Modal strain energy change ratio for damage identification in honeycomb sandwich structures

K. R. Pradeep a, B. Nageswara Rao b*, S. M. Srinivasan c, K. Balasubramaniam d

a Structural Design and Engineering Group, Vikram Sarabhai Space Centre, Trivandrum-695 022, India
b Department of Mechanical Engineering, School of Civil and Mechanical Sciences, KL University, Green Fields, Vaddeswaram – 522 502, India
c Applied Mechanics Department, Indian Institute of Technology Madras, Chennai-600 036, India
d Mechanical Engineering Department, Indian Institute of Technology Madras, Chennai-600 036, India

Keywords:
Debond, Finite element model, Frequency, Honeycomb sandwich panel, Modal strain energy change

Abstract

This paper deals with the generation of layered finite element models using a standard finite element package for simulation of undamaged and delaminated sandwich structures that are appropriate to vibration based structural health monitoring (SHM) and damage detection techniques. The intact regions are modelled using three layered elements. The skin-core debonded regions are modelled using single layer elements representing debonded top or bottom skin and two layer element representing the core and bottom or top skin either side of the debond along with contact definition between skin and core. Modal analysis has been carried out on honeycomb sandwich plate configurations and the changes in the frequencies due to debond calculated. A meaningful parameter called ‘Modal Strain Energy Change Ratio (MSECR) is used as an indicator for damage indicator in sandwich structures. The present numerical experiments and validation with the existing test results indicate the efficiency of the MSECR as an indicator for damage in sandwich structures.

1. Introduction

Honeycomb sandwich panels (viz., deck plate and payload components having low density and high stiffness to weight ratio) are extensively used in aircraft/aerospace industries. Debond is a frequently encountered damage in metallic honeycomb sandwich structures. Early detection of damage can prevent from catastrophic failure as well as structural deterioration in service beyond repair. Structural health monitoring (SHM) and damage assessment are inevitable for aerospace vehicles [1-3].

Vibration-based damage detection technique is an effective tool for damage identification of composite structures through the knowledge of frequency shifts, changes in mode shape, changes in curvature mode shape, modal force error, flexibility changes, modal strain energy change,
transmissibility, impedance change damping, etc [4-6]. Lestari and Qiao [7] have followed a procedure for damage identification and health monitoring based on dynamic response (curvature mode shape) and using piezoelectric sensor for fiber reinforced polymer sandwich composites. Han et al. [8] have performed the delamination buckling and propagation analysis of honeycomb panels using cohesive element approach. Aviles and Carlsson [9] have examined the local buckling of sandwich panel (skin with foam core) containing embedded debonds. Yam et al. [10] have suggested a vibration based method to locate the internal delamination in multilayered composites using a combination of measured modal damping change with computed modal strain energy distribution. Harney et al. [11] have utilized curvature mode shape and followed the damage index method and curvature factor method for damage detection in carbon/epoxy composite beams. Distributed PVDF sensors are used for direct measurement of the mode shape curvatures along with impact excitation using hammer and continuous excitation with piezoceramic actuators.

The strain energy based damage detection is found to be efficient compared to other methods (viz., change in mode shape curvature, change in flexibility and change in flexibility curvature) which considers the modal strain energy changes in structural element before and after the damage formation [12, 13]. Shi et al. [14] have considered the modal strain energy change ratio for identification of damage location. The optimal spatial sampling interval to minimize the effect of measurement noise and truncation errors is adopted for damage detection by curvature or strain energy mode shape [15]. Hu et al. [16] have utilized the modal strain energy method for identification of surface cracks in carbon/epoxy composite laminates. Cecchini [17] and Agosto et al. [18] have examined the effects of curvature changes for damage identification in sandwich composite cantilever beams.

Morassi and Rocchetto [19] have shown the flexural frequencies as high sensitivity to damage and considered as valid damage indicator for composite beams. Wei et al. [20] have examined the damage-induced energy variation of response signal and the mechanism of mode-dependent energy dissipation of composite plates due to delamination. Yan et al. [21] have adopted the cross modal strain energy (CMSE) and niche genetic algorithms (GAs) for the damage detection in composite structures. Guo and Li [22] integrated structural modal strain energy and frequency information using fusion theory for the damage identification. Particle swarm optimization (PSO) is used to identify the extent of structural damage.

Kulkarni and Frederick [23] have analyzed the de-bonded shell structure utilizing the reduced bending rigidity of the shell by summing the moment of inertia of delaminated layers. Mujumdar and Suryanarayan [24] have modeled delaminated composite beam in to four separate component
segments and analyzed each as an Euler beam and validated with experimental results. Schwartz-Givli et al. [25] have suggested a methodology for free vibrations of delaminated unidirectional sandwich panels accounting the model for the flexibility of the core in the out-of-plane (vertical) direction resulting high-order displacement, acceleration, and velocity fields within the core. Two approaches (viz., regional approach and layer-wise approach) are being followed for damage detection. In the regional approach, the delaminated laminate is divided into equivalent single-layer sub-laminates or segments and the continuity conditions are imposed at delamination junctions. In the layer-wise models, the sub-laminates are modelled using the layer-wise theories [26, 27].

Sandwich structures are more prone to stiffness reduction due to damages. The modal strain energy based techniques can serve as an effective damage detection tool, which demand computationally efficient damage models of sandwich structures. This paper deals with the generation of layered finite element models using ANSYS finite element package for accurate simulation of undamaged and damaged sandwich plate configurations. A critical view on the non-dimensional parameter called the 'Modal Strain Energy Change Ratio (MSECR)' that is often being used in recent literature to identify the debond location in the honeycomb sandwich panel is presented. It is observed from the present numerical experiments that MSECR is an efficient approach useful for damage identification in sandwich structures.

2. Modal Strain Energy Change Ratio (MSECR) Evaluation

The equations of motion for a multi-degree of freedom (MDOF) dynamic system can be written in the form

$$ [K][\phi] = \omega^2[M][\phi] $$

(1)

Here $K$ and $M$ are the stiffness and mass matrices. $\phi$ is the mode shape. $\omega$ is the frequency. Presence of debond in sandwich structures causes changes in the stiffness matrix thereby changes in modal frequencies and mode shapes. It should be noted that the sandwich skin sheets are thin compared to the low modulus thick core layer and the presence of debond significantly reduces the local stiffness. The stiffness matrix, modal frequencies and mode shapes of the structure with damage can be represented by

$$ [K'] = [K] - [\Delta K] $$

$$ \phi' = \phi + \Delta \phi $$

(2)

(3)

where $\Delta K$ is the fractional reduction in the stiffness matrix; $\Delta \phi$ is the fractional change in the mode shape and the superscript “d ”denotes the quantities for damaged case.

Modal strain energy of the element/region $j$ for the mode $i$ is given by
Modal strain energy of the damaged structure is

\[ \text{MSE}_d = \{ \phi_i \}^T [K] \{ \phi_i \} \]

(4)

The modal strain energy change ratio (MSECR) for the element \( j \) corresponding to the mode \( i \) is

\[ \text{MSECR}_j = \frac{\text{MSE}_d - \text{MSE}_0}{\text{MSE}_d} \]

(5)

For several modes, MSECR is defined as the average of their normalized MSECR values. The (MSECR) for the element \( j \) corresponding to \( m \) modes can be evaluated as

\[ \text{MSECR}_j = \frac{1}{m} \sum_{i=1}^{m} \frac{\text{MSECR}_i}{\text{MSECR}_{\text{max}}} \]

(6)

3. Finite Element Analysis

Finite element analysis has been carried out on sandwich plates made of aluminium skin and aluminium honeycomb core. Finite element models are generated using layered shell element (shell 99) of ANSYSTM for accurate simulation of undamaged (intact) and debonded sandwich structures. Figure 1 shows the layered shell element, which is based the classical lamination theory (CLT) and the first-order shear deformation theory (FSDT).

The debonded region is modeled using two sets of elements: one set representing the debonded skin and the other set for core and skin. Figure 2 shows the kinematic relations ships between debonded and undamaged regions, in which force and moment equilibrium at element junctions are satisfied. Contact of the debonded skin with core is simulated utilizing Cont170 and target175 element of ANSYSTM. Contact friction factor is specified as 0.15.
Properties of AA2014-T6 skin sheet material are: Young’s modulus, \( E_f = 68670 \text{ N/mm}^2 \); Poisson’s ratio, \( \nu = 0.3 \); Yield strength, \( \sigma_y = 360 \text{ N/mm}^2 \); Ultimate strength, \( \sigma_u = 400 \text{ N/mm}^2 \); and Density, \( \rho = 2800 \text{ kg/m}^3 \). Properties of the Honeycomb core (CR III 5056 141) material are: \( E_x = 41 \text{ N/mm}^2 \); \( E_y = 41 \text{ N/mm}^2 \); \( E_z = 1600 \text{ N/mm}^2 \); \( \nu_{12} = 0.44 \); \( \nu_{13} = \nu_{23} = 0 \); \( G_{xy} = 0 \text{ N/mm}^2 \); \( G_{xz} = 220 \text{ N/mm}^2 \); and \( G_{yz} = 410 \text{ N/mm}^2 \). Modal analysis has been carried out on the undamaged and debonded honeycomb sandwich structural configurations to obtain frequencies and strain energy. In the debonded region the strain energies of upper and lower elements are summed up to get the total energy of the region. The elemental level modal strain energy change ratio (MSECR) is evaluated using equation (7).
3.1. Honeycomb Sandwich Cantilever Beam type rectangular plates

The configuration of sandwich cantilever beam type plate shown in Figure 3 has 400mm length (L), 50mm width (B) and 15.6mm total thickness (which includes 0.3mm thickness of top and bottom skin sheets and 15mm thick core). Six case studies are made considering different damage sizes and locations in the sandwich cantilever beam type rectangular plates. Case studies 1 to 4 consist of 5% of beam length (20mm) through the width of panel as damage, which is close to the support, 100, 200 and 300 away from the support. Figure 4 shows different damaged sandwich beam type plates. Figure 5 shows the finite element model of intact (undamaged) and a damaged sandwich cantilever beam. Total number of elements and nodes in the intact model are 160 and 569 respectively. For the case of damaged models these are 168 and 606 respectively.
Modal analysis has been performed and extracted the first ten natural frequencies. It is noted from the results presented in Table 1 that there is a decreasing trend in frequencies of the damaged honeycomb sandwich beams due to reduction in stiffness. The MSECR plots for sandwich cantilever beams with 5% of their length as damage at different locations are shown in Figure 6. The maximum MSECR values of case-1, case-2, case-3 and case-4 are 0.6, 0.82, 0.42 and 0.84 respectively. The peaking up of MSECR can be seen at damage locations. The effect of increase in the size of damage on MSECR can be seen in Figure 7. The maximum MSECR values of case-5 and case-6 in Figure 7 are 0.71 and 0.23 respectively. The area under the peak MSECR is increased due to increase in the damage size. The plots clearly indicate the damage location at which modal strain energy change ratio (MSECR) is high.

### Table 1. Natural frequencies (Hz) of undamaged and damaged honeycomb sandwich beam type rectangular plates.

<table>
<thead>
<tr>
<th>Mode</th>
<th>Undamaged</th>
<th>Case-1</th>
<th>Case-2</th>
<th>Case-3</th>
<th>Case-4</th>
<th>Case-5</th>
<th>Case-6</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>105.4</td>
<td>105.23</td>
<td>105.25</td>
<td>105.28</td>
<td>105.32</td>
<td>104.13</td>
<td>101.25</td>
</tr>
<tr>
<td>2</td>
<td>197.47</td>
<td>197.48</td>
<td>197.47</td>
<td>197.45</td>
<td>197.40</td>
<td>197.49</td>
<td>197.49</td>
</tr>
<tr>
<td>3</td>
<td>653.16</td>
<td>641.16</td>
<td>646.02</td>
<td>652.92</td>
<td>650.28</td>
<td>570.68</td>
<td>459.39</td>
</tr>
<tr>
<td>4</td>
<td>1157.7</td>
<td>1157.70</td>
<td>1157.40</td>
<td>1157.60</td>
<td>1157.60</td>
<td>1068.80</td>
<td>861.07</td>
</tr>
<tr>
<td>5</td>
<td>1385.3</td>
<td>1292.00</td>
<td>1316.30</td>
<td>1344.10</td>
<td>1373.40</td>
<td>1157.60</td>
<td>1157.00</td>
</tr>
<tr>
<td>6</td>
<td>1797.9</td>
<td>1715.70</td>
<td>1793.60</td>
<td>1755.50</td>
<td>1788.60</td>
<td>1408.40</td>
<td>1225.80</td>
</tr>
<tr>
<td>7</td>
<td>2474.7</td>
<td>2474.80</td>
<td>2474.50</td>
<td>2474.00</td>
<td>2474.00</td>
<td>2537.40</td>
<td>1588.10</td>
</tr>
<tr>
<td>8</td>
<td>2969.9</td>
<td>2969.50</td>
<td>2966.30</td>
<td>2969.30</td>
<td>2968.40</td>
<td>2623.80</td>
<td>2168.70</td>
</tr>
<tr>
<td>9</td>
<td>3440.2</td>
<td>3189.70</td>
<td>3369.70</td>
<td>3418.30</td>
<td>3431.30</td>
<td>2623.80</td>
<td>2168.70</td>
</tr>
<tr>
<td>10</td>
<td>4161.4</td>
<td>3901.40</td>
<td>4120.60</td>
<td>3994.90</td>
<td>3931.40</td>
<td>2881.80</td>
<td>2474.80</td>
</tr>
</tbody>
</table>

### 3.2. Simply Supported Honeycomb Sandwich Square Plates

Honeycomb sandwich square plates of 400 x 400 x 15.6mm shown in Figure 8 are considered. To study the effect of element size (sensor density in experiment) coarse and refined finite element models (see Figure 9) are considered. Damage of 100 x 50mm is created at the centre of the plate. Total number of elements and nodes in the coarse mesh of the intact model are 64 and 81 respectively, whereas in the case of damage model these are 76 and 82 respectively. In the fine mesh, total number of elements and nodes of the intact model are 289 and 256 respectively, whereas these are 304 and 280 respectively in the damage model.
Case 1

Figure 6 (a). MSECR plot of sandwich cantilever beam type plates with 5% of its length as damage at different locations.

Case 2

Figure 6 (b). MSECR plot of sandwich cantilever beam type plates with 5% of its length as damage at different locations.

Case 3

Figure 6 (c). MSECR plot of sandwich cantilever beam type plates with 5% of its length as damage at different locations.

Case 4

Figure 6 (d). MSECR plot of sandwich cantilever beam type plates with 5% of its length as damage at different locations.

Case 5

Figure 7. MSECR plots of sandwich cantilever beam type plates with increasing damage size.

Case 6

Figure 8. Simply supported honeycomb sandwich plate.
Modal analysis results of both intact and damaged honeycomb sandwich simply supported square plates in Table 2 indicate decreasing trend in the frequencies due to reduction of stiffness.

**Table 2.** Natural frequencies (Hz) of undamaged and damaged honeycomb sandwich simply supported square plates.

<table>
<thead>
<tr>
<th>Mode</th>
<th>Undamaged model</th>
<th>Damaged model</th>
<th>Mode</th>
<th>Undamaged model</th>
<th>Damaged model</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>418.8</td>
<td>411.8</td>
<td>6</td>
<td>2024.3</td>
<td>2001.5</td>
</tr>
<tr>
<td>2</td>
<td>1012.7</td>
<td>1000.8</td>
<td>7</td>
<td>2370.8</td>
<td>2336.3</td>
</tr>
<tr>
<td>3</td>
<td>1036.8</td>
<td>1019.4</td>
<td>8</td>
<td>2454.5</td>
<td>2426.5</td>
</tr>
<tr>
<td>4</td>
<td>1555.8</td>
<td>1554.6</td>
<td>9</td>
<td>3008.4</td>
<td>2923.3</td>
</tr>
<tr>
<td>5</td>
<td>1913.0</td>
<td>1880.4</td>
<td>10</td>
<td>3142.8</td>
<td>3070.3</td>
</tr>
</tbody>
</table>

**Figure 9.** Finite Element model of intact and damaged honeycomb sandwich simply supported square plates.

**Figure 10.** MSECR plot of honeycomb sandwich simply supported square plates.
(a) Natural frequencies (Hz) of an intact sandwich composite cantilever beam

<table>
<thead>
<tr>
<th>Mode</th>
<th>Cecchini [17] Test</th>
<th>Analysis</th>
<th>Present Analysis</th>
</tr>
</thead>
<tbody>
<tr>
<td>First</td>
<td>25.625</td>
<td>24.903</td>
<td>24.96</td>
</tr>
<tr>
<td>Second</td>
<td>134.062</td>
<td>134.668</td>
<td>134.04</td>
</tr>
<tr>
<td>Third</td>
<td>314.062</td>
<td>318.859</td>
<td>316.36</td>
</tr>
</tbody>
</table>

(b) Natural frequencies (Hz) of damaged sandwich composite cantilever beam

<table>
<thead>
<tr>
<th>Mode</th>
<th>Cecchini [17] Test</th>
<th>Analysis</th>
<th>Present Analysis</th>
</tr>
</thead>
<tbody>
<tr>
<td>First</td>
<td>25.312</td>
<td>24.547</td>
<td>24.498</td>
</tr>
<tr>
<td>Second</td>
<td>129.687</td>
<td>122.079</td>
<td>130.73</td>
</tr>
<tr>
<td>Third</td>
<td>310.312</td>
<td>303.989</td>
<td>306.53</td>
</tr>
</tbody>
</table>

**Figure 11.** Comparison of analytical and experimental natural frequencies and mode shapes of the intact (undamaged) and the damaged sandwich composite cantilever beams.

Modal Strain Energy Change Ratio (MSECR) plots for the coarse and refined meshes are shown in Figure 10. For the case of refined mesh, maximum MSECR value at the damage location
is 0.5, whereas it is 0.3 for coarse mesh due to averaging effect. The damage location is clearly identifiable. For the refined mesh case the damage size is having a better clarity.

This study indicates the damage detection in two step processes. A coarse mesh experiment can locate the damage and refined mesh can provide the size of the damage. From the modal strain energy change ratio (MSECR) plots, it is clear that MSECR value is high at the damage locations.

3.3. Experimental Validation

Numerical experiments carried out in the previous sections indicate that any localized damage/imperfection in a structure produces variations in the dynamic response. The damage produces a localized reduction in stiffness and the natural frequencies have decreased compared to those corresponding to the perfect beam/plate configurations. Analytical and experimental studies have been made to study the effects of curvature changes and identify the presence of damage in sandwich composite cantilever beams [17, 18]. The vibration mode shapes (flexural mode shapes) of cantilever beams can be measured accurately. Hence, the first three flexural mode shapes and the corresponding natural frequencies of sandwich composite cantilever beams with and without damage are considered here to examine the adequacy of the present modal strain energy change ratio (MSECR) through comparison of analytical and experimental results.

The sandwich composite beams consisting of bi-directional woven [00/900] carbon fiber face-sheets bonded to polyurethane foam core with epoxy resin have 584mm length and 51mm width with 0.7mm face-sheet thickness and 6.4mm core thickness. Elastic properties of the face-sheet material are: Longitudinal modulus = 56054 N/mm²; Transverse modulus = 56054 N/mm²; Poisson’s ratio = 0.3; Shear modulus = 620 N/mm²; Mass density = 2105 kg/m³. Elastic properties of the foam core material are: Young’s modulus = 0.689 N/mm²; Poisson’s ratio = 0.3; Shear modulus = 38.6 N/mm²; Mass density = 166.7 kg/m³. Skin core separation (delamination) of 25.4mm long is considered at 101.6mm from the root of the sandwich composite cantilever beam. As in previous numerical experiments, finite element model of the sandwich composite cantilever beam is created utilizing ANSYS software. The undamaged model consists of 362 elements and 465 nodes. The damaged zone is modeled using two set of layered elements and each set consist of 16 elements.

Figure 11 shows the comparison of measured and computed natural frequencies as well as the corresponding mode shapes of intact and damaged sandwich composite cantilever beams. The present finite element analysis results are found to be reasonably in good agreement with existing
test results [17]. The numerical and experimental mode shape data is discrete in space. Change in slope at each node is estimated using a central difference approximation. The second derivative of the displacement \( \phi \) along the X direction at node \( i \)

\[
\left( \frac{\partial^2 \psi}{\partial x^2} \right)_i = \frac{\phi_{i-1} - 2\phi_i + \phi_{i+1}}{\Delta x^2}
\]

(8)

The modal strain energies are computed from the mode shapes as.

\[
U_i = \frac{D}{2} \int_i^{i+1} \left( \frac{\partial^2 \phi}{\partial x^2} \right)_i^2 dx
\]

(9)

Here D is the stiffness and \( u \) is the mode shape of the beam and \( \left( \frac{\partial^2 \phi}{\partial x^2} \right)_i \) is the curvature evaluated from the mode shape \( \phi \) corresponding to mode \( i \).

Figure 12. Comparison of modal strain energy change ratio (MSECR) from the finite element analysis and measured mode shapes of sandwich composite cantilever beams.
Figure 12 shows the modal strain energy change ratio (MSECR) evaluated for the first three modes from experimental and finite element data. Peak MSECR is noticed at the damage location (i.e., around 100mm away from the fixed end) of the sandwich composite beam. This shows that the modal strain energy based method is capable of identifying the damage and its location in honeycomb sandwich structures.

4. Concluding Remarks

This paper performs the modal analysis of honeycomb sandwich plates containing debonds and proposed to us the modal strain energy change ratio (MSECR) as an indicator for damage. Applicability of the MSECR concept for damage detection in the honeycomb sandwich structures is examined by introducing a debond. Finite element models of honeycomb sandwich rectangular plates are generated using shell element and damage is simulated by defining contact elements between skin and core elements. Modal analysis has been performed to extract frequencies and modal strain energies. Reduction trend is noticed in natural frequencies of debonded sandwich structures due to reduction in stiffness. However this trend may not be significant to identify the damage. The MSECR is found to be efficient and clearly identifies the damage in sandwich structures. It is customary to gain knowledge on the behavior of the undamaged structures in the Structural Health Monitoring (SHM) to understand the changes (if any) due to introduction of damage in service. The MSECR helps to indicate the damage from the measure mode shape. Otherwise, one has to measure the mode shape of the structure prior to its use in service to evaluate the strain energy and later on, evaluate MSECR during service from the measure mode shape to trace for the damage (if any).

References


DOI: 10.1177/1475921706067764

DOI: 10.1177/0583102406065898

DOI: 10.1006/jsv.1999.2624

DOI:10.1177/058310249803000201

DOI:10.1016/j.compstruct.2004.01.023

DOI: 10.1023/A:1016333709040

DOI: 10.1177/0021998305056387

DOI: 10.1177/0021998304045590

DOI: 10.1177/1475921704047502

DOI:10.1016/j.jsv.2005.07.036


DOI:10.1006/jsvi.1998.1878

DOI:10.1016/j.jsv.2004.08.021

DOI:10.1016/j.compstruct.2005.04.020


