solid elements are stiffer than shell element and have bad bending behavior [5]. Also, shell elements allow the thickness to be used as a design variable during the model updating process, because it is defined numerically while the thickness of solid elements is defined geometrically. However, in this study both shell and solid elements have been used.

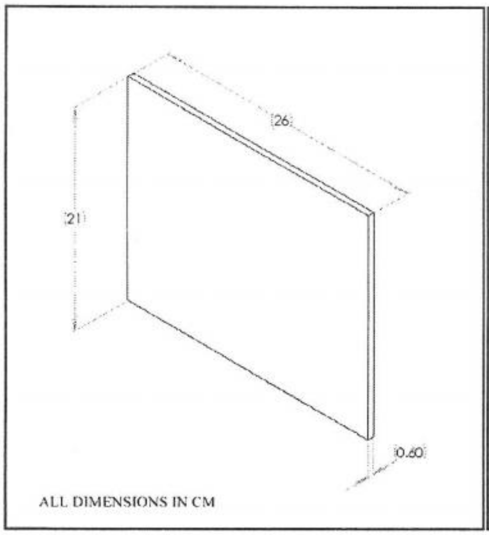

Figure 2 Aluminium Plate

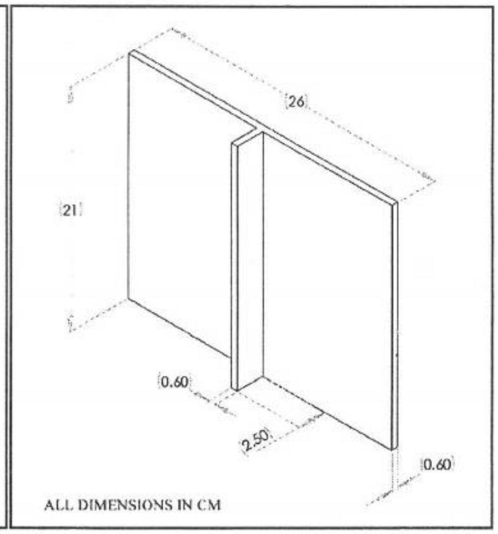

Figure 3 Aluminium Plate with Rib

The accuracy of an FE analysis is dependent on the finite element representation of the actual geometry of the structure. In this case, four FE models were constructed for both unstiffened and stiffened plates with different element topology. The first was Quad-4 shell model developed to simulate modal properties of the unstiffened plate with 256 shell elements, 289 nodes and 1445 d.o.fs. The thickness of the elements was defined as 0.6 cm. No constraints were assigned in an attempt to simulate free-free boundary condition. Thus the frequency range of interest was set between 1 to 2000 Hz. The starting frequency was set at 1 Hz to avoid the solver from calculating 6 rigid body motions which have the frequency of 0 Hz. Table 1 shows the complete models constructed for both unstiffened and stiffened plates.

Table 1 Finite element models: Number of elements, nodes and d.o.f.

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Unstiffened				Stiffened				
Quad-4 Shell	Hex-8 Solid	Tet-10 Solid	Tet-4 Solid	Quad-4 Shell	Hex-8 Solid	Tet-10 Solid	Tet-4 Solid	Properties
256	256	3308	3308	288	304	3806	3806	Elements
289	578	6889	6889	340	680	7927	1375	Nodes
1445	1734	20667	3588	1615	2040	23781	4125	d.o.f

<u>**Indicator**</u>

Quad-4 : Quadrilateral Element – 4 nodes
Hex-8 : Hexahedral Element – 8 nodes
Tet-10 : Tetrahedral Element – 10 nodes
Tet-4 : Tetrahedral Element – 4 nodes

The natural frequencies and the corresponding mode shapes of the models were calculated by using the finite element solver and the results are tabulated in Tables 2 to 5. The first mode is torsion and the remaining two are bending modes. It was observed that Tet-4 model gave higher values of natural frequencies for both plate conditions. This may be due to the high rigidity of the model. The mode shapes were found to be independent of the type of model.

Table 2 Natural Frequencies of Unstiffened Plate

Element Type		EMA	Quad-4	Hex-8	Tet-10	Tet-4
Natural Frequencies (Hz)	Mode 1	340.9	348.19	349.17	351.20	806.23
	Mode 2	456.3	447.99	447.31	454.29	966.57
	Mode 3	753.6	734.16	732.48	742.99	1464.10

Table 3 Natural Frequencies of Stiffened Plate

Element Type		EMA	Quad-4	Hex-8	Tet-10	Tet-4
Natural Frequencies (Hz)	Mode 1	354.0	363.42	372.63	372.05	877.28
	Mode 2	450.7	441.89	451.88	454.79	921.14
	Mode 3	781.4	780.98	792.17	799.05	1796.60

Table 4 Mode Shapes of UnStiffened Plate

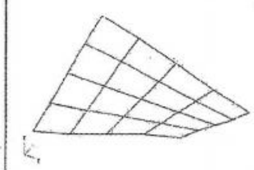
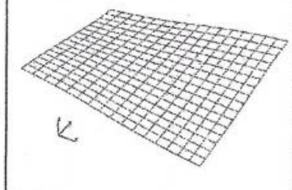
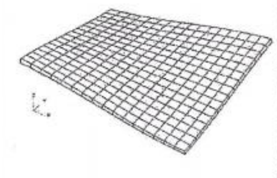
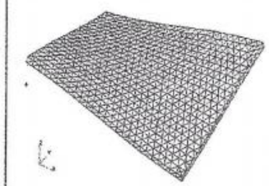
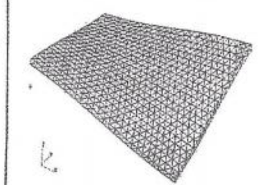
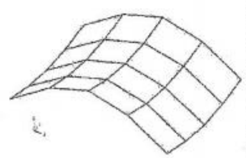
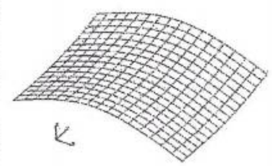
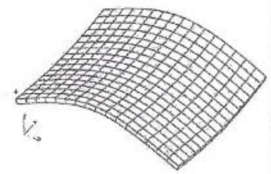
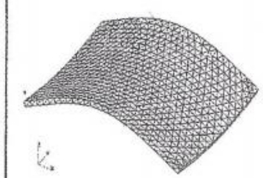
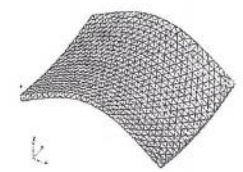
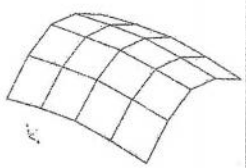
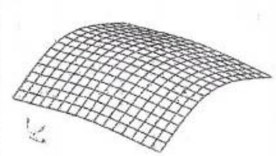
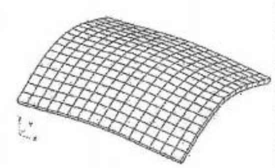
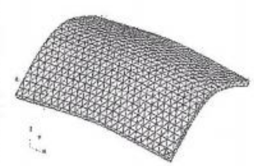
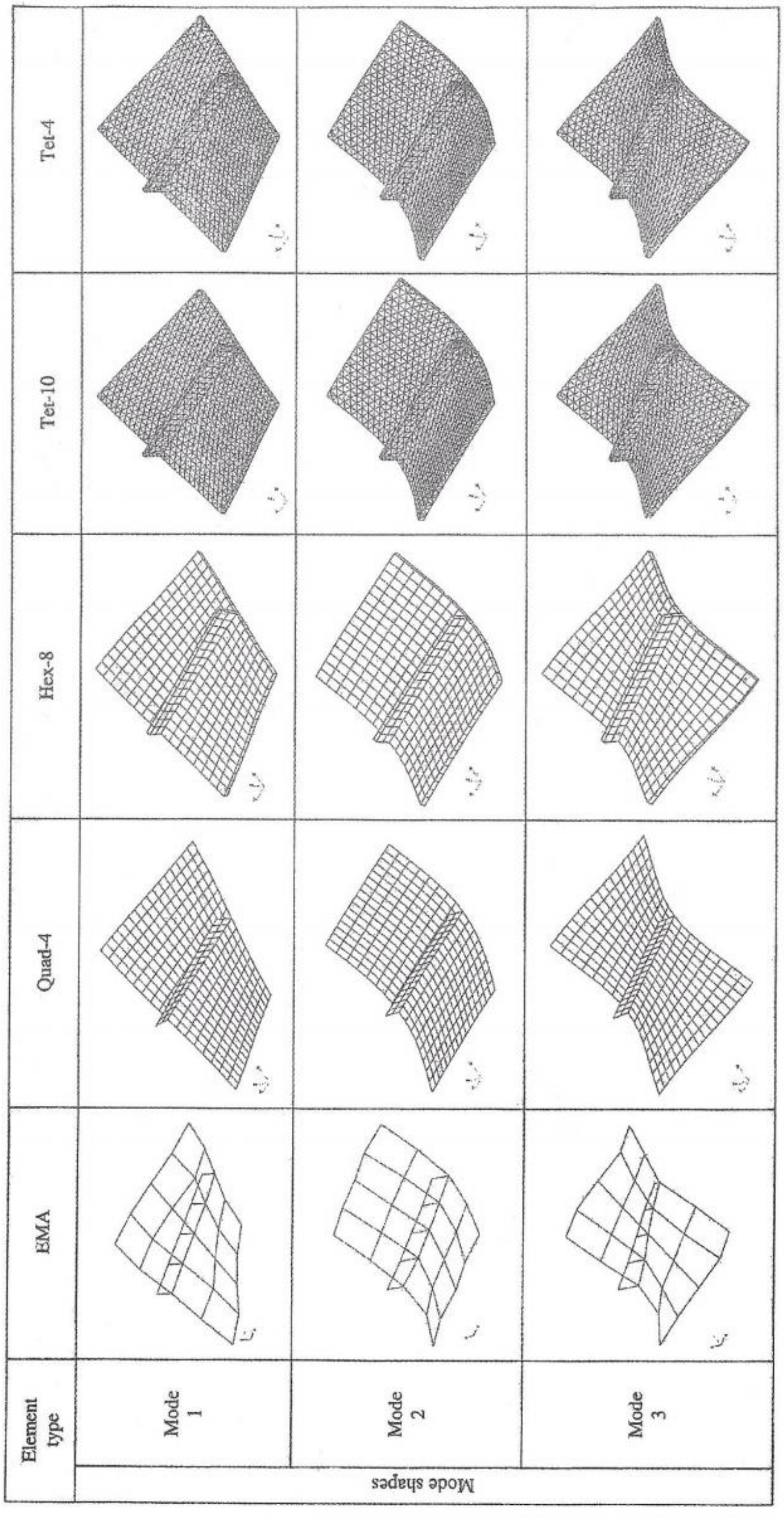
	Element type	EMA	Quad-4	Hex-8	Tet-10	Tet-4
Mode shapes	Mode 1					
	Mode 2					
	Mode 3					

Table 5 Mode Shapes of Stiffened Plate



3.0 EXPERIMENTAL MODAL ANALYSIS (EMA)

An experimental modal analysis was carried out to verify the analytical models and its results. The unstiffened plate was divided into 25 grid points where at these points Frequency Response Functions (FRF) were measured. A Roving Impact Hammer Method was applied on the plate whereby an accelerometer (Kistler Type 8630C50) was fixed at 1 point and the impact force was applied at all 25 points by the impact hammer (Bruel & Kjaer Force Transducer Type 8200).

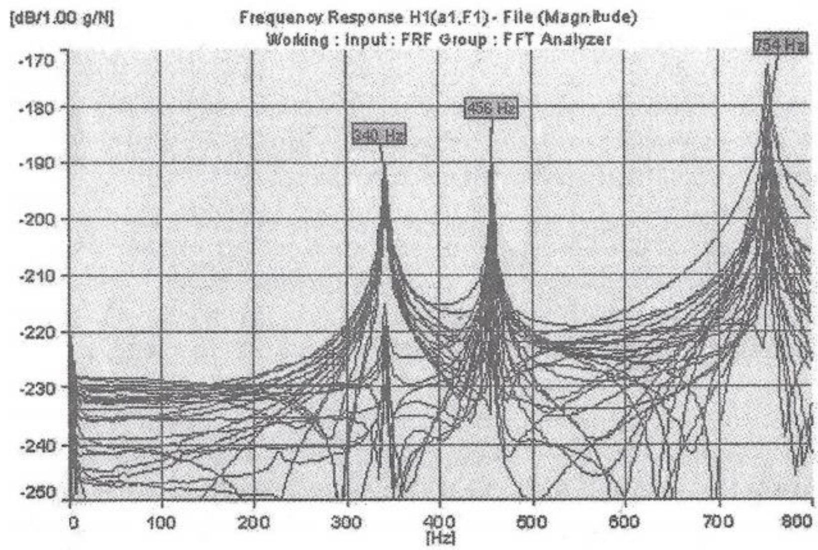


Figure 4 Superimposed FRF for unstiffened plate

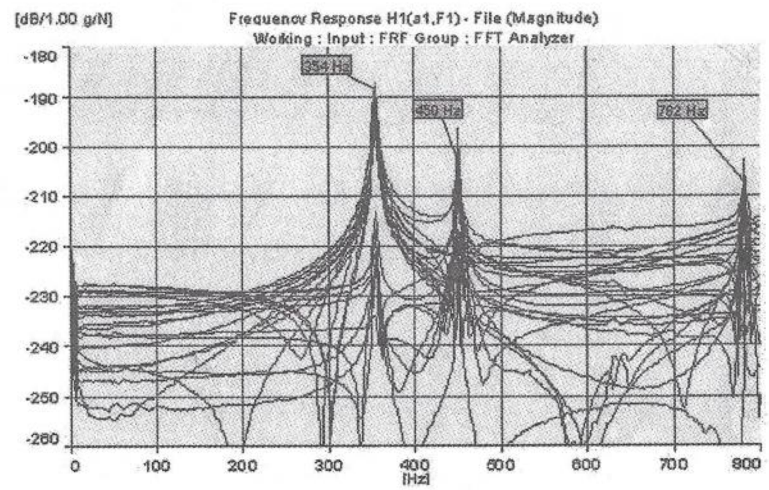


Figure 5 Superimposed FRF for stiffened plate

Figures 4 and 5 show the superimposed FRF at all points for both unstiffened and stiffened plates. These FRF were calculated by Bruel & Kjaer Pulse Analyzer Type 3560C and the frequency range of interest was set to 800 Hz. The experimental mode shapes of the plate were processed by MEScope Ves Modal Analysis software [6]. Curve fitting was done to all measured FRFs which was imported in Universal File Format by the Pulse to the MEScope software. The modal parameters were calculated using the Multi-Degree of Freedom Global Polynomial Modal Identification Method. These parameters are tabulated as shown in Tables 2 to 5.

4.0 TEST / ANALYSIS CORRELATION

Correlation is a process to determine how far the FE analysis results agree with the EMA. Discrepancies will always exist between the FE and the EMA model and there are at least three sources of discrepancies, [1]:

(1) Errors in experimental data – noise exists in the experimental data, the measurements are carried out at an imperfect set-up, and the original experimental data (FRF) are processed approximately to obtain the modal data (natural frequencies and mode shapes) that will be used in the updating process.

(2) Model parameter errors – some parameters in the FE model have values specified that are different from the actual structure such as thickness, material properties and damping.

(3) Model structure errors – some features that are important to the dynamic properties of the structure in the specified frequency range are replaced by different features in the FE model such as joints, etc.

Figure 6 shows the flowchart of the research methodology used in this study to produce a verified FE model. In this paper, only Quad-4 shell model and the results for stiffened plate are presented.

There are two methods of correlation and model updating that can be used. They can be categorized as either modal based or response based [7]. In this study, both methods were adopted and the calculations were carried out using FEMtools software [8].

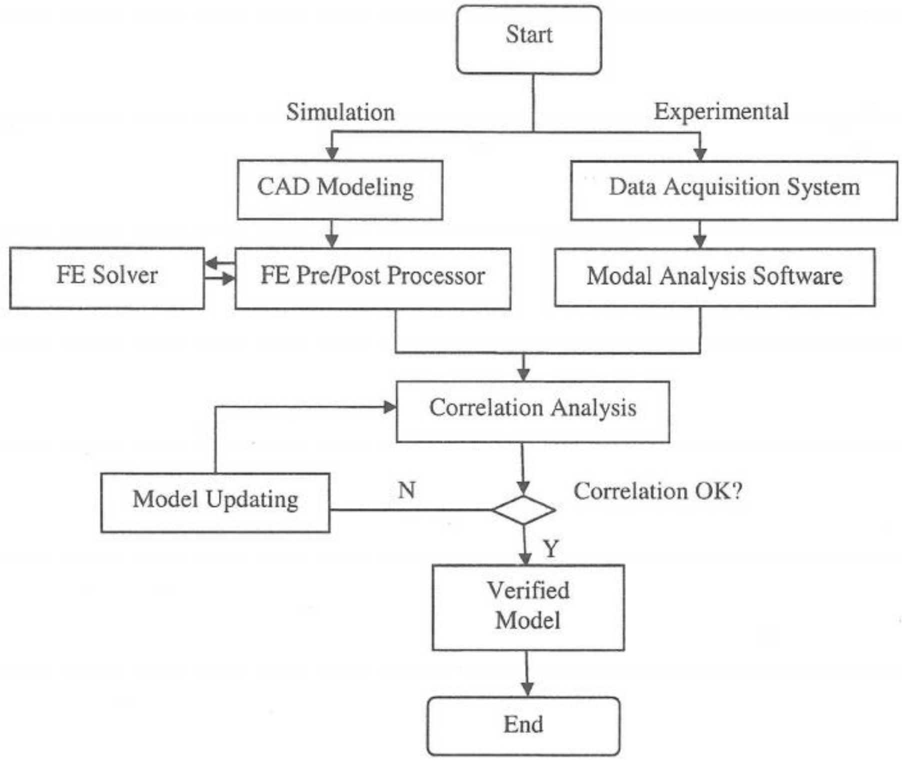


Figure 6 Correlation and Updating Flowchart

4.1 Correlation and model updating of Stiffened Plate

a) Modal Based Correlation

Modal based methods use the test modal parameters as targets in the correlation process. Correlation of the FEA and EMA mode shapes was quantified based on Modal Assurance Criterion (MAC). The MACvalue may range from 0 to 1. MACvalue close to 1 along the diagonal of the matrix indicates that the two sets of mode shapes are nearly identical. Table 6 shows the calculated MACvalue matrices of unstiffened plate. A high correlation of eigenmodes was achieved for the first two modes, with mode 3 showing a bad correlation. The correlation can be further improved by carrying out a model updating process.

Table 6 MAC matrix of stiffened plate before model updating

EMA FEA	1	2	3
1	93.8	0.1	0.1
2	0.1	92.8	0.6
3	0.5	0.2	73.8

Model updating is a step in model validation process that modifies the values of parameters in an FE model in order to bring the FE model prediction into a better agreement with the experimental data [9,10]. The test data were used as the target, and the FE parameters were updated. Parameters that can be 'updated' for the plate model are:

i. Plate thickness, h (global variable)
ii. Young's modulus, E (local variable)
iii. Mass density, ρ (local variable)
iv. Poisson ratio, ν (local variable)

Parameters such as E, ρ and ν were selected as local updating variables. This is justified by the type of modeling errors which have both local stiffness and mass effect [11]. The plate thickness, h, was selected as a global variable to represent the overall thickness of the plate. Table 7 shows the mode frequencies before and after model updating (M.U.). A tremendous improvement in the FE mode frequencies was observed. As for the MAC value, Table 8 shows the MAC matrix before and after model updating. There was a slight increase for mode 3 but a decrease for mode 2, with no changes occurring for mode 1.

Table 7 Mode frequencies of stiffened plate before and after model updating

Mode	EMA	Freq. (Before M.U.)	Freq. (After M.U.)	Percentage Error (Before M.U.)	Percentage Error (After M.U.)
1	354.0	363.42	355.06	2.67	0.31
2	450.7	441.89	449.3	-1.95	-0.30
3	781.4	780.98	781.33	-0.05	-0.01

Table 8 MAC diagonal matrix of stiffened plate before and after model updating

Mode	MAC Before M.U.	MAC After M.U.	Percentage Increase/ Decrease
1	93.8	93.8	0.0
2	92.8	92.7	-0.21
3	73.8	74.2	0.43

All four parameters (h, E, ρ and ν) were updated as illustrated in Figures 7 to 10. Thickness, h was reduced as much as 1.92 % (5.88 mm) from its original thickness of 6mm. The values of E, ρ and ν were reduced and increased locally. The FE Quad-4 shell model of the stiffened plate is thus updated and verified.

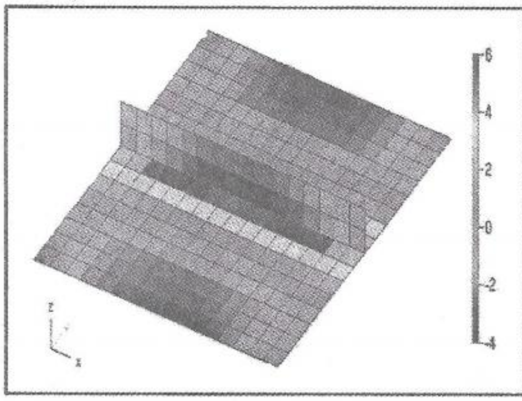


Figure 7 E localize modification percentage (%)

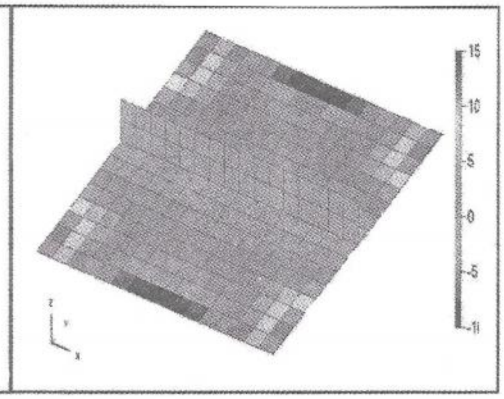


Figure 8 ρ localize modification percentage (%)

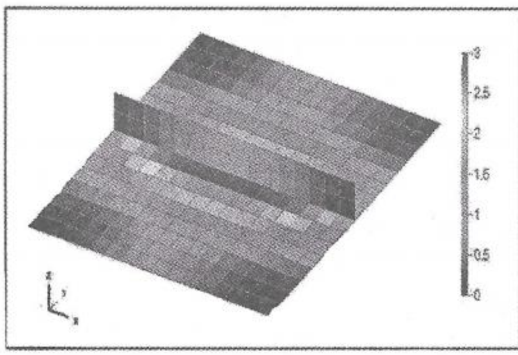


Figure 9 υ localize modification percentage (%)

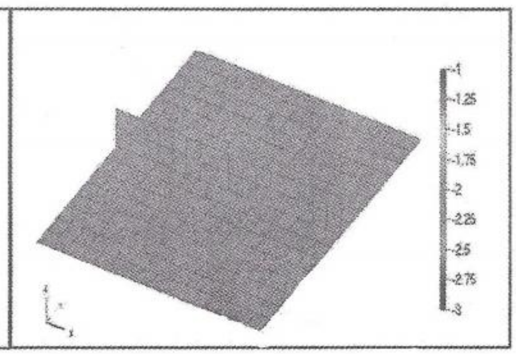


Figure 10 h global modification percentage (%)

b) Response Based Correlation

A response based model updating modifies the values of parameters in an FE model in order to bring the FE model prediction into a better agreement with the experimental data. It is based on establishing an agreement between the predicted and the measured FRFs [12]. One important advantage of this method is that the intermediate step of performing a modal extraction is unnecessary. The FRF based modal updating was carried out using the FEMtools software. Figures 11 and 12 show the FRF pair of the stiffened plate before and after modal updating. Only one node is presented here for model updating Selecting more nodes is possible but this will require more calculation time. The selected parameters for updating are similar to those of the modal based updating, i.e. h, E, ρ and ν. It was observed that, after updating, the first peak approach inline with the EMA peak but this move results in the second and third peaks shifting away from the EMA peaks. This is due to the FE spectrum shifting as a rigid curve when there are changes in their properties. Table 9 shows the mode frequencies correlation and percentage errors before and after updating. It can be seen that an improvement in one mode frequency may result in an error increase in the other mode frequency. Figures 13 to 16 show changes in the pattern for all four parameters (h, E, ρ and ν) after updating. The thickness, h was reduced by as much as 5.68 % (5.66 mm) from its original thickness of 6 mm. Like modal based results, the parameters for E, ρ and ν were reduced and increased locally.

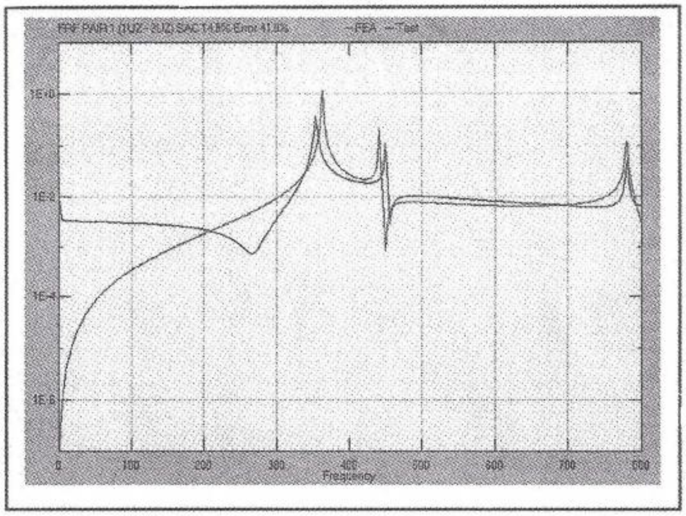


Figure 11 FRF pair (stiffened) before Model Updating

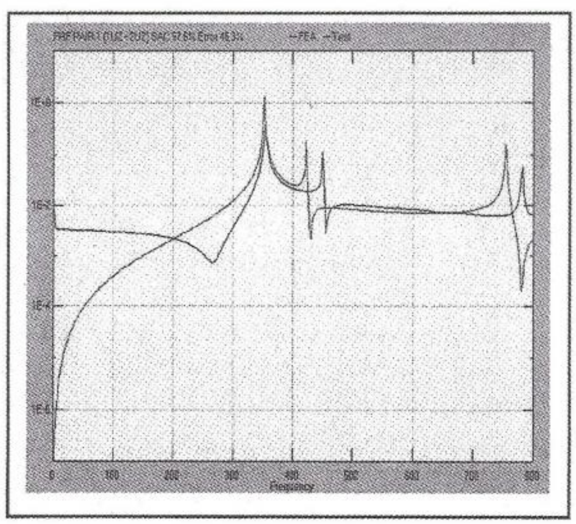


Figure 12 FRF pair (stiffened) after Model Updating

Table 9 Mode frequencies of stiffened plate before and after model updating

	EMA Natural Freq. (Hz)	FEA Natural Freq. Before Model Updating (Hz)	FEA Natural Freq. After Model Updating (Hz)	Error % Before Model Updating	Error % After Model Updating
Mode 1	354.0	363.42	352.5	2.67	-0.42
Mode 2	450.7	441.89	422.3	-1.95	-6.30
Mode 3	781.4	780.98	754.4	-0.05	-3.46

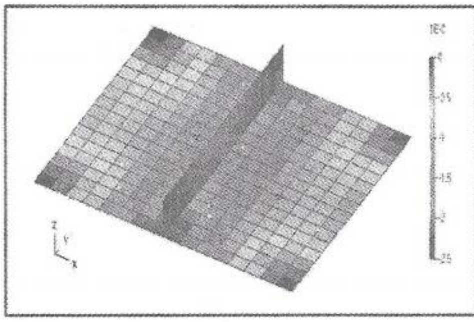


Figure 13 E localize modification percentage (%)

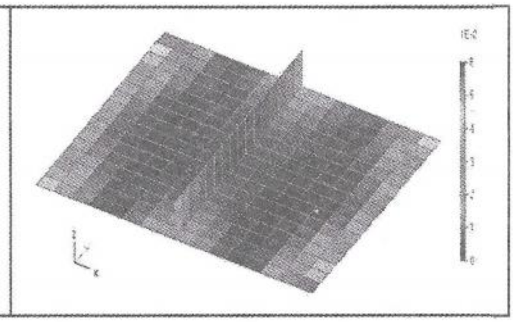


Figure 14 ρ global modification percentage (%)

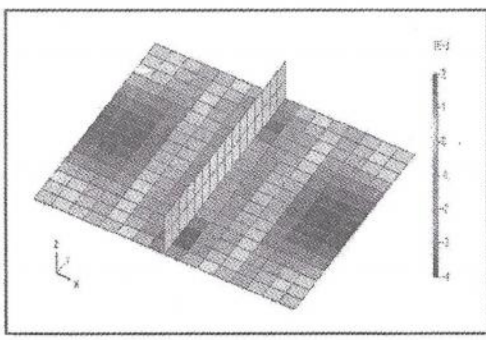


Figure 15 υ localize modification percentage (%)

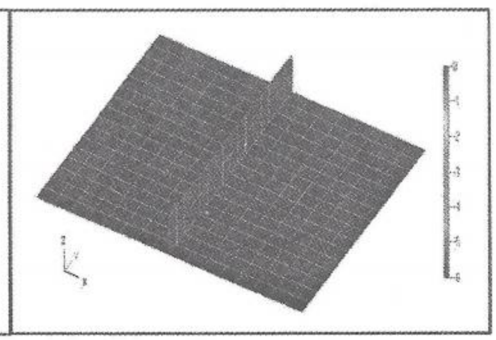


Figure 16 h global modification percentage (%)

5.0 DISCUSSION

The first part of the study showed that surface shell and solid elements can give a reasonably accurate simulation of dynamic properties of the structures being investigated. However, the use of Tet-4 elements should be avoided since it is too stiff and thus gives higher values of natural frequencies. The mode shapes are however acceptable. Solid elements however do not have rotational d.o.f., but the combination of surface and solid elements can be used to get an acceptable result. Surface elements are usually used for shell like parts while solid elements are used for parts like bracket, flange, bolt bosses, etc [4].

Adding a rib to the plate increases its natural frequencies. This may be due to the stiffening of the plate which dominates its vibration characteristics. In contrast, the second mode frequency decreases. As observed in Tables 4 and 5, the first and third modes were affected by the location of the stiffener. The deformations associated with these modes are prevented by the stiffener. For the second mode, adding the rib did not stiffened the deformation but only increases the total mass of the plate.This resulted in the decrease in the natural frequency. Thus from this stud, it can be concluded that to reduce the vibration and noise from a vibrating structure the dominant natural frequency and mode shape have to be identified for effective stiffening.

It can be seen that small discrepancies in the dynamic characteristics exist between FE and EMA analysis. This problem was rectified by executing the correlation and model updating process. A perfect correlation is achieved when the FE natural frequencies match the experimental natural frequencies within 2-10% and matched in mode shapes with MAC values above 90° [7]. Two methods of model updating were adopted the modal based and the response based methods. During the model updating process, the test data were used as the target, and the FE model parameters were updated to give better agreement between the model and the test. The FE model parameters being updated should represent parts of the model where there is less confidence of the actual value such as thickness, Young's modulus, density, etc. [4]

As a result of the modal based updating, the natural frequencies match was within 0.5% of the test data. The MAC values however did not improve much, with highest at 0.5 % increased in MAC. This may be due to several factors. The experimental mode shape was only in one d.o.f. since the accelerometer used was a single axial instead of a tri-axial type. Therefore, the displacement of the measurement point was measured only in one direction, thus giving an imperfect mode shape. The mode shapes of the FE model were calculated in 3 degrees of freedom and the correlation failed to get all MAC values above 90% for all three modes. The MAC values are even more unsatisfactory if the correlation was allowed up to ten modes since higher modes have complex mode shapes. Hence, it is recommended that a tri-axial accelerometer be used in EMA for complex structures.

The results of response based updating however are not as good as that of the modal based. The updated stiffened plate improved only by about 0.4% in the first mode but decreased up to 6.3 % for the next two modes. Thus only the first peak has shifted to a new improved value. The second and third peaks shifted away and worsen the correlation. Manual changes on several parameters indicates that the spectrum curve shifts as a rigid curve. Thus, it is quite impossible to have a good correlation at every mode frequency.

The pattern for the parameter changes for both modal based and response based updating differ significantly in terms of percentage modification. The response based updating modification percentages were small (<0.02%) compared to modal based updating (<20%) for localized parameters (E, ρ and ν). As for global parameter (h), the percentage modification was almost the same.

Hence, it can be said that both modal based and response based updating gave almost the same results and both methods are acceptable. But in terms of speed and time, response based modal updating is better. Modal based updating however does promise better results and high level of confidence.

The ribs do modify the dynamic properties of structures. They can be adjusted to suit the requirement to change certain modes of vibration. It can also be used to alter local or global modes. Generally, it will prevent bending from occurring in directions perpendicular to it. As for the plate, ribs can be added and overall thickness reduced. This will decrease the use and cost of material and thus the weight of the structure but increases the stiffness.

6.0 CONCLUSION

In conclusion, FE analysis is a reliable method for predicting the modal properties of a structure provided that the right element and method are used. In order to get a verified model, EMA needs to be done and EMA-FEA correlation should achieve a MAC level of at least 90%. In case of low level correlation, the model updating can be performed using either the modal base method or the response based method. Both methods have their own advantages. Structural modification by using ribs can add local or global stiffness to the structure and it is important to position the ribs correctly for effective stiffening.

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