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ASSESSMENT OF EQUIVALENT METHODS EFFECTIVENESS USED FOR HONEYCOMB PLATE STRUCTURE

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Abstract. Honeycomb plates are commonly used in aerospace structures for their lightweight and rigidity characteristics. However, for modeling honeycomb structure many equivalent methods are established to represent the behavior of detailed model. In the present paper, we focus on the equivalent methods precision compared to the miso-scale model. Therefore, we compare two equivalent assumptions regarding the detailed design of a honeycomb plate (miso-scale). Then, we perform modal analysis and microvibration assessments to verify the effectiveness of methods accuracy regarding modes values and microvibration transmissibility. It can be noticed that each method has the advantage to be a candidate to represent the honeycomb plate behavior due to the lower error percentage. The sandwich plate theory gives minimal difference error for both modal analysis and the measurement of disturbance transmissibility from the reaction wheel to the structures. Hence, the sandwich theory is more or less accurate for representing a detailed model for structural analysis of honeycomb plates.

Nomenclature

D	Bending stiffness
DOF	Degree of freedom
E	Young modulus
E_f	Skin young modulus
E_c	Core young modulus
E_{eq}	Equivalent modulus
G	Shear modulus
G_{xy}	In plane shear modulus
G_{yz}	Out of plane shear modulus
G_{eq}	Equivalent shear modulus
h_c	Core thickness
h_f	Skin thickness
h	Plate thickness

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H_{eq}	Equivalent plate thickness
I	Bending moment
K	Membrane stiffness
l	Cell size
MPC	Multi constraint point
NSM	Nonstructural mass
t	Cell wall thickness
w	Cell length
ρ	Density
ρ_{ca}	Core equivalent density
ρ_{eq}	Equivalent density
θ	Poison ratio

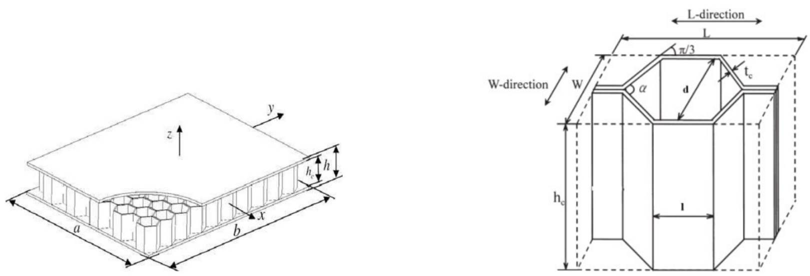
1. Introduction

Honeycomb plate structures are increasingly used in aerospace industry to satisfy the need for lighter structures with high-performance characteristics [1–3]. Usually, honeycomb sandwich plates are used for the primary and secondary structures of a spacecraft. Honeycomb plates consist of two thin-facing sheets separated by a core material. Consequently, understanding these sandwich structures, and the methods and solutions used to produce structural assemblies from them, is still a major concern in the aerospace industry. The design and analysis of complex geometries imply the use of finite element methods to accomplish an effective investigation. The accuracy of results obtained from structural analysis using finite element analysis depends on the modeling assumptions and the accurate application of boundary conditions [4]. For reasons of numerical efficiency, the mechanical analysis of honeycomb sandwich panels during the design process is performed in terms of effective properties rather than by means of a direct computational model. Consequently, it is necessary to select an appropriate equivalent method before the modal analysis or other assessments to qualify the structure. Many researchers deal with the equivalent methods for the modeling of the honeycomb sandwich structure whether directly or indirectly [5–7]. Hence, a homogenization method is introduced by Wang et al. [8]. Dong et al. contributed to developing a method for the definition of an equivalent elastic modulus of honeycomb core using experimental modal results [9]. Boudjemai et al. investigated honeycomb modal analysis made by the finite element method as well as the existing equivalent approaches and compared it to experimental tests [10].

In recent years, there has been extensive research on analyzing lightweight structures. Schwingshackl, C. W., et al. conducted a measuring of the dynamic shear moduli of honeycomb materials and compared the results with the material properties of honeycomb in the literature [11]. In addition, S. D. studied the flexural vibration of symmetric rectangular honeycomb panels and compared the results based on the classical and improved plate theories [12]. Kumar N. et al. presented a theoretical and computational procedure to predict the effective elastic properties of a periodic honeycomb structure using the strain-energy approach [13]. Yifeng Z.

et al. developed a semi-analytical model for evaluating the mechanical behavior of hybrid HSP based on the variational asymptotic theory by performing static and dynamic analysis carried out using the 2-D panel [14]. Xiufang Z. et al. investigated the frequencies and energies of vibrations for a honeycomb sandwich plate with negative Poisson's ratio based on Reddy's third-order shear deformation plate theory, von Karman type nonlinear theory, and Hamilton theory [15]. Georgios K. T. et al. studied the applications of auxetic structures subjected to dynamic loads in comparison to conventional honeycomb structures which represent an improved property of damping, indentation resistance, and fracture toughness [16].

On the other hand, a few researchers have dealt with the constancy of equivalent methods for microvibration which is considered as a tiny amount of disturbance. Micro-vibrations are generated from the moving part of satellite structure which affects the payloads image efficiency. However, Wei-Qing et al. [17] investigated two equivalent finite element dynamic models of honeycomb plates to build an accurate honeycomb plate FE dynamic model that can satisfy the requirements of accurate microvibration analysis. Moreover, Peifei et al. verified the validity of the results using different equivalent methods to analyze microvibration isolation of jointed sandwich plates [18]. This paper focuses on the evaluation of three equivalent theories' effectiveness relative to the miso-scale model, which includes the equivalent plate theory and sandwich plate theory to represent the honeycomb structure. On one hand, modal analysis was performed by comparing the natural frequencies of plate model to the miso-scale model which is considered as a reference; on the other hand, micro-vibrations assessments by deduction of transfer function of transferred load from wheel to the structure are used for the selection of the best equivalence modeling method with minimum difference error, representing the honeycomb structures. The 3D detailed honeycomb plate (miso scale) was designed with the reel geometric parameters and dimensions, which is composed of two identical skins and a honeycomb core. However, the miso-scale model was taken as a reference [9, 19] to compare the obtained results from the finite element analysis with respect to equivalent methods. In figure 1, we represent the details of the honeycomb structure and unit cell geometrical parameters.



(a) (b)

FIGURE 1. Miso-scale honeycomb panel: (a) Geometry parameters, (b) Unit cell geometry

2. Equivalence theories of honeycomb plate

Two equivalent methods will be used in our investigation to verify which approach is more effective to replace the miso-scale model for the complex honeycomb structures.

2.1. Equivalent plate theory assumptions. For this theory, both core and skins are homogenized as isotropic plate and modeled as solid [9, 20, 21] (see figure 2); which have the properties (H_{eq}, E_{eq}, G_{eq} and ρ_{eq}) calculated from the following equations:

$$E_{eq} = \frac{2h_f}{H_{eq}} E_f$$

$$H_{eq} = \sqrt{3h_c^2 + 6h_c h_f + 4h_f^2}$$

$$\rho_{eq} = \frac{2\rho_f h_f + \rho_{ca} h_c}{H_{eq}}$$

$$\rho_{ca} \cong \frac{8}{3\sqrt{3}} \frac{t_c}{t} \rho_c$$

$$G_{eq} = \frac{2h_f}{H_{eq}} G_f$$

where: H_{eq}, E_{eq} are the thickness and Young modulus of the equivalent plate, respectively. ρeq is the mass density of the equivalent plate; and G_f is shear modulus.

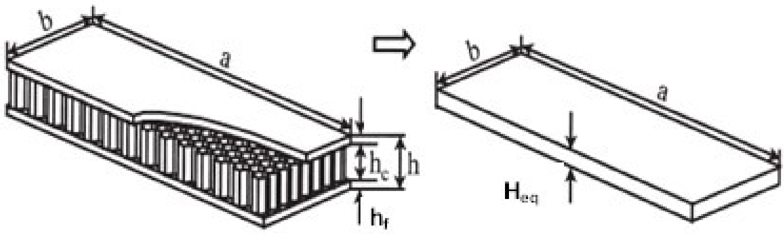


FIGURE 2. Equivalence of sandwich plate

2.2. Equivalent plate theory assumptions. In conventional sandwich theory, only the core will be equalized as 3-D orthotropic materials which have the same stiffness as the honeycomb core; and the face sheets are modeled as shells satisfying the Kirchhoff hypothesis. The properties of the equivalent honeycomb core parameters are calculated according to the following equations [22–25]:

$$E_{cx} = E_{cy} = \frac{4}{\sqrt{3}} \left(\frac{t_c}{l}\right)^3 E$$

$$E_{cz} = \frac{2}{\sqrt{3}} \left(\frac{t_c}{l}\right) E$$

$$G_{cxy} = \frac{\sqrt{3}}{2}\left(\frac{t_c}{L}\right)^3 E$$

$$G_{cyz} \frac{\sqrt{3}\gamma}{2l}\left(\frac{t_c}{l}\right) G$$

$$G_{cxz} = \frac{\gamma}{\sqrt{3}}\left(\frac{t_c}{l}\right) G$$

$$\rho_{ca} = \frac{8dt_c}{A} \cong \frac{8}{3\sqrt{3}}\frac{t_c}{l}\rho$$

Where: E_{cx}, E_{cy} are the in-plane equivalent Young's moduli. The out-of-plane modulus E_{cz} is obtained by equivalence in the z direction. G_{cxy}, G_{cxz} and G_{cyz} are the shear modulus. ρ_{ca} is the density of the core.

3. Materials and methods

3.1. Honeycomb plate model. The honeycomb sandwich plate that was analyzed had a squared shape of 200×200 mm and whole thickness of 15 mm; whereas, the core and skin thickness (h_c and h_f) were 14.4 and 0.3 mm, respectively.

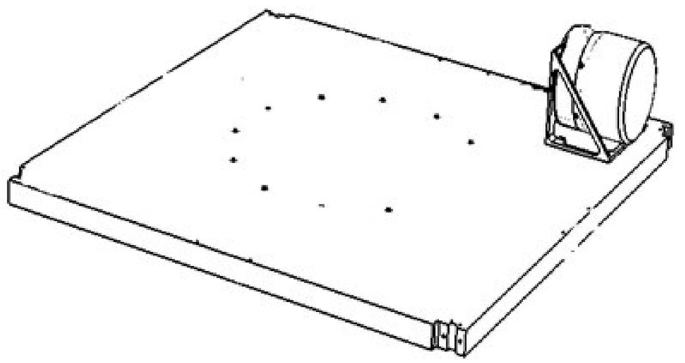


FIGURE 3. Hard mounted reaction wheel on a honeycomb plate

In addition, the unit cell geometric parameters are as follows: cell size (d) is 6.35 mm, cell length (l) is 3.6 mm and wall thickness (t_c) is 0.075 mm. We note that the plate skins are made of Al 7071-T6 and the core of Aluminum Al-5056; hence, materials properties are illustrated in tables 1 and 2. In addition, the equivalent properties calculated from different methods approach are presented in tables 3 and 4. Also, for micro-vibration assessment the same model of plate dimensions was considered which supports a reaction wheel fixed in the corner (figure 3).

The materials properties are calculated according to the two theories. Therefore, the tables below illustrate the values of the parameters:

Table 1. Skins Aluminium properties Al 7071-T6

Skin	E (GPa)	θ	G (GPa)	ρ (kg/^{3})
AI 5056	71	0.33	25.9	2700

Table 2. Orthotropic properties of the core Al 5056

Core material	ρ (kg/m^3)	cell size 1 (mm)	L shear modulus (MPa)	W shear modulus (MPa)
AI 5056	83	6.35	427	207

Table 3. Core equivalent parameters properties calculated from sandwich plate theory

$E_{cx} = E_{cy}$ (MPa)	E_{cz} (MPa)	G_{cxy} (MPa)	G_{cxz} (MPa)	G_{cyz} (MPa)	ρ_{eq} (kg/m^3)
1.48	1707	0.556	186.92	280.4	86.602

Table 4. Equivalent mechanical properties calculated from equivalent plate theory

H_{eq} (m)	E_{eq} (MPa)	ρ_{ca} (kg/m^3)	ρ_{eq} (kg/m^3)	G_{eq} (MPa)
0.02546	1673	86.6	112.6	610

4. Model mesh

Three finite element models representing the sandwich panel were analyzed according to the different equivalent approaches. However, MSc Patran/Nastran is used to investigate the natural frequencies of the plate based on modal analysis; in addition, the assessment of transfer function of micro-vibration induced by the reaction wheel disturbance was performed. Hence, figures (4, 5, 6 and 7) illustrate the mesh and the boundary conditions of different equivalence approaches models for both modal and micro-vibration analysis.

Concerning mesh generation, we used two types of elements: Quad, for meshing the surfaces, and Tetra for solid structures for both analysis cases. Thus, the choice of Mesh is very important for shell and solid modeling to get accurate results statically and dynamically [26]. However, the sensitivity test for selecting the suitable mesh size was conducted and a mesh convergence study was performed in which a series of runs were made for all models. Consequently, convergence was reached when modes values stabilized with mesh variation of less than 1 %. As a result, the meshing of the miso-scale, sandwich plate theory, and equivalent plate theory models require 24531, 20000, and 16000 elements, respectively. It should be noted that the modeling and meshing of the detailed miso-scale model with precise geometric parameters of the core and faces is more time-consuming than the equivalent models, primarily due to the complexity of the honeycomb plate. Furthermore, for some practical problems, modeling the entire core with the finite element method can significantly increase the total number of degrees of freedom (DOFs), which in turn increases computational effort without providing any tangible benefits.

In addition, the wheel and the supporting bracket are modeled as a lumped mass, which are hard mounted to the plate by a multi-point constraint (MPC). The boundary conditions concerning modal analysis of the plate is clamped-free-free-free (CFFF) (figure 5a). For micro-vibration all edges of the plates are free; consequently, a free-free analysis was performed for the equivalent theories of modeling (figure 5b), whereas the force F of magnitude ($1N$) was applied on the point mass, which represents the wheel and its supporting bracket, in (X, Y and Z) directions (figure 4). However, the output results of transfer function are measured in the center of the wheel connections to investigate displacements behaviors towards (x, y and z) in the center of the MPC (Fig. 4).

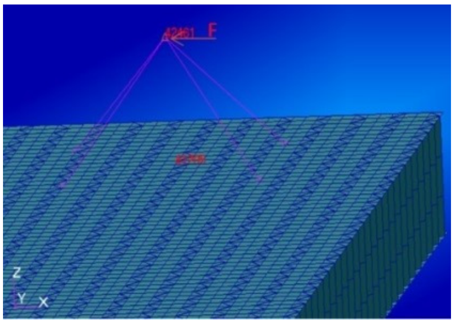
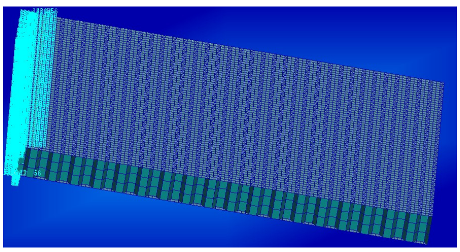
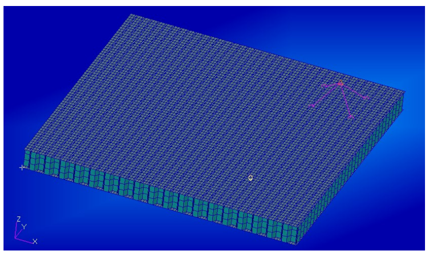


FIGURE 4. Applied force location and the output nodes of transfer function

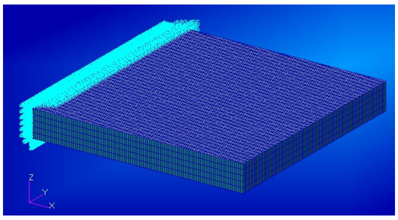


(a) Modal analysis

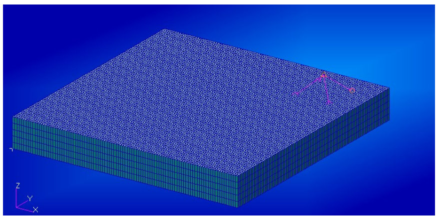


(b) Micro-vibration analysis

FIGURE 5. Boundary conditions and mesh of miso-scale models

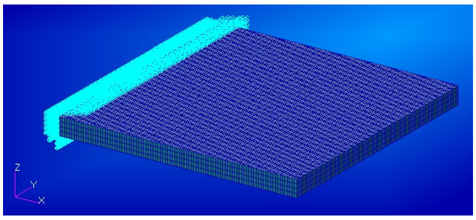


(a) Modal analysis

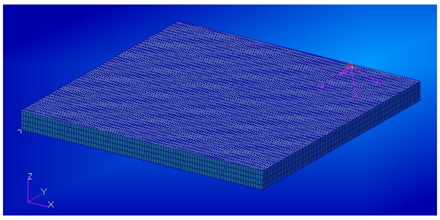


(b) Micro-vibration analysis

FIGURE 6. Boundary conditions and mesh of equivalent plate theory models



(a) Modal analysis



(b) Micro-vibration analysis

FIGURE 7. Boundary conditions and mesh of sandwich plate theory models

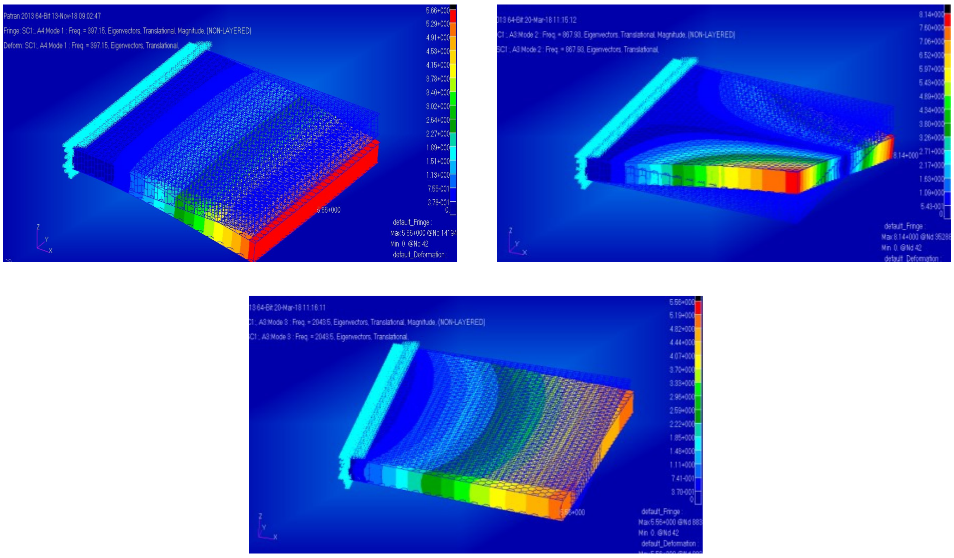


FIGURE 8. First 3 modes shape of miso-scale model

5. Results and discussions

In the first section of our analysis, we calculate the natural frequencies of miso-scale plate and the equivalent methods, in order to compare the exactness of each assumption theory.

5.1. Modal assessments. The same boundary conditions are applied for modal analysis results to investigate the natural frequencies. The results of the three modes generated from different models are illustrated in the figures bellow (see figures 8, 9 and 10) respectively:

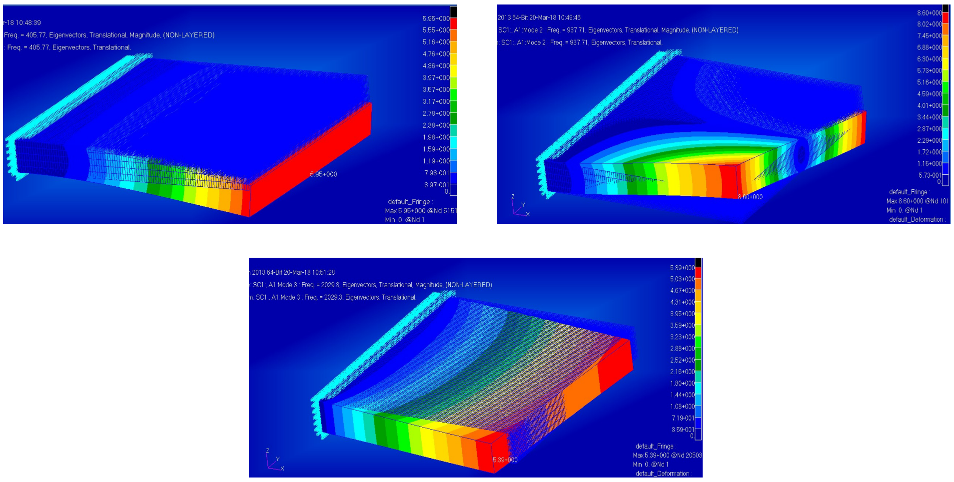


FIGURE 9. First three modes of equivalent theory model

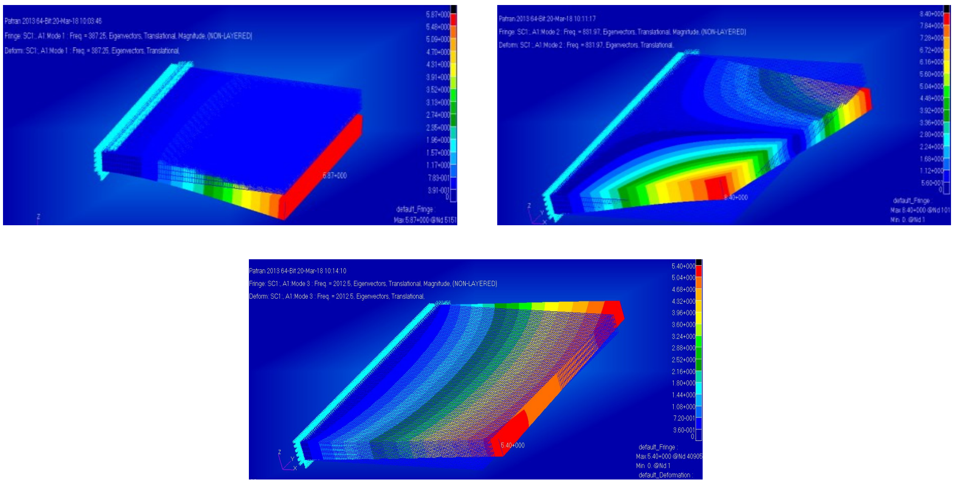


FIGURE 10. First three modes of sandwich plate theory

Figure 10 illustrates the comparison of the first 5 natural frequencies obtained from different equivalence method which are compared to the miso-scale model.

For a detailed comparison of the results, the table below shows regrouped values of first five modes of modal analysis using three models, stating the percentage error between the obtained results from the miso-scale and the two equivalence approaches (see table 5):

TABLE 5. Comparisons of differences variation of modal analysis for different model

	$f_1(Hz)$	$f_2(Hz)$	$f_3(Hz)$	$f_4(Hz)$	$f_5(Hz)$
Miso-scale model	397,15	867,92	2043,51	2152,7	2740,46
Equivalent plate theory model	405,77	937,71	2029,29	2319,24	2935,65
percentage error	2,12	7,44	0,7	7,185	6,65
Sandwich plate theory model	387,25	831,97	2012,53	2013,37	2560,22
percentage error	-2,55	-4,32	-1,54	-6,92	-7,04

The main focus of this analysis is the comparison of the first 5 natural frequencies of the honeycomb plate. However, for high frequencies the confidence of FEM prediction is progressively reducing [27]. Therefore, the differences in percentage errors between the main modes generated from the miso-scale method are compared with two equivalent methods. The results obtained by modal analysis using the three methods models give almost similar values, with a difference error varying from the equivalence approach to the other approaches. On the other hand, the application of the equivalent plate theory leads to higher values than when the miso-scale mode is used, which is contrary to the sandwich plate theory that leads to lower values. From table 5, the percentage errors vary according to the modes number and equivalent methods used ranging from 0.7 to 7.44 percent. Also, the

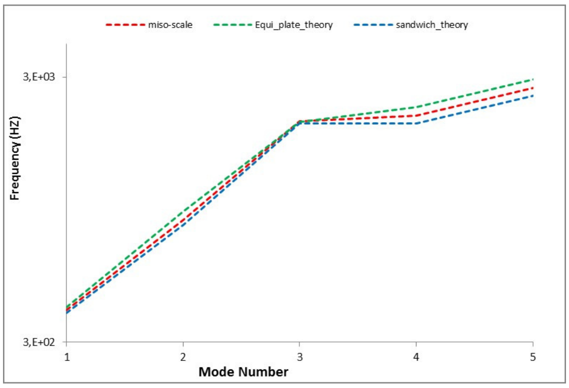


FIGURE 11. Frequency vs. Modes number according to the different equivalent methods

minimum difference values are noticed for the mode 3 for all models, contrary to mode 2 which represents the larger percentage error. However, the largest value of error percentage was marked for mode 2, when applying the equivalent plate theory which is about 7.44 percent (see table 5). Moreover, the closest values to the detailed model are registered with the application of the sandwich plate theory which presents a good correlation to the detailed model. We note that the calculation process was up to three times comparing to other two equivalence models. Thus, the process of modelling and analysis of resources using the equivalent methods is less time-consuming than using the miso-scale model, especially for the complex honeycomb structure.

5.2. Microvibrations assessment. In this section, we compare the results of transfer function of loads generated from the wheel to the honeycomb plate. The different methods of equivalence are compared to the miso-scale model to verify the accuracy of each method to predict the sensitivity of transferred micro-vibration from the wheel to the structure. For this analysis, the loads are applied towards X, Y and Z directions and the amount of transfer function are measured at the node situated in the middle of the modeled wheel (see figure 3). It can be seen that the load generates three displacement towards x, y and z.

Comparison results generated from X loading. The results in figure 12 illustrate a displacement amplitude with respect to frequency which are generated

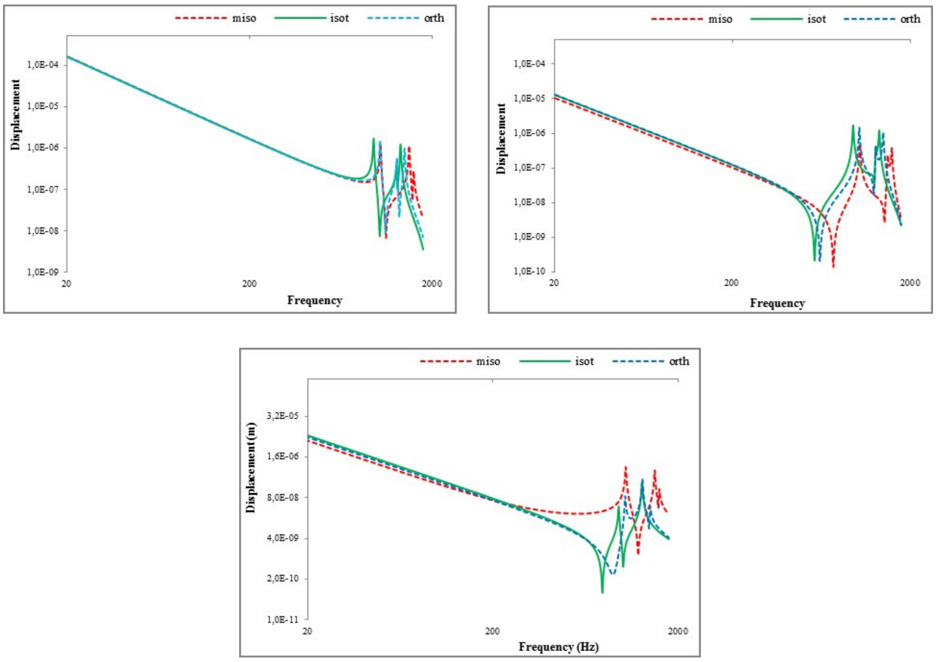


FIGURE 12. Displacement vs. frequency variation toward x, y, z

from an X direction disturbance force according to different equivalence assumptions compared to the miso-scale model.

Comparison results generated from Y loading. The illustrated results in figure 13 illuminate displacement amplitudes with respect to frequencies which were generated from a disturbance force towards Y direction according to the equivalence assumptions and miso-scale model.

Comparison results generated from Z loading. Figure 14 presents the results of displacements amplitude with respect to frequency response generated from Z loading according to different equivalence theories compared to the miso-scale model.

From figures 12, 13 and 14, we use the results generated from the miso-scale method as a reference to evaluate the effectiveness of different equivalent methods. It is clear for all loading cases that the disturbance amplitude is significantly marked towards applied force directions compared to the two other directions. We can also notice that the transfer functions have almost the same response behaviors generated according different equivalent method of honeycomb plates. However, the amount of resonance frequencies and their level differ slightly from an equivalent method to the other methods in terms of resonance frequencies and their magnitude of displacement.

For the three loading cases, the results of three modeling theories don't predict exactly the variation behavior of micro-vibration throw the frequency band (0–2000 Hz) generated from miso-scale model. We note that, the equivalent theory

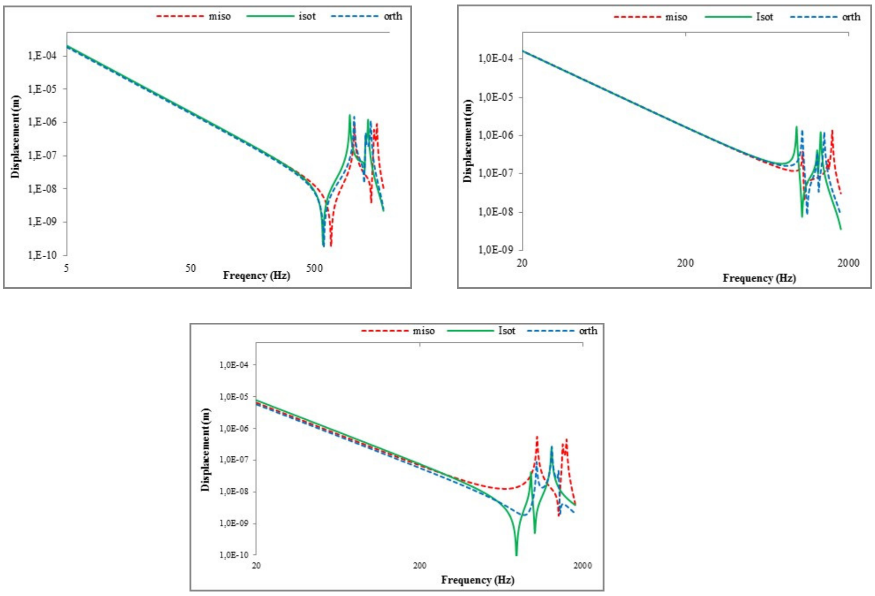


FIGURE 13. Displacement vs. frequency variation toward x, y, z

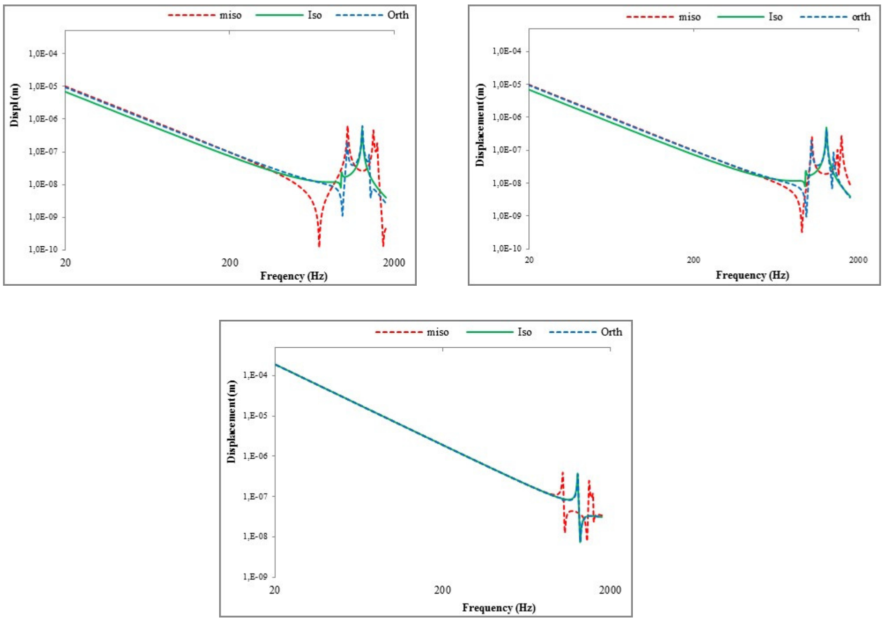


FIGURE 14. Displacement vs. frequency variation toward x, y, z

method presents the larger error shift in terms of resonance frequencies and correspondent magnitude. On the other hand, the sandwich plate theory presents the closest results according to detailed mod (miso-scale). Thus, it can be said that the modeling of honeycomb according to the later equivalence methods allows us to deduce the transferred disturbance from wheel to the supporting structure effectively with a minimum error of microvibrations.

6. Conclusion

In our investigation, two different models of regular hexagonal aluminum honeycomb cores used in sandwich structures are assessed and compared with detailed honeycomb plate geometry called the miso-scale model using the finite elements method.

On the other hand, equivalent method based on sandwich plate theory assumptions is more reliable and practical reflecting the finite element analysis for both modal assessments and evaluation of microvibration transfer function induced by reaction wheel. Hence, the sandwich plate theory has the ability to give the minimum error compared to the equivalent plate theory; and consequently, replacing the detailed model to obtain accurate results and reduce the time needed for pre- and post-processing of the results.

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ПРОЦЕНА ЕФИКАСНОСТИ ЕКВИВАЛЕНТНИХ МЕТОДА КОРИШЋЕНИХ ЗА СТРУКТУРУ ПЛОЧА У САЋУ

Резиме. Плоче са саћем се обично користе у ваздухопловним структурама због своје лакоће и крутости. Међутим, за моделирање структуре саћа уведене су многе еквивалентне методе које описују понашање детаљног модела. Наш фокус у овом истраживању је прецизност еквивалентних метода у поређењу са моделом мизо-скале. Због тога смо упоредили два еквивалентна метода у односу на детаљни дизајн плоче са саћем (мизо-скала). Сходно томе, извршили смо модалну анализу и процене микровибрација како бисмо проверили ефикасност тачности метода у односу на величину модова и преносивост микровибрација. Закључили смо да методе имају погодност да буду кандидати за представљање понашања плоче са саћем због малог процента грешке. Теорија сендвич плоче даје минималну грешку разлике и за модалну анализу и за мерење преносивости сметњи са реакционог точка на структуре. Дакле, сендвич теорија је мање-више тачна за представљање детаљног модела за структурну анализу плоча са саћем.

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