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Vibration in a satellite structure with a laminate composite hybrid sandwich panel

H.K. Cho [a,1], J. Rhee [b,*]

[a] Satellite Technology Research Center, Korea Advanced Institute of Science and Technology, 335 Gwahakro, Yuseong-gu, Daejeon 305-701, Republic of Korea
[b] Korea Aerospace Research Institute, 115 Gwahakro, Yuseong-gu, Daejeon 305-333, Republic of Korea

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ABSTRACT

The objective of this investigation is to study the complex vibration characteristics of an actual spacecraft structure using the FEA code in conjunction with experimental data. The body of a satellite consists of a monocoque structure formed by joining several composite sandwich panels composed of an aluminum honeycomb core with carbon fiber reinforced laminate skins on both sides.

An extensive random vibration test campaign was conducted on the satellite structure using an electro-dynamic shaker, and its results were compared with the FEA values. After this campaign, a database was established which correlates computational analysis modeling schemes with random acceleration spectra responses on selected locations obtained from the tests. These results can be successfully applied as reference data when a new satellite is developed and can provide excellent criteria in the vibration-proof design and analysis of a satellite. This paper demonstrates a modeling technique and discusses the applications of numerical analysis theories. It also gives the results achieved from the advanced vibration tests.

1. Introduction

Composite materials are commonly applied to spacecraft parts due to their high strength, feasible stiffness-to-weight ratios, and low thermal expansion coefficient. STSAT-III is Korea's first whole structure composite satellite. The structure of the earlier I and II models of the Science and Technology Satellite program of Korea consisted of an aluminum frame and aluminum sandwich panels. Despite the fact that the traditional aluminum honeycomb sandwich panel is the most commonly used panel type in satellite structures, the use of lightweight composite material is increasing. All or most parts of the structures of several well-known small satellites, including Forte [1], Mighty Sat [2], Proba [3] and Wire [4] were made of composite materials. Due advantages such as their light weight, high strength and durability, structural stability upon temperature variations when used in conjunction with metals, these composites are especially suitable for aerospace structures.

As their operating characteristics, satellites experience several types of mechanical, thermal, and electromagnetic disturbances during their development, manufacturing, and launch to their final operating position in space. Among them, vibration must be carefully considered. A number of studies on the subject of satellite vibration have been published; however, most of the studies focused on the launcher, electronic boxes or on a modal analysis of the entire structure.

In the present study, the dynamic responses of the entire structure as caused by random excitation were obtained by both finite element analysis and vibration testing. Moreover, random vibration theories that calculate the physical responses in terms of the frequency field and the time domain are introduced and the accuracy of the numerical random vibration spectrum analysis and FEA modeling techniques of the complex satellite structure are verified through a comparison with test data. Additionally, the vibration test profile defined in the frequency domain is transformed into an equivalent transient analysis input function based on the frequency-time transfer function theory suggested by Rice [5] and Engelhardt [6]. This method is effective to the point that it enables one to obtain the various dynamic responses of an actual vibration test through a numerical simulation. However, most previous studies did not attempt this. With this technique, the stress and displacement fields of a complex composite satellite structure can be determined.

2. Finite element analysis of composite structures

2.1. Mechanics of the composite material

The structure of the STSAT-III is made of sandwich panels having aluminum honeycomb core and composite laminate skins on both sides. For the finite element analysis, the sandwich panel is modeled with an eight-node composite laminate degenerated shell element. The stiffness matrix is calculated using Eqs. (1)–(4). Unlike the plate element, the shell element is able to represent curvature deflections and compute such values as the shear stress and

* Corresponding author. Tel.: +82 42 860 2443; fax: +82 42 860 2603.
E-mail addresses: marklee1@hanmail.net (H.K. Cho), jrhee@kari.re.kr (J. Rhee).
[1] Tel.: +82 42 350 8617; fax: +82 42 861 0064.

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bending stress. The material stiffness matrix of the shell element on a local coordinate system is written as

$$[C'] = \begin{bmatrix} \bar{C}_{11} & \bar{C}_{12} & 0 & \bar{C}_{14} & 0 & 0 \\ \bar{C}_{21} & \bar{C}_{22} & 0 & \bar{C}_{24} & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ \bar{C}_{41} & \bar{C}_{42} & 0 & \bar{C}_{44} & 0 & 0 \\ 0 & 0 & 0 & 0 & \bar{C}_{55} & \bar{C}_{56} \\ 0 & 0 & 0 & 0 & \bar{C}_{65} & \bar{C}_{66} \end{bmatrix} \quad (1)$$

where the material stiffness matrix $[C']$ refers to the local coordinate axis of an individual element. Individual entries of Eq. (1) are

$$\begin{aligned} \bar{C}_{11} &= m^4 Q_{11} + 2m^2n^2(Q_{12} + 2Q_{44}) + n^4 Q_{22} \\ \bar{C}_{12} &= m^2n^2(Q_{11} + Q_{22} - 4Q_{44}) + (m^4 + n^4)Q_{12} \\ \bar{C}_{14} &= mn(m^2Q_{11} - n^2Q_{22} - (m^2 - n^2)(Q_{12} + 2Q_{44})) \\ \bar{C}_{22} &= n^4Q_{11} + 2m^2n^2(Q_{12} + 2Q_{44}) + m^4Q_{22} \\ \bar{C}_{24} &= mn(n^2Q_{11} - m^2Q_{22} + (m^2 - n^2)(Q_{12} + 2Q_{44})) \\ \bar{C}_{44} &= m^2n^2(Q_{11} + Q_{22} - 2Q_{12}) + (m^2 - n^2)^2Q_{44} \\ \bar{C}_{55} &= m^2Q_{55} + n^2Q_{66} \\ \bar{C}_{56} &= mn(Q_{66} - Q_{55}) \\ \bar{C}_{66} &= m^2Q_{66} + n^2Q_{55} \end{aligned} \quad (2)$$

where $m = \cos\theta_{(j)}$, $m = \sin\theta_{(j)}$, angle θ is measured from the local x-axis to the principal material axis, and j denotes a typical ply of the laminate. When the principal material axes coincide with global coordinate axes, Q_s can be written in terms of material engineering quantities; i.e.,

$$\begin{aligned} Q_{11} &= \frac{E_{11}}{1 - v_{12}v_{21}} Q_{22} = \frac{E_{22}}{1 - v_{12}v_{21}} \\ Q_{12} &= \frac{v_{12}E_{22}}{1 - v_{12}v_{21}} = \frac{v_{21}E_{11}}{1 - v_{12}v_{21}} \\ Q_{44} &= G_{12}, Q_{55} = G_{23}K_p, Q_{66} = G_{13}K_p \end{aligned} \quad (3)$$

such that E and v are the Young's modulus and Poisson ratio, respectively. K_p is that value known as the "shear factor", which compensates for transforming a parabolic shear-stress distribution through the thickness into a constant stress profile. For an evaluation of the element stiffness, the material matrix in terms of local coordinates must be transformed to $[C] = [T]^T[C'][T]$ in the global system. Therefore, the stiffness matrix for a modal analysis is written as

$$[K] = \sum_{j=1}^{p} \int_V \{[B][C][B]^T\}(h_{(j)}/h)dV \quad (4)$$

where $h_{(j)}$ represents the jth layer thickness and where the mass matrix is a lumped mass matrix.

2.2. Vibration analysis

In structural dynamic analysis, spectrum analysis, a frequency response analysis, is different from a time-dependent analysis. It is especially effective for irregular random vibration such as seismic excitation and other time-consuming problems. Recently, these probabilistic power spectral density (PSD) random vibration functions were transformed into an equivalent transient vibration analysis input function that enable vibration tests to be simulated with numerical analysis methods to give results that are very close to the those with actual phenomena.

Response spectrum analysis is an efficient alternative to transient dynamic analysis for estimating the maximum response under different types of support excitation. Spectrum analysis essentially obtains solutions through the mode superposition method. The finite element analysis equilibrium equation considering damping is as follows:

$$[M]\{\ddot{x}(t)\} + [C]\{\dot{x}(t)\} + [K]\{x(t)\} = \{f(t)\} \quad (5)$$

By utilizing the basic Eq. (5), the theoretical background of the mode superposition as well as the random and transient analysis can be described. This is done in the following paragraphs [7–10].

2.2.1. Mode superposition

This method separates a structure's dynamic equilibrium equation into a number of independent modes and then calculates the contribution factors of the individual modes to determine the dynamic behavior of the entire structure by synthesizing the contributions. The fundamental concept of this method is mode separation and superposition using the orthogonality of the mode. By representing the displacement $\{x(t)\}$ of Eq. (5) through a multiplication of the generalized displacement $\{z(t)\}$ and mode vector $\{\phi\}$, the assumption is that the entire displacement of the structure can be expressed as the composition of each mode with an appropriate ratio.

$$\{x(t)\} = [\{\phi_1\}\{\phi_2\}\cdots\{\phi_p\}]\{z(t)\} = [\Phi]\{z(t)\} \quad (6)$$

Substituting Eq. (6) into Eq. (5) and rearranging it using mode orthogonality allows it to be divided into a p-number differential equation.

$$\ddot{z}_i(t) + 2\xi_i\omega_i\dot{z}_i(t) + \omega_i^2 z_i(t) = \frac{\phi_i^T f(t)}{\phi_i^T M \phi_i} \quad (7)$$

The mode equation, which is converted to a generalized displacement problem, can be solved with direct integration method as in a single-degree-of-freedom system or as in a Duhamel integral [11] in the time domain analysis. A general solution of Eq. (7) is

$$\begin{aligned} z_i(t) = &\frac{1}{\phi_i^T M \phi_i \omega_{di}} \int_0^t r_i(\tau) e^{-\xi_i\omega_i(t-\tau)} \sin\omega_{di}(t-\tau)d\tau + e^{-\xi_i\omega_i t}\{\alpha_i \\ &\times \sin\omega_{di}t + \beta_i\cos\omega_{di}t\} \end{aligned} \quad (8)$$

where $\omega_{di} = \omega_i\sqrt{1-\xi_i^2}$. The dynamic behavior of the entire structure is then obtained by substituting the solution of Eq. (8), $\{z(t)\}$, into Eq. (6).

2.2.2. Random vibration analysis

Random vibration effects among the satellite's vibration-proof design requirements are considered here because the spacecraft receives irregular vibration that originates from the launcher and from its ground transportation. These irregular vibrations are generally analyzed by the probability access method. PSD-type excitation functions include displacement, velocity, and acceleration, all of which can input excitation. Although there are several methods that can be used when analyzing random vibration, STSAT-III is assumed to be a linear system. Hence, a stationary random vibration analysis was performed. A random signal is stationary if its statistical properties do not change with time. The meal-square value of the random variable $x(t)$ is denoted by $\bar{x}^2$. It is defined as follows:

$$\bar{x}^2 = \lim_{T\to\infty} \frac{1}{T} \int_0^T x^2(t)dt \quad (9)$$

Another measure of interest related to random variables is how fast the value of the variable changes. The autocorrelation function, denoted by

$$R_{xx}(\tau) = \lim_{T\to\infty} \frac{1}{T} \int_0^T x(t)x(t+\tau)dt \quad (10)$$

provides a measure of how fast the signal $x(t)$ changes. The value τ is the time difference between the values at which the signal $x(t)$ is sampled. The Fourier series is useful and relatively straightforward to work with due to the special property of the non-periodic func-