Evaluation of the Dynamic Properties of an Aluminum Syntactic Foam Core Sandwich

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ABSTRACT

The purpose of this study is to evaluate the dynamic mechanical properties of syntactic foam core sandwich structures. Aluminum matrix syntactic foams with carbon fabric skins are studied. Syntactic foams incorporate porosity in their foam-like structure by means of hollow particles. The material examined here is composed of a core made of aluminum A356 matrix filled with alumina hollow particles. Two types of sandwich composites are studied containing: (a) a single layer of carbon fabric and (b) three layers of the same fabric in 0°/90° orientation. Dynamic

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characterization was conducted using a free vibration method to determine the resonant frequency of the sandwich beams. The dynamic modulus, storage modulus, loss modulus and damping ratio were calculated from the test results. The experimental results were used to validate the predictions of a theoretical model. The experimental values for Young's modulus of one- and three-layer fabric sandwiches were measured to be 32.57 ± 2.15 and 37.33 ± 1.12 GPa, respectively. The experimental results are in close agreement with the theoretical model.

Keywords: Metal matrix composite; metal matrix syntactic foam; hollow particles; vibration; dynamic property.

1. INTRODUCTION

Metal foams can be designed to exhibit lower density and higher specific properties including stiffness, energy absorption, and mechanical and acoustical damping capacities compared to fully dense material [1]. Metal matrix syntactic foams (MMSFs) are a specific class of metallic foams. They incorporate porosity in their foam-like structure by means of hollow particles dispersed in a matrix [2]. Syntactic foams are often used as core materials in sandwich composites for structural components in a wide range of applications. The high specific mechanical properties and the tailorability make them useful in weight sensitive applications [3]. Syntactic foams can provide benefits in many fields, for example in automobile and ship structures due to their light weight and high energy absorption ability under compression. A change in the filler type and content can provide the desired balance between the properties, performance and density of syntactic foams [4]. Syntactic foams show excellent properties under

compressive loading conditions because particles become load bearing elements under compression [5, 6]. Hollow particles of alumina (Al_2O_3), silicon carbide (SiC), boron nitride (BN), titanium oxide (TiO_2) and fly ash cenospheres have been used as reinforcements in syntactic foams [4]. Since marine and aerospace structures are continuously exposed to vibrations, the internal damping capabilities of these materials are extremely useful [7-10].

Metal matrix syntactic foams can be fabricated by different methods. The most common are the pressure or pressureless infiltration methods [11-13]. In these techniques the molten matrix infiltrates into a randomly packed preform of spheres followed by solidification under controlled conditions. Another fabrication method is the melt stirring method, where hollow particles are mixed into the matrix melt followed by solidification [11]. Aluminum matrix syntactic foams synthesized by these methods have been studied for a wide range of loading conditions and material parameters [14-16].

To enhance the mechanical properties of metallic foams, they have been sandwiched between two stiff sheets. The face-sheets (or skins) provide the material with increased tensile and flexural strength and stiffness while the foam core allows it to withstand large deformation and absorb energy [17]. Metal foams and foam core sandwich composites are promising protective materials in applications such as vehicle crumple zones and blast armors [18]. Although syntactic foams are also intended for use as core materials in sandwich structures, metal matrix syntactic foam core sandwich materials have only been studied recently because of challenges involved in their fabrication [19, 20]. Studies on compression and flexural response of aluminum matrix syntactic foam core sandwich composites are available [19-22]. These studies were conducted on sandwich specimens having one layer of fabric skin on each side.

Since MMSF core sandwich composites are only recently realized, their vibration response, to the best of authors' knowledge, has not been analyzed yet despite interest in such properties for their envisioned applications. Due to their ease of implementation, vibration-based methods are advantageous over wave propagation methods for dynamic characterization [23-26]. Free vibration of a cantilever beam was used in [23] to assess the dependence of dynamic mechanical properties of syntactic foams on hollow particle filler wall thickness and volume fraction. For functionally graded beams, the effects of constituent volume fractions, volume fraction gradient, and slenderness ratios on the natural frequency have been investigated using beam theories [27]. A theory for the vibration and stability of symmetric sandwich beams with elastic bonding has been developed by [28]. The main assumption was that the interfacial bond between the face layer and the core is thin and elastic and the bond shear tractions between the layers are proportional to the relative tangential displacement of the layers at the interface. Many other works based on the theoretical prediction of the free vibration response for sandwich composite materials are available in literature [29-31]. Availability of the natural frequency and dynamic properties of metal matrix syntactic foam core sandwich material can help in designing structures that can benefit from these properties. It is also noted in most previous studies on metal matrix syntactic foams that only the strength values are reported. The stiffness values are not reported mainly because the measured values are very low due to compliance of the test machine and other issues.

In the present study, the vibration test method is used to characterize the properties of syntactic foam core sandwich composites. The dynamic modulus, E ; the storage modulus, E' ; the loss modulus, E'' ; and the damping ratio, $\tan \delta$ are measured. A theoretical model is developed and validated with the experimental results. This model is based on Eshelby's inclusion theory

for the core and a sandwich rule of mixtures for the whole composite. The results obtained on storage modulus are important for such materials because the damping parameter is relatively small and these results can provide indication about the level of elastic modulus of such composites.

2. MATERIALS AND METHODS

2.1 Materials

The sandwich core is made of A356 aluminum alloy filled with alumina hollow spheres ($\text{Al}_2\text{O}_3\text{-HS}$). A356 alloy is attractive due to its low density, capability to be strengthened by precipitation hardening, high corrosion resistance, and high thermal and electrical conductivity [32]. The $\text{Al}_2\text{O}_3\text{-HS}$ are chosen because of high compatibility of alumina with the aluminum alloy matrix. A schematic representation of a syntactic foam core sandwich composite is presented in Figure 1. The average diameter and wall thickness of hollow particles are 3 ± 0.22 mm and 105 ± 7 μm , respectively. The true particle density of $\text{Al}_2\text{O}_3\text{-HS}$ is 8700 kg/m^3 and the nominal bulk density of the tapped bed is 5200 kg/m^3 . The carbon fabric used in the face-sheets (skins) of the sandwich composite has a simple weave pattern and a linear density of 0.193 kg/m^2 . The overall density of the beam, obtained by weighing the specimen and dividing by its volume, is $1650 \pm 10 \text{ kg/m}^3$.

The sandwich manufacturing process is schematically represented in Figure 2. To build the sandwich a $0^\circ/90^\circ$ carbon fabric layer is placed at the bottom of the mold cavity and covered with hollow ceramic shells. A second carbon fiber sheet is placed on top of the spheres and the mold is closed. The tapped bed of particles created in these composites is filled with the maximum possible volume of particles. Therefore, the particle volume fraction is estimated to be around 60% in the core [19, 20]. The mold is preheated to a temperature close to the melting point of the

metal matrix for a time that is sufficient to bring all contents to that temperature. Once the mold reaches an established temperature, it is filled with the matrix metal through feed sprues, while the gas escapes through the vent sprues. After the matrix solidifies, the sandwich panel is removed from the mold and cooled in a controlled oven. Panels with one and three layers of carbon fabric are produced by this method. In both sandwich panel architectures the carbon fiber layers are wet by the aluminum. In both cases, the carbon fabrics are placed in 0°/90° orientation. The one layer skin sandwich is the same as studied previously for compressive and flexural properties [19-22]. The three layer skin sandwich is a new material specifically synthesized for this study. Having these two different compositions can provide information about the roles played by the core and skin on the dynamic behavior of the sandwich. The dimensions of the vibration test specimens are 110×15×11 mm³ for the length, width and height, respectively. These specimens are machined from the cast slab using a waterjet cutter. The one- and three-layer fabric skin sandwich composites are called S1 and S3 composites, respectively, in the further discussion.

2.2 Microscopy

A Nikon D7000 DSLR camera equipped with an AF-S VR Micro-Nikkor 105 mm f/2.8G IF-ED macro lens is used to capture optical images of the specimens. An optical microscope Nikon Epiphot 200 fitted with a Nikon DS-Fil digital camera is used for optical microscopy.

A photograph of one- and three-layer skin sandwich composites is shown in Figure 3. A few Al₂O_{3-HS} appear to be broken during the composite manufacturing. However, most of the Al₂O_{3-HS} are intact as observed in this figure. Optical micrographs presented in Figure 4(a) show

a continuous interface between particles and matrix, which indicates that the particles are wetted by the matrix and there are no interfacial defects. In addition, the fabric tows are also found to be completely penetrated by the matrix as observed in Figure 4(b). Detailed microscopic observations on these sandwich composites are available in previous studies [19, 20].

2.3 Experimental setup

A specimen clamp is assembled by using two aluminum blocks with dimensions $101 \times 101 \times 19 \text{ mm}^3$. The bottom block is bolted to a Newport ST Series optical table with IQ Damping Technology and the top block is anchored to the bottom block. A torque of 15 N·m is applied to the screws holding the two blocks to minimize friction damping. The higher stiffness and considerably larger size of the aluminum blocks with respect to the test specimen minimizes structural perturbations. A LMI LSD Laser Distance Sensor 80/10 is used for amplitude measurement. This triangulation principle based laser does not need mirrors mounted on the specimen surface so there is no added mass for the beam. The laser system works with a position sensing detector (PSD) sensor installed inside the same equipment assembly. A position sensing detector is a semiconductor device, which gives electrical output based on the position of the light spot falling on the surface of the device [33]. The LDS 80/10 measurement range is 10 mm, the offset distance (OD) is 75 mm, the typical resolution is 0.001 mm, and the acquisition frequency is 3 kHz. The sensor is linear with an output of 0-10 Vdc with 1 V/mm. The laser requires a $\pm 15 \text{ V}$ DC power supply and to obtain the required voltage levels; a BK Precision 1672 Triple Output DC power supply is used. A schematic of the setup is shown in Figure 5. The signal cables of the impact hammer and laser are connected to the data acquisition board

(National Instruments DAQ). The board is connected to a computer and the signals are acquired by using LabView 2012 SP1 (32bit) software.

A Bruel & Kjaer Impact Hammer, Type 8204, 22mV/N, is used to excite the beam. It is connected to an Endeveco conditioner, model 4416B. The gathered signals are Fourier transformed by using a MATLAB R2013b routine to obtain the system transfer function, which is used to identify the storage modulus and the loss tangent.

Three specimens for each type of composite are tested. Each specimen is tested for three different free lengths in order to cover a wide frequency range. For each length three repetitions are performed.

2.4 Dynamic modal analysis

The scope of these experiments is to characterize the material by performing a dynamic modal analysis. The focus is on the Young's modulus and the damping parameter $\tan \delta$. The storage modulus E' and the loss modulus E'' are also investigated. The storage and loss moduli are related to the Young's modulus by

$$E = \sqrt{E'^2 + E''^2} \quad (1)$$

It is possible to express the storage modulus as a measure of the energy stored in the material during deformation. The loss modulus is a measure of the energy dissipated per loading cycle. The $\tan \delta$ is obtained by the ratio of loss and storage moduli and represents the lag between the response of the stress and the strain due to the applied load. The PSD measurement provides the beam deflection $y(\tau)$ by

$$y(\tau) = h(x_1, \tau) \quad (2)$$

where τ is the time. The deflection is given by

$$h(x, \tau) = \sum_{n=1}^N X_n(\tau) \varphi_n(x) \quad (3)$$

where $X_n(\tau)$ is the n^{th} modal coefficient and $\varphi_n(x)$ is the n^{th} mode function for the cantilever computed according to [34]

$$\varphi_n(x) = \sin\left(\frac{\lambda_n x}{e}\right) - \sinh\left(\frac{\lambda_n x}{e}\right) - \frac{\sin(\lambda_n) + \sinh(\lambda_n)}{\cos(\lambda_n) + \cosh(\lambda_n)} \left[\cos\left(\frac{\lambda_n x}{e}\right) - \cosh\left(\frac{\lambda_n x}{e}\right) \right] \quad (4)$$

where e is the beam free length and $\lambda_1 = 1.875, \lambda_2 = 4.694, \lambda_3 = 7.855$ are the first three solutions of the transcendental equation

$$\cos(\lambda e) \cosh(\lambda e) = -1 \quad (5)$$

The deflection of the beam is observed at the 1st iteration and the results are plotted in Figure 6. The Fast Fourier Transform of $X_n = a_1$ is then performed. The magnitude of the 1st modal coefficient allows computing the phase of the signal by acquiring the relative bin number. The resolution of the frequency response is 2.4 Hz.

The governing equation of motion for the specimen vibration in the Fourier domain, neglecting air damping, can be written according to [23]

$$E^*(\omega) \frac{I \partial^4 \tilde{h}(x, \omega)}{\partial x^4} - \rho A \omega^2 \tilde{h}(x, \omega) = P \psi(x - x_2) \quad (6)$$

where $E^*(\omega)$ is the complex Young's modulus, I is the cross-section moment of inertia with respect to the centroidal axis orthogonal to the vibration plane, ρ is the mass density, A is the area

of the cross section, ψ is the Dirac delta generalized function, and $P\psi(x-x_2)$ is the load applied by the impact hammer. Projecting Eq. 6 on the mode shapes set

$$\int_0^l E^*(\omega)\beta_n^4 X_n(x)X_n(x)\tilde{\phi}_n(\omega)dx - \int_0^l \rho A \omega^2 \tilde{\phi}_n(\omega)X_n(x)X_n(x)dx = \int_0^l P\psi(x-x_2)X_n(x)dx \quad (7)$$

Taking $X_n(x)X_n(x)=1$ and $\psi(x-x_2)X_n(x)=PX_n(x_2)$ provides

$$E^*(\omega)I\beta_n^4\tilde{\phi}_n(\omega) - \rho A \omega^2 \tilde{\phi}_n(\omega) = PX_n(x_2) \quad (8)$$

For moderately low values of structural damping, in the proximity of the m^{th} resonance frequency, the system can be modeled with a single degree of freedom and the Young's modulus can be assumed to be constant over the frequency range. The resonance frequency is given by

$$\omega_m^2 = \frac{E'(\omega_m)I\beta_n^4}{\rho A} \quad (9)$$

Since the Young's modulus can be assumed constant, especially for narrow frequency ranges, it can be computed by [35-37]

$$E = \left(\frac{M}{L}\right)\left(\frac{12}{bt^3}\right)(2\pi f_n)^2\left(\frac{l^2}{\beta_n^2 l^2}\right)^2 \quad (10)$$

where M is the total mass of the beam, L is the total length of the beam, b is the width of the beam, t is the thickness of the beam, f_n is the n-th mode of the natural frequency, l is the free length of the beam and $\beta_n l$ is the value corresponding to the n-th mode of the natural frequency.

During the study of a three-layered skin sandwich beam, [38] used the damping ratio ζ to approximate the loss factor. They consider $\tan(\delta) = \zeta$, where ζ is computed by

$$\tan(\delta) = \zeta = \frac{\log\left(\frac{x_1}{x_n}\right)}{2n\pi} \quad (11)$$

Since $\tan(\delta) = \frac{E''}{E'}$, the loss modulus E'' is

$$E'' = \tan(\delta)E' \quad (12)$$

For small damping ratios, the damping in the material can be incorporated into a complex modulus [36]:

$$E^* = E'(1 + 2i\zeta) \quad (13)$$

3. RESULTS AND DISCUSSION

3.1 Calibration and preliminary tests

In order to validate the adopted procedure and the acquisition system and the ad hoc developed MATLAB code, initially an Aluminum T6 6061 sample was tested. The specimen was a rectangular section beam 0.28 m long. The LDS sensor output shown in Figure 7 was used to obtain the deflection of the beam. The frequency response of the aluminum bar is shown in Figure 8; the natural frequency of the sample was 98.1 Hz. A 5% difference in Young's modulus was found between the expected and the experimental values.

The test was repeated on two additional materials. The first material was an austenitic steel bar, for which the difference between the expected and experimental values was found to be 7%. Further testing of the setup was conducted on a certified titanium bar with two different free lengths. The bar was cut to be equal to the used specimens (110×15×11 mm³) in order to reproduce the same experimental conditions. The nominal Young's modulus of this bar was

114 GPa. The average value obtained from four tests was 118.5 GPa. A 3.8% difference is considered acceptable. Following this validation, the aluminum matrix syntactic foam core sandwich composite was studied.

3.2 Experimental results on sandwich composites

Three specimens of S1 and three specimens S3 sandwich were tested for three different free lengths of 0.095 m, 0.1 m and 0.105 m, and each length was tested three times. In each single test the specimen was hit three times with the impact hammer. Figure 9 presents the impact hammer and position sensing detector outputs. The oscillation frequency of the specimen is shown in Figure 10. The resonant frequency, storage modulus and $\tan \delta$ were obtained through the MATLAB code. The mean and standard deviation values for the S1 and S3 specimens are provided in Table 1. The S3 sandwich exhibited 13% higher storage modulus than the S1 sandwich, which is attributed to the three layers of skin that provide increased stiffness to the S3 specimen. The increased stiffness is achieved at the expense of damping and the damping ratio is higher in the S1 sandwich by 11.5%. A longer specimen could help in reducing the noise by increasing the amplitude. As shown in literature the damping ratio exhibits both frequency and amplitude dependence [36]. Equation 1 presents the relation between Young's modulus and storage and loss moduli. In the strict sense the Young's modulus presented by this equation is a complex quantity. However, the ratio of storage to loss moduli is on the order of 10^3 for the stiff sandwich beams tested in this work. In this case, the loss modulus can be neglected to consider storage modulus to be the Young's modulus of the material.

The acquisition frequency of the LDS Laser (3 kHz) is limited when compared to the sample resonance frequency (720 – 930 Hz). This limitation could also affect the results with aliasing phenomena. Nevertheless the acquisition frequency is inside the range suggested by the

Shannon-Hartely theorem. Also the limited slenderness ratio of the beam (which varies from 8.6 to 9.5) is not well suited to the simple beam theory which usually refers to $L/t > 10$. However, the absence of relevant differences between the tests performed with the shortest and longest free lengths of the specimen, together with the calibration tests results, exclude any relevant aliasing related issues.

3.3 Theoretical validation

The Young's modulus of the syntactic foam core has been computed by Eshelby's inclusion theory [39]. The data presented in Table 2 were used for the computation. The obtained output value for the core Young's modulus was 27.8 GPa. In order to evaluate the effect of the external layer, the rule of mixtures for a sandwich material from [40] was used.

Considering the schematic in Figure 11 and introducing the thickness ratio parameter $\nu = \frac{h_c}{h}$ it is possible to use a rule of mixtures to compute the tensile modulus of the sandwich material, given by [40]

$$E_t = E_c \nu + E_s (1 - \nu) \quad (14)$$

Where E_c is the core Young's modulus and E_s is the skin Young's modulus. Equation 13 shows that the skin and core contribute to the tensile stiffness in proportion to their thicknesses. It is assumed that transverse strains arising from differences in Poisson ratio are negligible.

Similarly, by starting with the flexural rigidity

$$D = E_c I_c + 2E_s I_s \quad (15)$$

where I_c and I_s are the moments of inertia of the core and skin. In order to compute the value of E_s , a rule of mixtures with an efficiency factor (or Krenchel factor) needs to be applied [41]. In

fact, the sandwich skin is a composite material itself: the carbon fibers are wet in Aluminum A356 and the properties can be estimated by

$$E_s = \eta_\theta \eta_L E_f V_f + E_m V_m \quad (16)$$

where η_θ is the composite efficiency is factor, η_L is the fiber-length correction factor, E_f and V_f are the fiber's Young's modulus and volume fraction respectively, E_m and V_m are the same parameters for the matrix. For a $0^\circ/90^\circ$ orientation of the fibers the value for η_θ is 0.5 [41]. η_L is equal to 1 for continuous fibers. By using a value of 235 GPa for E_f , the result obtained from (16) is $E_s = 103.6$ GPa. Hence, the values listed in Table 3 are used in order to compute the Young's modulus of the MMSF sandwich composite. From Eq. 14 the theoretical values of Young's modulus are predicted to be 31.2 and 38.1 GPa, for S1 and S3 sandwich composites, respectively. The comparison between the experimental and the theoretical values shows a difference of 4.3 and 2% for S1 and S3 sandwich composites, respectively.

The same vibration response test, with a similar procedure, was performed on other materials by [36] and [42]. Aluminum 2024-T3, pure epoxy, and glass-filament reinforced epoxy composite beams were tested by [36]. Aluminum 6061-T4 and a graphite/aluminum metal matrix composite were tested by [42]. The graphite/aluminum composite was made of 6061 aluminum matrix reinforced with graphite fibers P55 or P100. The results for comparable frequencies are shown in Table 4. From the previous results can be noticed that the syntactic foam core sandwich composite has more than twice the damping ratio of the two types of aluminum, the glass reinforced epoxy and the graphite/aluminum composites. The damping capabilities together with the low density make aluminum matrix syntactic foam core sandwich material suitable for many different applications.

4. CONCLUSION

The present work was focused on studying syntactic foam core sandwich composites for vibration response. The sandwich core consisted of A356 alloy filled with alumina particles, and the external skins were made of one or three layers of 0°/90° carbon fabric sheets. The material is characterized by using the vibration response. The main conclusions can be summarized as follows:

- The sandwich has storage modulus of 32.57 ± 2.15 GPa for the single layer configuration and 37.33 ± 1.12 GPa for the three layer skin configuration. The difference between the two samples is 13%. This difference is due to the increased stiffness provided by the carbon fabric sheets. The outcomes of the theoretical computations showed a Young's modulus of 27.8 GPa and 103.6 GPa for the core and the external skin composite materials, respectively.
- The damping ratios were 0.0052 ± 0.0036 for the single layer carbon fabric sandwich and 0.0046 ± 0.0036 for the three layer sandwich. The difference is 11.5%.
- The experimental results obtained are in agreement with the theoretical computations for the composite sandwich material. The theoretical results have provided a value of 31.2 and 38.1 GPa for the single and the three layered sandwich Young's modulus. The differences with the experimental values are 4.2% and 2.0% respectively.
- As the loss modulus is three orders of magnitude smaller than the storage modulus, the storage modulus is taken as the estimate of the Young's modulus.

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Figure Captions

Figure 1. Schematic representation of the metal matrix syntactic foam core sandwich composite.

Figure 2. Scheme of the mold with all its components [20].

Figure 3. Sandwich composites with (a) one and (b) three layers of carbon fabric skins.

Figure 4. (a) Optical micrograph showing $\text{Al}_2\text{O}_3\text{-HS}$, (b) details of the single carbon fabric layer specimen [19].

Figure 5. Schematic of the experimental setup for vibration testing of the sandwich specimens.

Figure 6. First mode deflection for the single carbon fabric layer specimen. The curves show the position of the beam in 50 consecutive intervals of time.

Figure 7. LDS sensor output for the aluminum 0.28 m specimen. It is used to acquire the deflection of the beam.

Figure 8. Frequency response of the aluminum 0.28 m bar obtained by performing a FFT in MATLAB. The natural frequency of the specimen is 98.1 Hz.

Figure 9. Example of a single test on the S1 specimen 0.105 m long, with the laser spot at 0.102 m, hit at 0.09 m. (a) Signal from the impact hammer for three consecutive hits and (b) signal from the position sensing detector laser.

Figure 11. Fast Fourier Transform of the 1st modal coefficient for the S1 specimen.

Figure 12. Schematic of sandwich beam [40].

Tables

Table 1. Mean and standard deviation values of properties obtained for the metal matrix syntactic foam core sandwich composite.

Sandwich type	Nomenclature	E' (GPa)	$\tan(\delta)$	E'' (GPa)	E (GPa)
One layer skin	S1	32.57 $\pm$ 2.15	0.0052 $\pm$ 0.0036	0.169	32.57
Three layer skin	S3	37.33 $\pm$ 1.12	0.0046 $\pm$ 0.0036	0.171	37.33

Table 2. Data used for the Eshelby's model computation.

Material	Young's Modulus (GPa)	Density (kg/m³)	Poisson Ratio	Volume fraction
A356 alloy	71	2685	0.35	--
Al₂O₃-HS	250	3690	0.21	60%

Table 3. Data used to perform the computation of (16), to find the Young's modulus of the sandwich.

Sandwich type	Material	Young's Modulus (GPa)	Thickness (m)
S1	Core	27.8	0.0105
	Skin	103.6	0.00025
S3	Core	27.8	0.0095
	Skin	103.6	0.000750

Table 4. Performances comparison, in terms of Young's modulus and damping ratio, of other materials tested with the same vibrations method [36].

	Reference	Dynamic Modulus (GPa)	Damping ratio
Aluminum 2024-T3	[36]	74	0.0008
Pure Epoxy	[36]	2.9	0.0117
0° Reinforced Epoxy	[36]	35.6	0.0016
Aluminum 6061-T4	[42]	--	0.00196
P100/6061	[42]	--	0.00205

Figures

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Figure 11. Schematic of sandwich beam [40].