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ORIGINAL PAPER

An Equivalent Modeling Method for Honeycomb Sandwich Structure Based on Orthogonal Anisotropic Solid Element

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Abstract

An equivalent modeling method for honeycomb sandwich structure is presented in this paper. Honeycomb core is regarded as an interlayer and orthogonal anisotropic solid elements are used to model it, while the panels of honeycomb sandwich structure are represented by shell elements. This method not only controls model size and ensures computational efficiency, but also solves the problem that two-dimensional model cannot represent the internal stress distribution and local deformation. Based on the orthogonal anisotropy of honeycomb and the actual cellular size, 9 independent elastic parameters of the interlayer are given, so that the physical properties of the interlayer are described completely. In the example, the displacement errors under typical static loadcases are less than 3.12% and the frequency errors of the first six orders are less than 4.07%, compared with the precise model. A modal tapping test was carried out on a payload mounting panel with honeycomb sandwich structure. By comparing the test data with the analysis data of the equivalent model, it was shown that the frequency errors of the first six orders were all within 5%, and the analysis modes were consistent with the experimental fitting modes, which further verified the validity of the equivalent method.

Keywords Honeycomb sandwich structure · Equivalent modeling · Orthogonal anisotropic material · Finite element analysis · Modal tapping test

1 Introduction

Honeycomb sandwich structure is a special composite material with low surface density, high specific stiffness and good fatigue resistance, which is an ideal material for lightweight design in aerospace engineering and has been widely used in modern remote sensors and satellite structures [1–4]. In the newly developed remote sensors, such as Japan's ALOS-3 high image resolution camera, ESA's CHEOPS space telescope, honeycomb sandwich structure was used in their main bearing structure [5, 6]. In addition, honeycomb sandwich structure is also widely used in payload mounting panels, solar panels and other structural parts in satellite platform.

In recent years, the research on the modeling methods of honeycomb sandwich structure has been continuously deepening. Tanimoto and co-workers proposed a new modeling method for honeycomb sandwich structure, in which orthogonal anisotropic shell elements and two kinds of beam elements were used to represent the panel, bonding layer and honeycomb core, respectively [7]. It was pointed out that the vibration characteristics of honeycomb sandwich structure are closely related to the stiffness of the panel and the geometrical shape of the honeycomb core. Guj and Sestieri established an orthogonal anisotropic equivalent model of honeycomb using multi-scale asymptotic technique, and the model was verified by the method of finite element numerical simulation [8]. Jiang et al. analyzed the sensitivity of the modal frequency of honeycomb sandwich structure to the constitutive parameters, and pointed out that the normal shear modulus has the greatest influence on the modal frequency, and proposed a method to determine the equivalent elastic parameters of honeycomb core by means of modal test [9]. Qin et al. constructed a response surface model for the

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two-dimensional equivalent model of honeycomb sandwich structure based on sandwich theory to replace the original finite element model, using design of experiment (DOE) method and optimization algorithm, to improve the analysis efficiency of optimization design of honeycomb sandwich structure [10].

At present, the two-dimensional modeling method based on sandwich theory is considered to be the most widely used equivalent method with relatively high accuracy in the engineering analysis of honeycomb sandwich structures [10–12]. Similar modeling methods treat the actual three-dimensional structure as a two-dimensional plate with a certain thickness. Although the shell element can satisfy certain engineering calculation accuracy in a small scale model, it only reflects the bending stiffness and in-plane stiffness of the thin plate structure, and cannot represent the stress distribution and deformation along the thickness direction of honeycomb sandwich structure.

In the simulating calculation of satellite platforms or remote sensors, the finite element model of the key components with honeycomb sandwich structure, such as payload mounting panels in satellites and main bearing panels in space cameras, must be able to faithfully represent the overall stress state and force transferring path, especially the local deformation and stress under static loadcases. In view of this special demand, this paper introduces the concept of interlayer and deduces the equivalent parameters of interlayer material, and a finite element modeling method of honeycomb sandwich structure is put forward, which combines three-dimensional elements with two-dimensional elements. Compared with the simulation of precise model, the computational accuracy of this method is tested. Finally, a modal tapping test on a honeycomb panel is carried out to further verify the accuracy of the equivalent modeling method proposed in this paper.

2 Equivalent Modeling Method

2.1 Modeling Description

Honeycomb sandwich structure, commonly used in aerospace engineering in China, generally adopts aluminum honeycomb core, and its upper and lower panels are usually laid with carbon fiber, as shown in Fig. 1a. The honeycomb core consists of numerous basic units, as shown in Fig. 1b, and the main direction of honeycomb sandwich structure is generally defined as the *X*-axis direction in the same figure. Considering the corresponding manufacturing technique in China, honeycomb sandwich structure usually adopts regular hexagonal honeycomb, and the length of vertical rib h is equal to that of diagonal rib l, $\theta = \pi/6$, and as the honeycomb core is usually made of two layers of aluminum foil after being pasted and stretched, the vertical rib is twice as thick as the diagonal rib.

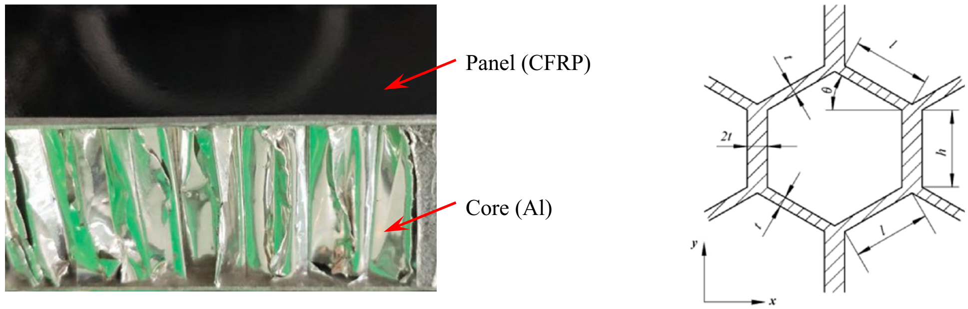
The most commonly used method in engineering analysis of honeycomb sandwich structure is the two-dimensional modeling method based on sandwich theory, in which the whole honeycomb panel is represented by only one layer of two-dimensional shell elements with the property of composites, as shown in Fig. 2a. The upper panel, lower panel and honeycomb core are regarded as three layers in the composite, respectively, and the honeycomb core layer is endowed with orthogonal anisotropic property for shell elements. In contrast, the method proposed in this paper is regarding the actual loose and discontinuous aluminium honeycomb core as a uniform and continuous 'interlayer', of which the thickness and size are consistent with that of the honeycomb core, and the solid element with corresponding orthogonal anisotropic property is used to characterize the interlayer in modeling. Meanwhile, the upper and lower panels made of carbon fiber are mainly subjected to in-plane loads, so that they can be simulated using two-dimensional shell elements. The finite element model of honeycomb sandwich structure based on the method above is shown in Fig. 2b. According to the relevant knowledge of material mechanics, honeycomb has the mechanical properties of orthogonal anisotropy, and the equivalent mechanical parameters of the interlayer can be deduced.

For orthogonal anisotropic materials, the stress–strain relationship is characterized by the constitutive Eq. (1), in which the strain component is expressed as $[\varepsilon]_m = \left[\varepsilon_{11}\varepsilon_{22}\varepsilon_{33}\gamma_{12}\gamma_{23}\gamma_{31}\right]^{\mathrm{T}}$, the stress component is expressed as $[\sigma]_m = [\sigma_{11}\sigma_{22}\sigma_{33}\tau_{12}\tau_{23}\tau_{31}]^{\mathrm{T}}$, and the expression of flexibility matrix is described as Eq. (2). According to the symmetry of stiffness matrix, it can be known that, $E_1\mu_{21} = E_2\mu_{12}$,$E_2\mu_{32} = E_3\mu_{23}$, $E_3\mu_{13} = E_1\mu_{31}$, so that there are only nine independent elastic parameters in the flexibility matrix S, which are $E_1, E_2, E_3, G_{12}, G_{23}, G_{31}, \mu_{12}, \mu_{23}, \mu_{31}$:

$$[\varepsilon]_m = S \cdot [\sigma]_m, \tag{1}$$

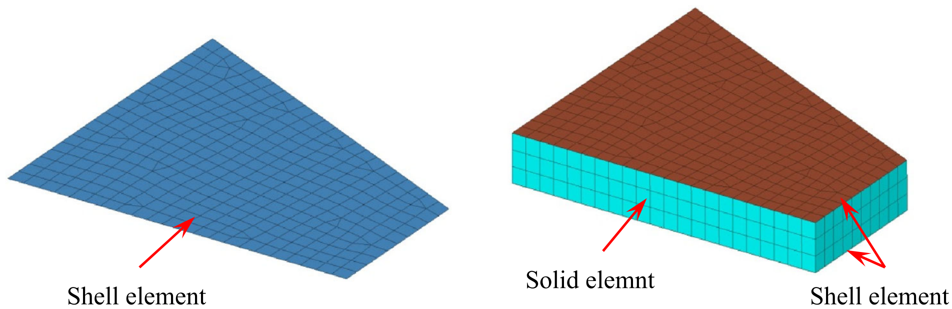
$$S = \begin{bmatrix} \frac{1}{E_1} & -\frac{\mu_{21}}{E_2} & -\frac{\mu_{31}}{E_3} & \cdots & \cdots & \cdots \\ -\frac{\mu_{12}}{E_1} & \frac{1}{E_2} & -\frac{\mu_{32}}{E_3} & \cdots & \cdots & \cdots \\ -\frac{\mu_{13}}{E_1} & -\frac{\mu_{23}}{E_2} & \frac{1}{E_3} & \cdots & \cdots & \cdots \\ \cdots & \cdots & \cdots & \frac{1}{G_{12}} & \cdots & \cdots \\ \cdots & \cdots & \cdots & \cdots & \frac{1}{G_{23}} & \cdots \\ \cdots & \cdots & \cdots & \cdots & \cdots & \frac{1}{G_{31}} \end{bmatrix}. \tag{2}$$

Based on the geometric parameters of honeycomb core, the expressions of the nine parameters above can be deduced, and the physical properties of the interlayer material can be described completely. With the help of orthogonal



(a) internal structure **(b)** basic unit of honeycomb core

Fig. 1 Schematic of honeycomb sandwich structure



(a) common two-dimensional modeling method **(b)** equivalent modeling method in this paper

Fig. 2 Comparison between different modeling methods for honeycomb sandwich structures

anisotropic material property cards for three-dimensional solid elements in commercial FEA software, such as the MAT9ORT card in HYPERMESH, the equivalent material properties of the interlayer, which is regarded as the equivalent of honeycomb core, can be defined.

2.2 Derivation of Equivalent Elastic Parameters

The structural parameters of the hexagonal honeycomb discussed in this paper are shown in Fig. 1b, manufacturing process contributes to the phenomenon that vertical rib is twice as thick as diagonal rib. In the equivalence process, the Euler–Bernoulli beam theory and the small deformation hypothesis are adopted to analyze the cellular thin-walled structure. The in-plane equivalent elastic parameters of this specific realistic honeycomb configurations are given by Gibson et al. and Burton et al. as follows [13–15]:

$$E_1 = E_s \frac{t^3}{l^3} \frac{\cos\theta}{(\beta + \sin\theta)\sin^2\theta}, \tag{3}$$

$$\mu_{12} = \frac{\cos^2\theta}{(\beta + \sin\theta)\sin\theta}, \tag{4}$$

$$E_2 = E_s \frac{t^3}{l^3} \frac{(\beta + \sin\theta)}{\cos^3\theta}, \tag{5}$$

$$\mu_{21} = \frac{(\beta + \sin\theta)\sin\theta}{\cos^2\theta}, \tag{6}$$

$$G_{12} = E_s \frac{t^3}{l^3} \frac{(\beta + \sin\theta)}{\beta^2(\beta/4 + 1)\cos\theta}. \tag{7}$$

In the equations above, E_s is the elasticity modulus of honeycomb material, θ is the characteristic angle of honeycomb, l is the length of diagonal rib, h is the length of vertical rib, t is the thickness of diagonal rib, and β is the ratio of h to l.

In Refs. [13–15], the form of G_{12} changes with the specific configurations of the honeycomb structure. As mentioned above, the vertical rib is twice as thick as the diagonal rib in the basic unit of honeycomb core structure discussed in this paper, which is consistent with the situation in Ref. [15], so the expression of shear modulus G_{12} is chosen as Eq. (7).

However, it can be found from Eqs. (4) and (6) that $\mu_{12} \cdot \mu_{21} = 1$, which will lead to the singularity of stiffness matrix in finite element analysis. Considering the influence of tensile deformation in honeycomb walls on the in-plane stiffness, the above Gibson equations about E_1, E_2, μ_{12}, μ_{21}, are modified [16, 17]:

$$E_1 = E_s \frac{t^3}{l^3} \frac{\cos\theta}{(\beta + \sin\theta)\sin^2\theta}\left(1 - \cot^2\theta \frac{t^2}{l^2}\right), \tag{8}$$

$$\mu_{12} = \frac{\cos^2\theta}{(\beta + \sin\theta)\sin\theta}\left(1 - \csc^2\theta \frac{t^2}{l^2}\right), \tag{9}$$

$$E_2 = E_s \frac{t^3}{l^3} \frac{(\beta + \sin\theta)}{\cos^3\theta}\left[1 - \left(\beta\sec^2\theta + \tan^2\theta\right)\frac{t^2}{l^2}\right], \tag{10}$$

$$\mu_{21} = \frac{(\beta + \sin\theta)\sin\theta}{\cos^2\theta}\left[1 - (\beta + 1)\sec^2\theta\frac{t^2}{l^2}\right]. \tag{11}$$

By means of the modifications above, the problem of matrix singularity is solved. In the honeycomb panels discussed in this paper, the cellulars are regular hexagon, so h is equal to l, and $\beta = 1$, $\theta = \pi/6$. Thus, the expressions of in-plane equivalent modulus of elasticity E_1, E_2, in-plane Poisson's ratio μ_{12}, and in-plane shear modulus G_{12} are converted as follows:

$$E_1 = \frac{4}{\sqrt{3}}E_s\left(1 - 3\frac{t^2}{l^2}\right)\frac{t^3}{l^3}, \tag{12}$$

$$E_2 = \frac{4}{\sqrt{3}}E_s\left(1 - \frac{5t^2}{3l^2}\right)\frac{t^3}{l^3}, \tag{13}$$

$$\mu_{12} = 1 - 4\frac{t^2}{l^2}, \tag{14}$$

$$G_{12} = \frac{4\sqrt{3}}{5}E_s\frac{t^3}{l^3}. \tag{15}$$

Through the calculation of strain energy, the equation of the normal equivalent elastic modulus E_3 and the out-of-plane equivalent shear modulus G_{23}, G_{31} can be derived as follows [13, 18]:

$$E_3 = E_s\left(\frac{t}{l}\right)\frac{1 + \beta}{\cos\theta(\beta + \sin\theta)}, \tag{16}$$

$$G_{31} = G_s\left(\frac{t}{l}\right)\frac{\cos\theta}{(\beta + \sin\theta)}, \tag{17}$$

$$G_{23} = G_s\left(\frac{t}{l}\right)\frac{\beta + \sin\theta}{(\beta + 1)\cos\theta}. \tag{18}$$

In the equations above, G_s is the shear modulus of honeycomb material. As for the shear modulus G_{23}, Kelsey derived the expressions of both the upper and lower limits for G_{23} in Ref. [18], using two different approximate theoretical approaches, respectively, as shown in Eq. (19). It was also pointed out that the exact shear modulus depends on the thickness of the faces in a specific sandwich structure. Sandwich with thin faces tends to give results closer to the lower limit of G_{23}, which is described as Eq. (18), while sandwich with thick faces favours more the upper limit of G_{23}:

$$G_s\left(\frac{t}{l}\right)\frac{\beta + \sin\theta}{(\beta + 1)\cos\theta} \le G_{23} \le G_s\left(\frac{t}{l}\right)\frac{\beta + \sin^2\theta}{(\beta + \sin\theta)\cos\theta}. \tag{19}$$

The honeycomb sandwich structure discussed in this paper is commonly used in aerospace engineering, and the thickness of faces is usually very small compared to the total height of the entire honeycomb structure, which is consistent with the situation of sandwich with thin faces proposed by Kelsey, so that the lower limit of G_{23} is adopted in our equivalent modeling method. By substituting in the actual parameters of the honeycomb, the above three equations are converted as follows:

$$E_3 = \frac{8\sqrt{3}}{9}E_s\left(\frac{t}{l}\right), \tag{20}$$

$$G_{31} = \frac{\sqrt{3}}{3}G_s\left(\frac{t}{l}\right), \tag{21}$$

$$G_{23} = \frac{\sqrt{3}}{2}G_s\left(\frac{t}{l}\right). \tag{22}$$

Apparently, the normal Poisson's ratio μ_{31} and μ_{32} of the cellular structure is the same as that of the honeycomb core material itself, that is:

$$\mu_{31} = \mu_{32} = \mu_s. \tag{23}$$

Considering the symmetry of the stiffness matrix of orthogonal anisotropic material, $E_2{\cdot}\mu_{32} = E_3{\cdot}\mu_{23}$, the following equation can be obtained:

$$\mu_{23} = \frac{E_2 \cdot \mu_{32}}{E_3} = \frac{3}{2}\left(1 - \frac{5t^2}{3l^2}\right)\frac{t^2}{l^2}\mu_s. \tag{24}$$

The density of honeycomb material is ρ_s. According to the geometric relationship, the equation of equivalent density ρ_c of honeycomb material can be deduced:

$$\rho_c = \rho_s\left(\frac{t}{l}\right)\frac{\beta + 1}{(\beta + \sin\theta)\cos\theta} = \frac{8\sqrt{3}}{9}\rho_s\frac{t}{l}. \tag{25}$$

In summary, the expressions of the equivalent elastic parameters of the interlayer are given by Eqs. (12)–(15), and Eqs. (20)–(25).

2.3 Comparison of Analysis Accuracy

Assuming that the size of a honeycomb panel is 109 mm × 96 mm, the total thickness is 20 mm, the upper and lower panels are made of carbon fiber composite (T700) with the thickness of 1 mm, the honeycomb core material is aluminum alloy (5A02), the wall thickness is 0.05 mm, and the rib length is 3 mm, as shown in Fig. 3. The physical properties of honeycomb core and panel materials are shown in Table 1. According to the equivalent equations of honeycomb core given above, the physical properties of the interlayer can be obtained as shown in Table 2. In the study of this paper, the finite element analysis software used was HyperWorks, the computer used was a DELL T5810 workstation with 16 GB memory and an CPU of Intel Xeon E5-1603.

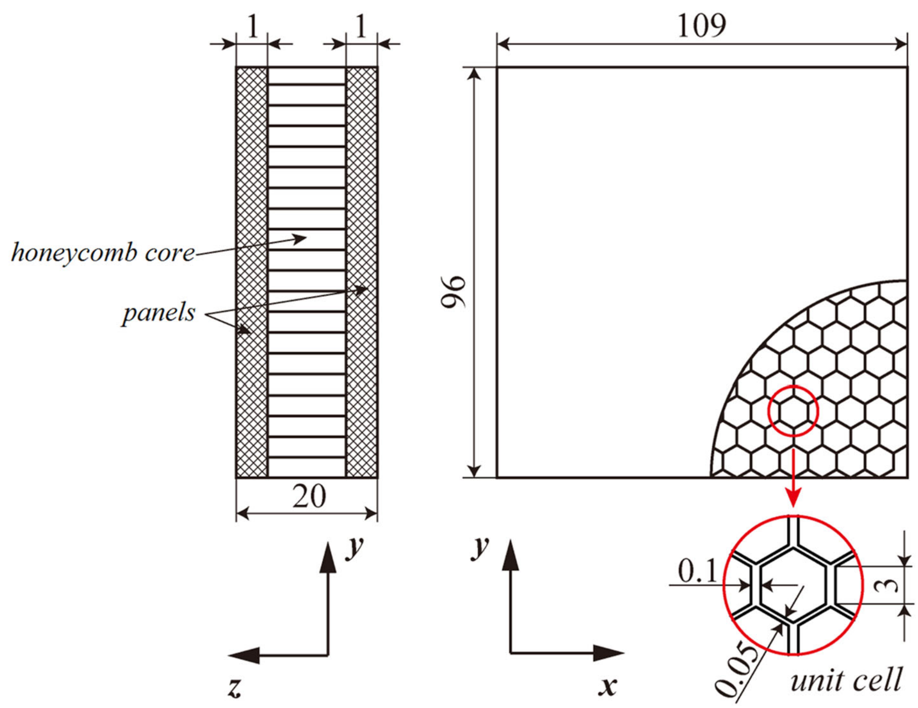


Fig. 3 Schematic diagram of a honeycomb panel

Table 1 Physical properties of the materials used in typical honeycomb sandwich structure

Structure	Materials	E (Gpa)	G (Gpa)	ρ (g/cm^3)	μ
Panels	T700	55	–	1.8	0.30
Honeycomb core	5A02	70	26	2.8	0.33

Although the precise model is rarely used in large-scale engineering analysis because of its large size and low computational efficiency, it is generally believed that it can relatively accurately represent the structural characteristics and actual deformation of honeycomb sandwich structure in the mechanical simulation, and the analysis results of the precise model can be regarded as exact solutions in the absence of practical test results.

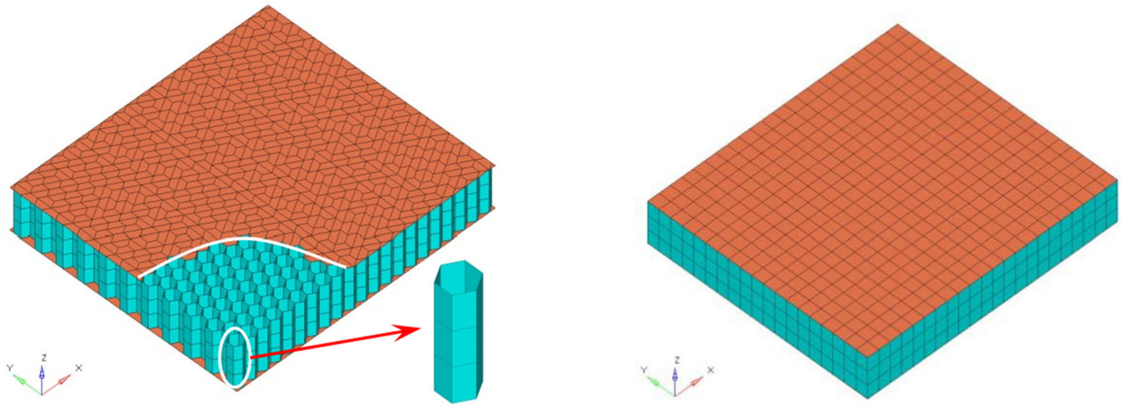
Deformation under typical static loadcases and free modes of the honeycomb panel described above are analyzed using the equivalent modeling method proposed in this paper, and then these statistics are compared with the results of the precise model, in this way, the accuracy of the modeling method proposed in this paper can be evaluated.

In the precise modeling method, two-dimensional shell elements are used to simulate both the thin-walled cellular structure and the two panels of honeycomb sandwich structure. As shown in Fig. 4a, the precise model contains 3814 nodes and 5814 shell elements of type CTRIA (triangular three-node shell element) or CQUAD4 (quadrilateral four-node shell element). According to the method in this paper, the finite element model of the honeycomb panel in the example is obtained as shown in Fig. 4b, there are 1880 nodes, 836 shell elements of type CQUAD4 and 1254 solid elements of type CHEXA (six-sided solid element with eight nodes) in the equivalent model. A uniform element size of 6 mm was used in the meshing of both modeling methods. In this example, the main direction of honeycomb structure is the X direction of the coordinate system in the Fig. 4.

In the analysis, six degrees of freedom of all nodes at the end of + X side in the honeycomb panel are constrained, uniform tensile load P_x (force that along X axis), in-plane shear load P_y (force that along Y axis), normal shear load P_z (force that along Z axis), pure bending load M (torque that around Y axis) are applied at the other end, respectively. The maximum displacements of the honeycomb panel under different loadcases are calculated using different models, as shown in Table 3. In the loadcases of P_x, P_y, P_z, the displacements are the components along the direction of loading forces,

Table 2 Equivalent elastic parameters of interlayer in the example

E_1 (MPa)	E_2 (MPa)	E_3 (MPa)	G_{12} (MPa)	G_{23} (MPa)	G_{31} (MPa)	μ_{12}	μ_{23}	μ_{31}	ρ_c (g cm^{-3})
0.748	0.748	1796	0.449	375.3	250.2	0.9989	0.0001	0.33	0.0718



(a) precise model **(b)** equivalent model proposed in this paper

Fig. 4 Finite element models of the honeycomb panel

Table 3 Analysis results of displacement under typical static loadcases

Load	Value	Precise model (mm)	Equivalent model (mm)		Relative value (%)
			G23 upper limit	G23 lower limit	
P_x	1000 N	9.698×10^{-3}	9.815×10^{-3}	9.816×10^{-3}	1.22
P_y	1000 N	8.083×10^{-2}	8.207×10^{-2}	8.208×10^{-2}	1.55
P_z	1000 N	7.085×10^{-1}	7.306×10^{-1}	7.306×10^{-1}	3.12
M	10 N m	6.720×10^{-2}	6.792×10^{-2}	6.793×10^{-2}	1.69

respectively, while in the loadcase of M, the displacement is the magnitude.

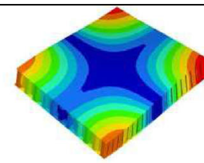
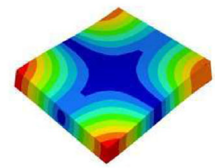
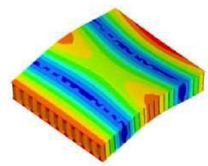
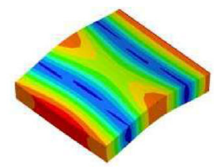
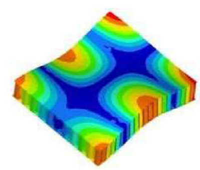
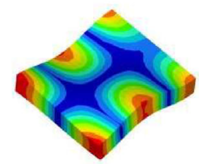
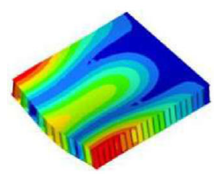
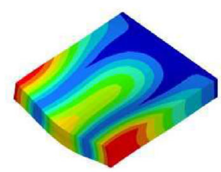
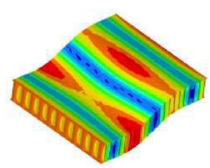
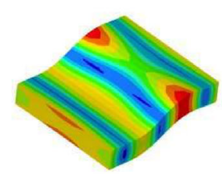
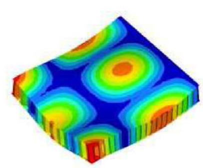
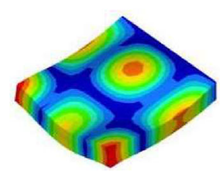
From the statistics in Table 3, it can be seen that the maximum displacement relative error of the equivalent model is less than 3.12% among all the typical static loadcases compared with the precise model, which shows that the equivalent modeling method proposed in this paper has high accuracy in static analysis. According to Eq. (19), the lower limit of G_{23} in this example is 375.3 Mpa, which was adopted in the analysis, while the upper limit is 417 Mpa. With the G_{23} value in the above equivalent model replaced by the upper limit, the deformation under typical static loadcases was also calculated, and it can be concluded that the difference between these two cases is so small that can be ignored.

As shown in Table 3, the maximum displacements of the equivalent model under typical loadcases are all slightly smaller than the results of the precise model, indicating that the stiffness of finite element model is slightly weakened in the equivalent process of honeycomb sandwich structure. Compared with other loadcases, the error between the results of equivalent model and precise model under the loadcase of P_z is relatively large. The reason may be that, in the loadcases of P_x, P_y and M, the honeycomb sandwich structure is mainly subjected to in-plane load, in which the carbon fiber panels with good in-plane stiffness on the upper and lower surfaces play the key role in enhancing the structural stiffness, while the honeycomb core plays a relatively small role. However, under the loadcase of P_z, the honeycomb sandwich structure mainly bears normal load, and the thin carbon fiber panels have weak bearing capacity to this load type, so the normal load is mainly borne by the honeycomb core, resulting in a relatively large error.

In order to check whether the equivalent modeling method proposed in this paper can accurately represent the dynamic characteristics of honeycomb panel or not, free mode analysis of the honeycomb panel in this example is carried out using the precise model and the equivalent model, respectively. The analysis results of the first six free modes are shown in Table 4.

From the statistics in Table 4, we can see that the first six order frequencies data calculated by the proposed equivalent model are slightly smaller than those by the precise model, which also shows that the equivalent modeling method in this paper can weaken the stiffness of honeycomb sandwich structure to a certain extent. Numerically, the maximum difference of the first six order frequencies is 4.07%, and from the comparison of deformation color-mappings, the mode shapes of each order frequency are basically the same with precise model, which shows that the equivalent modeling method in this paper can basically meet the accuracy requirements of engineering analysis in the dynamic analysis of honeycomb sandwich structure, and meanwhile the modal characteristics can be well preserved.

Table 4 Results of free mode analysis using different models

Order	Frequency (Hz)		Relative Error (%)	Mode shapes	
	Precise model	Equivalent model		Precise model	Equivalent model
1	5574.23	5377.39	− 3.53		
2	7306.59	7140.55	− 2.27		
3	10133.60	9721.34	− 4.07		
4	10280.18	9995.52	− 2.78		
5	11718.87	11332.13	− 3.30		
6	12722.42	12234.77	− 3.83		

In the example above, with the same software settings and hardware configurations, the calculating process of the model built with the equivalent modeling method proposed in this paper, which contains both the static loadcases and the free mode analysis, only took 2 s (measured by CPU time), far less than the 6 s of the precise model.

Furthermore, the model scale of precise modeling method is mainly determined by the honeycomb size, while the model size of the equivalent modeling method is mainly determined by the overall size of the honeycomb panels. Therefore, using the equivalent modeling method, larger element size can be adopted in practice to further reduce the number of nodes, which makes the advantage of improving computational efficiency more obvious in the analysis of complex structure with honeycomb panels.

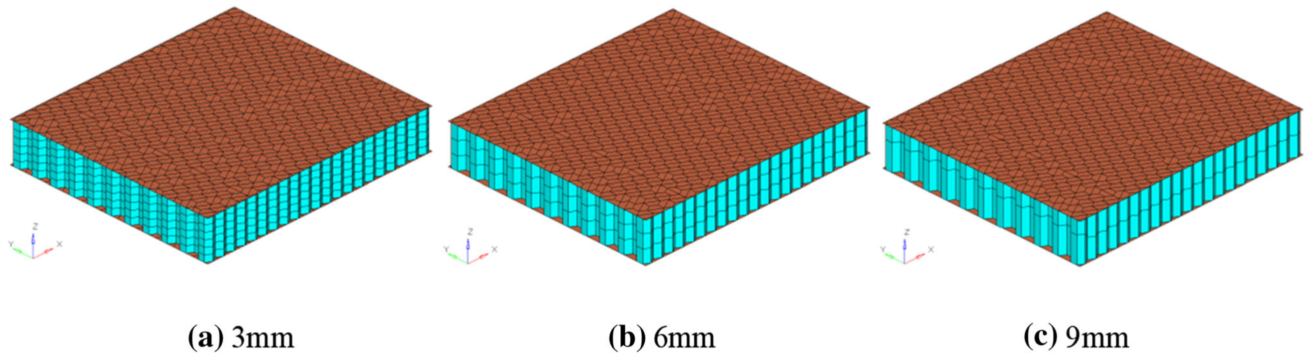


Fig. 5 Precise models of the honeycomb panel with different element size

Table 5 Convergence test of precise model in static analysis

Element size (mm)	Load			
	P_x	P_y	P_z	M
Displacement (mm)				
3	9.771×10^{-3}	8.035×10^{-2}	7.019×10^{-1}	6.722×10^{-2}
6	9.698×10^{-3}	8.083×10^{-2}	7.085×10^{-1}	6.720×10^{-2}
9	9.717×10^{-3}	8.053×10^{-2}	7.033×10^{-1}	6.683×10^{-2}
Relative error with respect to 3 mm model (%)				
6	− 0.75	0.60	0.94	− 0.03
9	− 0.55	0.22	0.20	− 0.58

Table 6 Convergence test of precise model in free mode analysis

Element size (mm)	Order					
	1	2	3	4	5	6
Frequency (Hz)						
3	5563.44	7284.92	10,103.31	10,232.86	11,681.06	12,749.08
6	5574.23	7306.59	10,133.6	10,280.18	11,718.87	12,722.42
9	5600.03	7341.62	10,193.25	10,344.79	11,770.65	12,807.2
Relative error with respect to 3 mm model (%)						
6	0.19	0.30	0.30	0.46	0.32	− 0.21
9	0.66	0.78	0.89	1.09	0.77	0.46

2.4 Discussion on the Convergence of Precise Model

As meshing configurations, such as element size and mesh quality, will affect the simulation results of finite element models to a great extent, a convergence test was carried out in order to clarify the influence of element size on the analysis accuracy of the precise model used in Sect. 2.3.

In the convergence test, three typical precise models of the honeycomb panel with element size of 3 mm, 6 mm and 9 mm (as 1 time, 2 times, 3 times the length of the honeycomb rib) were built, respectively, as shown in Fig. 5, static analysis and free mode analysis with the same loadcase settings as the example in former section were carried out. The analysis results are shown in Tables 5 and 6, and the relative deviations of the 6 mm and 9 mm models with respect to the 3 mm model were also calculated.

According to the statistics, the maximum deviation of the 6 mm and 9 mm models with respect to the 3 mm model is no more than 0.94% in static analysis, while in the free mode analysis, the maximum deviation is no more than 1.09%. It can be concluded that these data is basically convergent, and thus for a sandwich panel with the honeycomb rib length of 3 mm, there is no significant difference among the analysis results of precise model with element size from 3 to 9 mm.

However, the smaller the element size, the more nodes there will be in the model, which consumes more computing resources and solution time. On the contrary, the bigger the element size, the more coarse the model will be, which leads to the lack of much useful information in the interpo-

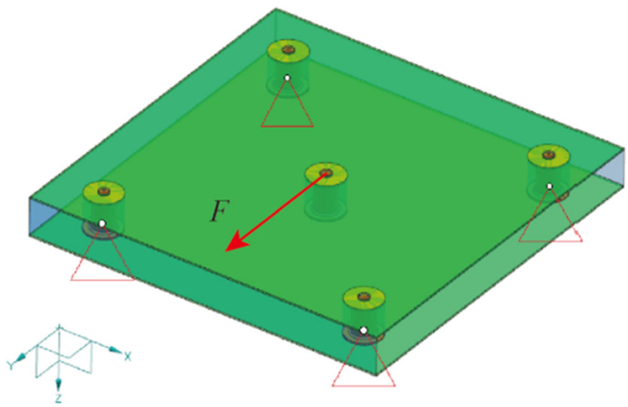


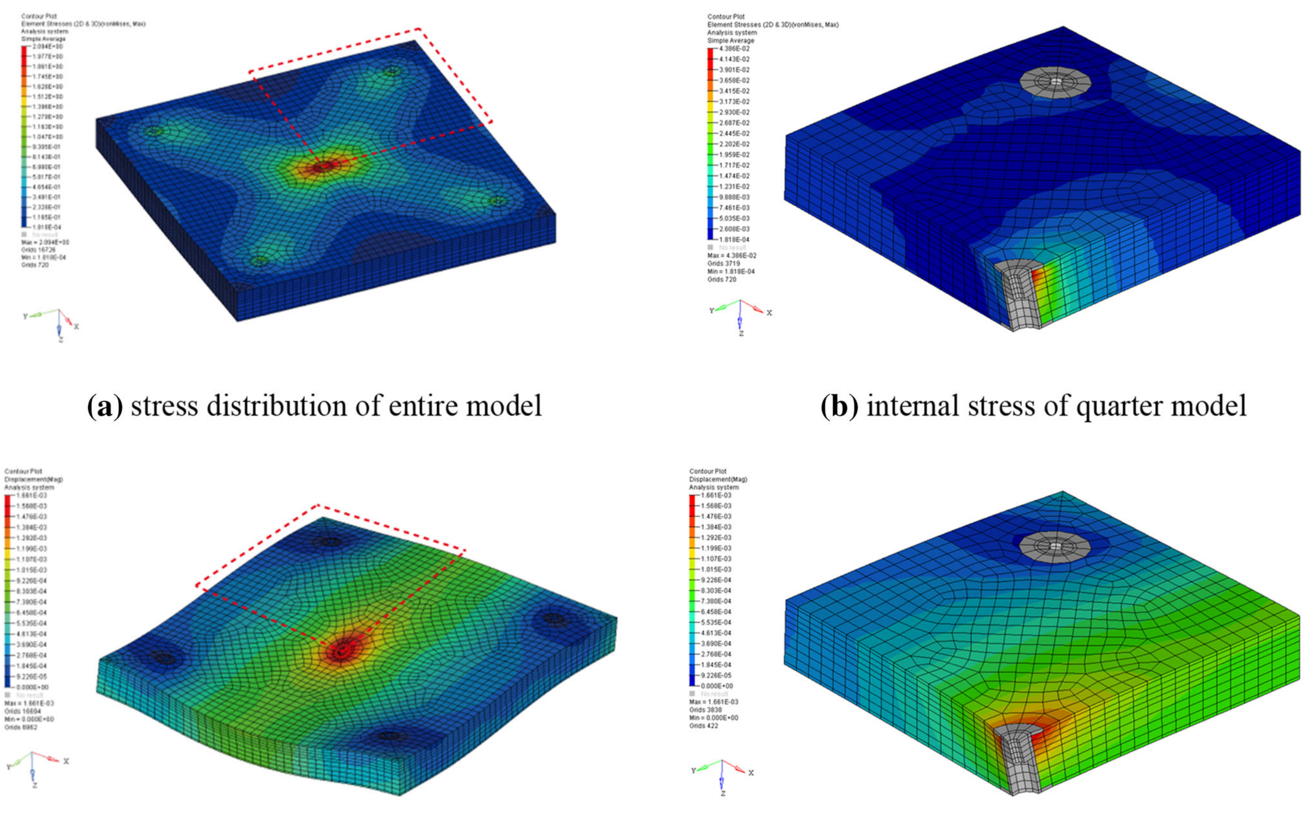
Fig. 6 Boundary conditions and typical loadcase of a conventional honeycomb panel

lation calculation of finite element analysis. Therefore, the moderate 6 mm model was adopted in the example, and the recommended element size of the precise model in this paper is twice the length of honeycomb rib.

It must be emphasized that the mesh quality must be guaranteed in these calculations, otherwise abnormal data will appear and destroy the above range. This requirement can be realized using the quality check function of the software. By optimizing the meshing of elements, the quality indexes such as aspect ratio, warpage, skew, and Jacobian value of all shell elements must comply with the element check criterion of RADIOSS/OptiStruct, which is built into the software.

2.5 Demonstration of Stress and Deformation

A square conventional honeycomb panel with thickness of 20 mm and side length of 200 mm is shown in Fig. 6, of which the parameters of sandwich structure are consistent with the example in the former section. There are four inserts on each corner and one insert in the center of the panel. The equivalent modeling method proposed in this paper was used to model the panel, and the bonding position between inserts and honeycomb sandwich structure was simulated by node fitting. Assume a simplified typical loadcase: the honeycomb panel is placed vertically, the inserts at four corners are fixed, and a weight of 10 kg (converted into a force along Y axis direction, marked as F in Fig. 6) is mounted on the central insert. Under the above conditions, the stress and deformation of the entire honeycomb sandwich structure were analyzed, as shown in Fig. 7.



(a) stress distribution of entire model

(b) internal stress of quarter model

(c) deformation distribution of entire model

(d) local deformation of quarter model

Fig. 7 Stress and deformation of the honeycomb panel under typical loadcase

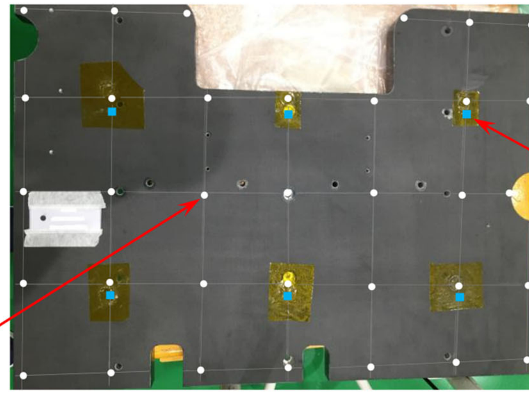


Fig. 8 Modal tapping test on the payload mounting panel

Table 7 Equivalent elastic parameters of interlayer in the payload mounting panel

E_1 (MPa)	E_2 (MPa)	E_3 (MPa)	G_{12} (MPa)	G_{23} (MPa)	G_{31} (MPa)	μ_{12}	μ_{23}	μ_{31}	ρ_c (g cm^{-3})
0.035	0.035	646.63	0.021	135.10	90.07	0.99986	0.00002	0.33	0.0259

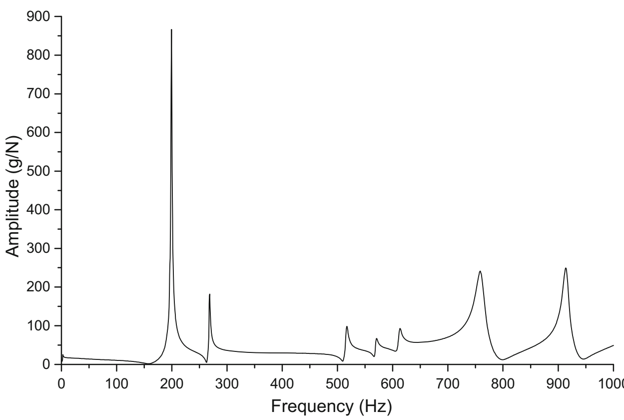


Fig. 9 Dynamic information of the payload mounting panel obtained in the modal tapping test

To distinguish the inserts from the honeycomb sandwich structure, the inserts are not color-mapped in the quarter model of Fig. 7. It can be seen from Fig. 7b, d that the stress distribution in the honeycomb sandwich structure varies along the normal direction, and the deformations of upper and lower surfaces of the panel are obviously different. With these results, the detailed internal structure of honeycomb panels, including inserts, can be improved and optimized. However, in the common two-dimensional model as shown in Fig. 2a, as there is only one layer of shell elements, the deformation difference between the upper and lower surfaces and the internal stress information cannot be distinguished.

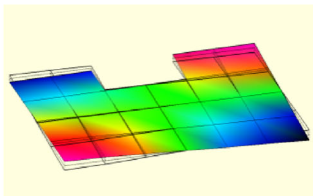
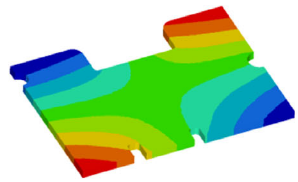
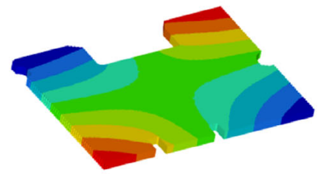
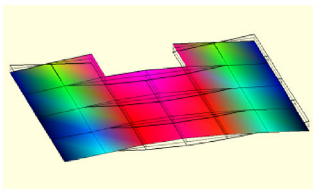
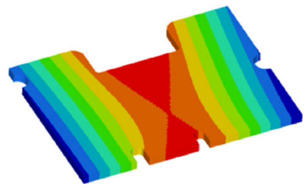
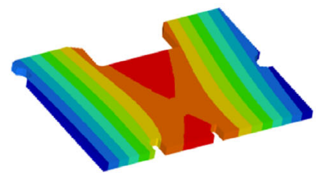
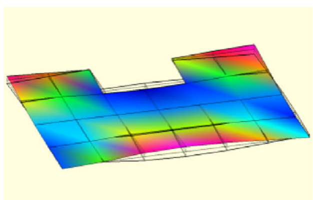
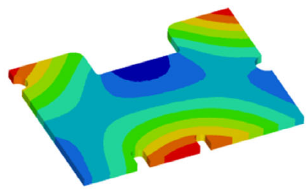
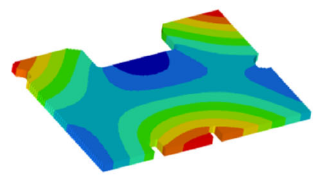
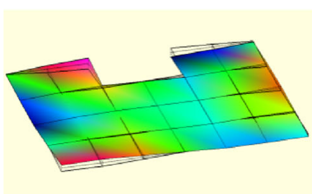
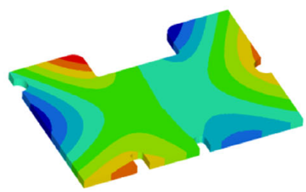
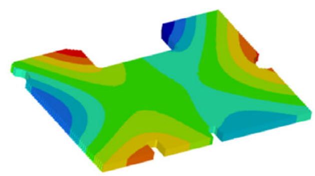
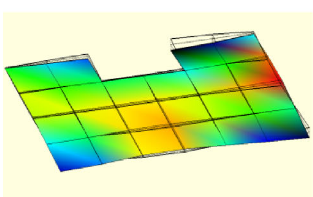
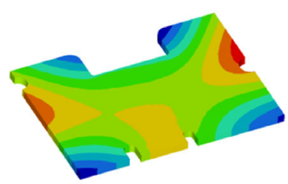
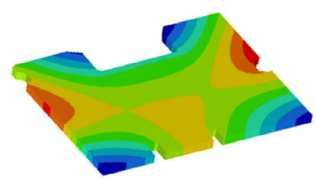
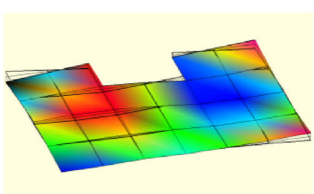
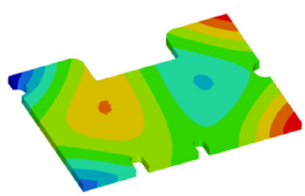
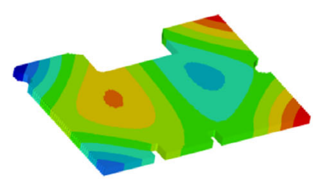
3 Test Verification

The payload mounting panel used in a satellite platform is made of honeycomb panel with the size of 610 mm × 444 mm and the total thickness of 25 mm. The honeycomb core is made of regular hexagonal aluminum honeycomb with the wall thickness of 0.03 mm and the rib length of 5 mm. The upper and lower panels are made of CFRP with the thickness of 0.8 mm. The physical properties of each material are the same as those in Table 1. In order to further verify the accuracy of the equivalent modeling method in this paper, the modal tapping test was carried out on this payload mounting panel, and the experiment data was compared with the free mode analysis data obtained by the equivalent modeling method in this paper.

Table 8 First six order frequencies of the payload mounting panel (simulation A: modeling method proposed in this paper; simulation B: precise modeling method)

Orders	1	2	3	4	5	6
Test (Hz)	199.739	268.375	516.253	569.623	612.115	763.808
Simulation A (Hz)	197.511	275.831	490.877	564.093	599.224	776.016
Simulation B (Hz)	200.900	276.527	477.343	558.537	579.329	767.317
Relative error between test and simulation A (%)	− 1.12	2.78	− 4.92	− 0.97	− 2.11	1.60
Relative error between test and simulation B (%)	0.58	3.04	− 7.54	− 1.95	− 5.36	0.46

Table 9 Comparison of mode shape between test results and simulations (simulation A: modeling method proposed in this paper; simulation B: precise modeling method)

Orders	Test	Simulation A	Simulation B
1			
2			
3			
4			
5			
6			

During the test, the honeycomb panel was suspended in the air by a nylon rope. Six acceleration sensors were attached to the component. Specific positions on the honeycomb panel was struck by rubber hammer, and the response data of these sensors were collected. A total of 34 striking points were evenly distributed on the surface of the component, and the distribution of sensors and striking points is shown in Fig. 8.

According to the structure parameters of the payload mounting panel and equations mentioned above, the physical properties of the interlayer, equivalent to honeycomb core, were calculated as shown in Table 7. Then the finite element models of the payload mounting panel were established using the equivalent modeling method proposed in this paper and the precise modeling method, respectively, and the free mode analysis was carried out. In order to ensure the accuracy of the analysis, with the reference to the conclusion of convergence test, a uniform element size of 10 mm was chosen for both finite element models, which is twice the length of honeycomb rib in the tested payload mounting panel.

The frequency information of the payload mounting panel within 1000 Hz obtained in the modal tapping test is shown in Fig. 9. The comparison of the first six order frequencies and mode shapes between the test results and the finite element analyses using different methods is shown in Tables 8 and 9. Based on the frequency response of each sensor position collected during the test while tapping at different striking points, the mode shape information of each order of the payload mounting panel can be fitted by the test instrument automatically, as shown in the first column of Table 9. All the color-mappings in Table 9 represent the displacement component along the normal direction of the honeycomb panel.

According to the data in Table 8, it can be seen that with the equivalent modeling method proposed in this paper, the maximum error of the first six order frequencies of the payload mounting panel is not more than 5% (4.92%, the third order frequency), and the others are no more than 3%, while the maximum error reached 7.54% using the precise model. By comparing the color-mappings in Table 9, the first four modes and the sixth mode of the payload mounting panel fitted by the test instrument are consistent with the finite element analysis, and the fifth mode is more complex, but the finite element analysis results can also basically represent the deformation trend of the component. The experimental data further proves that the equivalent modeling method in this paper can accurately represent the structural characteristics of honeycomb sandwich structure, and the analysis accuracy can meet the engineering requirements.

4 Conclusion

In the equivalent modeling of honeycomb sandwich structure, the honeycomb core is equivalent to the interlayer and endowed with orthogonal anisotropic material properties, the expressions of nine independent elastic parameters of the interlayer are given, solid elements and shell elements are used to represent the interlayer and the panels, respectively, in the finite element model. The introduction of interlayer makes the finite element model well represent the internal stress and local deformation of honeycomb sandwich structure, and provide more accurate simulation results.

For the honeycomb panel in the example, compared with the results of precise model, the maximum relative errors of displacement under static loadcases and first six order frequencies are 3.12% and 4.07%, respectively, using the equivalent model, which shows that the analysis accuracy of the equivalent modeling method meets the engineering requirements. In the modal tapping test of the payload mounting panel on a satellite platform, the relative errors of the first six order frequencies measured are less than 5% compared with the analysis data. The experimental data are in agreement with the analysis results, which further verifies the validity of this modeling method. The equivalent modeling method proposed in this paper is of reference significance for the modeling of honeycomb sandwich structure in engineering analysis, and meanwhile can be used to analyze the stress and deformation of honeycomb sandwich structure, so as to guide the detailed design of such structural components.

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