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Effects of Structural Damage on Dynamic Behavior at Sandwich Composite Beams – Part II- FEM Analysis

This paper presents results obtained by modal analysis on composite beam like structures in healthy and damaged state. The aim is to obtain damage "signatures" for all possible damage scenarios and to use these data to assess transversal cracks based on vibration techniques, by involving natural frequency shifts. The analysis was performed in SolidWorks software for a five-layer composite, 20 vibration modes being obtained by numerical simulation.

***Keywords**: beam, damage, FEM, frequency, structure*

1. Introduction

The composites materials (or composites) consist of two or more constituents with quite different physical and/or chemical properties, separately and distinctive identifiable within the structure.

Unlike to natural materials which have predefined properties, composites are elaborated to fulfill predefined needs, permitting a new approach in structural design. Among composites sandwich structures are a special class, manufactured by attaching two thin stiff faces to a lightweight thick core [1], [2], [3].

The faces are usually made of steel or aluminum, while the core is low strength material like foam. The spatial distribution of the three components together with their mechanical and physical properties provides the sandwich composite with high shear stiffness to weight ratio and high bending stiffness to weight ratio respectively [4], [5].

Sandwich structures related problems, for static and dynamic loads in various environments, are studied from the mid of the last century. The main attempts are focused on bending and buckling optimal design shock-resistance [6], [7], [8] and vibrational behavior with identification of natural frequencies and/or elastic parameters of the sandwich structures. Recent works are focused on detection of damage in sandwich beams or plates, with different core types and damage configurations [8]. Most actual damage detection methods, even for isotropic structures, are difficult to be applied, due to the fact that no analytic relation to quantify frequency changes caused by the damage was found.

2. 3D modeling and numerical simulation

The numerical method chosen for modal analysis is the finite element method (FEM) implemented by SolidWorks software. Geometry modeling analysis was performed in CAD module of the same program. In order to determine the proper frequency, usually the modal analysis module is used.

The ambient temperature has been set to 22 ºC. By using the modal analysis of first 20 proper frequencies were determined. From there the first 10 proper frequencies of the transversal vibration modes, after the minimum moment of inertia have been extracted.

3D model meshing was computed by the tetrahedral finite elements, with an average element size of 1 mm. The materials were chosen from the database of the program, namely: structural steel AISI 1045, and PVC rigid. Mechanical and geometrical characteristics of the 5-layer structure were described into the first script part. Mechanical features of the materials are indicated in table 1 and 2.

Table 1. Mechanical features of steel

Yield strength	Tensile strength	Mass density	Elastic modulus	Poisson's ratio	Thermal expansion coefficient
[N/mm^2]	[N/mm^2]	[kg/m^3]	[N/m^2]	[-]	[K^{-1}]
4,5e+07	4.07e+07	1,300	2.41e+09	0.3825	0.000028

Table 2 Mechanical features of rigid PVC

Yield strength	Tensile strength	Mass density	Elastic modulus	Poisson's ratio	Thermal expansion coefficient
[N/mm^2]	[N/m^2]	[kg/m^3]	[N/m^2]	[-]	[K^{-1}]
5,30e+08	6.25e+08	7,850	2.05e+011	0.29	0.000012

The simulations were performed for the beam fixed at one end and with a defect placed at x = 274 mm from the fixed end. Defect was 1 mm width, gradually increased defect of 0.2 mm depth for each simulation. Figure 1, 2 and 3 show a portion of defective rod and the mesh mode for beam with 1x0.2 mm damage dimension. Table 3 provides information related to the mesh for all type of defected beams.

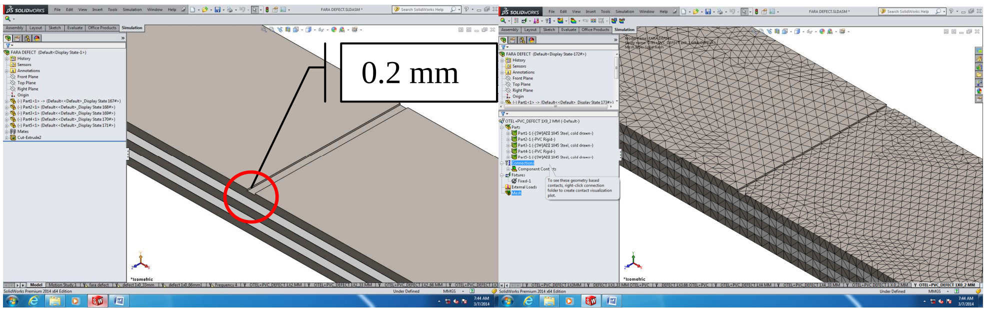

Figure 1. Portion of defective rod and the mesh mode for beam with 1x0.2 mm damage dimension

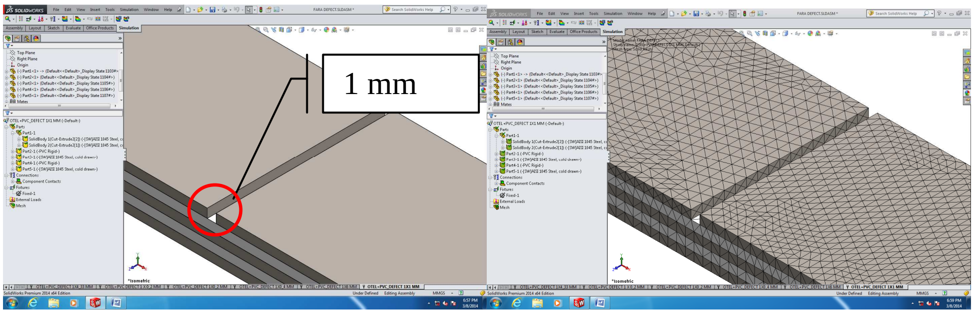

Figure 2. Portion of defective rod and the mesh mode for beam with 1x1 mm damage dimension

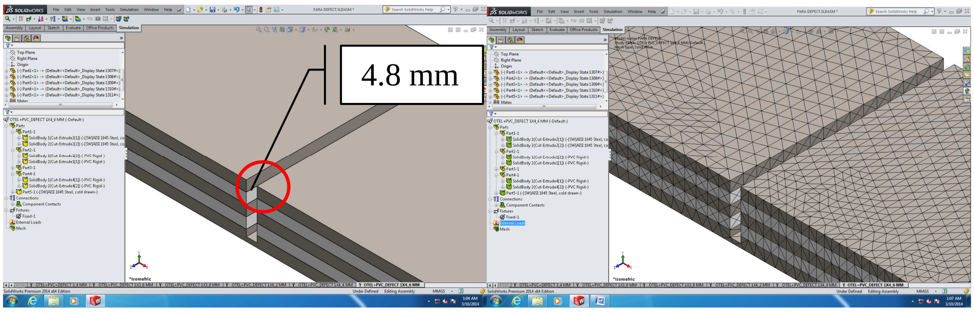

Figure 3. Portion of defective rod and the mesh mode for beam with 1x4.8 mm damage dimension

Table 3. Mesh information

Type of structure	Total number of nodes	Total number of elements	Time to complete mesh [min:s]
Beam without defect	1,006,553	676,702	01:03
Beam with defect 1x0.2 mm	1,308,697	898,548	01:15
Beam with defect 1x0.4 mm	1,308,624	898,479	01:18
Beam with defect 1x0.6 mm	1,305,538	896,304	01:18
Beam with defect 1x0.8 mm	1,306,166	896,716	01:17
Beam with defect 1x1 mm	1,361,040	937,379	01:18
Beam with defect 1x1.2 mm	1,363,461	939,,130	01:17
Beam with defect 1x1.4 mm	1,363,331	939,010	01:17
Beam with defect 1x1.6 mm	1,361,757	937,832	02:24
Beam with defect 1x1.8 mm	1,361,850	937,831	01:17
Beam with defect 1x2 mm	1,492,351	1,032,590	01:21
Beam with defect 1x2.2mm	1,495,077	1,034.604	01:19
Beam with defect 1x2.4mm	1,494,960	1,034,491	02:30
Beam with defect 1x2.8mm	1,494,274	1,03,3915	01:22
Beam with defect 1x3 mm	1,603,949	1,11,4514	03:47
Beam with defect 1x3.2 mm	1,604,994	1,115,243	01:28
Beam with defect 1x3.4 mm	1,604,804	1,115,099	01:39
Beam with defect 1x3.6 mm	1,605,371	1,115,488	05:27
Beam with defect 1x3.8 mm	1,605,679	1,115,694	01:25
Beam with defect 1x4.4.8 mm	1,708,830	1,191,221	03:04

3. FEM analysis

The results indicate the nature of proper frequencies (quantitative assessment) and modal forms (qualitative assessment pointing only the curve shape, not the actual values). Examples of the modal shapes for the transversal vibrations of beam, without defect, respective with defect (defect of 1.6 mm depth), are pictured into the Figures 4 and 5. Values of the first 10 natural frequencies of vibration modes rack are shown in table 4, as for the FEM analysis and analytical study.

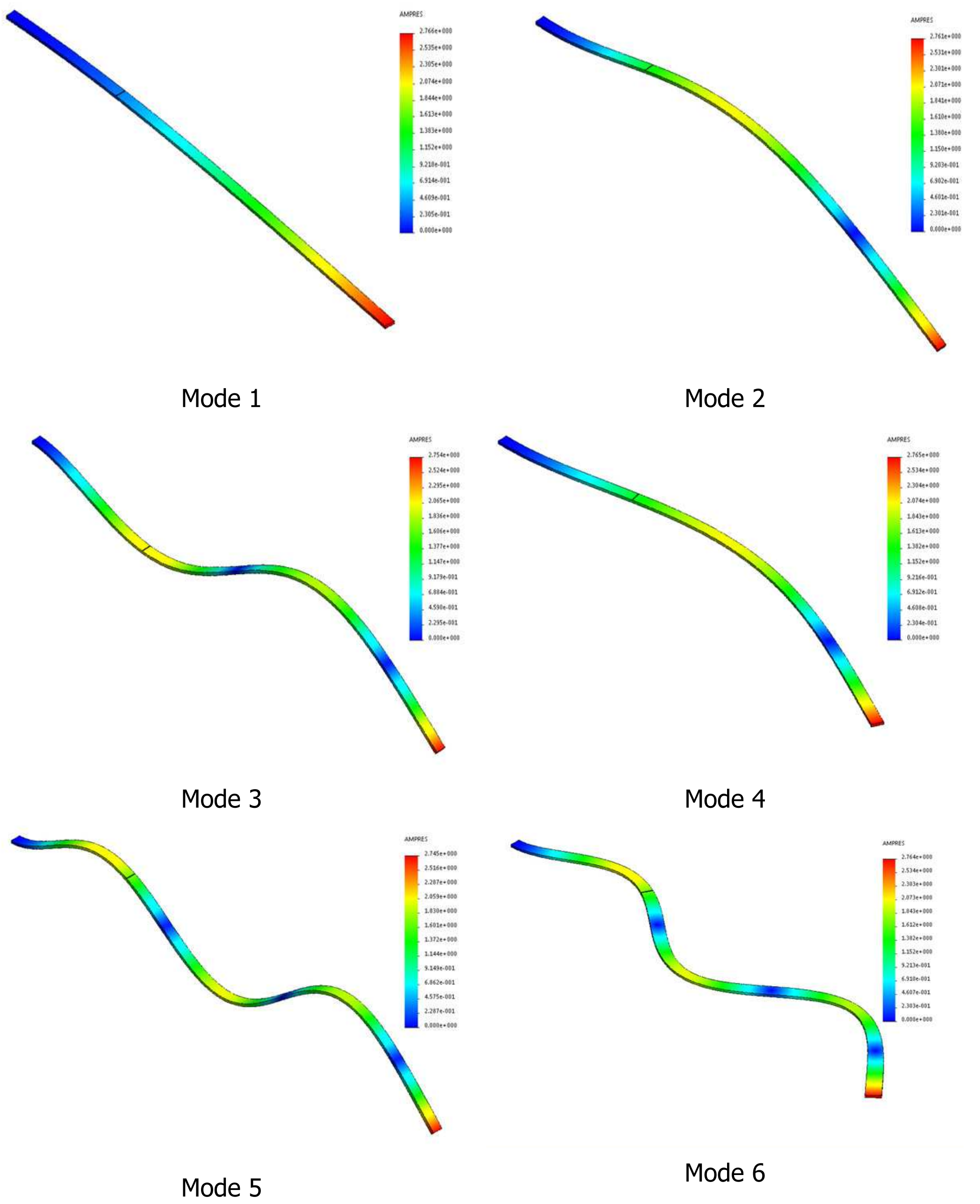

Figure 4. First six vibration modes for damaged beam

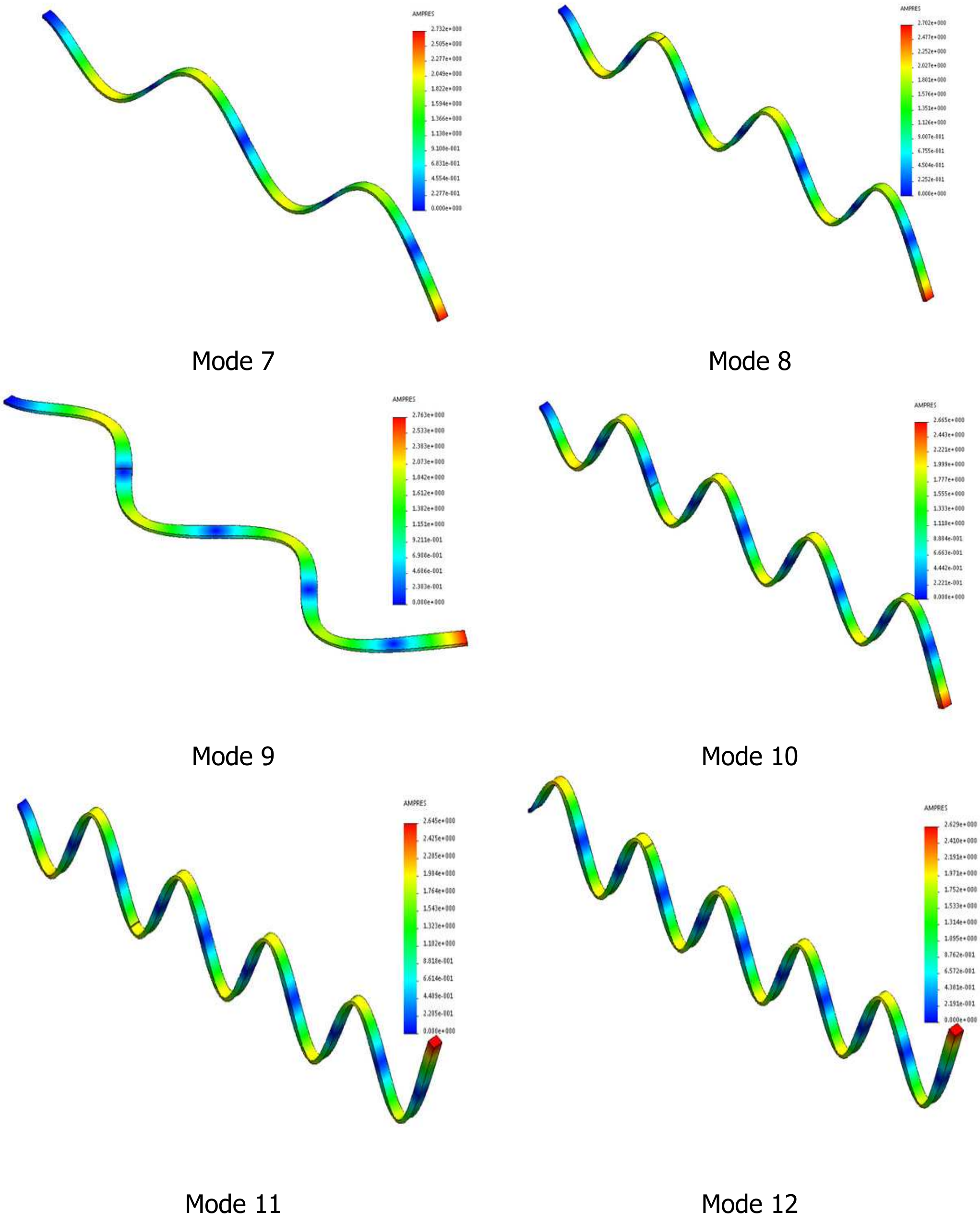

Figure 5. Vibration modes seven to twelve for damaged beam

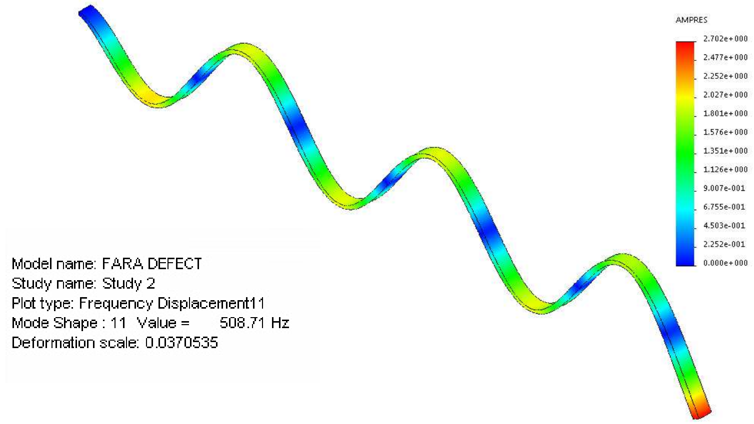

Figure 6. Eighth vibration mode shape of the beam intact

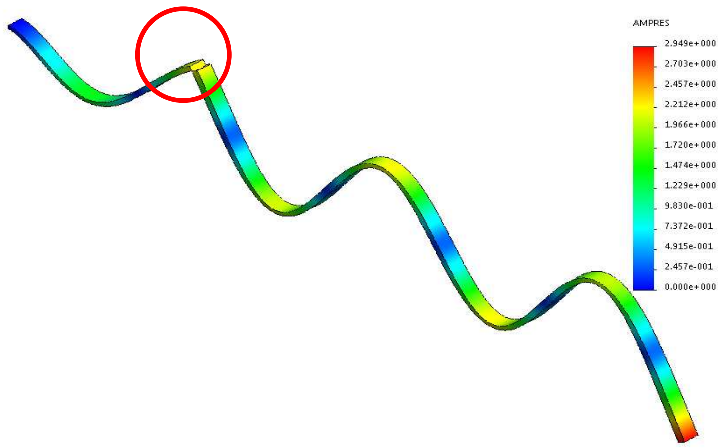

Figure 7. Eighth vibration mode shape of the beam with defect at x / L = 0.274 m

Figure 7 highlights for a larger scale on the y-axis movements the influence of the defect located at 274 mm flush end, on modal form, with repercussions on proper frequency in that way.

In fact, a much smaller deformation was determined. Often this clue is difficult to observe, even by the laser vibrations. In addition to another fact that the natural frequencies change significantly (Table 4), the global control methods based on changing frequencies is recommended.

Table 4. Values of frequency

Mode *n*	Analytical without damage [Hz]	FEM without damage f_u [Hz]	Error ε [%]	FEM with damage f_d [Hz]	Relative change *f** [%]
1	4.466	4.5076	-0.927	4.3908	2.591
2	27.989	28.158	-0.603	28.073	0.302
3	78.370	78.439	-0.087	76.537	2.425
4	153.575	152.578	0.649	150.09	1.631
5	253.871	249.857	1.581	249.85	0.003
6	379.239	369.034	2.691	363.27	1.562
7	529.681	508.711	3.959	495.98	2.503
8	705.197	667.375	5.363	662.63	0.711
9	905.787	843.447	6.882	841.71	0.206
10	1131.45	1035.34	8.494	1013.9	2.071

In table 4 is analyzed a comparison of the natural frequencies obtained analytically via the relation proposed by the authors, those have been achieved by numerical simulation. It was noted that the ε errors are negligible at low mode and increase with the mode of vibration, the fact which suggests that the model is the model of Shear (Figure 9).

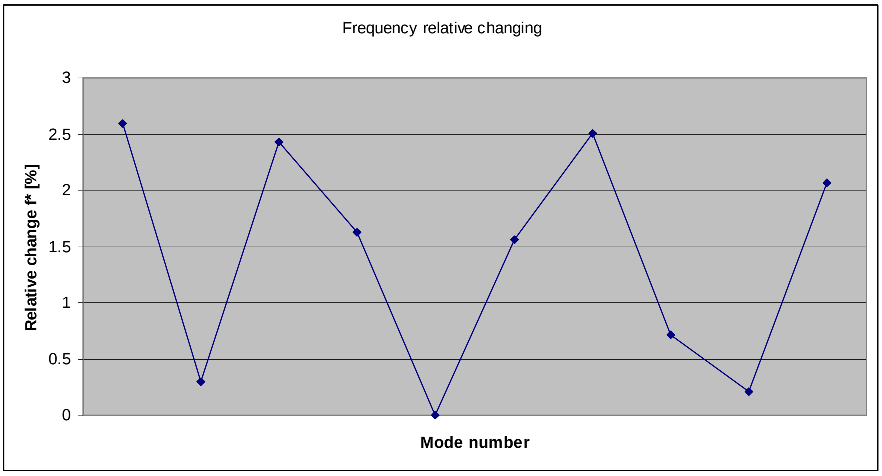

Figure 8. Relative frequency changes

Also, Table 4 and Figure 8 show the variation of natural frequency, because of the defect (relative variation in natural frequency is actually

$f^* = \frac{f_u - f_d}{f_u} \cdot 100[\%]$). Here can be observed significant changes in frequencies, certain ones identifiable and usable for fault location.

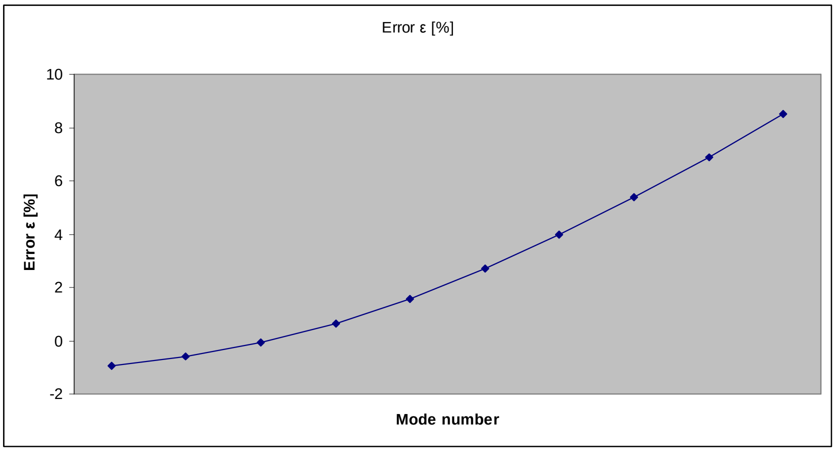

Figure 9. Error evolution as function the node number

4. Conclusion

Researches performed and presented in this paper by the authors, reveal that the damage location in sandwich beams and similar composite structures can be successfully accomplished via the relation contrived by the authors for isotropic materials. The difference appears in the case of damage evaluation, where the curves represent the damage severity vs. dimensionless damage depth. Characteristic evolution of the experimental relative frequency has a more complex allure, given by the mechanical and geometrical features of the beam's constitutive layers.

From another point of view, the relative frequency shifts for sandwich type beams with rigid PVC core, attending higher values as than the steel beams. This fact makes the vibration-based damage detection and location in composite beams more facile comparative to steel beams, but the estimation of damage severity more difficult.

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