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Research Paper

Miscellaneous Modeling Approaches and Testing of a Satellite Honeycomb Sandwich Plate

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Abstract. The honeycomb sandwich structures are commonly and efficiently adopted in the development of light mass satellite structures as a result of their inherent high stiffness and strength properties. Through a comprehensive study, the equivalent finite element modeling of honeycomb sandwich structures utilizing miscellaneous modeling approaches is introduced. For the sake of validating results, both theoretical analysis and experimental modal testing are implemented upon a honeycomb sandwich plate utilizing free-free boundary conditions. Based on the results, the sandwich theory and its related shell-volume-shell approach introduce a good match with the experimental results. The aforementioned approach is utilized extensively during the process of satellite structural design and modeling. In addition, a parametric study is executed so as to relate the geometric and material variations to the resonant modal frequencies. The study results indicate a crucial influence of both honeycomb core and facing sheets thicknesses on the modal frequency values.

Keywords: Sandwich structure, Equivalent plate theory, Modal analysis, Satellite structure, Sandwich theory.

1. Introduction

The design of the satellite structure is a complicated process as it includes the trade-off between multiple conflicting design requirements. Minimum structural mass is considered as a main structural requirement that should be satisfied. Such minimization leads to increment in the payload portion and reduction in the launch cost. Thus, the utilization of honeycomb sandwich structures as the main constituent in the build-up of satellite structures is of crucial importance [1, 2, 3, 4]. In the past decades, many researchers investigated the modeling, testing, and applications of sandwich structures [5, 6, 7, 8]. Detailed modeling of such structures represents the main challenge due to the involved computational expense. Thus, developing an equivalent model either for the core or for the whole structure is regarded as the most efficient way for modeling such structures. A brief survey of the equivalent modeling of such structures follows next.

Xia et al. [9] proposed extensively three equivalent modeling methods, namely sandwich theory, equivalent plate theory, and honeycomb plate theory. They estimated the modal frequencies of a honeycomb sandwich plate based on the aforementioned theories. The obtained results indicated the reliability of such theories in the finite element modeling (FEM) of such structures.

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In 2010, Zheng et al. [10] presented thoroughly the equivalent modeling of a microsatellite honeycomb sandwich floor by utilizing both equivalent plate theory and sandwich theory. They compared the computational results with the analytical solution, and a good agreement was received.

Hao et al. [11] built the FEM of a spacecraft structure using the equivalent honeycomb plate theory and carried out both modal and harmonic response analyses. They consider their results as bases for the optimum structural design of the aforementioned spacecraft.

Boudjemai et al. [12] implemented and analyzed a detailed FEM for honeycomb panels based on the equivalent plate theory. Moreover, they carried out experimental modal testing for the sake of comparing and validating results. They concluded that such a theory can be used efficiently in modeling sandwich structures.

Jiang et al. [13] constructed an initial FEM of a honeycomb panel using sandwich theory. They calculated the equivalent mechanical properties of the honeycomb core analytically and carried out an experimental modal testing. Based on comparing both numerical and experimental results, they proved that such theory can predict accurately the honeycomb core equivalent elastic properties.

Fu et al. [14] carried out the dynamic analysis of a satellite solar array that is based upon the Aluminum honeycomb sandwich structure. They estimated the equivalent mechanical properties of honeycomb core via sandwich theory and compared both modal and frequency response analyses result with experimental testing results. A close matching between both numerical and experimental results was assessed.

Recently, Sun and Cheng [15] discussed the equivalent FEM of a honeycomb sandwich plate employing two approaches based on a well-developed sandwich theory. They validated the two equivalent numerical models through the implementation of an experimental modal testing. Finally, they performed a FEM updating to minimize the difference between numerical and experimental data and thus, enhancing the behavior of the numerical finite element model

It is evident from the aforementioned survey that utilizing equivalent modeling methods is inevitable in the FEM of honeycomb sandwich structures. In this article, Equivalent modeling theories namely sandwich theory and equivalent plate theory are thoroughly discussed. In order to reach a proper applicable theory for satellite structures, a comprehensive comparison is carried out between five miscellaneous modeling approaches using a modal analysis module via ANSYS workbench software [16]. Theoretical analysis and experimental modal testing of a honeycomb sandwich plate are performed in order to compare and validate results. Based on the results, the sandwich theory and its corresponding approach shell-volume-shell (SVS) is suggested to model the whole satellite structure. Finally, a parametric study is performed to relate the modal analysis results with several aspects including material and geometrical variations.

2. Modal Analysis and Testing of Honeycomb Sandwich Structure

FEM of a honeycomb sandwich structure is introduced in this section using both sandwich and equivalent plate theories. Moreover, an experimental modal testing is carried out and both computational and experimental results are compared. The structure under investigation is a square honeycomb sandwich plate of 400 mm side length and 10 mm thickness. Such a plate consists of a honeycomb core of 8 mm thickness and two Aluminum facing sheets AL2024-T3, each of thickness 1 mm, and. The utilized honeycomb core is a regular hexagon core ($l=h$, $\theta=30^o$) with the geometric properties and density listed in Table 1 and shown in Fig. 1. The core material (AL-5052) specifications are presented in Table 2.

Table 1. Geometric properties and density of utilized honeycomb core

Designation	Material	Wall length ($l=h$)	Wall thickness (t)	Density (ρ_{core})
1/8 – 5052 – 0.002	AL-5052	1.83 mm	0.05 mm	130 kg/m^3

Table 2. Core material (AL-5052) specifications

E_c	ρ_c	υ_c
70 GPa	2700 kg/m^3	0.33

where h represents the cell vertical wall length, l is the cell inclined wall length, ϑ is the honeycomb cell angle, t is the cell inclined wall thickness, and ρ_{core} is the honeycomb core density. In Table 2, E_c represents the core material modulus of elasticity, ρ_c is the core material density, and υ_c is the core material of the poison's ratio. The first four resonant modal frequencies and the related mode shapes are estimated under free-free boundary conditions.

2.1 FEM of Honeycomb Sandwich Plate Using Sandwich Theory

The sandwich theory represents accurately the inherent layered nature of sandwich structures. For such a theory, only the core is homogenized. The theory assumptions state that the core layer is capable of bearing out-of-plane extensional and shear loadings while maintaining a little in-plane stiffness. The upper and lower facing sheets carry the in-plane extensional and shear loads while complying with Love- Kirchhoff assumptions. Consequently, the heterogeneous honeycomb core can be tailored as a homogenized continuum with orthotropic properties. Hence, the sandwich theory requires the calculation of the equivalent homogenized properties of honeycomb core with double thickness walls. The calculation of such properties relies basically on the geometric properties of the core cell in addition to the mechanical properties of the original core material. The formulas employed for estimating such equivalent elastic properties are thoroughly discussed in Appendix A.1.

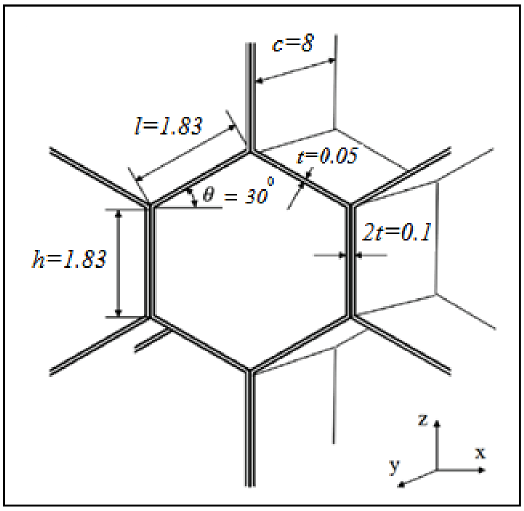

Fig. 1. Geometric properties of utilized honeycomb core

To apply such theory during FEM via ANSYS workbench software, different approaches are utilized for the sake of finding the most appropriate approach from the point of view of results accuracy and computational time. Three approaches are utilized as follows:

- **First approach**: Each honeycomb sandwich structure is composed of three distinct volumes, where the upper and lower volumes represent the facing sheets and the middle volume represents the honeycomb core. The three volumes are modeled using solid finite elements "solid 186" [17]. This approach is denoted by (VVV).
- **Second approach**: The upper and lower facing sheets are modeled using shell finite elements "shell 181", while the core volume is modeled via solid finite elements "solid 186" [18]. The solid shell elements are related together by multiple point contact to avoid mismatching of degrees of freedom. This approach is denoted by (SVS).
- **Third approach**: The honeycomb structure is modeled as a 2D sandwich plate shell using shell finite elements "shell 181" [19]. This approach is denoted by (SPS).

In this article, the modeling of the adhesive layer between facing sheets and honeycomb core is neglected due to its minor effect and for simplicity [20, 21]. The connection between facing sheets and honeycomb core is implemented using the multiple point connections (MPC) via bonded contact option in ANSYS workbench [13, 15].

The nine equivalent elastic parameters of honeycomb core given in Table 3 are estimated based upon Appendix A.1 formulas starting from eq.(3) up to eq. (16). The mechanical properties of the facing sheets are listed in Table 4.

Table 3. Equivalent material properties of the honeycomb core

E_x	3.3 MPa	E_y	2944.6 MPa
E_z	3.3 MPa	G_{xy}	426 MPa
G_{xz}	1.98 MPa	G_{zy}	651.7 MPa
υ_{xz}	1	$\upsilon_{xy} = \upsilon_{zy}$	0

Table 4. Material properties of the facing sheets (AL2024-T3)

E	ρ_f	υ
70 GPa	2800 kg/m^3	0.33

where E represents the facing sheets material modulus of elasticity, ρ_f is the facing sheets material density and υ is the facing sheets material poison's ratio. Meshing is carried out using a fine mesh with element size 8 mm, at which results converge and less element size keeps the same results. A sample of the mesh sensitivity analysis concerning the second and third modeling approach is shown in Fig. 2 and Fig. 3 respectively.

Table 5 lists the total number of nodes and elements used for meshing the honeycomb sandwich plate.

2.2 FEM of Honeycomb Sandwich Plate Using Equivalent Plate Theory

In such a theory, the three-layered honeycomb sandwich structure is compensated by an equivalent isotropic single-layered structure. This is obtained by equating the axial and bending stiffness of both the sandwich structure and the single isotropic structure [22]. Figure 4. depicts the discussed theory schematic drawing.

Table 5. Number of nodes and elements used to mesh the honeycomb plate

Modeling approach	Element size, mm	Number of elements	Number of nodes
VVV	8	7500	54009
SVS	8	7500	23205
SPS	8	2500	2601

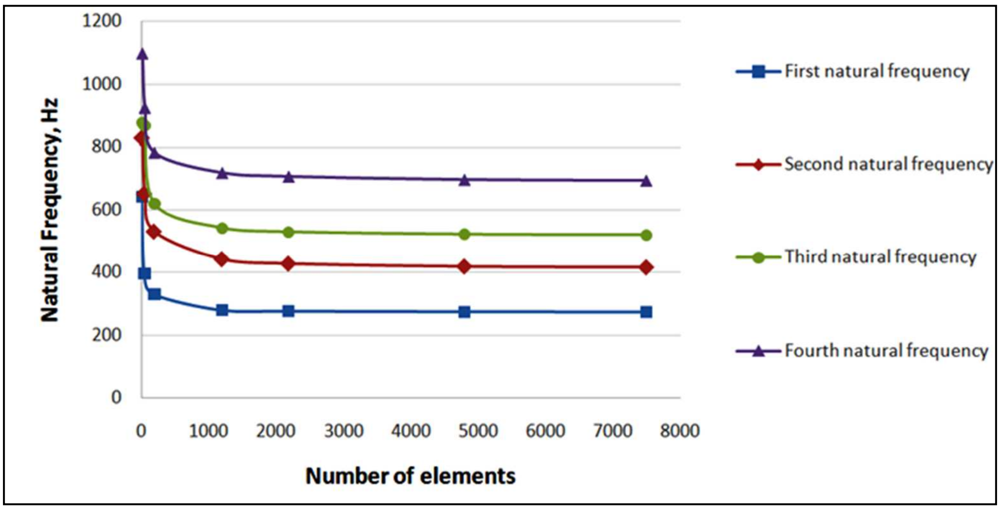

Fig. 2. Mesh sensitivity analysis of plate modal analysis (SVS)

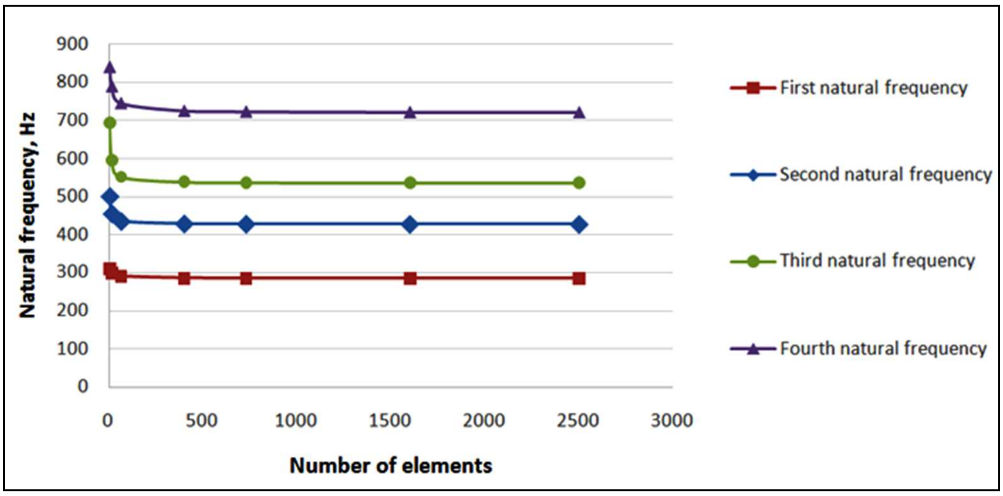

Fig. 3. Mesh sensitivity analysis of plate modal analysis (SPS)

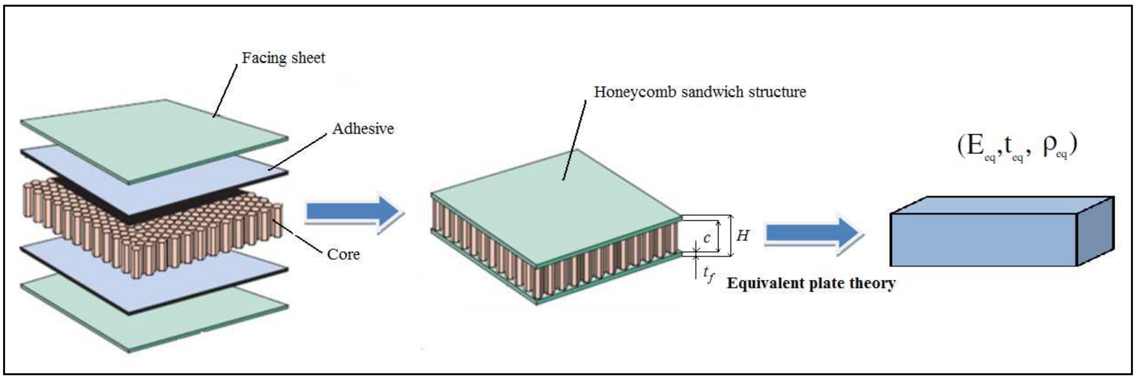

Fig. 4. Equivalent properties schematic drawing

Two different approaches are utilized as follows:

- **First approach**: The equivalent isotropic single-layered plate is modeled using solid finite elements "solid 186". This approach is denoted by (EPT_solid).
- **Second approach**: The equivalent isotropic single-layered plate is modeled using shell finite elements "shell 181" [12]. This approach is denoted by (EPT_shell).

The equivalent properties of the honeycomb sandwich plate are estimated based upon Appendix A.2 formulas according to eq. (17), eq. (18), and eq. (19) as shown in Table 6.

Table 6. Material properties of the facing sheets

$t_{eq.}$	$E_{eq.}$	$\rho_{eq.}$
15.62 mm	9.091 GPa	425.09 kg/m^3

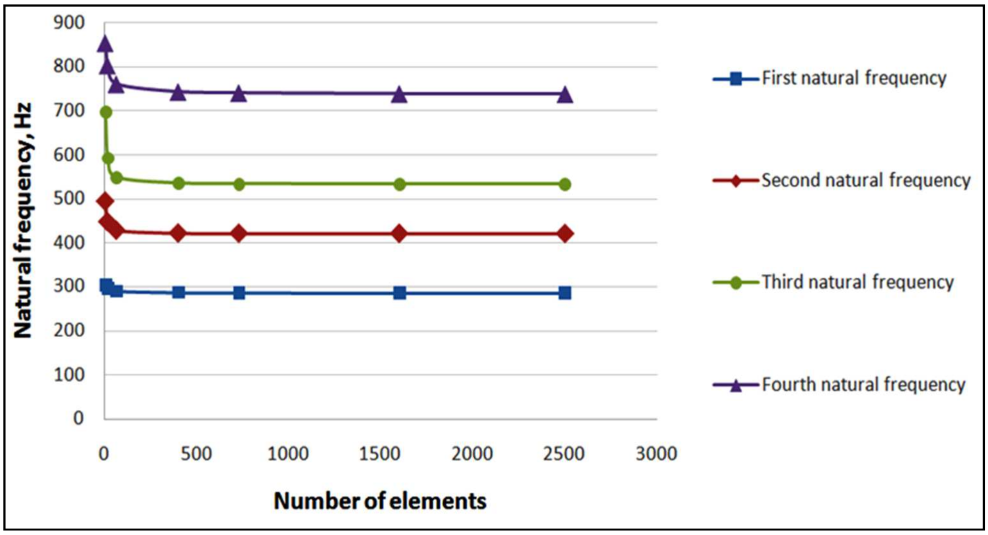

Fig. 5. Mesh sensitivity analysis of plate modal analysis (EPT_shell)

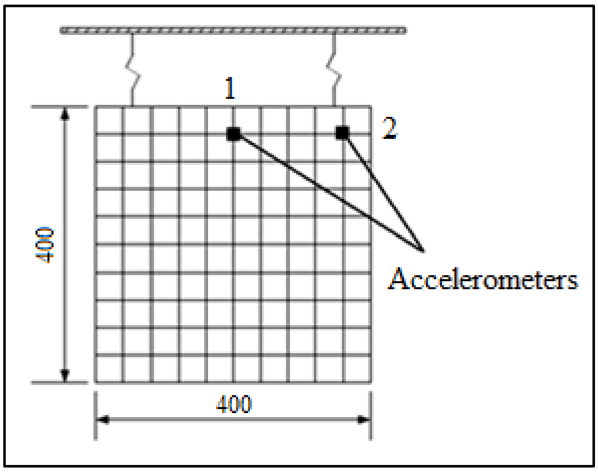

Fig. 6. Specimen fixation and accelerometers location

Table 7. Number of nodes and elements used to mesh the honeycomb plate

Modeling approach	Element size, mm	Number of elements	Number of nodes
EPT_solid	8	5000	28305
EPT_shell	8	2500	2601

In Table 6, $t_{eq.}$ is the equivalent plate thickness, $E_{eq.}$ is the equivalent plate modulus of elasticity, $\rho_{eq.}$ is the equivalent plate density. Meshing is carried out using a fine mesh with element size 8 mm, at which results converge. Figure 5. presents a sample of the mesh sensitivity analysis concerning the second modeling approach (EPT_shell). Table 7 lists the total number of nodes and elements used for meshing the honeycomb sandwich plate.

2.3 Experimental Modal Testing

An experimental modal testing, specifically frequency response function (FRF) testing, is carried out in the structure and vibration lab in the Military Technical College. A frequency response function is a mathematical model defining the relation between the input parameter (force in Newton's) and the output parameter (acceleration in g's). In this article, such testing is implemented for the sake of measuring the first four resonant natural frequencies of a square honeycomb sandwich plate of 400 mm side length and 10 mm thickness. The plate detailed dimensions and material properties are thoroughly discussed in Section 2.

2.3.1 Test Procedure

The Free-Free boundary conditions are simulated through hanging the plate under investigation via soft bands. Such boundary conditions are selected to avoid the presence of boundary errors. The soft bands are attached to a horizontal stand and wrapped around the plate at a distance of 5 cm from both edges.

The specimen is subjected to impacts from a hammer equipped with a soft tip. Such a hammer includes a force transducer with a sensitivity of 22 mV/N. The response is measured using two piezoelectric accelerometers of sensitivity 10 mV/g. The accelerometers are attached to the specimen under investigation via a special type of wax in the location shown in Fig. 6. Accelerometer no. (1) is located at the mid-distance of one side, while the second accelerometer is located near one edge. The location of accelerometers is selected such that they can capture accurately the first four mode shapes. Meanwhile, nine excitation points are determined at the plate center, four corners, and the four midpoints of each side. Both impact force and response acceleration are analyzed on a computer-aided Fast Fourier Transformation (FFT) data acquisition system capable of extracting the modal parameters with the help of LabVIEW signalExpress software. The whole experimental modal testing setup is shown in Fig. 7.

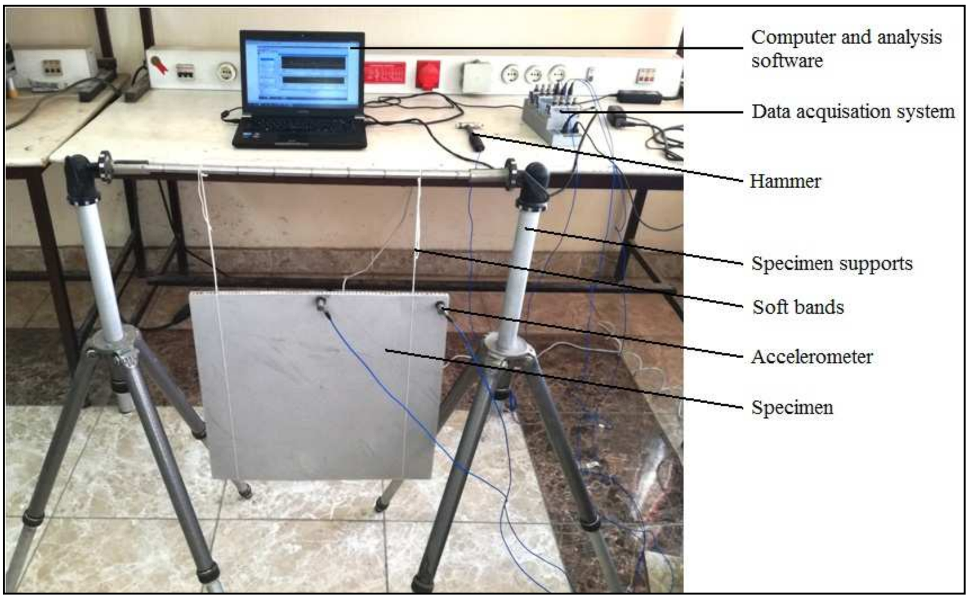

Fig. 7. Experimental modal testing setup

Table 8. Devices employed in experimental modal testing

Device	Description	Part Number	Photo
Data acquisition	NI cDAQ-9188	782323-01	
Impact hammer	PCB Modally Tuned	780991-01	
Accelerometer	PCB General Purpose, 10mV/g	780988-01	
Software	LabVIEW signalExpress		

Table 9. Experimental modal testing results (Hz)

Mode order	First mode	Second mode	Third mode	Fourth mode
Resonant frequency	259	391	494	664

The frequency span is adjusted as 0–700 Hz, and the frequency resolution is set at 0.5 Hz. The experiment is performed such that the plate is subjected to ten sequential impacts at each excitation point so as to decrease the variance of random noise and also to minimize the nonlinearities influence. The test is repeated for each excitation point to check the consistency of results. The FRF and its related resonant natural frequencies are recorded in such a manner to be compared with the FEM (numerical) results for validation purposes. The kit employed in the experimental modal testing belongs to "NATIONAL INSTRUMENTS (NI)", the utilized devices are shown in Table 8.

2.3.2 Test Results

Table 9 introduces the results of the experimental modal testing of the honeycomb sandwich plate represented in the first four resonant frequencies. The FRF of the implemented modal testing is presented in Fig. 8. The first accelerometer measurement is displayed with white color; such accelerometer captures modes with maximum deformation at plate corners. The second accelerometer measurement is shown with red color; it extracts modes with maximum deformation at sides' mid-distance. The first four resonant modal frequencies of the honeycomb sandwich plate are clearly indicated at the four peaks.

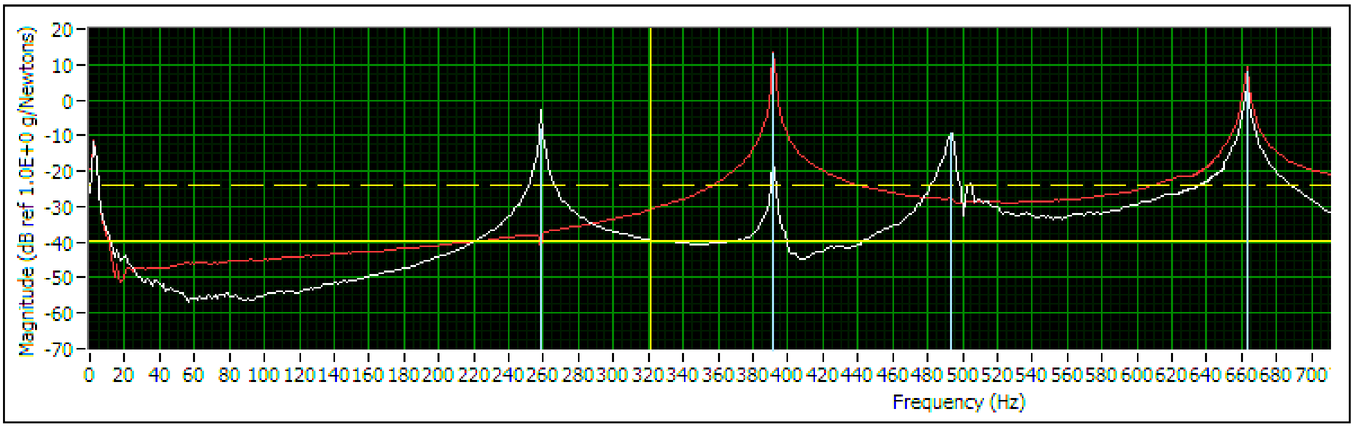

Fig. 8. Experimental modal testing results

Table 10. Frequency parameter for a square plate with free-free boundary conditions [23]

Mode order	First mode	Second mode	Third mode	Fourth mode
Frequency parameter, λ	13.419	19.559	24.212	34.646

Table 11. Experimental and computational natural frequencies comparison concerning sandwich theory approaches

Mode order	Mode Type	Experimental, Hz	Numerical (computational), Hz					
			VVV	Deviation, %	SVS	Deviation, %	SPS	Deviation, %
Mode 1	Torsion	259	271.91	4.98	272.66	5.27	283.77	9.56
Mode 2	Bending	391	411.67	5.286	415.65	6.3	426.99	9.2
Mode 3	Bending	494	516.5	4.55	519.88	5.238	535.33	8.37
Mode 4	Torsion	664	690.23	3.915	694.08	4.53	721.55	8.67
Mean deviation, %			-	4.69	-	5.34	-	8.95
Computational time, sec.			71	-	52	-	6	-

Table 12. Experimental and computational natural frequencies comparison concerning equivalent plate theory approaches

Mode order	Mode Type	Experimental, Hz	Numerical (computational), Hz			
			EPT_solid	Deviation, %	EPT_shell	Deviation, %
Mode 1	Torsion	259	284.68	9.92	285.12	10.08
Mode 2	Bending	391	420.85	7.63	421.01	7.68
Mode 3	Bending	494	532.7	7.83	533.06	7.91
Mode 4	Torsion	664	736.45	10.91	737.7	11.1
Mean deviation, %			-	9.09	-	9.15
Computational time, sec.			8	-	6	-

2.4 Theoretical Analysis

Theoretical analysis of the square plate with free-free boundary conditions is carried out based upon the discrete singular convolution (DSC) method [23] according to the following formula:

$$\lambda = \omega\, a^2 \sqrt{\frac{m}{D}} \tag{1}$$

where λ is the frequency parameter, ω is the circular frequency, a is the plate side length (400 mm), m is the plate mass per unit area, D is the plate flexural rigidity. Based upon the DSC method, the values of frequency parameter for the first four modes of a square plate with free-free boundary conditions are listed in Table 10 according to the published data [23].

2.5 Results Comparison and Discussion

The results of the numerical modal analysis concerning the five miscellaneous approaches compared with both experimental and theoretical results are listed in Table 11, Table 12, Table 13 and Table 14.

Table 13. Theoretical and computational natural frequencies comparison concerning sandwich theory approaches

Mode order	Mode Type	Theoretical, Hz	Numerical (computational), Hz					
			VVV	Deviation, %	SVS	Deviation, %	SPS	Deviation, %
Mode 1	Torsion	270.9	271.91	0.4	272.66	0.65	283.77	4.75
Mode 2	Bending	394.85	411.67	4.26	415.65	5.27	426.99	8.14
Mode 3	Bending	488.78	516.5	5.67	519.88	6.36	535.33	9.5
Mode 4	Torsion	699.4	690.23	1.3	694.08	0.76	721.55	3.167
Mean deviation, %			-	2.91	-	3.26	-	6.4

Table 14. Theoretical and computational natural frequencies comparison concerning equivalent plate theory approaches

Mode order	Mode Type	Theoretical, Hz	Numerical (computational), Hz			
			EPT_solid	Deviation, %	EPT_shell	Deviation, %
Mode 1	Torsion	270.9	284.68	5.09	285.12	5.25
Mode 2	Bending	394.85	420.85	6.58	421.01	6.63
Mode 3	Bending	488.78	532.7	8.99	533.06	9.06
Mode 4	Torsion	699.4	736.45	5.3	737.7	5.5
Mean deviation, %			-	6.49	-	6.61

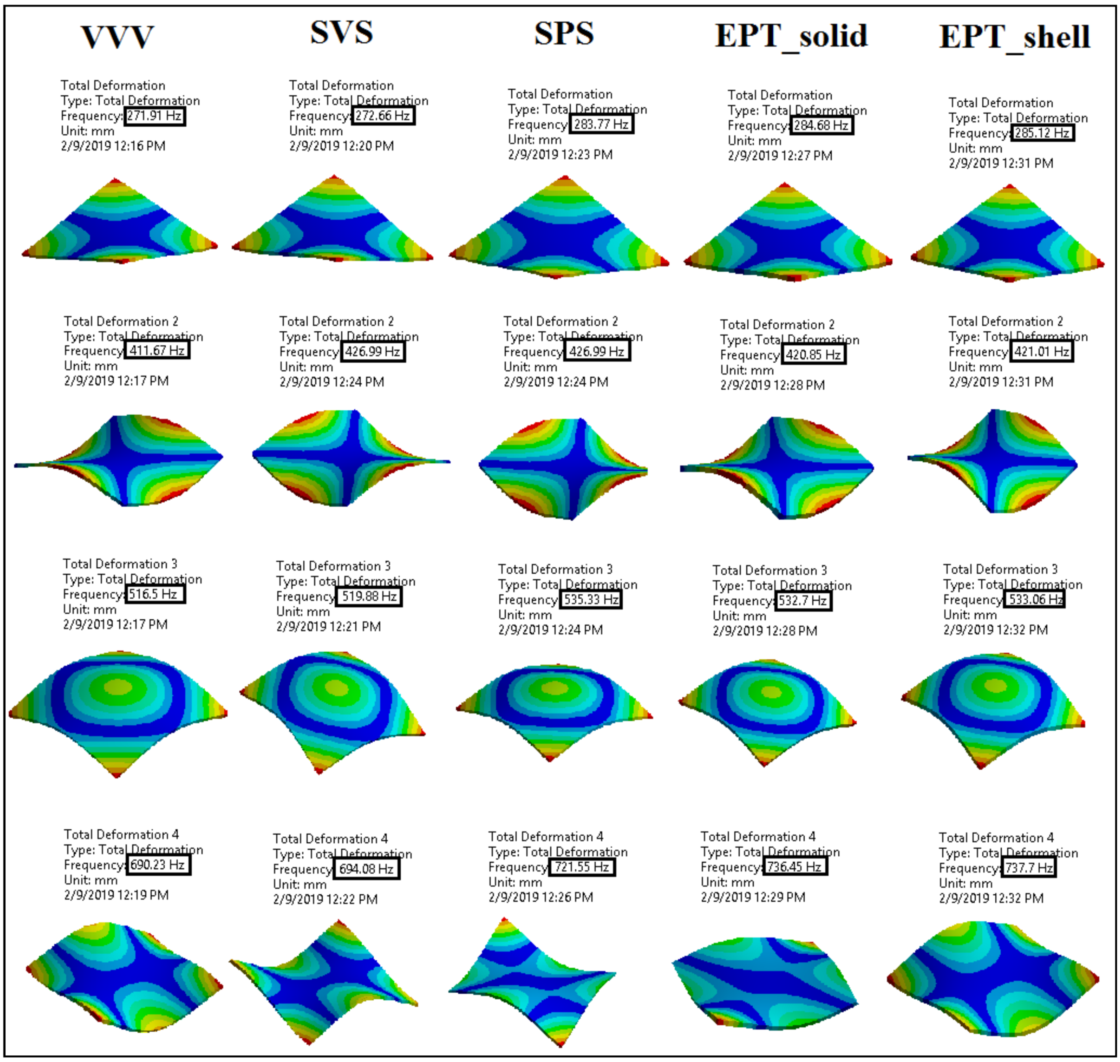

Fig. 9. Natural frequencies and related mode shapes of the honeycomb sandwich plate

The deviation in natural frequency for each mode is calculated as follows:

$$error = \frac{|f_{num} - f_{exp/th.}|}{f_{exp/th.}} \times 100\% \tag{2}$$

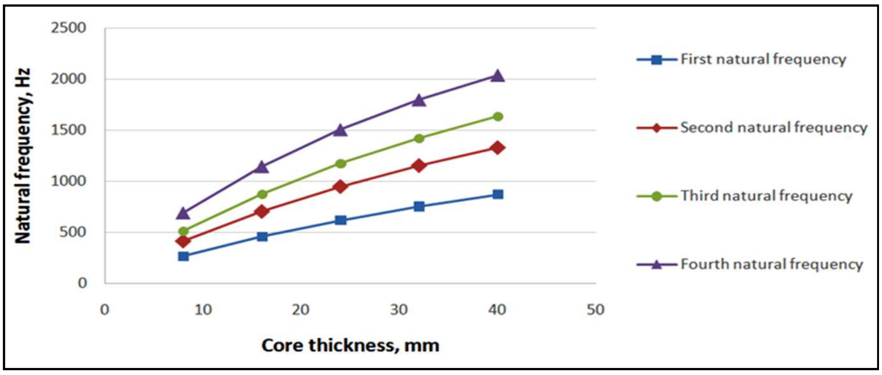

Fig. 10. Influence of core thickness on natural frequency

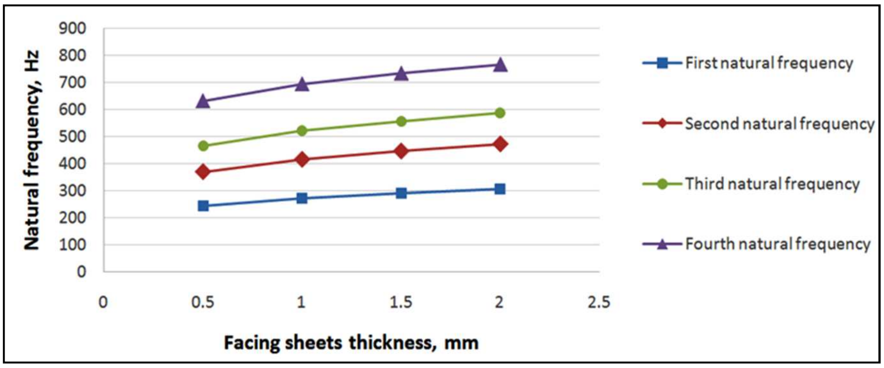

Fig. 11. Influence of facing sheets thickness on the natural frequency

Figure 9 shows the calculated natural frequencies and the related mode shapes of the plate under investigation concerning the aforementioned five modeling approaches. The computed values for resonant natural frequencies are highlighted for each modeling approach. In addition, the figure investigates the first four mode shapes for each modeling approach. The figure reveals that the five modeling approaches have similar mode shapes concerning each mode type. The first and fourth modes are purely torsion, while the second and third represent bending mode.

It can be observed that both theories and their corresponding approaches introduce an acceptable agreement with the experimental and theoretical results. Although the equivalent plate theory is simple and provides computational time saving, it does not account for the layered nature of honeycomb sandwich structures as it deals with this layered structure as an equivalent isotropic single layer. On the contrary, the sandwich theory represents clearly such layered nature of sandwich structures. Thus, it is capable to provide the stress distribution along with each layer, and consequently have clear failure criteria for each layer separately in case of static analysis implementation. Consequently, the sandwich theory is proposed to efficiently model honeycomb sandwich structures as it offers a closer agreement with both experimental and theoretical results.

A quick comparison between the miscellaneous approaches of sandwich theory reveals that the first approach (VVV) has the best agreement with the experimental results. Nevertheless, such a modeling approach exhibits a high computational expense which will form a huge obstacle while building the whole satellite finite element model. The second approach (SVS) exhibits a close agreement with the experimental and theoretical results and a reasonable computational time-saving. On the other hand, using 2-D finite elements in the third approach (SPS) results in a huge computational time saving but with a significant drop in results accuracy. Based on the aforementioned comparison, it is assessed that the sandwich theory and its related approach (SVS) can be utilized efficiently in honeycomb sandwich structures modeling.

3. HONEYCOMB SANDWICH PLATE PARAMETRIC STUDY

The goal of such a study is to relate the modal analysis results with several aspects including the honeycomb core thickness, facing sheets thickness, and facing sheets material, and thus, identifying the influence of such parameters on the plate natural frequency. The sandwich theory and its related approach (SVS) will be utilized during the implementation of this study.

4.1 Influence of Honeycomb Core Thickness

Modal analysis is carried out upon the aforementioned square honeycomb sandwich plate (side length=400 mm, each facing sheet thickness=1 mm) while permitting the honeycomb core thickness to vary from 8 mm up to 40 mm. the free-free boundary conditions are applied to the specimen under investigation. Figure 10. illustrates the influence of the honeycomb core thickness variation on the sandwich plate modal frequency.

It can be observed that increasing the core thickness leads to a significant increment in the modal frequencies regardless of the type of mode shape.

Table 15. Material properties of the selected alloys

Alloy Type	E, GPa	ρ_f, kg/m3	υ
AL2024-T3	70	2800	0.33
Ti-6AL-4V	110	4500	0.33
Steel 4340	200	7800	0.3

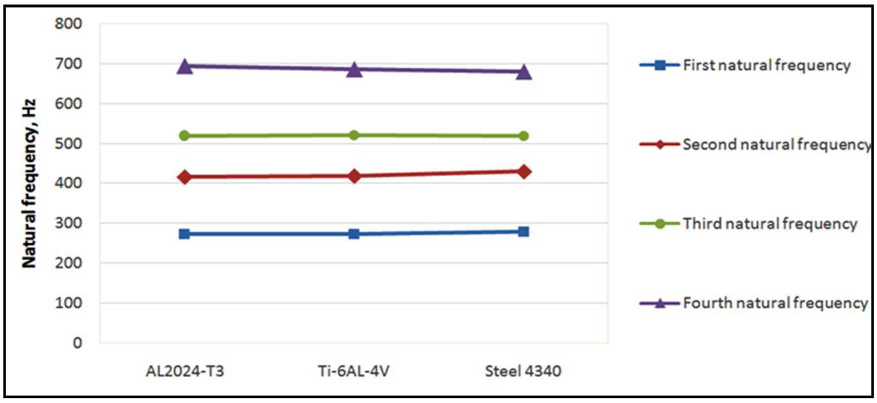

Fig. 12. Influence of facing sheets material on natural frequency

4.2 Influence of Facing Sheets Thickness

Modal analysis is implemented upon the specimen under investigation (side length=400 mm, core thickness=8 mm) while allowing the facing sheets thickness to vary from 0.5 mm up to 2 mm. The free-free boundary conditions are applied to the specimen under investigation. Figure 11. illustrates the influence of the facing sheets thickness variation on the sandwich plate modal frequency. It is obvious that the increment in facing sheets thickness results in a considerable increase in the modal frequencies regardless of the type of mode shape. Nevertheless, the core thickness variation is more effective on the values of the natural frequencies than the facing sheets thickness variation effect.

4.3 Influence of Facing Sheets Material

Modal analysis is performed on the specimen under investigation with its original dimensions (side length=400 mm, core thickness=8 mm, and each facing sheet thickness=1 mm) while utilizing two different isotropic alloys for the facing sheets in addition to the original material AL2024-T3. Table 15 lists the material properties of the selected alloys which are Aluminum, Titanium, and high Steel alloys.

Figure 12. depicts the influence of the facing sheets material variation on the sandwich plate natural frequency. It can be deduced that the facing sheets material variation has a slight influence on the modal frequencies value. This is due to the fact that altering the facing sheets materials means altering the modulus of elasticity and density values. Increasing the modulus of elasticity leads to increasing the modal frequencies value, on the contrary, increasing the density values results in lowering the modal frequencies value. Such contradiction results in an approximately stable modal frequencies value. This conclusion is valid only for metallic alloys with isotropic properties [12].

4. Conclusion

Equivalent finite element models of a honeycomb sandwich plate utilizing miscellaneous modeling approaches were thoroughly discussed based on well-developed sandwich theory and equivalent plate theory, regarding the simulation for modal analyses. Theoretical analysis and experimental modal testing were carried out on the aforementioned plate for the sake of comparing computational and experimental results. Finally, a parametric study was performed to investigate the influence of the sandwich plate geometric properties and material variations on the resonant modal frequencies. The following conclusions were evolved during this work:

- The sandwich theory and its related approach shell-volume-shell exhibit an acceptable agreement with both experimental and theoretical results with an averaged deviation of 5.34 % and 3.26% respectively. Simultaneously, they provide reasonable computational time-saving.
- The shell-volume-shell (SVS) approach will be utilized efficiently in the modeling of the whole satellite structure.
- Increasing the core thickness leads to a significant increment in the modal frequencies regardless of the type of mode shape.
- The increment in facing sheets thickness results in a considerable increase in the modal frequencies regardless of the type of mode shape.
- The facing sheets material variation has a slight influence on the modal frequencies value in case of employing metallic alloys with isotropic properties.
- The accomplished work can be considered as a reliable basis for the design process of honeycomb sandwich structures.

Author Contributions

Ali Aborehab initiated the project, suggested and participated in the experiments and analyzed the results; Mohammed Kassem participated in the experiments and theoretical analysis; Ahmed Farid Nemnem examined the theory validation; Mohamed Kamel planned the scheme. The manuscript was written through the contribution of all authors. All authors discussed the results, reviewed and approved the final version of the manuscript.

Conflict of Interest

The authors declared no potential conflicts of interest with respect to the research, authorship, and publication of this article.

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Nomenclature

h	Cell vertical wall length, [mm]	l	Cell inclined wall length, [mm]
ϑ	Honeycomb cell angle, °	t	Cell inclined wall thickness, [mm]
ρ_{core}	Honeycomb core density, [kg/m^3]	ρ_c	Core material density, [kg/m^3]
E_c	Core material modulus of elasticity, [MPa]	υ_c	Core material poison's ratio
ρ_f	Facing sheets material density, [kg/m^3]	$t_{eq.}$	Equivalent plate thickness, [mm]
$E_{eq.}$	Equivalent plate modulus of elasticity, [MPa]	$\rho_{eq.}$	Equivalent plate density, [kg/m^3]
f_{num}	Numerical natural frequency, [Hz]	f_{exp}	Experimental natural frequency, [Hz]
c	Honeycomb core layer thickness, [mm]	λ	Frequency parameter
ω	Circular frequency, [rad./sec.]	m	Plate mass per unit area, [kg/m^2]
a	Plate side length, [mm]	D	Plate flexural rigidity, [N.m]
H	Total sandwich structure thickness, [mm]	t_f	Facing sheet thickness. [mm]
E	Facing sheets modulus of elasticity, [MPa]	υ	Facing sheet poison's ratio

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Appendix A

Theoretical Approaches for Modeling of Sandwich Structures

A.1 Sandwich Theory

A.1.1 In-plane Equivalent Properties

The in-plane equivalent properties are obtained by applying uniaxial loading in (x-z) plane. Each cell wall is dealt with as a beam of length (l), thickness (t), and modulus of elasticity (E_c) [13]. Four elastic properties; two moduli of elasticity (E_x, E_z), shear modulus (G_{xz}), and Poisson's ratio (υ_{xz}) are calculated from static analysis as indicated in the published references [24, 25]:

$$E_x = \frac{\cos\theta}{\left(\frac{h}{l}+\sin\theta\right)\sin^2\theta}\left(\frac{t}{l}\right)^3 E_c \tag{3}$$

$$E_z = \frac{\left(\frac{h}{l}+\sin\theta\right)}{\cos^3\theta}\left(\frac{t}{l}\right)^3 E_c \tag{4}$$

$$G_{xz} = \frac{\left(\frac{h}{l}+\sin\theta\right)}{\left(\frac{h}{l}\right)^2\left(1+\frac{h}{4l}\right)\cos\theta}\left(\frac{t}{l}\right)^3 E_c \tag{5}$$

$$\upsilon_{xz} = \frac{\cos^2\theta}{\sin\theta\left(\frac{h}{l}+\sin\theta\right)} \tag{6}$$

υ_{zx} is a dependent parameter and can be calculated from the following equivalency:

$$\frac{E_x}{\upsilon_{xz}} = \frac{E_z}{\upsilon_{zx}} \tag{7}$$

$$\upsilon_{zx} = \frac{\sin\theta\left(\frac{h}{l}+\sin\theta\right)}{\cos^2\theta} \tag{8}$$

A.1.2 Out-of-plane Equivalent Properties

The out-of-plane equivalent elastic properties are obtained when loading in y-direction. Five elastic moduli are needed to describe the out-of-plane properties. The first obvious modulus is the modulus of elasticity (E_y) which is calculated by utilizing equivalence in y-direction when subjected to extension or compression [24]. In addition, modulus of elasticity (E_y) is proportional to the core equivalent density $\rho_{eq.}$ [15] as follows:

$$\frac{E_y}{E_c} = \frac{\rho_{eq.}}{\rho_c} = \frac{\left(\frac{h}{l}+1\right)}{\left(\sin\theta + \frac{h}{l}\right)\cos\theta}\left(\frac{t}{l}\right) \tag{9}$$

$$E_y = \frac{\left(\frac{h}{l}+1\right)}{\left(\sin\theta + \frac{h}{l}\right)\cos\theta}\left(\frac{t}{l}\right)E_c \tag{10}$$

where ρ_c represents the core material density, the two Poisson's ratios (υ_{yx}), (υ_{yz}) are straightforward and are equal to the core material Poisson's ratio (υ_c) as follows:

$$\upsilon_{yx} = \upsilon_{yz} = \upsilon_c \tag{11}$$

The two Poisson's ratios (υ_{xy}), (υ_{zy}) are calculated as follows:

$$\upsilon_{xy} = \frac{E_x}{E_y}\upsilon_c \approx 0 \tag{12}$$

$$\upsilon_{zy} = \frac{E_z}{E_y}\upsilon_c \approx 0 \tag{13}$$

The calculation of shear moduli is more complicated and computational methods are the only possible way for receiving precise calculations [24]. However, the upper and lower bound for such moduli can be formulated using the published method [26] by estimating the strain energy associated with both strain distribution and stress distribution. If they coincide, then it is an exact solution, if not, then the exact solution lies between them.

For the shear modulus (G_{xy}), both bounds coincide as follows [25]:

$$G_{xy} = \frac{\cos\theta}{\left(\frac{h}{l}+\sin\theta\right)}\left(\frac{t}{l}\right)G_c \tag{14}$$

For the shear modulus (G_{yz}), both bounds differ as follows [15]:

$$\frac{\left(\frac{h}{l}+\sin\theta\right)}{\left(1+\frac{h}{l}\right)\cos\theta}\left(\frac{t}{l}\right)G_c \le G_{zy} \le \frac{\left(\frac{h}{l}+\sin^2\theta\right)}{\left(\frac{h}{l}+\sin\theta\right)\cos\theta}\left(\frac{t}{l}\right)G_c \tag{15}$$

A finite element analysis of a honeycomb core cell was performed and the calculated shear modulus (G_{zy}) was compared with different core thickness (*c*) to wall length (*l*) ratios, and finally an approximation of (G_{zy}) using the least-squares regression was obtained [27] as follows:

$$G_{zy} = G_{zy}{}^{lower} + \frac{0.787}{\left({}^{c}/_{l}\right)}\left(G_{zy}{}^{upper} - G_{zy}{}^{lower}\right) \tag{16}$$

A.2 Equivalent Plate Theory

The equivalent properties of the sandwich structure are estimated as follows:

$$t_{eq.} = \sqrt{t_f^2 + 3\left(H - t_f\right)^2} \tag{17}$$

$$E_{eq.} = \frac{2Et_f}{t_{eq.}} \tag{18}$$

$$\rho_{eq.} = \frac{2t_f\rho_f + \left(H - 2t_f\right)\rho_{core}}{t_{eq.}} \tag{19}$$

where *H* represents the total sandwich structure thickness, t_f is the facing sheet thickness, *E* is the facing sheets modulus of elasticity, ρ_f is the facing sheets density, and ρ_{core} is the honeycomb core density.

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