IDENTIFICATION OF MATERIAL PROPERTIES OF ORTHOTROPIC COMPOSITE PLATE USING HYBRID NON-DESTRUCTIVE EVALUATION APPROACH

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Identification of material properties is one of the key issues in composite materials research. This information is the most important database for an accurate Computer-Aided Engineering (CAE) simulation and various design enhancement purposes. For example, it can be used to estimate deflections and stress state of composite structure under static or dynamic load. The mechanical properties of composite materials depend on diverse factors such as configuration of the laminates, constituent materials used, production method adopted, etc. Hence, it is generally impossible to find these properties in standard tables. Conventional testing approach tends to be time-consuming, expensive and destructive. Moreover for properties such as shear modulus, these tests often yield poor results. As an alternative, a hybrid approach which utilises experimental and numerical techniques is proposed. This approach is a rapid, inexpensive and non-destructive evaluation of the mechanical properties of composite materials which involves both Experimental Modal Analysis (EMA) and Finite Element Analysis (FEA). Experimental modal data which consists of natural frequencies and mode shapes of an orthotropic composite plate are utilised for correlation purpose with its Finite Element (FE) model in CAE environment. This finite element model of the composite plate is continuously updated and achieves less than 5% in difference of natural frequencies and over 70% in Modal Assurance Criterion (MAC). Material properties such as Young's moduli, in-plane shear modulus and Poisson ratio of the composite plate are then successfully determined using the well-correlated FE model.

Keywords: orthotropic composite plate, non-destructive evaluation, experimental modal data, Experimental Modal Analysis, Finite Element Analysis, model updating

Introduction

Composite materials are composed of two or more different materials at macro-scale with each of the materials will contribute to the final properties. When the materials are combined, composites are generally more superior as compared to the individual components. Composite structures are usually constructed in multiple laminates where each layer is oriented specifically to achieve optimal strength and stiffness performance. Composites are preferred in applications due to their high strength-to-weight ratio, high-stiffness-to-weight ratio, high wear resistance, high corrosion resistance.

The determination of mechanical properties is one of the important parts in composite materials research. Production method, materials used, and laminates configuration of the composites are among the contributing factors in changing the mechanical properties of composite materials. These properties are could not be found in any standard databases and tables. Conventional test procedures based on static loading are destructive, time and cost consuming. In addition, these tests often produce poor results especially in determining properties such as shear modulus. This has encouraged the development of specific methods to improve the accuracy in the identification method.[1]

Alternatively, mixed of numerical and experimental techniques is gaining popularity among researchers to be applied in their research. Modal analysis is a rapid, cheap and non-destructive evaluation approach to determine the mechanical properties of composite materials. The elastic constant that represents the local and global properties of a structure panel can be determined in a single test.

Researchers like Fayyadh et al.[2], Ismail[3], Ismail et al.[4, 5], Ismail and Ong[6], Ong et al.[7] and Rahman et al.[8] have demonstrated the application of modal parameters from modal analysis using vibration theory in fault identification of various structures. Vibration theory can also be used to develop a non-destructive test to determine the dynamic elastic constant namely longitudinal Young's modulus, transverse Young's modulus, in-plane shear modulus and Poisson's ratio of a composite plate. The analytical approaches based on the Rayleigh–Ritz[9-11] or Rayleigh[12, 13] methods and the numerical approaches based on the finite element method[14-17] have been applied to determine the elastic constants of materials. The calculated analytical and numerical natural frequencies are compared with the measured natural frequencies to find the corresponding elastic constants. Deobald and Gibson[18] analysed a thin orthotropic composite plate with different boundary conditions. Rayleigh-Ritz technique was used to model the vibration of rectangular orthotropic plates. Natural frequencies from the analytical model were verified by Experimental Modal Analysis (EMA) and Finite Element Analysis (FEA). The plate vibration model was then used to determine the elastic constants. Zubaidah et al.[19] used an inverse method based on Fourier series to determine the material properties of orthotropic plates. An example was presented to demonstrate the accuracy and convergence of the solutions by the combined use of analytical model and FEA modal data. Hwang et al.[20] and Maletta and Pagnotta[21] proposed a non-destructive numerical method based on a hybrid genetic algorithm.

Vibration testing combined with this numerical method is a potential alternative approach for determining elastic constants of materials. Wolf and Carne[22], Yu et al.[23], DeWilde et al.[24, 25], Khov et al.[26] are other researchers who involve in determining the elastic constants of composite materials using vibration technique. Other techniques such as lamb and ultrasonic waves are other possible approach used in determination of elastic constants of composites[27] . Current researches only focused and utilised one modal parameter i.e. natural frequency in determining the material properties of composite plates. Modal parameter such as mode shape or the combination of both natural frequencies and mode shapes are rarely used in elastic constants determination.

In this paper, a non-destructive evaluation of the mechanical properties of composite materials which involves EMA and Finite Element (FE) model updating method is proposed. This approach utilises both measured natural frequencies and mode shapes and compares with the corresponding numerical data. The FE model of the composite plate is continuously updated in Computer-Aided Engineering (CAE) environment based on the experimental modal data i.e. natural frequencies as well as mode shapes. Model updating process will stop once the FE model show close correlation with the experimental modal data in terms of percentage difference in natural frequencies and mode shape correlation in Modal Assurance Criterion (MAC). Material properties such as Young's moduli, in-plane shear modulus and Poisson ratio of the composite plate could be determined based on the updated FE model.

Background theories

Experimental Modal Analysis

Experimental Modal Analysis (EMA) is an investigation on vibration characteristics of elastic structures. It involves experimental methods in investigating the oscillation behaviour of component structures by describing a system with its modal parameters; its natural frequencies, natural damping and mode shapes.

Carrying out measurement using Fast Fourier Transformation (FFT) analyser on a continuous system, the Frequency Response Function (FRF) being an estimated quantity is obtained over a number of averages. In general, the measured FRF matrix $[\mathbf{Z}(\omega)]$, relates the displacement response vector $\{\boldsymbol{X}(\omega)\}$ of a linear and time-invariant structure due to excitation force vector $\{\boldsymbol{Q}(\omega)\}$ as,

$$\{\boldsymbol{X}(\omega)\}=[\mathbf{Z}(\omega)]\{\boldsymbol{Q}(\omega)\}, \quad (1)$$

$$Z_{ij}(\omega)=\sum_{r=1}^{n} \frac{\varphi_{ir}\varphi_{jr}}{-\omega^2+2\sigma_r\omega \mathrm{i}+\omega_{or}^2} \quad . \quad (2)$$

Equation (2) describes the relationship between the element of FRF matrix $Z_{ij}(\omega)$ and the modal parameters, namely the r th mode of natural frequency ω_{or}, mode shape φ_r and damping σ_r. Number of mode r = 1, ..., n. ω is angular frequency. i is imaginary part. Subscripts i and j are response Degree of Freedom (DOF) and force DOF respectively.

Accelerometers are commonly used for EMA. By using differentiation in frequency domain[28], displacement response vector can be converted to acceleration response vector as follows: $\{\ddot{\boldsymbol{X}}(\omega)\}=-\omega^2\{\boldsymbol{X}(\omega)\}$. Hence, the Accelerance FRF matrix can be related to Admittance FRF matrix as follows: $[\mathbf{A}(\omega)]$ = -$\omega^2[\mathbf{Z}(\omega)]$. The measured Accelerance FRF matrix $[\mathbf{A}(\omega)]$ relates the acceleration response matrix and force matrix as follows: $\{\ddot{\boldsymbol{X}}(\omega)\}=[\mathbf{A}(\omega)]\{\boldsymbol{Q}(\omega)\}$. Element of Accelerance FRF matrix is shown as follows:

$$A_{ij}(\omega)=\sum_{r=1}^{n} \frac{-\omega^2\varphi_{ir}\varphi_{jr}}{-\omega^2+2\sigma_r\omega \mathrm{i}+\omega_{or}^2} \quad . \quad (3)$$

Finite Element Modal Analysis

For a system with n degrees of freedom, the action equation for undamped free vibrations take the general form

$$[\mathbf{M}]\{\ddot{\boldsymbol{X}}\}+[\mathbf{K}]\{\boldsymbol{X}\}=0. \tag{4}$$

Assume that in natural vibration all masses follow the harmonic function to produce a set of algebraic equations, equation (4) can be stated as

$$(K_{ij} - \omega_{or}^2 M_{ij})\varphi_r = 0, \tag{5}$$

where $H_r = K_{ij} - \omega_{or}^2 M_{ij}$ is the element of characteristic matrix. For non-trivial solutions of equation (5), the determinant of the characteristic matrix is set equal to zero giving

$$\begin{vmatrix} K_{11} - \omega_{or}^2 M_{11} & K_{12} - \omega_{or}^2 M_{12} & \cdots & K_{1n} - \omega_{or}^2 M_{1n} \\ K_{21} - \omega_{or}^2 M_{21} & K_{22} - \omega_{or}^2 M_{22} & \cdots & K_{2n} - \omega_{or}^2 M_{2n} \\ \cdots & \cdots & \cdots & \cdots \\ K_{n1} - \omega_{or}^2 M_{n1} & K_{n2} - \omega_{or}^2 M_{n2} & \cdots & K_{nn} - \omega_{or}^2 M_{nn} \end{vmatrix} = 0. \tag{6}$$

If the polynomial cannot be factored, its n roots ω_{o1}^2, ω_{o2}^2, ..., ω_{or}^2, ..., ω_{on}^2 could be determined by numerical procedure. Such roots, which were referred to previously as characteristic values, are also called eigenvalues. Vectors of modal amplitudes, any one of which is represented by φ_r are called characteristic vectors or eigenvectors.

It is important to remark that the stiffness depends on the linear elastic parameters that are to be identified

$$K = K(E_x, E_y, G_{xy}, \upsilon_{xy}), \tag{7}$$

where E_x is longitudinal Young's modulus. E_y is transverse Young's modulus. G_{xy} is in-plane shear modulus. υ_{xy} is Poisson's ratio.

Sensitivity Analysis

When the relation between a non-proportional parameter and the structural matrices is too complicated to obtain an explicit formula, a finite difference approach can be used. This is for example the case for parameters like orthotropic material constants. The sensitivity matrix $[\mathbf{S}]$ is usually a rectangular matrix and is obtained as:

$$[\mathbf{S}] = \left[\frac{\Delta R_\alpha}{\Delta P_\beta}\right] = \left[\frac{R_\alpha(P_\beta + \Delta P_\beta) - R_\alpha(P_\beta)}{\Delta P_\beta}\right], \qquad (8)$$

where R_α is the responses or resonance frequency and P_β is parameter consist of elastic constants. α = 1, …, total number of responses and β = 1, …, total number of parameters. Δ is finite difference.

Using finite differences to derive element structural matrices is a much faster approach than computing finite differences of the response values. For example, when the derivatives of the mass and stiffness matrices in equation (8) are approximated using finite differences, the following form of the equation is obtained:

$$[\mathbf{S}] = \left[\frac{\Delta f_\alpha}{\Delta P_\beta}\right] = \left[\frac{\{\boldsymbol{\varphi}_\alpha\}^{\mathrm{T}}\left(\dfrac{\Delta[\mathbf{K}]}{\Delta P_\beta} - 4\pi^2 f_\alpha{}^2 \dfrac{\Delta[\mathbf{M}]}{\Delta P_\beta}\right)\{\boldsymbol{\varphi}_\alpha\}}{8\pi^2 f_\alpha\left(\{\boldsymbol{\varphi}_\alpha\}^{\mathrm{T}}[\mathbf{M}]\{\boldsymbol{\varphi}_\alpha\}\right)}\right], \qquad (9)$$

where f_α is the resonance frequency in Hz.

Finite Element Model Updating

Values of selected parameters are adjusted through model updating to minimize the correlation coefficient of reference. Linear term of Taylor series expansion is used to show the functional relationship between predicted analytical parameters and experimental modal characteristics. The relation can be expressed as:

:

$$\{\boldsymbol{R_e}\} = \{\boldsymbol{R_a}\} + [\mathbf{S}](\{\boldsymbol{P_u}\} - \{\boldsymbol{P_o}\}), \tag{10}$$

$$\{\boldsymbol{\Delta R}\} = [\mathbf{S}]\{\boldsymbol{\Delta P}\}, \tag{11}$$

where:

$\{\boldsymbol{R_e}\}$: reference system responses vector (experimental data).

$\{\boldsymbol{R_a}\}$: predicted system responses vector with initial parameter values, $\{\boldsymbol{P_o}\}$

$\{\boldsymbol{P_u}\}$: updated parameter values.

$[\mathbf{S}]$: sensitivity matrix.

It is to be noted that equation (10) implies that responses occur in pairs, i.e. the corresponding analytical response must exist if experimental response is used as reference.

Procedures

Experimental Modal Analysis

Figure 1 shows the set-up of the composite plate. Composite type used is Carbon Fibre – Epoxy Resin Matrix with square dimension of 300mm in width. The thickness of the composite plate is 3.5mm. The composite plate was clamped at all 4 edges. In this research, modal analysis using Frequency Response Function (FRF) measurement techniques based on Fast Fourier Transformation (FFT) was used in determining the dynamic characteristics namely the natural frequencies, mode shapes and damping. The test method used is called impulsive excitation techniques[29]. The measured input is force from a modally tuned impact hammer and the measured output is acceleration from a uni-axial accelerometer. Data was obtained by using a DASYLab-National Instruments data acquisition system together with the impact hammer and the accelerometer. Impact hammer was connected to channel 1 of the National Instrument (NI) dynamic analyser and accelerometer was connected to channel 2 for capturing vertical direction response. The accelerometer was set at point 16 as the fixed response. The experiment was carried out by fixing the accelerometer and roving the impact hammer from point 1 until point 25. The sampling rate used was 2048 samples per seconds with block size of 4096 samples. This yields frequency resolutions of 0.5 Hz and 2 seconds of time record length to capture every response signal. Five averages or impacts were taken at each measurement point. The signals were processed by a self-developed virtual instruments application programme in DASYLab to generate FRF and Coherence Functions. The acquired data was then sent to modal analysis software called ME'Scope to perform post-processing and curve-fitting for the extraction of modal frequency, modal damping and modal shape.

Figure 1: Composite Plate Set-up

Finite Element Modal Analysis

A Finite Element (FE) model was built according to the geometrical properties, boundary conditions and material properties of the composite plate. The model was constrained at the 4 edges as in actual set-up. Initial values of material properties were assigned accordingly in the FE model. Longitudinal and transverse Young's moduli were initially set as 70GPa respectively. Meanwhile, initial values for shear modulus and Poisson's ratio were assigned at 5GPa and 0.1. When the FE model was completely defined, the structural element matrices and the numerical natural frequencies and mode shapes were computed.

Modal Correlation

Both the numerical and experimental databases are now complete and automatic mode shape pairing can be done. The standard requirement is to have a discrepancy of less than 5% in terms of resonance frequencies of matching mode shapes. The common target is for natural frequency predictions to be within 5% of the measured value. This is considered to be very good, while natural frequency predictions within 10% were considered marginal, but acceptable.[30] The Modal Assurance Criterion (MAC) was used as a quantitative approach to show the correlation of mode shape vectors between Experimental Modal Analysis (EMA) and Finite Element Modal Analysis. Computation of MAC is fast and does not require the mass and stiffness matrices. The off-diagonal terms of the MAC-matrix provide a mean to check linear independence between modes. MAC values range between 0 and 1, and should be interpreted as follows: MAC = 1 means the two mode shapes are identical or perfect correlated and MAC = 0 means the two mode shapes are completely different. Historically, when MAC values are greater than 0.7, the correlation of two modes are considered to be reasonable. Values greater than 0.85 are considered to be very good, it indicates close correlation of mode shapes.[30]

Finite Element Model Updating

The purpose of this analysis is to adjust the global orthotropic material properties. This is because, compared to other possible parameter types like mass density or shell thickness, these properties are most uncertain. Initial values were assigned from the manufacturers specifications.

The sensitivities for non-proportional parameter types like $E_x, E_y, G_{xy}, \upsilon_{xy}$ cannot be computed directly using a differential formulation as it is done for proportional parameters for example, the isotropic Young′s modulus. Therefore, the sensitivities are computed using a perturbation method. This method requires a perturbation coefficient (in percentage) which will be applied on the parameter value to compute a finite difference sensitivity value.

Results and Discussions

When initial values of elastic properties were assigned on the FE model, total of 6 experimental modes were used as reference response. In this case, the model updating procedure only requires 4 reference responses i.e. mode shapes to be selected to obtain a unique solution to obtain the 4 elastic constants namely longitudinal Young's modulus, transverse Young's modulus, in-plane shear modulus and Poisson's ratio (4 global parameters and 4 responses).

The mode shapes pairing as shown in Table 1 shows that all the 6 modes are considered well correlated, giving MAC values of over 70%. The highest experimental mode shape that can be paired with computational mode is mode 4 with MAC value of 98.8%. However, second and third modes show disagreement in the modal correlation process. Mode 2 and 3 of FE Modal Analysis correlated with Mode 3 and 2 of EMA respectively. MAC values chart is shown in Fig. 2(a). Meanwhile, the percentage error of natural frequencies above 10% showing bad agreement between the FE model and actual composite plate set up. FE model updating was performed in order to obtain a better correlation results. As shown in Table 1, the correlation results show significant improvement in percentage error in natural frequencies between FE Modal Analysis and EMA with the discrepancy of less than 5%. On the other hand, mode shape pairing shows minimal changes where the MAC values remain the achievement of 70% and above. The MAC values chart after FE model updating is shown in Fig. 2(b). It is worthwhile to mention here that first mode is showing slightly lower correlation in MAC because the mode itself is not sensitive enough to be excited. The fixed response at Point 16 was attached at the nodal point of the first mode where minimal response due to impact was captured at the particular point. Detailed mode shape pairings of all the 6 modes are shown in Fig. 2(c). Overall modal correlation results after FE model updating are considered acceptable and good correlation between FE modal analysis and EMA has been achieved.

Table 1: Modal Correlation between EMA and FEA before and after model updating

	Before Model Updating				After Model Updating			
	Natural Frequencies (Hz)				Natural Frequencies (Hz)			
Mode	FEA	EMA	Difference (%)	MAC	FEA	EMA	Difference (%)	MAC
1	113.93	98.43	15.75	69.1	99.83	98.43	1.42	68.7
2	204.74	177.66	15.25	76.1	178.66	170.95	4.51	74.1
3	204.74	170.95	19.77	75.8	184.93	177.66	4.09	74.3
4	318.46	286.13	11.3	98.8	281.65	286.13	-1.56	98.6
5	408.69	371.87	9.9	96.7	358.96	371.87	-3.47	96
6	507.7	454.58	11.68	95.9	445.3	454.58	-2.04	95.5

Figure 2: (a) MAC before model updating, (b) MAC after model updating

(c) Mode shapes correlation between EMA and FEA

The material properties, i.e. longitudinal Young's modulus, transverse Young's modulus, in-plane shear modulus and Poisson's ratio were determined with the updated and well-correlated FE model. The results are shown in Table 2. There are around 10% to 30% changes in these properties as compared to the initial assigned values. These predicted material properties shall reflect the actual elastic constants of the composite plate without destroying the material.

Table 2: Updated material properties of the composite plate

Type	Old (Pa)	Actual (Pa)	Difference (%)
E_x	7.00E+10	5.60E+10	-19.97
E_y	7.00E+10	4.90E+10	-30.00
G_{xy}	5.00E+09	4.01E+09	-19.72
υ_{xy}	1.00E-01	7.89E-02	-21.06

Conclusion

A hybrid non-destructive approach has been presented for determining the material properties of an orthotropic composite plate based on experimental modal data and FEA. This approach is fast and inexpensive in updating the FE model. The convergence between the experimental modal data and FEA is very satisfactory after model updating which achieves less than 5% in difference of natural frequencies and over 70% in MAC for all the 6 vibration modes. Material properties of the orthotropic composite plate have been successfully determined using this hybrid approach.

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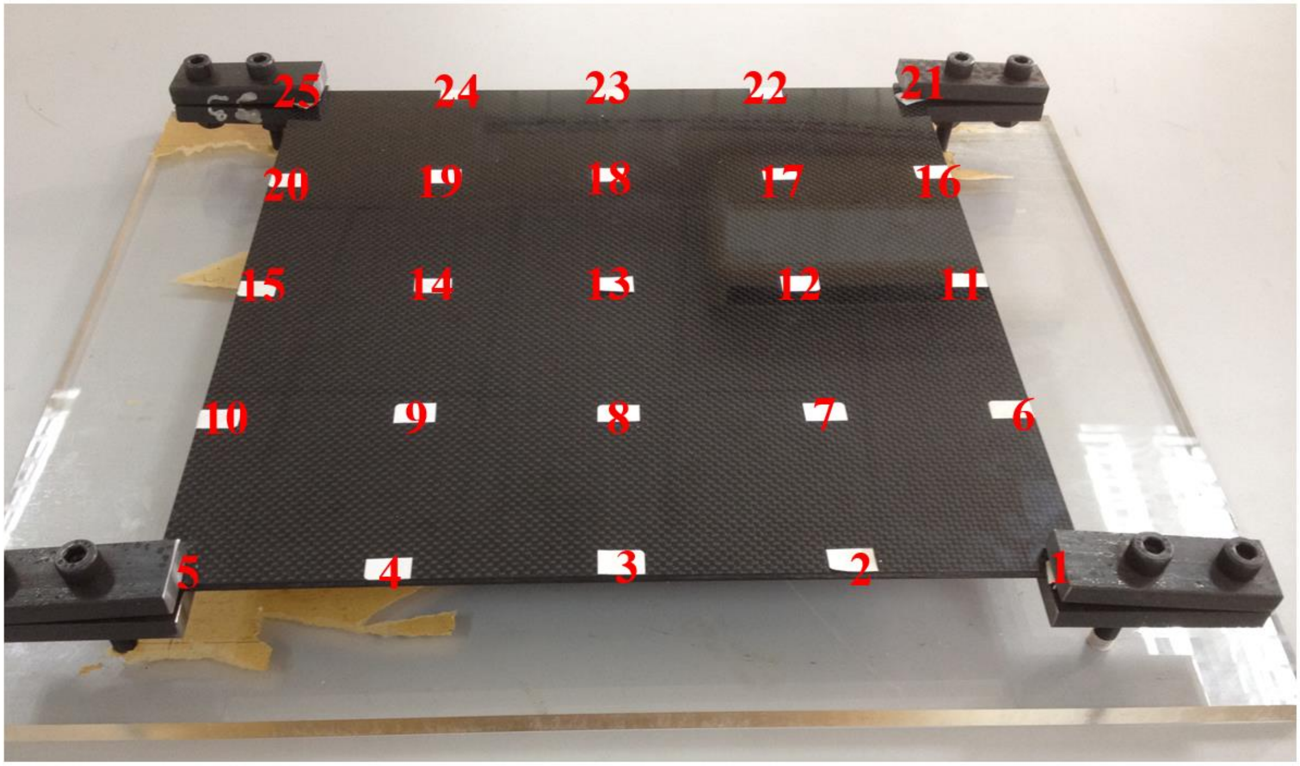

Figure 1: Composite Plate Set-up

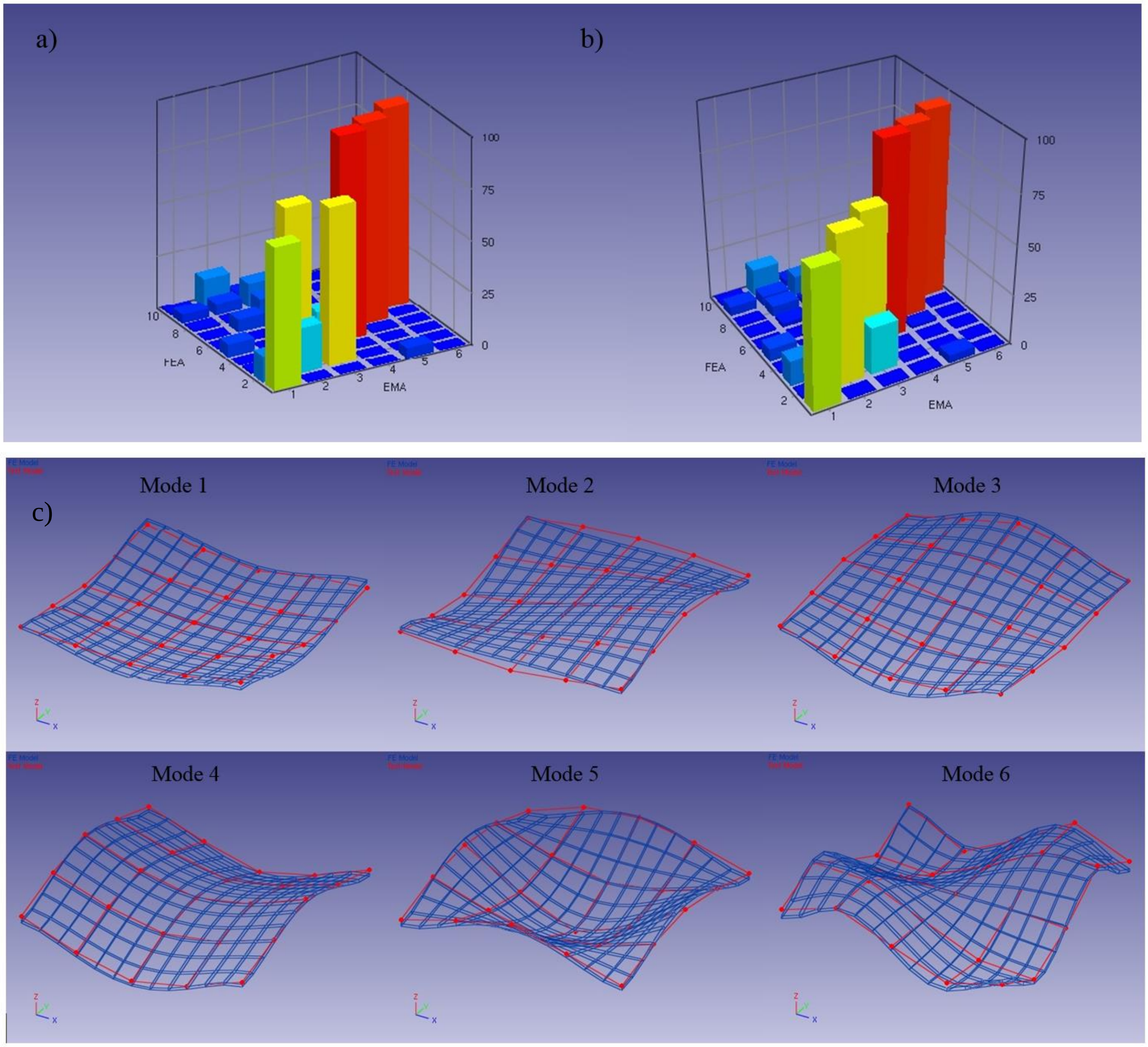

Figure 2: (a) MAC before model updating, (b) MAC after model updating

(c) Mode shapes correlation between EMA and FEA