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Monitoring the Effects of Impact Damages on Modal Parameters in Carbon Fiber Entangled Sandwich Beams

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Abstract: The aim is to study the impact toughness of two types of entangled sandwich materials (heavy and light) with the help of vibration testing. A simple case of symmetrical impacts is studied in this article as no literature is available regarding impact tests on entangled sandwich materials. The variation of modal parameters with two levels of damage (BVID and Damage not apparent on the surface) is studied. Vibration test results show that the light entangled specimens possessing good damping capabilities seem more sensitive to impact damage than the heavy ones. Furthermore, damping is found to be more sensitive to damage than the stiffness variations, so it is reasonable to assume that damping may be used instead of natural frequency as a damage indicator tool for structural health monitoring purposes.

Keywords: Entangled Sandwich Materials, Vibration Testing, Impact Testing, Modal Parameters

1. Introduction

Laminated honeycomb sandwich materials are being used widely in weight sensitive structures where high flexural rigidity is required, such as in the aerospace industry. By inserting a light weight core between the two face sheets, the bending stiffness and strength are substantially increased compared with a single layer homogeneous structure, without adding much weight. When the beam or plate undergoes flexural vibration, the damped core is constrained primarily to shear. This shearing causes energy to be dissipated and the flexural motion to be damped. However damage in these structures may negate many of the benefits of sandwich construction. Impact can induce various types of damage in the structure. The facesheets can be damaged through delamination and fibre breakage; the facesheet and core interface region can be debonded and the core can be damaged through crushing and shear failure mechanisms. Safe and functional effectiveness of stressed sandwich structures can often depend on the retention of integrity of each of the different materials used in its manufacture. Therefore lightweight sandwich materials used in next generation of more advanced aircraft, marine craft, road and rail vehicles must possess the capability to absorb higher impact energy and retain a high degree of structural integrity. For aeronautical structures, a field where this problem has been quite studied, the components have to undergo (i) low energy impacts caused by dropped tools, mishandling during assembly and maintenance, and in-service impacts by foreign objects such as stones or birds, and (ii) medium to high energy impacts in military aircrafts caused by weaponry projectiles. In a low energy impact (but high enough to produce damage), only a very small indentation will be seen on the impact surface. This level of damage is often referred to as barely visible impact damage (BVID). There has been considerable research on the impact performance and damage development in carbon fiber composite materials and sandwich composite materials; see for example references [1-5].

The vibration characteristics of sandwich materials have drawn much attention recently in order to attenuate vibrations and to cushion impact force for structural components and mechanical parts. The dynamic parameters of a structure i.e. natural frequency, damping and mode shapes are determined with the help of vibration testing which provides the basis for rapid inexpensive dynamic characterization of composite structures [6]. D.J Ewins gave a detailed overview of the theory behind vibration testing [7]. A wide amount of literature is present related to vibration testing of composite sandwich structures [8-12]. The equations that explain the dynamic behavior of sandwich beams are described extensively in the literature and notably in the references [13,14]. One way of damage detection is with the help of vibration testing, as the presence of delamination effect the vibration characteristics of a structure (e.g., natural frequency, damping ratio and mode shape).

In order to carryout effective structural health monitoring (SHM), it is therefore important to understand the performance of delaminated composites and debonded sandwich composites in a dynamic environment [15-19]. Comprehensive reviews on vibration-based damage detection methods have been presented by Zou et al. [20] on the model-dependent delamination identification methods for composite structures, and by Yan et al. [21], Doebling et al. [22] and Sohn et al. [23] on general vibration-based damage detection methods. Most of the traditional SHM methods are based on the fact that damage leads to appreciable reduction in the stiffness of a structural element (changes in natural frequencies) [24-26]. However, in structures made of composite materials there seems to be a tendency to use damping as a damage indicator tool, as it tends to be more sensitive to damage than the stiffness variations, mainly when delamination is concerned. When a delamination or debonding failure mode is concerned, friction between the interacting surfaces may occur for small bending deformations. As friction is an energy dissipation mechanism, it is reasonable to assume that damping may be used for SHM, when this type of damage is concerned [27-30].

Enhanced study on the vibration characteristics of composite structures has singled out the importance of material damping in the design process in recent years, as the control of noise and vibration in high precision, high performance structures and machines has become more of a concern. At the same time, composites sandwich researchers have focused more attention on damping as a design variable [31,32]. One way of increasing damping in sandwich materials is by putting a viscoelastic layer as core between the two laminates [33,34]. Jueng and Aref reported that sandwich structures with combined honeycomb-foam cores have higher damping than those with individual honeycomb or solid viscoelastic cores [35].

These advancements have led to the need for developing materials possessing better damping characteristics. Newer materials like fiber entangled materials can be used as potential dampers and sound absorbers in specific applications like the inner paneling of a helicopter, where structural strength is not the primary requirement. Entangled materials are made from natural materials (wool, cotton etc) as well as artificial ones (carbon, steel, glass etc) and are quickly becoming of widespread use as sound absorbers [36]. Bonded metal fibers entangled materials offer advantages for use as heat exchanger [37] or insulation [38]. These materials possess low relative density, high porosity and are cost-effective. Sandwich structures normally consist of two thin skins separated by a thick core. Traditional core materials are usually honeycomb, foam or balsa. Recently, a novel type of sandwich has been developed with bonded metallic fibers as core material [39-41]. This material presents attractive combination of properties like high specific stiffness, good damping capacity and energy absorption. Entangled materials with carbon fibers have also been studied for use as core material [42]. Entangled cross-linked carbon fibers present many advantages as core materials i.e., open porosity, multifunctional material or the possibility to reeve electric or control cables on core material. Mechanical testing has also been carried out on entangled specimens made of wood fibers [43], glass fibers [44] and various matted fibers [45]. Characterization of carbon and glass fiber entangled sandwich materials has been carried out both statically (compression and bending tests) and dynamically (vibration test) by L. Mezeix and A. Shahdin [46-48]. The compression and bending test results show that the entangled sandwich specimens have a relatively low compressive and shear modulus as compared to honeycomb and foam sandwich materials [47,48]. Vibration tests verify the presence of high damping in the entangled sandwich specimens making them suitable for specific applications like the inner paneling of a helicopter cabin, even if the structural strength of this material is on the lower side.

The main motivation of the work presented in this article, is to carry the research process on entangled sandwich materials one step further by carrying out impact tests on the carbon fiber entangled sandwich materials in order to understand the relation between damage level density and modal parameters. Monitoring the dynamic characteristics of these sandwich specimens allows us in future to study the dectability of impact and to verify that whether damping can be used as a damage indicator tool as it tends to be more sensitive to damage than the stiffness variations.

2. Materials and fabrication procedure

2.1. Core and skin materials

The core of the entangled sandwich test specimens used in this article consists of carbon fibers (HTS-5631) that are made of a yarn of standard carbon filaments having a diameter of 7µm. The length of the carbon fibers is 10 mm and their elastic modulus is 240 GPa. The fibers are provided by the company Toho-Tenax. For the cross-linking of carbon fibers, epoxy resin is used. The epoxy resin SR 8100 and injection hardener SD 8824 are used provided by SICOMIN. All the test specimens presented in the article are carefully weighed using METTLER balance. A better vaporization is achieved if the resin is heated up to 35°C before being sprayed on the carbon fibers. This allows the mixture of resin and hardener to become less viscous.

Resin-containing carbon-fiber/epoxy prepreg of T700/M21 is used to fabricate the skin materials [49]. The material is supplied by Hexcel composites, the physical properties are set out in Table 1. The upper and lower skins consist of four plies each with a stacking sequence of [0/90/90/0] [50]. The thickness of each ply is 0.125mm.

Table 1
Physical properties of carbon/epoxy prepreg T700/M21 used as skin material

Young’s modulus in fiber direction (E_1)	125000 MPa
Young’s modulus in transverse direction (E_1)	9000 MPa
Shear Modulus (G_{12})	5000 MPa
Poisson Ratio (v_{12})	0.4
Volume density (ρ)	1550 kg/m^3

2.2. Fabrication methodology

The fabrication of entangled sandwich specimens is often a tedious and complex process. As these types of materials are still mostly in the research phase, so as such standard fabrication processes do not exist. The fabrication procedure used in this article is the one developed by L. Mezeix et al. at CIRIMAT/LGMT in France [46]. For the test specimens used in this article, approximately 900 g of fibers of 10mm length are cut manually. The carbon fibers are then treated in a solution of dichloromethane for 24 hours and are then cleaned for 2 hours in methanol. These uncoated carbon fibers are then separated by a blow of compressed air. The mixture of resin and hardener is then sprayed on the separated carbon fibers by a spray paint gun. In case of larger test specimens, the volume of carbon fibers is large, so with the current technology it is not easy to spray the resin equally on the carbon fibers. The fibers vaporized by the resin are then placed in the mold between the two skins of unidirectional composites. In order to polymerize the fiber network sandwich specimens, the mold is then heated in an oven up to 180°C for two hours in a press [49]. The dimensions of the test specimens used for vibration and impact testing are 480 x 50 x 11 mm. The thickness of the entangled sandwich specimen core is 10 mm and that of each skin is 0.5 mm.

In order to verify the process of fabrication, a single test specimen is fabricated first by using a small mold (510 x 65 x 11 mm) with a fiber core density of 100 kg/m^3. This specimen showed that 100 kg/m^3 fiber core density is relatively insufficient for a volume of 480 x 50 x 10 mm^3, as there are places in the core of the sandwich beam that lack sufficient quantity of fibers. So for the next specimen, a fiber core density of 150 kg/m^3 is chosen. As compared to the previous specimen (fiber core density of 100 kg/m^3), the specimen with 150 kg/m^3 fiber core density has a far better fiber distribution. Finally a test specimen with a fiber core density of 200 kg/m^3 is fabricated by using the same small mold. It is seen that a fiber density of 200 kg/m^3 for the core is on the higher side and it is not possible to close the mold properly. So for the fabrication of the next batch of specimens, a fiber core density of 150 kg/m^3 is chosen.

In the next step a larger mold (510 x 250 x 11 mm) is used to produce multiple test specimens having the same characteristics. First the large mold is used to fabricate four relatively identical test specimens referred to as heavy specimens in the article. The average composition of these four heavy test specimens is presented in Table 2. Next, the same large mold is used to produce relatively lighter specimens. They have approximately 25 g less mixture of resin and hardener than the previously produced heavier specimens. These specimens shall be referred to as light specimens in future discussions (Table 2). The ratio of the mixture of resin and hardener in the heavy specimens is approximately 2.5 times more than that of the light specimens. The weight comparison of the vibration test specimens fabricated from the small and large mold is presented in Table 4 (in Section 4.2). The difference in weights observed between the specimens is due to the uneven distribution of the manually sprayed resin.

Table 2.

Composition of vibration test specimens having fiber core density of 150 kg/m^3 fabricated from the small and large mold (Average of four specimens)

Density	150 kg/m^3 (Light)	150 kg/m^3 (Heavy)
Weight of Fibers	39 g	39 g
Weight of Resin & Hardener	18 g	43 g
Weight of Skin (Upper + Lower)	38 g	38 g
Total Weight of Specimen	95 g	120 g

3. Experimental procedure

3.1. Vibration tests

The experimental equipment used to obtain the vibration test results discussed in this paper is shown in Fig. 1. The experimental set-up is that of a free-free beam excited at its center, based on Oberst method [51]. The Oberst method states that a free-free beam excited at its center has the same dynamical behavior as that of a half length cantilever beam. The test specimen is placed at its center on a B&K force sensor (type 8200) which is then assembled on a shaker supplied by Prodera having a maximum force of 100 N. A fixation system is used to place the test specimens on the force sensor. The fixation is glued to the test specimens with a HBM X60 rapid adhesive. The response displacements are measured with the help of a non-contact and high precision Laser Vibrometer OFV-505 provided by Polytec. The shaker, force sensor and the laser vibrometer are manipulated with the help of a data acquisition system supplied by LMS Test Lab.

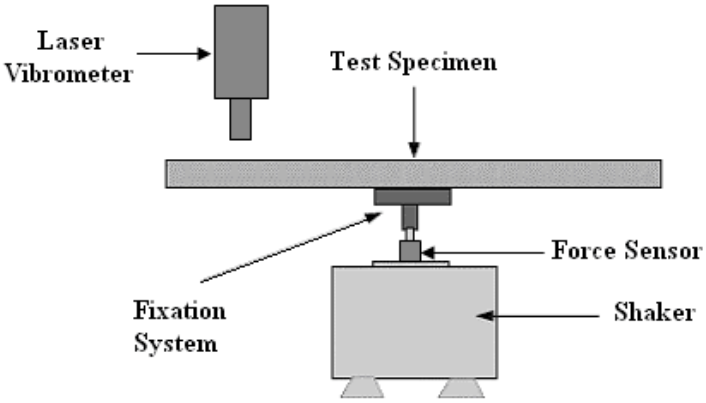

Fig. 1. Diagram of the experimental set-up

The center of the test specimens is excited at Point 17 as shown in Fig. 2. Burst random excitation is used which is a broadband type excitation signal (0-2650 Hz). The signal is averaged 10 times for each measurement point. The level of the excitation signal is chosen as 1 N. Hanning windows are used for both the output and the input signals. The linearity is checked and a high frequency resolution ($\Delta f = 0.25$Hz) for precise modal parameter estimation is used. Response is measured at 33 points that are symmetrically spaced in three rows along the length of the beam. The modal parameters are extracted by a frequency domain parameter estimation method (Polymax) integrated in the data acquisition system.

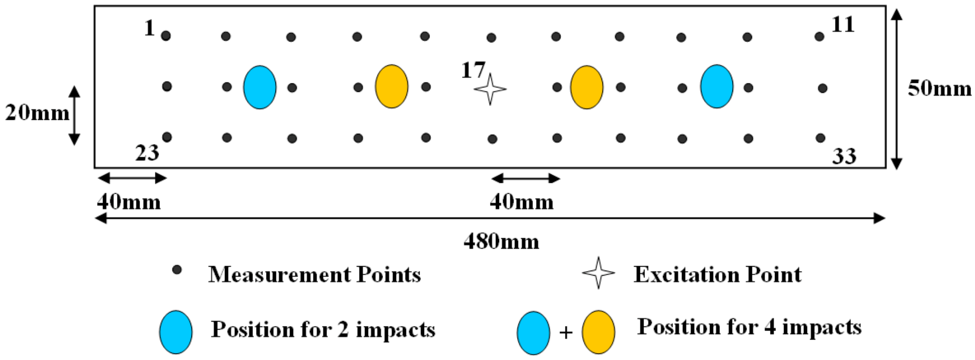

Fig.2. Sandwich test specimen with location of damage, excitation and measurement points

3.2. Impact tests

The impact test system used to damage the entangled sandwich beams is a drop weight system as shown in Fig. 3, and a detailed cut away of the drop assembly is shown in Fig. 4.

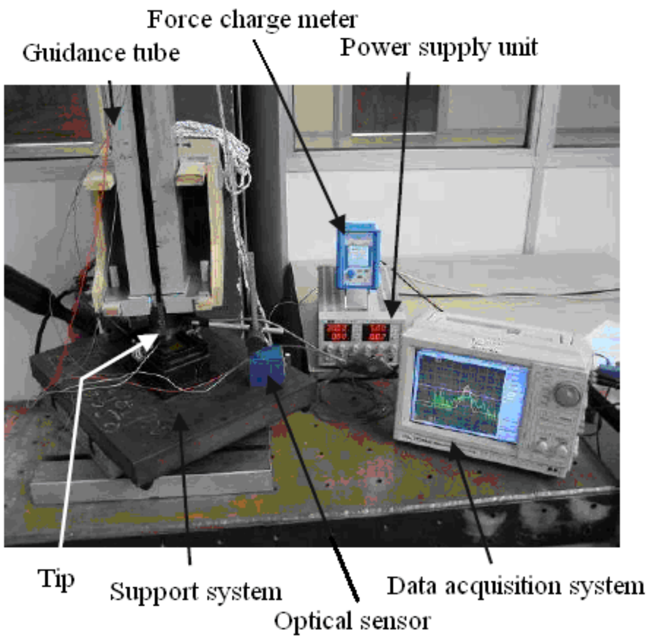

Fig. 3. Arrangement of the test equipment for the impact test

The impactor tip has a hemispherical head with a diameter of 12.7 mm. A force sensor (type 9051A) provided by Kistler is placed between the impactor tip and the free falling mass of 2 kg. The impact velocity is measured with the help of an optic sensor. The combined weight of the impact head, freefalling mass, force sensor and the accelerometer is 2.03 kg. In the calculation of impact height, a factor of 1.1 is used to compensate for the losses due to friction between the guidance tube and the drop assembly. The size of the impact window is 80 x 40 mm^2 which allows all the impact points to have the same boundary conditions and all the four ends are clamped. Further details on the impact test methodology of this drop tower can be found in the reference [3,17].

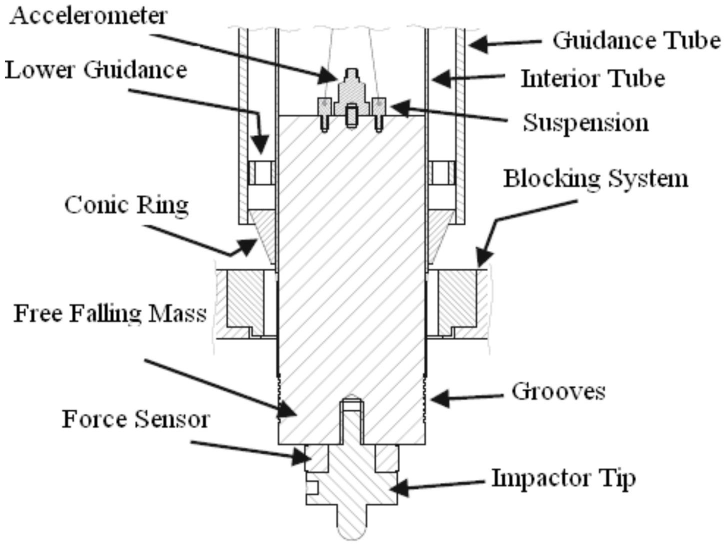

Fig. 4. Detailed cutaway of the drop assembly, the guidance tube and the blocking system

The entangled sandwich specimens tested in this article are impacted by taking into account the barely visible impact damage limit (BVID). BVID corresponds to the formation of an indentation on the surface of the structure that can be detected by detailed visual inspection and can lead to high damage. In the aeronautical domain, BVID corresponds to an indentation of 0.3 mm after relaxation, aging etc (according to Airbus certifications). In this study, it is decided to take 0.6-0.8 mm of penetration depth as detectability criterion just after the impact [3].

As no scientific literature can be found related to impact testing of entangled sandwich materials, so two test specimens of each of the four heavy and four light specimens are used for trial impact tests, in order to determine the BVID levels. The impact energies are chosen in such a way that the heavy and light specimens have the same level of damage. In general, two levels of damage are studied in this studied

- Barely visible impact damage (0.6-0.8 mm of indentation measured just after the impact)
- Damage not apparent on the surface (0.1-0.2 mm of identation measured just after the impact)

These trial impact tests revealed that in case of the heavy specimens, impact energy of 8 J corresponds to the BVID limit. However, in case of light specimens, they have to be impacted at 12 J in order to induce damage corresponding to the BVID limit. As discussed previously, that sometimes damage that is not visually apparent on the surface can prove quite detrimental to the load bearing capacities of sandwich structures. Therefore, during the trial impact tests on the heavy and light entangled sandwich specimens, an indentation depth of 0.1-0.2 mm is found to be undetectable through visual inspection. This indentation depth corresponds to impact energy of 6 J in case of heavy specimens and 8 J for the light ones. After the trial impact tests, two remaining specimens of each heavy and light specimens are used for the real impact and vibration tests. Each specimen is impacted at four different points (Fig. 2), but the impact energy level is kept the same for each of the specimens. The two heavy specimens are impacted at 6 J (0.1-0.2 mm indentation depth) and 8 J (BVID limit), whereas the two light specimens are impacted at 8 J (0.1-0.2 mm indentation depth) and 12 J (BVID limit). The impact parameters for the two heavy and two light entangled sandwich specimens studied in this article are listed in Table 3.

Table 3

Impact test parameters

Beam Name	Energy of Impact (J)	Height (mm)	Indentation just after impact (mm)	Velocity of impact Measured (m/s)
Entangled Heavy 1 (EH 1)	6	331.8	0.1 - 0.2	2.49
Entangled Heavy 2 (EH 2)	8 (BVID)	442.3	0.6 - 0.8	2.83
Entangled Light 1 (EL 1)	8	442.3	0.1 - 0.2	2.83
Entangled Light 2 (EL 2)	12 (BVID)	663.5	0.6 - 0.8	3.52

The data obtained during the drop weigh impact tests is shown in Fig. 5.

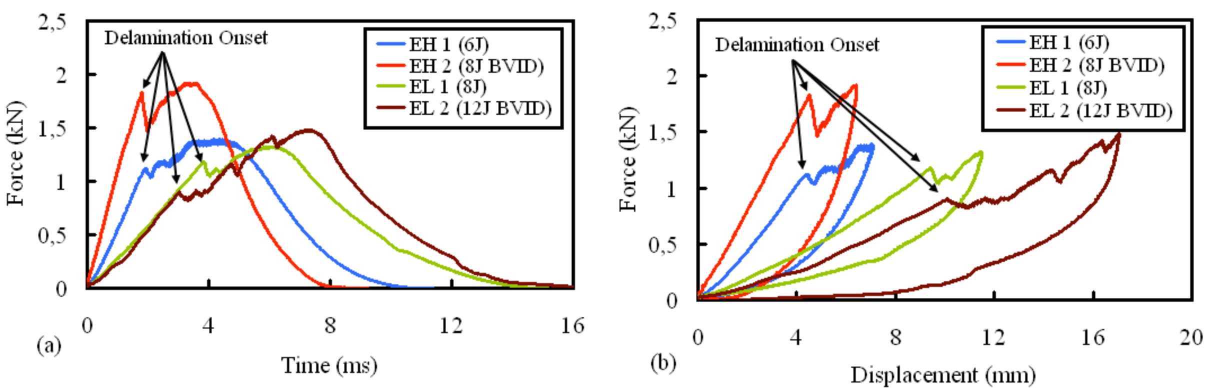

Fig. 5. Impact test data (a) force-time (b) force-displacement

Four similar impacts have been performed on each specimen. However, in order to clarify these plots, only one impact test result for each specimen is plotted. All the impact curves presented in Fig. 5 are filtered at 15 kHz to avoid a free frequency of the impactor at about 20 kHz. These curves, representative of all performed impact tests, are very classic in the literature [4,5]. In Fig. 5 a, the impact forces are drawn as a function of time during impact tests. These curves are globally smooth and almost sinusoidal at low impact energy, with little oscillation due to natural frequencies of the panel. They show an important force signal fall followed by oscillations which is characteristic of delamination onset. This phenomenon is more prominent for higher impact energies. The force-displacement plot (Fig. 5 b) shows the same force signal peak as soon as the delamination begins. These curves also give us an indication about the static strength of the heavy and light entangled sandwich specimens. They underline the facts that as the light specimens are less dense having less resin as compared to the heavy ones, so in order to attain peak force or maximum energy, more time and displacement are required. The results of the static tests (compression and bending) carried out on the heavy and light specimens have been discussed in detail in the reference [46].

The entangled sandwich specimens have three states. First one is the undamaged state (UD), the second is the damage state due to two impacts (D1) and the third is the damage state due to four impacts (D2). As impact tests are carried out the first time on entangled sandwich materials, so a simple case with symmetric impacts is chosen. These impact points are shown in Fig. 2. Vibration tests are carried out on the four entangled sandwich specimens after each of these three states.

4. Results and discussion

4.1. Tracking of poles for damage detection

Modal parameter estimation is a special case of system identification where the a priori model of the system is known to be in the form of modal parameters. The identification process consists of estimating the modal parameters from frequency response function (FRF) measurements. Modal identification uses numerical techniques to separate the contributions of individual modes of vibration in measurements such as frequency response functions. Each term of the FRF matrix can be represented in terms of pole location and a mode shape. The FRF matrix model is represented mathematically by:

$$\left[H(\omega)\right]=\sum_{k=1}^{modes}\left\{\frac{\left[R(k)\right]}{(j\omega(k)-p(k))}+\frac{\left[R(k)^{*}\right]}{(j\omega(k)-p(k)^{*})}\right\} \qquad (1)$$

The numerator R(k) is the residue of the FRF and is a function of the product between mode shape components at all points. The denominator gives the modal frequency and modal damping (second term in Equation (1) is the complex conjugate term). The poles p(k), are the roots that satisfy this equation and are related to modal frequency and damping as follows:

$$p(k) = -\sigma(k) + j\omega(k) \tag{2}$$

The magnitude of each pole is the undamped natural frequency (ω_n). The undamped natural frequency (ω_n) and the modal damping (σ) are related to mass, stiffness and damping as follows: given by

$$\omega_n = \sqrt{\omega_d^2 + \sigma(k)^2} = \sqrt{\frac{K}{M}} \tag{3}$$

$$2\sigma(k) = \frac{C}{M} \tag{4}$$

The effect of physical properties on poles in the complex s-plane is illustrated in Fig. 6.

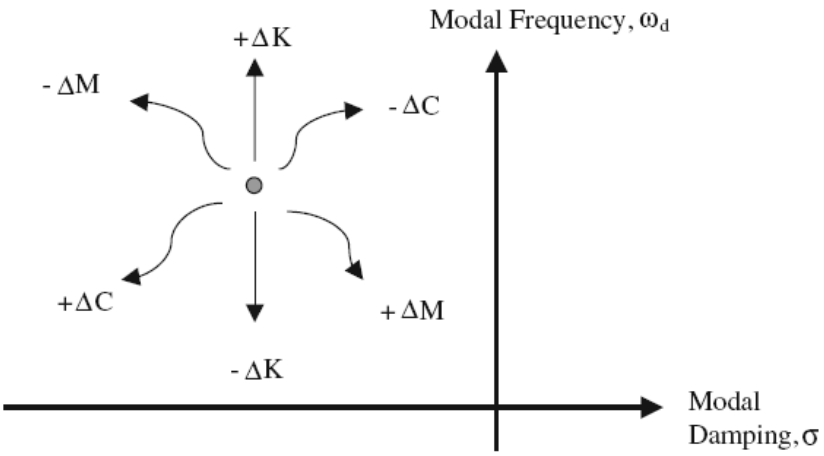

Fig. 6. Movement of pole due to mass stiffness and damping effect

From Fig. 6, it can be observed that a change in stiffness affects only the frequency, while changes in mass and structural damping affect both modal damped frequency (ω_d) and modal damping (σ). For this study, the primary interest is to study the decrease in the modal damped frequency (ω_d) and the increase in modal damping (σ) due to damage in the entangled specimens [52].

4.2. Monitoring through frequency and damping changes

Monitoring of the impact damage in the entangled sandwich specimens is carried out through frequency and damage changes. Frequency and damping results presented in this article are the global parameters of the specimen, and are extracted from the measurements carried out on the 33 measurement points. The frequency and damping changes are studied with the help of bending modes as they have the largest amplitudes for the type of test configuration presented in this article. For the first four bending modes, the variation of damped natural frequency as a function of the undamaged (UD) and the two damage states (D1 and D2) is presented in Fig. 7.

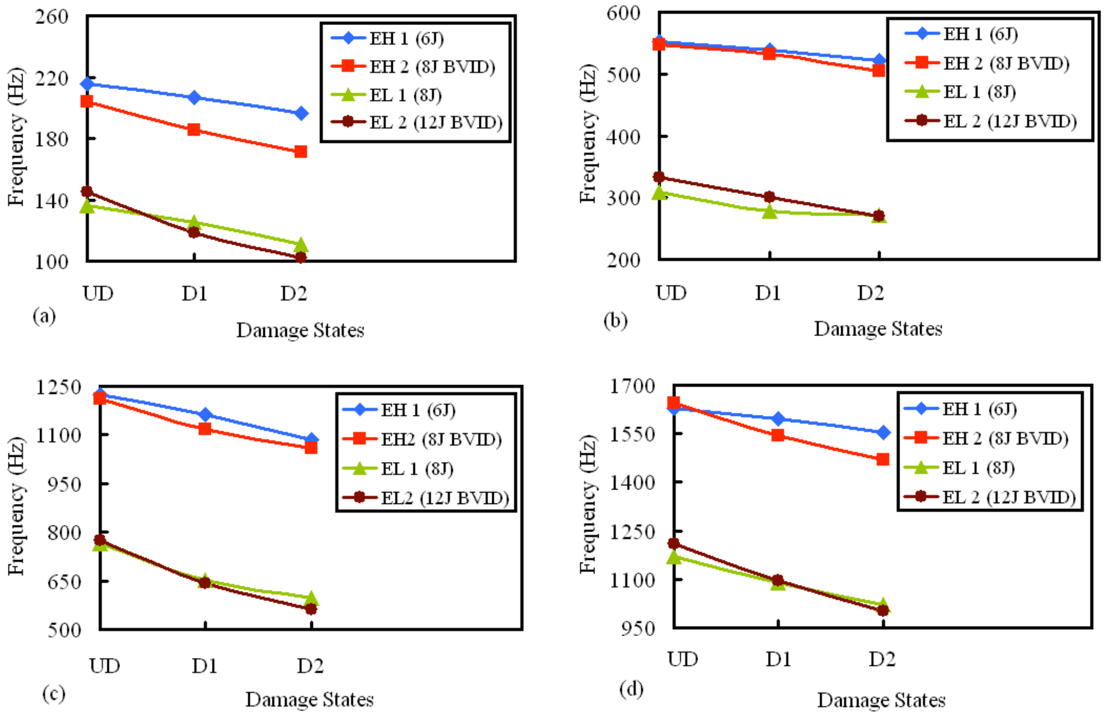

Fig. 7. Variation of damped natural frequencies with damage states for (a) 1st bending mode, (b) 2nd bending mode, (c) 3rd bending mode and (d) 4th bending mode: UD is undamaged state, D1 is damaged at 2 points of impact and D2 is damaged at 4 points of impacts, for the four entangled sandwich specimens

As discussed before in section 4.1, that damage in the specimens prompts a decrease in natural frequencies. So from Fig. 7, it is clear that the decrease in the natural frequencies for both the heavy and light specimens is more prominent in case of the higher impact energies i.e., 8 J in case of the heavy (EH 2) and 12 J in case of the light specimens (EL 2). But the interesting fact is that, for the heavy specimen (EH 1) impacted at 6 J which does not produce a visible damage on the surface, the average change in frequency for the first four bending modes between the undamaged and the damaged cases is 6 %. Similarly, for the light specimen (EL 1) impacted at 8 J this change in frequency ratio is 13 %. So it can be seen that the damage not visually apparent can affect the modal parameters resulting in a certain loss of rigidity. Therefore, vibration testing can be an effective tool to carry out non destructive tests for structural health monitoring purposes.

Fig. 7 also shows a dispersion in the natural frequencies between the two heavy (EH 1 and 2) and the two light specimens (EL 1 and EL 2) at the undamaged state. Each of the two heavy and light entangled sandwich specimens is fabricated from the same mold. However, this dispersion is a result of the differences in weight (Table 4) which as outlined previously, is due to the uneven distribution of the manually sprayed resin which highlights the complexity of the fabrication process.

The shift in the natural frequencies between the undamaged and the damaged cases is more prominent at higher frequencies. This is evident in Fig. 8, which shows a comparison of the sum of the frequency response functions (FRF) for the entangled sandwich specimen EH 2 (impacted at 8 J) for the undamaged case (UD), damaged at 2 points (D1) and damaged at 4 points (D2). The sum of the FRF can be compared as for each entangled sandwich beam 33 symmetric measurement points have been chosen and the four impact points are also symmetric on both sides of the two major axes of symmetry.

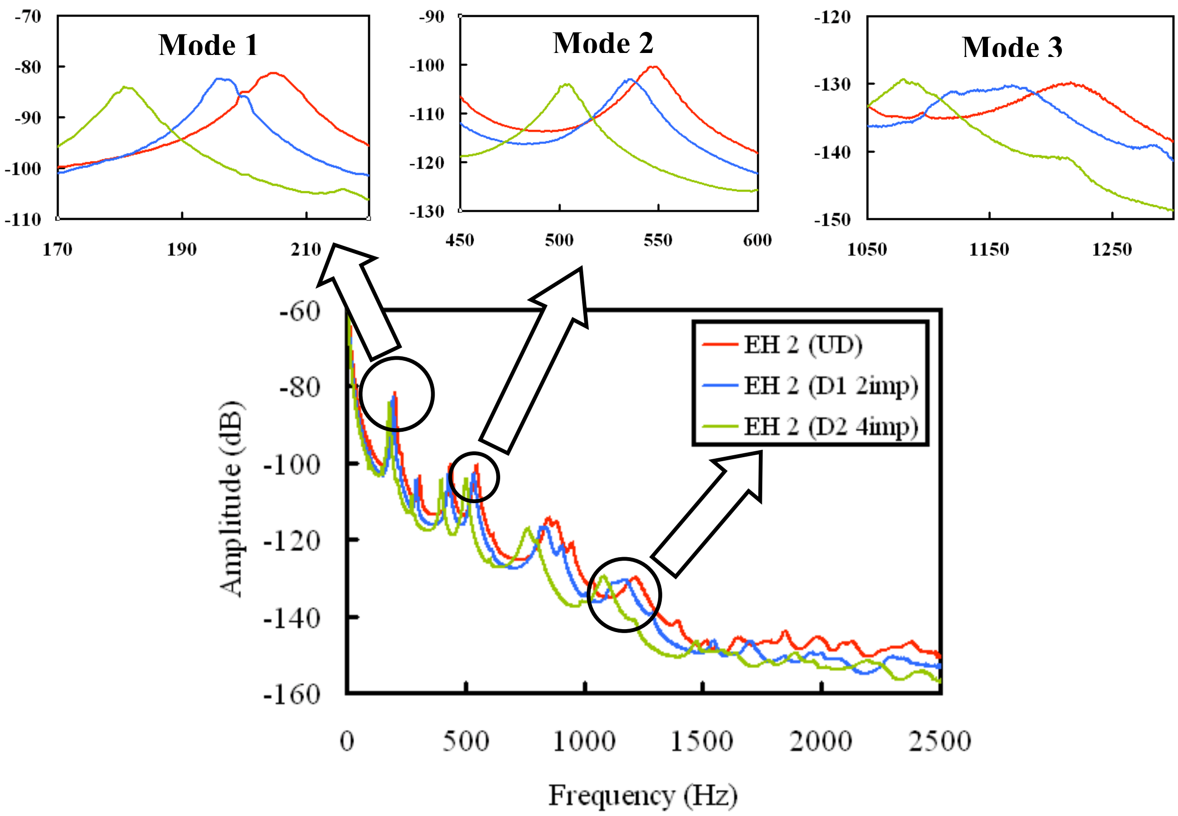

Fig. 8 Comparison of the sum of the frequency response functions for EH 2 for the undamaged case (UD), damaged at 2 points (D1) and damaged at 4 points (D2)

The damping ratios estimated by Polymax algorithm for the two heavy and two light entangled sandwich specimens for the first four bending modes are shown in Fig. 9.

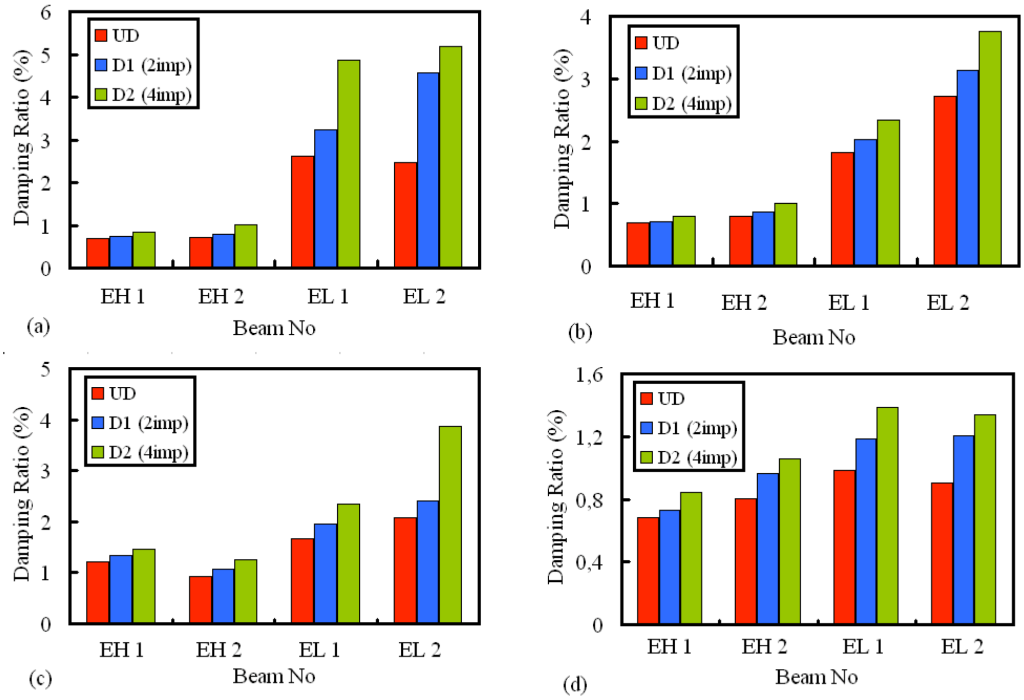

Fig. 9. Variation of damping ratios for the three damage states for (a) 1st bending mode, (b) 2nd bending mode, (c) 3rd bending mode and (b) 4th bending mode: UD is undamaged state, D1 is damaged at 2 points of impact and D2 is damaged at 4 points of impacts, for the four entangled sandwich specimens

Fig. 9 also shows a similar dispersion at the undamaged state between the two heavy and the two light specimens in case of the damping ratios. However as discussed in section 4.1, the damping increases with the increase in damage in the entangled sandwich specimens. Damping ratios are considerably higher in case of the light specimens, as they are more dissipative in nature due to lesser amount of resin. It can be noticed that with the exception of the 4th bending mode (Fig. 8d), the change in damping ratio between the undamaged and the damaged states for the two heavy specimens (EH 1 and EH 2) is smaller as compared to the two light specimens (EL 1 and EL 2), which shows that the light specimens are more sensitive to damage than the heavy ones. The affect of damage on the frequencies and damping ratios can be further elaborated by studying the frequency and the damping change ratios between the undamaged (UD) and the damaged cases (D1 and D2) for the two heavy and the two light entangled sandwich specimens, presented in Table 4.

Table 4 Frequency and damping change ratios between the undamaged (UD) and the two damaged states (D1 and D2) for the two heavy (EH1 and EH2) and the two light (EL1 and EL2) entangled sandwich specimens

Type of Specimen	Specimen Weight (g)	Between States	Frequency Change Ratios (%)				Damping Change Ratios (%)			
			1st Mode	2nd Mode	3rd Mode	4th Mode	1st Mode	2nd Mode	3rd Mode	4th Mode
EH1 (6J)	127	UD and D1	4.26	2.48	4.89	2.04	7.01	2.21	9.54	31.38
		UD and D2	8.83	5.61	11.33	4.73	23.98	15.95	19.74	47.30
EH2 (8J)	124	UD and D1	9.11	2.96	7.62	6.18	10.13	9.23	14.17	19.63
		UD and D2	16.11	7.98	12.45	10.73	40.58	27.69	33.06	30.88
EL1 (8J)	95	UD and D1	7.96	9.78	14.65	7.04	24.18	11.43	16.16	20.73
		UD and D2	18.73	12.07	21.76	12.82	86.15	29.29	39.45	40.56
EL2 (12J)	98	UD and D1	18.20	9.95	17.23	9.36	85.10	15.32	15.96	32.88
		UD and D2	29.70	18.85	27.35	17.11	109.6	38.66	86.92	47.49

The percentage change in frequency and damping ratios between the undamaged and the damaged cases is calculated with the help of Eq. 5 and Eq. 6

$$\text{Change in frequency between UD and D1, } (\Delta f) = \frac{f_{UD}(k) - f_{D1}(k)}{f_{UD}(k)} \tag{5}$$

$$\text{Change in damping between UD and D1, } (\Delta\zeta) = \frac{\zeta_{D1}(k) - \zeta_{UD}(k)}{\zeta_{UD}(k)} \tag{6}$$

where $f_{UD}(k)$ is the damped natural frequency for the undamaged specimen for the k_{th} mode and $f_{D1}(k)$ is the damped natural frequency for the specimen damaged at two impact points (D1) for the k_{th} mode. Nomenclature in case of Eq. 6 is the same. Furthermore, in order to calculate the frequency and damping change ratios between UD and D2 the same procedure is used.

For all the four specimens studied in this article, it can be seen from Table 4, that the damping change ratios are more prominent than the frequency change ratios. The maximum damping change ratio is 109.6 % and the maximum frequency change ratio is 29.7 % which occur in the case of EL 2 specimen impacted at 12J. It can concluded from the above results that damping seems more sensitive to damage than the natural frequency variations even in the case of entangled sandwich materials. So it is reasonable to assume that damping may be used instead of natural frequency as a damage indicator tool for structural health monitoring purposes. However, the fact that damping is a parameter that is relatively difficult to estimate as compared to natural frequency has to be taken into account.

Furthermore, if the frequency and damping change ratios are compared for the two heavy and light specimens (EH 2 and EL 2) impacted at the BVID limits i.e., having the same damage (0.6 – 0.8 mm indentation depth), it can be seen from Table 4 that the change in modal parameters is more significant in case of the light entangled specimen. This shows that even if the same level of damage (BVID) is imparted to the two specimens, the lighter specimens seem to be more affected. Similarly, by comparing the heavy (EH 1) and the light entangled specimen (EL 1) having the same lower level of damage that is not visible (0.1 – 0.2 mm indentation depth) in Table 4, it is evident that again the light specimen (EL 1) is seen more sensitive to damage than the heavy one. The only exception is the damping change ratio for the 4^{th} bending mode, which is higher in case of the heavy specimen.

So overall, it can be concluded that the light specimens having lesser amount of resin possess good damping capabilities as seen in Fig. 9, but are more sensitive to impact damage than the heavy ones, even when they have the same amount of damage. Therefore, while selecting the application of these light entangled sandwich materials, their sensitivity to impact damage should be taken into consideration. The vibration results also prove that the damage is more localized in the heavy specimens as they are denser in nature as compared to the light specimens, which results in smaller variations of modal parameters in case of heavy specimens. Furthermore, the damage in light specimens is less restricted to a certain zone and thus the light specimens are more globally affected than the heavy specimens.

5. Conclusion

Entangled sandwich materials possesses high damping characteristics and can be used for specific applications like the inner paneling of a helicopter cabin as their structural strength is on the lower side. So in this article, impact toughness of these entangled sandwich materials is studied. Due to the lack of available literature on the behavior of entangled sandwich materials due to impact damage, a simple case of symmetrical impacts is studied. The impact energies are chosen in such a way that the heavy and light specimens have the same level of damage. Vibration tests are carried out after each of the undamaged and damaged states to study the variation of modal parameters with damage. Two types of entangled sandwich specimens (heavy and light) are studied in this article. The light specimens have 2.5 times less resin than the heavy ones. Results show that with the accumulation of damage in the specimens, there is a decrease in natural frequency accompanied by an increase in the damping ratio. Vibration test results prove that the light specimens having better damping characteristics are more sensitive to impact damage than the heavy ones. Therefore, while selecting the application of these light entangled sandwich materials, their sensitivity to impact damage should be taken into consideration. In the heavy specimens, the damage seems to be more localized as compared to the light ones. Furthermore, it can concluded that damping seems more sensitive to damage than the stiffness variations. So it is reasonable to assume that damping may be used instead of natural frequency as a damage indicator tool for structural health monitoring purposes.

In this article impact toughness has been compared for two types of entangled sandwich specimens only. In the future, the impact toughness of these entangled sandwich materials shall be compared with classical sandwich materials, having honeycomb and foam as cores. The sensitivity of both the energy of impact and density of damage shall be established by making use of the design of experiments (DOE).

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Nomenclature

EH = Heavy Entangled Specimens
EL = Light Entangled Specimens
UD = undamaged state
D1 = damaged state at 2 points
D2 = damaged state at 4 points
FRF = Frequency Response Function
$H(\omega)$ = Frequency Response Function matrix
j = Imaginary axis in the complex plane
$*$ = Complex conjugate
$\omega(k)$ = Modal damped frequency for kth mode (rad/s)
$p(k)$ = Pole location for the kth mode
$R(k)$ = Residue magnitude (FRF/s)
$\sigma(k)$ = Modal damping for kth mode
ω_n = Undamped natural frequency (rad/s)
ω_d = Damped natural frequency (rad/s)
C = Structural damping matrix (force/velocity)
K = Stiffness matrix (force/displacement)
M = Mass matrix

f_k = Resonance frequency (Hz) for the kth mode

ζ_k = Damping ratio (%) for the kth mode

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