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THE INVESTIGATION FOR ACOUSTIC FATIGUE ON COMPOSITE SOLAR PANEL

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In the launch process, the spacecraft has been subjected to huge sound excitation. The investigation on the fatigue behaviour and its evolution law of spacecraft under sound excitation is very significant to ensure the safety of the spacecraft in the launch. In this paper, the acoustic fatigue of composite solar panel is analysed and investigated. Finite Element Method (FEM) and Boundary Element Methods (BEM) are used to establish numerical analysis models for acoustic response in composite solar panel. The numerical analysis model is verified and validated by acoustic test of the solar panel in reverberation chamber. Furthermore, the fatigue life of composite solar panel under acoustic excitation is calculated and predicted. The results show that the canter location in composite solar panel has the shortest fatigue life due to high stress level caused by several vibro-acoustic coupling modes. This research provides an analysis and prediction method to acoustic fatigue life of some key components in spacecraft such as satellites, solar panel and others.

Keywords: EF/BEM, acoustic response, fatigue life

1. Introduction

In spacecraft such as satellites, the solar panel is always subject to wideband noise at launching. Frequency range covers from 20Hz to 8000Hz and sound pressure level (SPL) is over 140dB or more. Noise loads will lead to dynamic stress on thin-walled structures of spacecraft. Furthermore, the dynamic stress will cause fatigue damage on spacecraft [1-2].For example, joints or skins of spacecraft could be broken, which affects the safety and service life of the spacecraft seriously. Therefore, investigation of fatigue behaviour for spacecraft under acoustic excitation and its evolution becomes a key point in its structural design and experimental verification.

In general, acoustic fatigue analysis for thin-walled panels, diaphragm-wall structures in aerospace is difficult. In most cases, it depends on experimental analysis. Usually, modal frequency and stress are developed by using the Empirical formula, and then fatigue life can be calculated with monogram or a formula, such as widely used DSR (Detail Sonic Rating) method [3]. Fast-DSR method is used by Boeing to predict the fatigue life of parts that consist of skin-web and honeycomb core [4]. Liu [5] estimated the fatigue life of Titanium alloy plate using new method that combines finite element method with DSR and the result was proved to be more accurate. However, DSR method has some limitations. Some parameters in DSR are based on experimental results. At the same time, it costs a lot.

In recent years, numerical analysis method is used in the fatigue analysis of the complex spacecraft structure widely because of its low cost and high efficiency. The numerical fatigue analysis can be carried out in time domain and frequency domain [6-8]. The time domain method is used to simulate the response of stochastic process at first, and then the magnitude, mean value and the probability distribution of stress are obtained from the time-domain stress response curve using classical “Rain-flow Counting "method. Time domain method is usually able to get good cumulative fatigue damage. However, it needs a sufficiently long time signal. Bai [9] used this method and proposed two kinds of algorithms: power spectrum input and time domain input and estimated the fatigue life of T-shaped plate under white noise excitation. The algorithm feasibility was verified. The frequency domain method is based on the power spectral density (PSD) analysis, and data processing is simple compared with time domain method. The PSD function contains the most important parameters to describe the stationary ergodic processes. The PSD function can be used to obtain the frequency, peak probability distribution and root- mean-square (RMS) value of stochastic stress signal, and then the structural fatigue life is calculated with S-N curve. Sha [10] proposed a new stochastic fatigue life prediction method that is based on stress probability density and power spectral density method, and used it to estimate the fatigue life of aircraft engine. Based on the definition of rain circulation and Dirlik empirical formula, Bishop [11] gave a method of fatigue analysis method that based on power spectral density, which is of great reference value for engineering application. All these cases show that fatigue analysis in the frequency domain presents strong engineering applicability and high efficiency.

The spacecraft, like solar panel, mainly consists of carbon fibre surface and aluminium honeycomb core. The core layer is hexagonal aluminium honeycomb, and the surface panel is generally made of carbon-fibre composites. To meet the needs of different regions of the strength and rigidity, different areas of the panel have different number of layers and laying direction, resulting in a complex non-continuous layer. The complex structural design of the solar panel meets the requirements of lightweight and high strength, but it also brings great difficulty for accurate response calculation and fatigue life analysis. Yang Jiang [12] established a solar panel simulation model based on the FE-SEA (SEA, statistical energy analysis method) hybrid method and the coupled FE / BEM. The acceleration response in the solar panel is obtained. The simulation result corresponds to the noise test result. Liu [13] established a finite element model for a honeycomb sandwich structure in antenna cover. Some typical structures and their boundary conditions are investigated through a series of tests. The result shows that the honeycomb core is the key element to fatigue failure. This method supplies a high accuracy for fatigue prediction. However, its cost is also relatively high. It is still a difficult work to predict fatigue life of composite structures.

In this paper, sandwich panel equivalent theory is used to simplify solar panel that consists of carbon fibre skins and aluminium honeycomb core. The FE/BEM model is established in order to analyse acoustic fatigue life of solar panel. The proposed fatigue life prediction method for solar panels will help engineers to understand if their design is successful and evaluate if their design can suffer from severe noise excitation during launch.

2. Mechanical equivalent of composite solar panel

2.1 Sandwich Equivalent Theory

Common equivalent methods of composite honeycomb sandwich panel are equivalent plate theory, theory of honeycomb panels, sandwich plate theory. Hu [14] carried on simulation and verification using above three methods. The results show that mechanical properties and vibration responses obtained by sandwich plate equivalent theory are closest to the real value. That means that sandwich plate theory is the most accurate among them. According to sandwich plate theory, aluminium honeycomb sandwich panel is composed of upper and lower isotropic surface layer and middle anisotropy core，then the sandwich panel can be analysed by finite element method

The theory assumes that the core layer can resist lateral shear deformation and has certain in-plane stiffness. The upper and lower skin layers obey the Kirchhoff hypothesis. The honeycomb core layer can be equivalent to a homogeneous layer. The thickness of the orthotropic layer in the middle unchanged when their ability to resist lateral shear stresses is ignored. Thus the model can reflect the structure of sandwich plate.

2.2 Equivalent process

In this paper, solar panel is composed of two skin layers of carbon fibre with honeycomb aluminium inside. According to the sandwich equivalent theory, the core layer is equivalent to an orthotropic material, and its geometric parameters remain unchanged, while the upper and lower panel parameters unchanged, as shown at Fig.1.

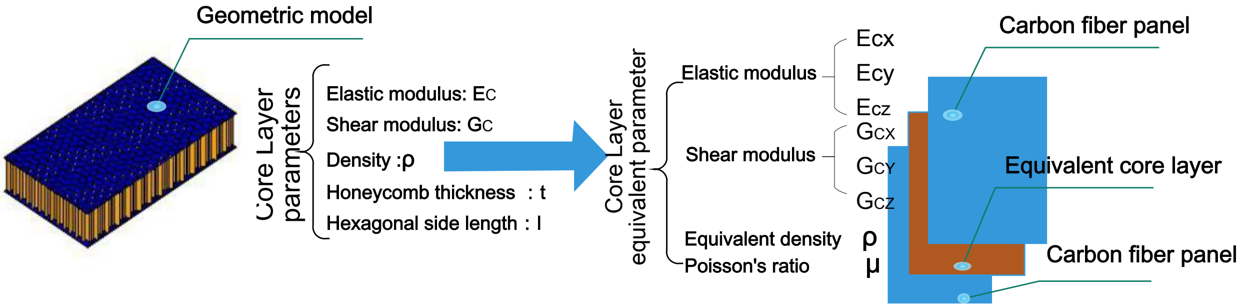


Figure 2: Equivalent schematic diagram in sandwich solar panel

For a hexagonal cell, the elastic constant of the equivalent honeycomb plate are expressed as follows:

$$E_{cx} = E_{cy} = \frac{4}{\sqrt{3}} E_c (1 - 3\frac{t^2}{l^2}) \frac{t^3}{l^3} \quad (1), \qquad E_{cz} = \frac{8\sqrt{3}}{9} E_c \frac{t}{l}. \quad (2), \qquad G_{cxy} = \frac{4\sqrt{3}}{5} E_c (1 - \frac{12}{5}\frac{t^2}{l^2}) \frac{t^3}{l^3}. \quad (3)$$

$$G_{cxz} = \frac{\sqrt{3}}{3} G_c \frac{t}{l}. \quad (4), \qquad G_{cyz} = \frac{2\sqrt{3}}{3} G_c \frac{t}{l}. \quad (5), \qquad \rho = 1.54 \rho_c \frac{t}{l}. \quad (6)$$

where: ρ_c , E_c , G_c are core material density, elastic modulus and shear modulus respectively. E_c represents the elastic modulus of honeycomb core, and subscript *x/y/z* mean three axial directions. *z* direction perpendicular to the honeycomb core layer plane, and the shear modulus subscript has the same meaning. *"l"* is the length of the core layer and *"t"* the thickness of the hexagon. The equivalent core layer parameters are shown in Table 1.

Table 1: Equivalent parameters of the core in sandwich solar panel

Equivalent parameters	Value	Equivalent parameters	Value
$E_{cx} = E_{cy}$	0.15 MP	G_{cxy}	0.09 MP
E_{cz}	1000 MP	G_{cyz}	153 MP
ρ	37.7 kg/m³	G_{cxz}	306 MP

3. FE/BEM model of composite solar panel

3.1 FE/BEM model

According to above geometry model, FE / BEM model of solar panel was established. The coupling matrix of acoustic boundary element and structural finite element can be established:

$$\begin{bmatrix} [K]+i\omega[C]-\omega^2[M] & [L]^{\mathrm{T}} \\ -\rho_0\omega^2[L] & [D] \end{bmatrix} \begin{Bmatrix} \{u\} \\ \{q\} \end{Bmatrix} = \begin{Bmatrix} \{F_{\mathrm{v}}\} \\ \{F_{\mathrm{a}}\} \end{Bmatrix} \tag{7}$$

where:[K],[C],[M] correspond to stiffness matrix, damping matrix and mass matrix; $\{q\}$ is node's pressure vector; $\{u\}$ is node's displacement; $\{F_{\mathrm{v}}\}$ is Load vector of the structure; $\{F_{\mathrm{a}}\}$ is the load vector of the fluid; $[L]$ is acoustic boundary element and structural finite element coupling matrix.

As mentioned before, the solar panel is regarded as a laminated structure of different materials, and the honeycomb core can be simplified as an orthotropic material. The finite element model of the solar panel was established by using the laminated layer structure method. The different colour represent different laying angles and thicknesses. As shown at Fig.2, the size of the grid is determined according to the different ply areas and load conditions, the total number of units is 5245. For acoustic fatigue analysis, the boundary condition is free and the five reference points A1-A5 are exactly the same as the experimental condition.

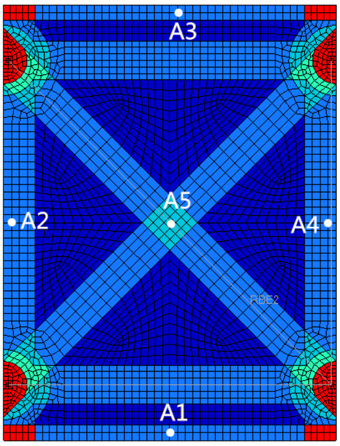


Figure 2: Finite element model of composite solar panel

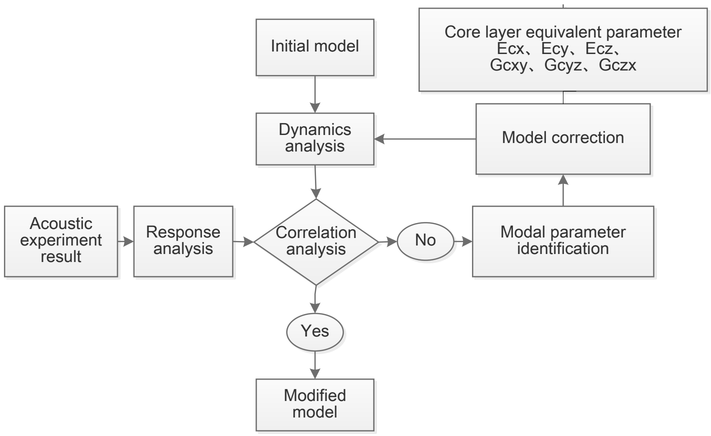


Figure 3: Model verification chart

3.2 FE model verification

In order to improve the accuracy of model analysis in the solar panel under acoustic excitation, finite element model for solar panel is verified by acoustic test. The process of model verification is shown in Fig3.

Model verification process can be written as follows:

1) FE / BEM was used to establish the initial model of solar panel, and the stochastic acoustics response analysis was carried out to obtain the response PSD and dynamic characteristics.

2) MAC (Modal Assurance Criterion) is used to evaluate the consistency of the FE analysis results with the experimental results.

$$MAC_{ij} = \frac{\left|\phi_{mi}{}^{\mathrm{T}}\phi_{aj}\right|^2}{(\phi_{aj}{}^{\mathrm{T}}\phi_{aj})(\phi_{mi}{}^{\mathrm{T}}\phi_{mi})}. \tag{8}$$

where ϕ_{mi} denotes the i-th test modal vector; ϕ_{aj} denotes the j-th modal vector. MAC value is between [0 1], and correlations between two modals is higher while MAC is closer to 1. If MAC> 0.90, that is qualified, otherwise the model parameter needs to be corrected.

3) when correlation analysis is unqualified, the frequency domain modal parameters are identified from simulation data, and the modal main mode which mainly contributes to the structural response, is determined. It is the main reference factor.

4) Model parameters are modified to minimize the deviation of response between the simulation model and the real model under noise excitation. The correction parameters should be among the equivalent parameter of core layer: E_{cx}, E_{cy}, E_{cz}, G_{cxz}, G_{cyz} and G_{cxy}. The inverse modal sensitivity

was used to determine the major corrections affecting the structural modality. The response deviation includes the peak and frequency

5) Once model is corrected, it is return to step 1) until the relevant analysis meets the requirements

After model is modified and verified. Simulation results are basically corresponding with the experimental results. Considering structural symmetry, A2 and A5 are analyzed, as shown in Fig.4. The results show that the peak and the peak frequency obtained by simulation basically correspond to those obtained by experiment. The simulated values coincide well with the experimental values at middle and low frequency bands while the simulation value is slightly larger than the experimental value in the high frequency range. The main reason is that the accuracy of FE/BEM is decreased due to the large modal density of solar panel at high frequency.

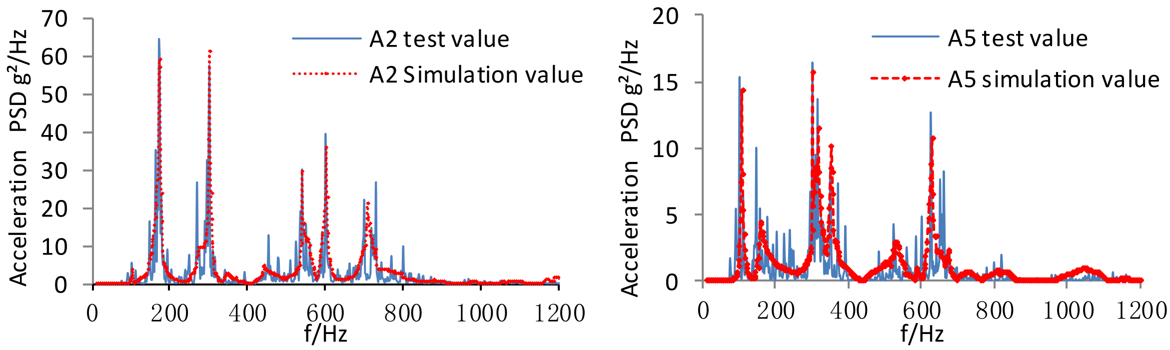


Figure 4: Comparison between experimental and numerical acceleration responses

4. Acoustic fatigue analysis

4.1 Structural response under acoustic excitation

The modal superposition method is used to calculate the structural response of composite solar panel under acoustic excitation. The local stress under acoustic excitation can be calculated by Eq. (9).

$$\sigma_{i,j}(xt) = \sum\nolimits_{K} C_{i,j}^{(k)}(x) L_k(t). \tag{9}$$

where: $C_{i,j}^{(k)}(x)$ denotes the modal stress for a location, $L_k(t)$ denotes Modal Participation Factor.

In acoustic simulation, 24 plane waves were used to simulate the reverberant sound source, as shown in Fig.5. Table 2 is sound pressure spectrum in experiment. The solar panel is in a free state.

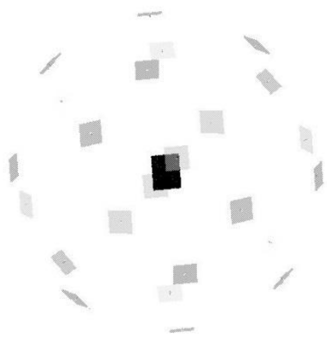
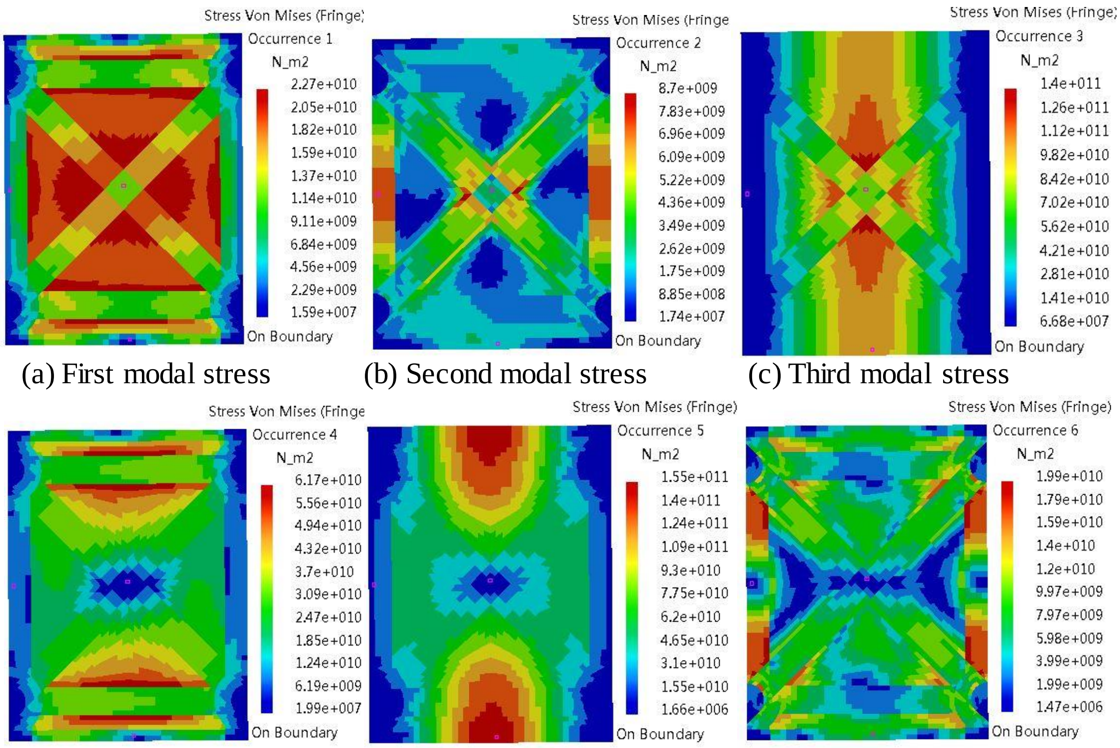


Figure 5: Reverberation sound source

Table 2: Sound pressure spectrum in test

Center frequency of octave bandwidth /Hz	SPL/dB	Deviation allowed/dB
31.5	118	±5
63	131	±3
125	134.5	
250	135	
500	133.5	
1000	127	
2000	122	
Total SPL	140	±1.5
Excitation time/min	1	—

Fig.6 shows the modal stress distribution of solar panel obtained by modal analysis. According to Eq. (9), the stress response of solar panel under noise excitation can be calculated.



(a) First modal stress (b) Second modal stress (c) Third modal stress

(d) Fourth modal stress (e) Fifth modal stress (f) Sixth modal stress

Figure 6: Modal stress distribution of the first six modes

4.2 Fatigue damage theory

Before fatigue failure occurred, solar panel is usually subject to periodic load, which may be constant amplitude or variable amplitude. Amplitude is difficult to measure. The time history of the stress and strain cycles is counted, and the complex variable amplitude load history is simplified into a set of discrete simple constant amplitude loading processes. Rain flow counting is considered to be a good method to fatigue life analysis. For complex random loads such as noise randomization, rain flow counting can identify events similar to those of constant amplitude fatigue data in complex load sequences, and filter out events with smaller stress magnitude.

In this paper, Miner linear accumulation damage theory is used to analyse the fatigue life of solar panel. According to Miner's linear damage assumption, the damage caused by each stress cycle can be superimposed linearly. The stress is independent each other, regarding of the loading order. When the damage accumulates to a certain critical value, fatigue failure of the specimen occurs. For the *i*-th load that repeated n_i times, the fatigue damage caused is D_i :

$$D_i = \frac{n_i}{N_i}. \tag{10}$$

The fatigue damage caused by cumulative loading at all levels can be expressed as :

$$D = \sum D_i = \sum_{i=1}^{n} \frac{n_i}{N_i}. \tag{11}$$

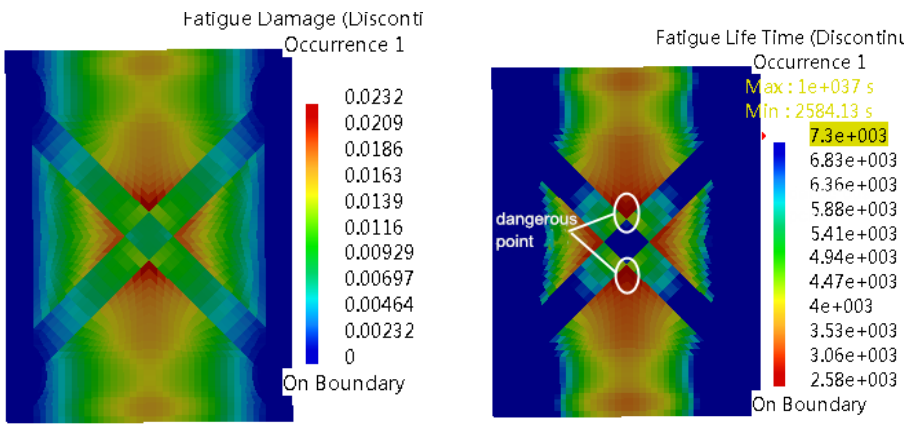
The fatigue life T is equal to :

$$T = \frac{D_{CR}}{D} t. \tag{12}$$

where: N_i means the number of cycles when the i-level fatigue failure occurs, generally obtained by the experiment; D_{CR} is fatigue damage threshold, for most of time, it is one.

4.3 Results

After subjected to noise excitation for 60s, fatigue damage and fatigue life of solar panel are shown in Fig. 7. Fatigue damage distribution shows symmetry along the long axis of the panel due to structural symmetry in solar panel and uniform distribution of sound source. The fatigue analysis result also indicates that central region of the solar panel is the most dangerous. The stress of the dangerous point in the multi-order modes is always at a high level, especially obvious for second-order and third-order modes, as shown in Fig8.



(a) fatigue damage (b) fatigue life time

Figure 7: Random acoustic fatigue for solar panel under 60s

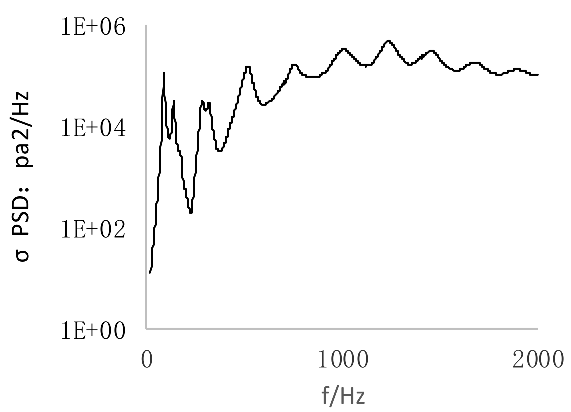


Figure 8: Power spectral density of stress in dangerous location

It is seen that the maximum damage rate is 0.0232 at the fatigue risk point and the shortest fatigue life is $2.58*10^3$ s. No fatigue failure appeared in solar panel.

5. Conclusion

Based on sandwich equivalent plate theory, FE / BEM model of a composite solar panel was established. Compared with noise test, uncertain parameters in FE/BEM model are modified and verified. Furthermore, fatigue life of solar panel was analysed by means of miner linear accumulation damage theory and rain flow counting method. In this research, some important conclusions are summarized:

1) The sandwich theory is used to simplify solar panel with carbon fibre skin and honeycomb aluminium core. It is used to reduce computation cost and improve analysis efficiency.
2) According to noise test results, numerical model was validated by frequency response date. Some equivalent parameter in honeycomb core was modified to improve model prediction precision.
3) The fatigue risk of solar panel appears in the central region which is at high stress level due to multi-level modal stress, especially under second-order and third-order modes.

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