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Finite Element Modelling and Updating of a Thin Plate Structure using Normal Mode Analysis

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Abstract. Structural design and analysis require accurate and precise finite element model in order to have a structure with high safety conditions such as aerospace, offshore and civil structures. However, the predicted results of the finite element model are often considerably different from the experimental results due to the modelling assumptions and simplifications on the properties and parameters of the actual structure. This paper presents the finite element modelling and updating methods on the thin plate structure. The modal parameters obtained from the plate are measured by experimental modal analysis by using an impact hammer and roving accelerometers method under free-free boundary conditions. The initial modal parameters are predicted by numerical analysis using normal mode analysis via NASTRAN SOL 103. The accuracy of the natural frequencies and the mode shapes obtained from the initial finite element model are compared with the measured counterparts. The modal based updating methods using NASTRAN SOL 200 is applied to update the properties of the plate. In conclusion, updating the material properties of the plate is successfully achieved. The total error of the initial finite element model of the plate is reduce from 45.12 percent to 6.28 percent respectively.

Keywords: structural design, model updating, experimental modal analysis, normal mode, natural frequency

1. Introduction

In engineering field practices, structural design and analysis is an essential task that deals with the use of structures to support and resist loads during the operations. From the large-scale engineering structures such as bridges, platforms and aircrafts to the medium scale engineering structures such as vehicles and industrial equipment, the design and analysis processes require accurate and effective finite element (FE) model [1]. Structural analysis encounter modelling and load analysing which are depending on the quality of the model. In addition, an implementation of the advance structural design and analysis of the components and structures are crucial in order to verify the functional and operations with the structural safety condition and serviceability.

Finite element method is a tool for the numerical prediction using computational processes on the discrete model to obtain the objective solutions. This finite element model is normally utilizing uncertain system parameters such as geometry, material properties, boundary conditions and kinematic interactions [2,3]. Several types of sources which contribute to the inaccuracy of the model are model

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structure errors, model parameters errors and model order errors [4]. There is a considerable discrepancy between the numerical predictions and the corresponding test results [5]. Modifications on the modelling assumptions and parameters are crucial until the correlation satisfy the requirements [6,7]. A process of adjusting and refining certain parameters and incorrect assumptions of the finite element model based on the experimental results is known as finite element model updating [8]. This reliable method is widely used to solve and minimize numerous sources of modelling error which can be divided into three categories: idealization errors, discretization errors and erroneous assumptions for the modal parameters [9].

In this paper, the modal parameters (natural frequencies and mode shapes) of the thin bending plate structure are measured using the experimental modal analysis (EMA) by impact testing. Meanwhile the finite element method by normal mode analysis is used to predict the dynamic behaviour of the initial model. The objective function and the sensitivity analysis are discussed and the finite element model updating are performed to produce accurate updated finite element model.

2. Experimental Set Up and Modal Testing

Modal testing was conducted to determine the modal parameters of the thin plate that is fabricated from a cold rolled mild steel. The modal parameters involve natural frequencies, mode shapes and damping ratios [10,11]. The EMA set up is shown in Figure 1. The plate was suspended on the test rig by using soft bands to simulate free-free boundary conditions. Figure 2 shows the equipment used for the modal testing. An impulse excitation force was applied to the plate by using an impact hammer for the broadband vibration excitation. The applied force was measured using the force transducer installed at the impact hammer. The vibration response of the plate was measured using three accelerometers. The method of roving accelerometers was used to avoid mass loading issue during the experimental modal analysis [12,13]. Two accelerometers were used for roving and one accelerometer was used as a reference at the driving point to measure the acceleration response.

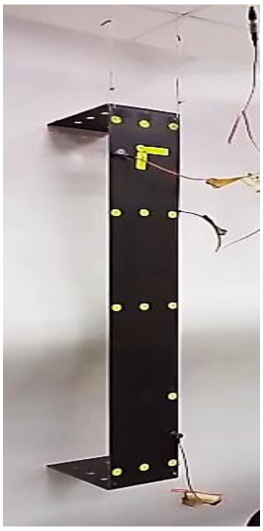


Figure 1. Modal testing setup for the plate

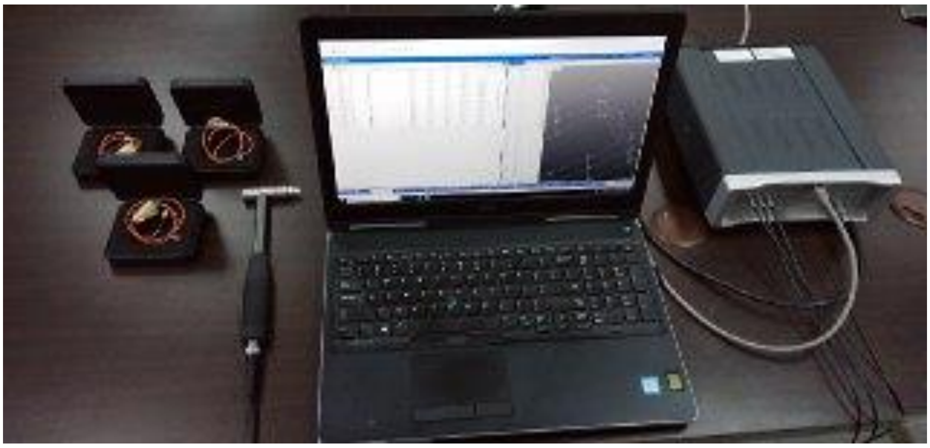


Figure 2. Testing equipment for experimental modal analysis

The schematic diagram of the modal testing is shown in Figure 3. The data acquisition equipment used to calculate the modal parameters is LMS SCADAS data acquisition system. The measured force and acceleration responses were processed using LMS Test.Lab analyser. The bandwidth for the frequency of interest is below 200 Hz.

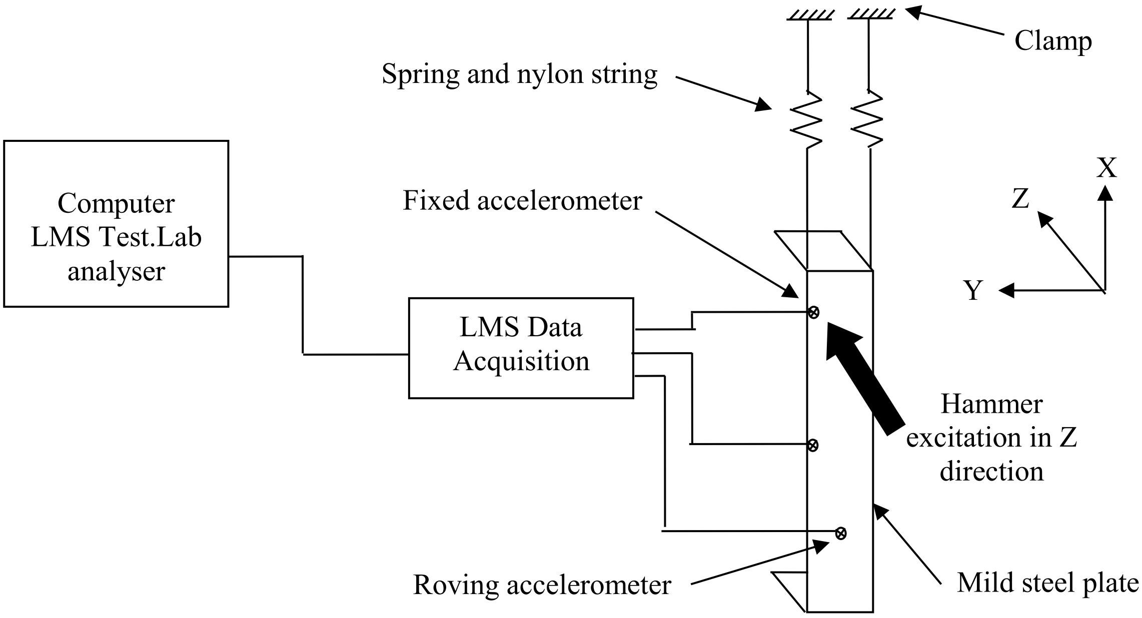
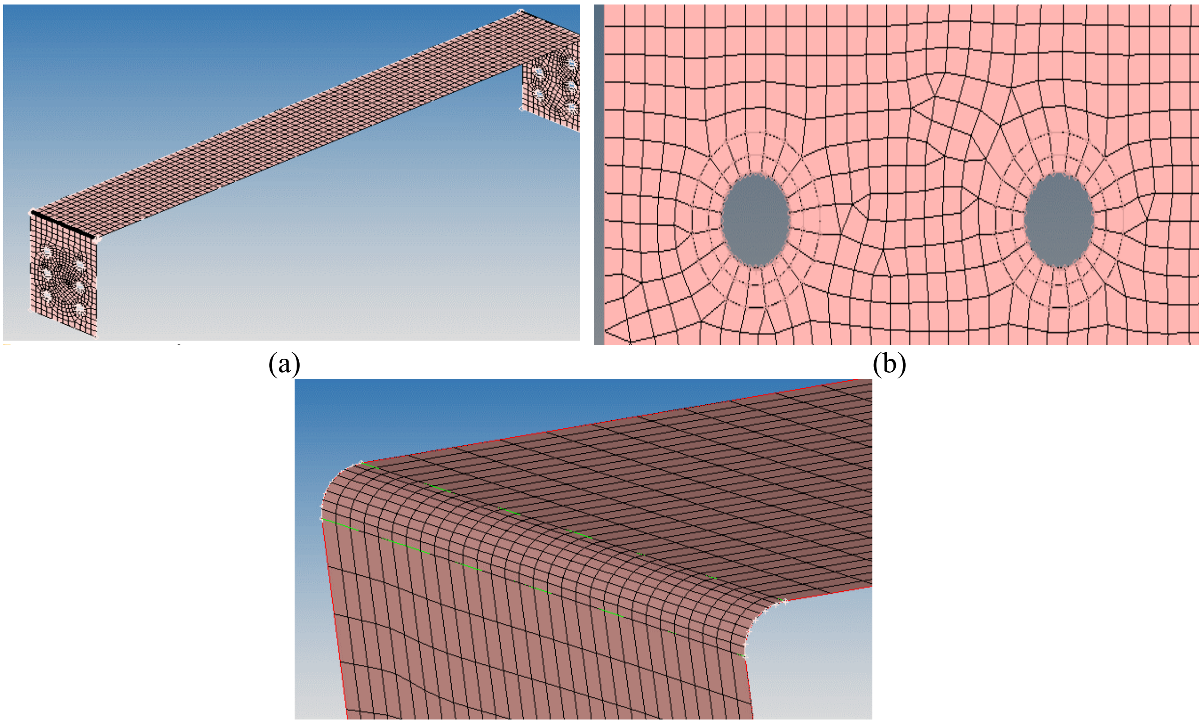


Figure 3. Schematic diagram of the modal testing of the plate

3. Finite Element Analysis

The three main steps in the numerical simulation works which are pre-processing, processing and post-processing were performed using OptiStruct from HyperWorks in order to obtain natural frequencies and mode shapes. The FE model of the plate is shown in Figure 4. The model was developed which consists of 10,172 elements of 2D mixed mesh (combination of triangle and quadrilateral) and 10,555 nodes. The property of the mesh is thin shell elements of size 5 mm. The nominal value of material properties [14] that was assigned to the FE model are tabulated in Table 1. Normal modes analysis with free–free boundary conditions was utilized to determine the modal parameters which are natural frequencies and mode shapes. The bandwidth for the frequency of interest is from 1 Hz to 200 Hz.



(a) (b) (c)

Figure 4. Finite element model (a) plate model (b) meshing of the bolt hole (c) meshing at the bend location

Table 1. Material properties of the plate for the FE normal mode analysis

Properties	Values
Young's Modulus, E (MPa)	200,000
Poisson's Ratio, υ	0.3
Density, ρ (kg/m^3)	7800
Thickness, t (mm)	1.5

The equation of motion for the free vibration of an undamped finite element model [15] can be calculated using Eq. (1)

$$M\ddot{x}(t) + Kx(t) = 0 \quad (1)$$

where K and M are the stiffness and mass matrices. Meanwhile, $\ddot{x}$ and x are the vector of acceleration and displacement respectively. These system matrices are calculated based on the geometry and properties of the FE model. The natural frequencies and mode shapes of the structure are obtained by solving Eq. (2) where ω is a natural frequency and ϕ is a mode shape [16,17].

$$(K - \omega^2 M)\varphi = 0 \quad (2)$$

In this study, Lanczos method was used for algorithm computing free vibration modes. The free vibration characteristics depend on the magnitude and distribution of the structural stiffness and mass [18,19]. All structures display several different mode shapes at different frequencies, both at a local and a global level.

4. Finite Element Model Updating

Normal mode analysis using Lanczos method was used to determine the modal parameters. The natural frequencies and mode shapes obtained from the FE model are compared with the measured counterparts in order to validate the accuracy of the initial FE model. Table 2 shows the comparison of natural frequencies between EMA and initial FE analysis. It can be seen that the predicted natural frequencies from the FE analysis are higher compare to the measured natural frequencies from the EMA. The range value of the error is between 6.0 per cent and 10.0 per cent which contributing to the high total error of 45.12 per cent.

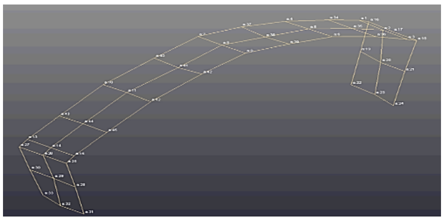
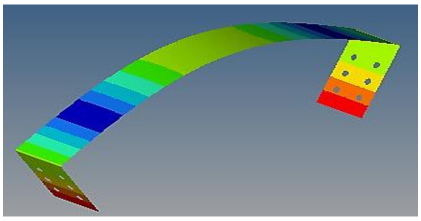
Table 2. Comparison of EMA and initial FE results for the plate

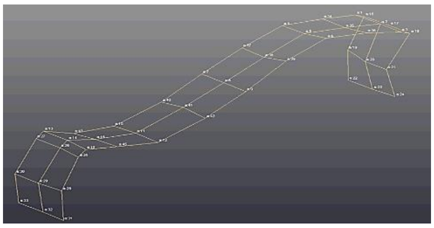
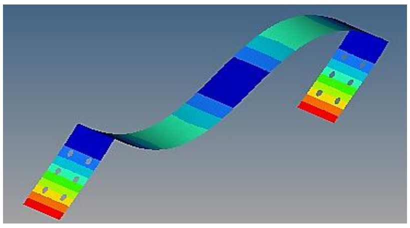
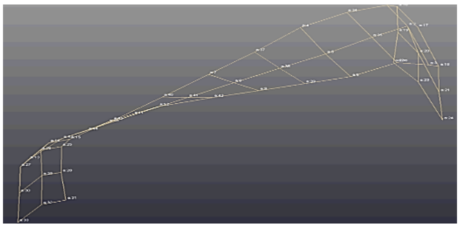
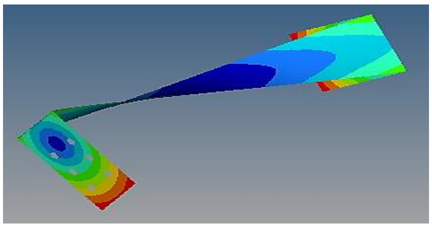
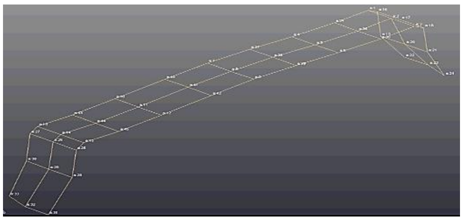
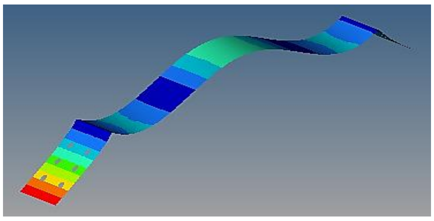
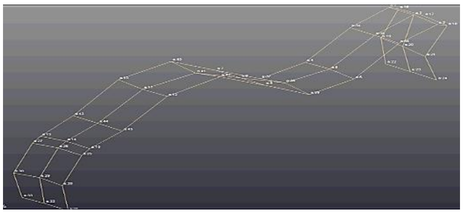
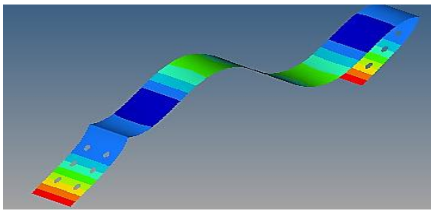
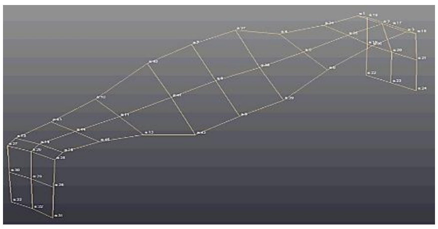
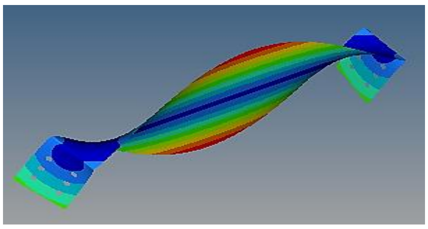
Mode	EMA (Hz)	Initial FE Analysis (Hz)	Error (%)
1	22.49	24.17	7.42
2	57.56	62.09	7.87
3	64.02	68.13	6.42
4	85.34	92.55	6.80
5	136.18	145.62	6.92
6	185.89	203.79	9.69
	Total Error		45.12

Table 3 shows the mode shapes from EMA and FEA corresponding to each natural frequency. It is observed that four modes which are mode 1, mode 2, mode 4 and mode 5 are elastic bending modes in x and z direction. Two modes are the elastic torsional modes which are mode 3 and mode 6 that is in y and z direction. The accuracy of the mode shapes between the experimental and finite element model has been evaluated by pairing each mode correctly. The modal assurance criterion (MAC) can be calculated using Eq. (3) where experimental (ϕ_m) and finite element (ϕ_a) mode shapes in matrix form. MAC is used to quantify level of correlation of mode shapes between the FE models and the experimental models. Table 3 also shows the MAC value for each mode which has a good correlation above 0.8.

$$MAC = (\varphi_m \varphi_a) = \frac{|{\varphi_m}^T \varphi_a|^2}{({\varphi_a}^T \varphi_a)({\varphi_m}^T \varphi_m)} \quad (3)$$

Table 3. Comparison of the mode shapes of the plate between the measured and predicted natural frequencies

Measured EMA Mode Shape	Predicted FEA Mode Shape	MAC
 Mode 1	 Mode 1	0.98

 Mode 2	 Mode 2	0.93
 Mode 3	 Mode 3	0.95
 Mode 4	 Mode 4	0.91
 Mode 5	 Mode 5	0.92
 Mode 6	 Mode 6	0.87

From the high value of total error shown in Table 2, the model updating processes has been performed on the initial FE model to minimise the error. This is important for the finite element model to be as close enough to the test structure prior to be used for further structural engineering analysis. Model updating is widely used to improve the accuracy of finite element models by correcting the invalid assumptions about the finite element model. This is carried out by changing some uncertain finite modelling parameters, which have the potential to influence modal properties and subsequently to improve the accuracy of the models. Finite element model updating is performed using NASTRAN optimisation code (SOL 200). The optimisation algorithm, which is a sensitivity-based iterative procedure, allows the objective function (J) to be minimised by adjusting the eigenvalues of the initial finite element model until the objective function is converged. The objective function based on eigenvalues is calculated using Eq. (4) where ${\lambda_i}^{exp}$ is the ith experimental eigenvalue and ${\lambda_i}^{fe}$ is the ith predicted eigenvalue from the finite element model and n is the number of eigenvalues involved in the updating procedure [20].

$$J = \sum_{i=1}^{n\Sigma} W_i \left(\frac{{\lambda_i}^{fe}}{{\lambda_i}^{exp}} ()^2 \right) \tag{4}$$

Computation of sensitivity analysis (SOL 200) is performed to identify the sensitive parameters of the initial FE model. Four potential parameters are listed in this sensitivity analysis which are Young's modulus, Poisson's ratio, density and thickness of the plate. Comparison of experimental and updated FE analysis was performed to study the accuracy of the finite element model updating. Table 4 shows the discrepancies between the updated FE analysis and the measured natural frequencies of the plate. The high value of natural frequencies from the initial FE analysis were decrease to the values which close to the measured value from EMA. For the total error of the six modes, the large reduction in the discrepancies is seen from 45.12 per cent to 6.28 per cent. The updated values of the updating sensitive parameters are shown in Table 5. An increment on the updated values have been obtained on Young's modulus (from 200,000 MPa to 203,800 MPa) and density (from 7800 kg/m^3 to 7998 kg/m^3). Meanwhile, the initial thickness of the plate has been reduced from 1.5 mm to 1.43 mm, and also the reduction of the Poisson's ratio value from 0.30 to 0.29.

Table 4. Comparison of experiment and updated FE analysis of the plate

Mode	EMA (Hz)	Initial FE Analysis (Hz)	Error (%)	Updated FE Analysis (Hz)	Error (%)
1	22.49	24.17	7.42	22.43	0.31
2	57.56	62.09	7.87	57.75	0.33
3	64.02	68.13	6.42	63.27	1.17
4	85.34	92.55	6.80	86.71	0.06
5	136.18	145.62	6.92	137.06	0.64
6	185.89	203.79	9.69	192.79	3.77
		Total Error	45.12	Total Error	6.28

Table 5. Updated values of the parameters of the plate

Parameters	Initial Value	Updated Value	Unit
Young's modulus	200,000	203,800	MPa
Poisson's ratio	0.30	0.29	
Density	7800	7998	kg/m^3
Thickness	1.5	1.43	mm

5. Conclusions

The finite element model updating of a thin bend plate has been performed successfully in this study. A 2D shell element FE model has been constructed using HyperMesh OptiStruct software. A numerical simulation work via NASTRAN SOL 103 normal mode analysis has been carried out on the model which was assigned with nominal values of material properties. The measured modal parameters (natural frequencies and mode shapes) have been obtained from the conducted experimental modal analysis on the real plate under free-free boundary conditions with lightly damped conditions. The six natural frequencies from the FE analysis showed large discrepancies in comparison with the measured results showing 45.12 per cent relative error. Sensitivity analysis via NASTRAN SOL 200 has been utilised and listed the sensitive parameters. The discrepancies have been reduced by updating the value of the material properties of the initial FE model of the plate which are Young's modulus, Poisson's ratio, thickness and density. The relative error has been reduced to 6.28 per cent, thus produce better validated FE model for further engineering analysis.

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