Dynamic Characteristics of Cylindrical Composite Panels with Co-cured and Constrained Viscoelastic Layers*

In LEE**, Il-Kwon OH***, Won-Ho SHIN†, Ki-Dae CHO†† and Kyo-Nam KOO‡‡

Dynamic characteristics of cylindrical composite panels with co-cured and constrained viscoelastic layers are investigated using the finite element method based on the layerwise shell theory. The natural frequencies, loss factors and frequency response functions of cylindrical viscoelastic composites are computed considering the effect of transversely shear deformation and material degradation of viscoelastic layers. Various damping characteristics for unconstrained layer damping, constrained layer damping, and symmetrically co-cured laminates are compared with an original base panel in view of damping efficiency. The layerwise in-plane displacements and shear deformations greatly influence vibration and damping characteristics of the hybrid composite panels. Also, the present results show that it is very important to consider the variation of damping properties according to exciting frequency and environmental temperature for the accurate damping evaluation of cylindrical hybrid composite panels with viscoelastic damping layers.

Key Words: Cylindrical Panel, Viscoelastic Layers, Layerwise Theory, Material Degradation, Dynamic Characteristics

1. Introduction

As the hybrid composite structures with co-cured, unconstrained and constrained damping layers have been used in the aerospace, automotive and electronic products recently, more accurate finite element models have been required to describe the effects of transverse shear mechanism, general ply orientation, thick cross-sectional thickness, various boundary condition and material degradation. The effectiveness of viscoelastic damping materials in forms such as free layer coating or constrained layer treatments has been investigated for many years. There are many previous works for sandwich beams and plates with embedded or co-cured viscoelastic damping material. Also, it is already well known that the surface damping treatments may significantly improve acoustic and dynamic performance of flexible composite structures.

Among the previous studies on the damping mechanisms of composite structures, Saravanan and Pereira developed a finite element based on a discrete layer laminate damping theory to predict the damped dynamic characteristics of specially composite plates with embedded damping layers. Cho et al. investigated vibration and damping characteristics of laminated plates with fully and partially covered damping layers by applying layerwise displacement plate theory. Lee and Kosmatka suggested layerwise zig-zag theory for passively damped vibration of composite plates. The literature survey reveals that a few papers were devoted to investigate the damping capacity of cylindrical composite shells with damping layers. Ramesh and Gaines studied the harmonic response of cylindrical shells with constrained damping.

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ing treatment. Chen and Huang\(^7\) investigated the damping effect of constrained layer damping treatment of strip type on a cylindrical shell. Recently, the research on the active constrained layer damping of composite shells is a current issue of smart structures\(^8\).

Extensive efforts have been put forth to suppress the vibration of composite structures using passive vibration control, like unconstrained and constrained layer damping treatments, as well as active vibration control for various plate panels and beams. However, only a few studies on the damping characteristics for cylindrical composite shell models, neglecting the material degradation dependent of exciting frequency and environmental temperature of viscoelastic layers, are found in the open literature. In this study, an accurate finite element method based on the layerwise shell theory has been developed to investigate damping characteristics of cylindrical hybrid shells considering the effects of transversely shear deformations and viscoelastic material properties dependent on exciting frequency and environmental temperature. The present paper attempts to observe the general tendency of natural frequencies and loss factors of cylindrical hybrid composite panels with various damping treatments, and to explain the dynamic characteristics in view of frequency response functions of constrained layer damping and co-cured sandwich models.

2. Layerwise Finite Element Formulations

2.1 Description of layerwise displacement fields

Based on the layerwise laminate theory, the displacement fields \((u, v, w)\) on the \(x-y-z\) coordinate system shown in Fig. 1, can be expressed by introducing the following piecewise continuous approximations.

\[
\begin{align*}
  u_l &= \sum_{p=1}^{N_l} U_l(x, \phi, t) \Phi_l(z), \\
  v_l &= \sum_{p=1}^{N_l} V_l(x, \phi, t) \Phi_l(z) \\
  w_l &= \sum_{p=1}^{N_l} W_l(x, \phi, t) \Phi_l(z)
\end{align*}
\] (1)

where \(U_l'\) and \(V'_l\) are the in-plane displacements at the \(l\)-th interface; \(N_l\) is the number of degrees of freedom for the in-plane displacement along the thickness direction for element \(l\); \(\Phi_l(z)\) is the Lagrange interpolation function and is assumed to be linear through the thickness direction as given below and is described in Fig. 1.

\[
\begin{align*}
  \Phi_l(z) &= \begin{cases} 
    0 & \text{for } z < z_{l-1} \\
    \frac{z - z_{l-1}}{z_l - z_{l-1}} & \text{for } z_{l-1} < z < z_l \\
    \frac{z - z_{l+1}}{z_l - z_{l+1}} & \text{for } z_l < z < z_{l+1} \\
    0 & \text{for } z_{l+1} < z
  \end{cases}
\end{align*}
\] (2)

The linear relationships between strains and displacements of cylindrical composite shells based on the layer shell theory can be written as follows:

\[
\begin{align*}
  \varepsilon_{xx} &= \frac{\partial u}{\partial x} + \frac{v}{g} \frac{\partial V_l'}{\partial x} \\
  \varepsilon_{yy} &= \frac{\partial v}{\partial y} + w \frac{\partial U_l'}{\partial y} + \frac{W}{g} \\
  \gamma_{xy} &= \frac{\partial u}{\partial y} + \frac{\partial v}{\partial x} + \frac{\partial U_l'}{\partial y} + \frac{\partial V_l'}{\partial x} \\
  \gamma_{xz} &= \frac{\partial w}{\partial x} + \frac{\partial u}{\partial z} + \frac{\partial V_l'}{\partial x} \\
  \gamma_{yz} &= \frac{\partial w}{\partial y} + \frac{\partial v}{\partial z} + \frac{\partial U_l'}{\partial y} \\
  \gamma_{zz} &= -\frac{\partial W}{\partial z} - \frac{\partial V_l'}{\partial z} - \frac{\partial U_l'}{\partial z}
\end{align*}
\] (3)

2.2 Constitutive equations of viscoelastic materials

Mechanical properties of viscoelastic materials are defined by complex modulus of describing hysteric damping dependent of exciting frequencies and environmental temperature.

\[
\begin{align*}
  E_{li}(\omega, T) &= E_{li}(1 + i\eta_i) \quad \text{for } I = 1, 2, 3 \\
  G_{li}(\omega, T) &= G_{li}(1 + i\eta_i) \quad \text{for } I = 1, 2, 3 \\
  G_{li}(\omega, T) &= G_{li}(1 + i\eta_i) \quad \text{for } I = 1, 2, 3
\end{align*}
\] (4)

The linear constitutive equations between stresses and strains can be written as

\[
\begin{pmatrix}
  s_{1} \\
  s_{2} \\
  s_{3} \\
  s_{23} \\
  s_{13} \\
  s_{31} \\
  s_{12}
\end{pmatrix} =
\begin{pmatrix}
  Q_{11} & Q_{12} & Q_{13} & 0 & 0 & 0 & s_{1} \\
  Q_{12} & Q_{22} & Q_{23} & 0 & 0 & 0 & s_{2} \\
  Q_{13} & Q_{23} & Q_{33} & 0 & 0 & 0 & s_{3} \\
  0 & 0 & 0 & Q_{33} & 0 & 0 & s_{23} \\
  0 & 0 & 0 & 0 & Q_{33} & 0 & s_{13} \\
  0 & 0 & 0 & 0 & 0 & Q_{33} & s_{31} \\
  Q_{21} & Q_{31} & Q_{12} & 0 & 0 & 0 & s_{12}
\end{pmatrix}
\] (5)

and

\[
\begin{align*}
  Q_{11} &= (1 - v_{12}v_{23})E_{11} / \Delta, \quad Q_{12} = (v_{12} + v_{23})E_{22} / \Delta, \\
  Q_{13} &= (v_{13} + v_{23})E_{33} / \Delta, \quad Q_{23} = (v_{23} + v_{13})E_{33} / \Delta, \\
  Q_{22} &= (1 - v_{13}v_{32})E_{22} / \Delta, \quad Q_{33} = (1 - v_{13}v_{23})E_{33} / \Delta \\
  Q_{31} &= G_{31}, \quad Q_{32} = G_{32}, \quad Q_{36} = G_{36} \quad \Delta = 1 - v_{12}v_{23} - v_{23}v_{32} - v_{12}v_{32} - 2v_{13}v_{23}v_{32}
\end{align*}
\] (6)

The corresponding constitutive relation for an
anisotropic lamina referred to the initial configuration \(x-\Phi-z\), can be obtained by the coordinate transformation with a fiber angle \(\theta\).

\[
[s]^{\text{ref}}=[Q(\omega, T)][s]^{\text{ref}}
\]

\[-[Q_{0}(\omega, T)+\frac{1}{2}Q_{0}(\omega, T)][s]^{\text{ref}}
\]

\[
(7)
\]

### 2.3 Derivation of Governing Finite Element Equations

In order to derive the governing equation of motion for the cylindrical composite panels with viscoelastic layers, Hamilton's principle was applied in the following form:

\[
\int_{\Omega} \rho \ddot{u} \partial u \, dV + \int_{\Omega} \sigma \ddot{\varepsilon} \, dV = \int_{\Omega} \tau \partial u \, dS
\]

\[
(\delta T) \text{(Kinetic Energy)} - E V \text{(Strain Energy)}
\]

\[
= \int_{\Omega} \tau \partial u \, dS + \int_{\Omega} \sigma \ddot{\varepsilon} \, dV
\]

\[
(8)
\]

Here, infinitesimal surface and volume are given as \(dV=g(\mathbf{r})d\Phi dx dz\) and \(dS=g(\mathbf{r})d\Phi dx dz\), respectively. Over each finite element, the displacements are expressed as a linear combination of shape functions and nodal values in the following form:

\[
(W, U', V') = \sum_{k=1}^{N_k} (W_k, U_k, V_k) \bar{\Phi}_k
\]

\[
(9)
\]

where \(N_P\) is the number of nodes per element. The shape function used here is nine node C\(^0\) Lagrange elements. Let us define the nodal displacement vector for an element \(i\) as

\[
\mathbf{u}_i = \begin{bmatrix} u_i^1 & u_i^2 & \cdots & u_i^{N_v} \end{bmatrix}^T
\]

and

\[
\mathbf{u}' = \begin{bmatrix} W_i & W_{i+1} & \cdots & W_{i+N_{PE}} \end{bmatrix}^T
\]

\[
\mathbf{v}' = \begin{bmatrix} U_i & U_{i+1} & \cdots & U_{i+N_{PE}} \end{bmatrix}^T, J = 1, \cdots, N_i
\]

\[
\mathbf{v}' = \begin{bmatrix} V_i & V_{i+1} & \cdots & V_{i+N_{PE}} \end{bmatrix}^T, J = 1, \cdots, N_i
\]

By using Hamilton's principle and finite elements, the governing finite element equation of motion for the cylindrical composite panel can be obtained in the following form:

\[
M_{\omega} \ddot{u}_e + (K_{\omega}(\omega, T) + iK_{\omega}(\omega, T)) \mathbf{u}_e = F_e(\omega)
\]

\[
(12)
\]

The detailed components of Eq. (12) can be written for mass and stiffness matrices.

\[
M_{\omega} = \int_{\omega} \begin{bmatrix} M_{\omega_{11}} & 0 & \cdots & 0 \\ 0 & M_{\omega_{22}} & \cdots & 0 \\ \vdots & \vdots & \ddots & \vdots \\ 0 & 0 & \cdots & M_{\omega_{N_{\omega}N_{\omega}}} \end{bmatrix} \, d\omega d\Phi
\]

\[
K_{\omega} = \int_{\omega} \begin{bmatrix} K_{\omega_{11}} & K_{\omega_{12}} & \cdots & K_{\omega_{1N_{\omega}}} \\ K_{\omega_{21}} & K_{\omega_{22}} & \cdots & K_{\omega_{2N_{\omega}}} \\ \vdots & \vdots & \ddots & \vdots \\ K_{\omega_{N_{\omega}1}} & K_{\omega_{N_{\omega}2}} & \cdots & K_{\omega_{N_{\omega}N_{\omega}}} \end{bmatrix} \, d\omega d\Phi
\]

\[
(13)
\]

In the infinitesimal surface integral \((dS=g(\mathbf{r})d\Phi dx dz)\), \(g\) is included in the integration in the thickness direction. The detailed elements of above equations are expressed by integration terms of thickness direction, shape function and nonlinear strains. The constitutive elements of mass matrix are given by

\[
M_{\omega_{11}} = h_1^1 h_1^1, \quad M_{\omega_{12}} = h_1^1 h_2^1, \quad M_{\omega_{1N_{\omega}}} = h_1^1 h_{N_{\omega}}^1
\]

\[
(15)
\]

where

\[
\Gamma = \int_{\Omega} \rho g d\omega, \quad \Gamma' = \int_{\Omega} \rho f d\omega
\]

\[
(16)
\]

The sub-components of stiffness matrix are given as follows:

\[
K_{\omega_{11}} = \begin{bmatrix} \frac{1}{2} & \frac{1}{2} & \cdots & \frac{1}{2} \\ \frac{1}{2} & \frac{1}{2} & \cdots & \frac{1}{2} \\ \vdots & \vdots & \ddots & \vdots \\ \frac{1}{2} & \frac{1}{2} & \cdots & \frac{1}{2} \end{bmatrix}
\]

\[
K_{\omega_{12}} = \begin{bmatrix} \frac{1}{2} & \frac{1}{2} & \cdots & \frac{1}{2} \\ \frac{1}{2} & \frac{1}{2} & \cdots & \frac{1}{2} \\ \vdots & \vdots & \ddots & \vdots \\ \frac{1}{2} & \frac{1}{2} & \cdots & \frac{1}{2} \end{bmatrix}
\]

\[
(17)
\]

Here, the thickness integration terms with complex forms are defined as

\[
A_{\omega} = \int_{\omega} \Phi_{\omega} g d\omega, \quad A_{\omega}^* = \int_{\omega} \Phi_{\omega}^* g d\omega,
\]

\[
A_{\omega}^p = \int_{\omega} \Phi_{\omega} g d\omega, \quad B_{\omega} = \int_{\omega} \Phi_{\omega} g d\omega,
\]

\[
B_{\omega}^p = \int_{\omega} \Phi_{\omega} g d\omega, \quad B_{\omega} = \int_{\omega} \Phi_{\omega} g d\omega,
\]

\[
C_{\omega} = \int_{\omega} \Phi_{\omega} g d\omega, \quad C_{\omega} = \int_{\omega} \Phi_{\omega} g d\omega,
\]

\[
D_{\omega} = \int_{\omega} \Phi_{\omega} g d\omega, \quad D_{\omega} = \int_{\omega} \Phi_{\omega} g d\omega,
\]

\[
F_{\omega} = \int_{\omega} \Phi_{\omega} g d\omega,
\]

\[
(18)
\]
2.4 Natural frequency, loss factor and frequency response functions

Through the assembly procedure of finite elements, global finite element equations of cylindrical hybrid composite shells considering the dependences of frequency and temperature can be obtained as:

\[ \{M\dot{U} + (K(\omega, T) + ik\beta(\omega, T))U = F(\omega) \} \]  \hspace{1cm} (19)

Here, \( \omega \) and \( T \) indicate exciting frequency and temperature, respectively. Because it takes much computational time to calculate the dynamic characteristics of free vibration and frequency response functions of the complex system matrices, the modal approach is applied to reduce the order of system matrices. The eigen-system matrices can be written in the following form with proper frequency \( \omega_n \) and temperature \( T \).

\[ (K(\omega_n, T) - \omega_n^2M)\Psi = 0 \]  \hspace{1cm} (20)

Consequently, the reduced equations of motion with generalized modal coordinate system can be obtained as follows:

\[ M^*\dot{U}^* + (K^*(\omega, T) + ik\beta^*(\omega, T))U^* = F^*(\omega) \]  \hspace{1cm} (21)

where the reduced system matrices and vectors are given as:

\[ M^* = \Psi^TM\Psi \]
\[ K^* = \Psi^TK\Psi \]
\[ K_\beta^* = \Psi^TK_\beta\Psi \]
\[ F^* = \Psi^TF \]
\[ U^* = \Psi U \]

The natural frequencies and modal loss factors can be determined using following eigenvalue equations with a general complex form.

\[ (K^*(\omega, T) + ik\beta^*(\omega, T) - \lambda_n^2M^*)U^* = 0 \]  \hspace{1cm} (23)

From Eq.(23), the natural frequencies and loss factors for each mode are defined by real and imaginary parts of complex eigenvalue \( \lambda_n^* \).

\[ \omega_n^2 = \text{Real} \left[ \lambda_n^* \right], \quad \eta_n = \frac{\text{Imag} \left[ \lambda_n^* \right]}{\text{Real} \left[ \lambda_n^* \right]} \]  \hspace{1cm} (24)

The frequency response function is used to investigate the steady state dynamic characteristics of the linear system subject to harmonic excitation. Due to a harmonic excitation at a certain point, the reduced force vector with respect to modal coordinates is given as follows:

\[ F^* = \Psi F = F_0 \Psi F_{\text{input}} e^{i\omega t} = F_0 F_{\text{input}} e^{i\omega t} \]  \hspace{1cm} (25)

Finally, the frequency response function applying modal approach can be obtained in the following form.

\[ H = \frac{U}{F_0} \frac{\Psi U^*}{F_{\text{input}}} = \Psi (K^*(\omega, T) + ik\beta^*(\omega, T) - \alpha^2M^*)^{-1}F_{\text{input}} \]  \hspace{1cm} (26)

In the computation of the frequency response function, the variation of material properties of viscoelastic layers dependent on exciting frequency and environmental temperature is taken into account as addressed in Eq.(26). The magnitude and phase of frequency response functions are defined as

\[ \text{Magnitude} = |H| \]
\[ \text{Phase}, \varphi = \arctan \left( \frac{-\text{Imag}(H)}{\text{Real}(H)} \right) \]  \hspace{1cm} (27)

3. Results and Discussion

3.1 Numerical validation of sandwich beam with viscoelastic core

Since there are few finite element studies dealing with damping characteristics of cylindrical composite shell including the effects of transversely shear deformation and viscoelastic layer, the numerical validation of sandwich beam with viscoelastic core as shown in Fig.2 is performed instead of shell structures. Material properties of the sandwich beam composed of isotropic facesheet and viscoelastic core are given as follows:

- Elastic Facesheet: \( E = 69 \text{ GPa}, \quad v = 0.3, \quad \rho = 2800 \text{ kg/m}^3, \quad \text{thickness} = 1.524 \text{ mm}, \quad \eta = 0.0 \)
- Viscoelastic Core: \( E = 2.1 \text{ MPa}, \quad v = 0.499, \quad \rho = 970 \text{ kg/m}^3, \quad \text{thickness} = 1.524 \text{ mm}, \quad \eta = 0.1, 1 \)

Based on the layerwise shell theory, the present results of natural frequencies and modal loss factors are compared with a sixth order beam theory solution of Rao(3), 3-D finite element solution of Soni and Bogner(2), and layerwise displacement theory solution of Cho et al.(4). The 7×2 mesh with nine-node elements and three sub-layers through the thickness were used in the present layerwise finite element model. The long radius of \( R = 17.78 \text{ m} \) and the tiny shallowness angle of \( \phi = 0.01 \text{ rad} \) are chosen to minimize the geometric curvature effect with the relation of \( b = R\phi \). In the computation of structural damping and stiffness matrices, the material properties dependent on frequency and temperature are neglected.

As shown in Table 1, the present results show good agreements with the previous results given in references for low and high loss factors of the viscoelastic core. Table 1 shows that the present finite element method based on the cylindrical layerwise shell theory can be successfully applied to investigate the dynamic characteristics of flat beams and plates as well as curved cylindrical panels.

Fig. 2 Geometry of sandwich beam with viscoelastic layer
Table 1  Natural frequencies and modal loss factors of viscoelastic sandwich beam

<table>
<thead>
<tr>
<th>Mode</th>
<th>Res. (Hz)</th>
<th>Sint. (Hz)</th>
<th>Cho. (Hz)</th>
<th>Present Results (Hz)</th>
<th>Core loss factor</th>
<th>λu</th>
</tr>
</thead>
<tbody>
<tr>
<td>1st</td>
<td>64.1</td>
<td>64.2</td>
<td>64.2</td>
<td>64.2</td>
<td>0.0280</td>
<td>0.2</td>
</tr>
<tr>
<td>2nd</td>
<td>296.4</td>
<td>297.0</td>
<td>297.4</td>
<td>297.2</td>
<td>0.0243</td>
<td>0.2</td>
</tr>
<tr>
<td>3rd</td>
<td>743.7</td>
<td>747.2</td>
<td>747.9</td>
<td>747.9</td>
<td>0.0183</td>
<td>0.2</td>
</tr>
<tr>
<td>4th</td>
<td>1389.9</td>
<td>1413.0</td>
<td>1413.0</td>
<td>1414.5</td>
<td>0.0089</td>
<td>0.2</td>
</tr>
<tr>
<td>5th</td>
<td>2260.1</td>
<td>2335.5</td>
<td>2335.5</td>
<td>2339.0</td>
<td>0.0055</td>
<td>0.2</td>
</tr>
</tbody>
</table>

Core loss factor = 0.1

![Diagram](image.png)

Fig. 3 Geometry and construction of cylindrical composite panels with viscoelastic layer

Original panel

UCLD

CLD

Co-cured sandwich

laminate, b,

viscoelastic core, b,

Fig. 4 Boundary conditions of hybrid panels for various damping treatments

3.2 Description of structural models with viscoelastic layers

In this section, the structural models for various damping treatments such as unconstrained and constrained layer damping and co-cured sandwich models are described. Figure 3 shows the general cylindrical hybrid composite panels with embedded viscoelastic layers. Two curved edges of AB and CD are all clamped and the boundaries of AC and BD are free. In Fig. 4, detailed clamping conditions of AB and CD are given according to viscoelastic damping treatments, and the information of lamination and construction of cylindrical hybrid panels is given in Table 2. The composite material of lamina is Graphite/Epoxy and material properties and loss factors are given as follows:

\[ E = 119 \text{ GPa}, \quad E_s = 8.67 \text{ GPa}, \quad G_{12} = G_{13} = 5.18 \text{ GPa}, \]

\[ G_{23} = 3.9 \text{ GPa}, \quad \nu_{12} = 0.31, \quad \rho = 1570 \text{ kg/m}^3 \]

\[ \eta_1 = 0.118\%, \quad \eta_s = 0.520\%, \quad \eta_{12} = \eta_{33} = 0.812\%, \]

\[ \eta_{23} = 0.84\% \]

3M-SDI10 and 3M-SD112 are used for viscoelastic damping layers in this study and the shear modulus and loss factor of viscoelastic materials are supplied by the 3M companies in the monogram form. Two viscoelastic materials have the variation of material properties dependent on the environmental temperature and driving frequency. For the simple calculation of material properties of 3M-SDI10 and 3M-SD112, the curve fitting model proposed by Drake is used and the coefficients of mathematical equations are arranged in Table 3.

\[
\log_{10}(M) = \log_{10}(ML)
\]

\[
+ \frac{2}{1 + (QFROM/FR)^{2/3}} \log_{10}(ETA) = \log_{10}(ETAFROL)
\]

\[
+ C((SH + SL)A + (SL - SH)(1 - \sqrt{1 + A^2})
\]

\[
\log_{10}(FR) = \log_{10}(F) - \frac{12(T - T_0)}{525 + T - T_0}
\]

\[
A = \frac{\log_{10}(FR) - \log_{10}(FROL)}{C}
\]

Here, \( M \) means Young's or shear modulus and \( ETA \) indicates loss factor of the viscoelastic layer. Also, \( T \) and \( F \) are an exposed temperature and an exciting frequency, respectively.
3.3 Natural frequencies and loss factors for various damping treatments

The material and structural parameters, such as frequency, temperature, length, thickness of laminated structures and so on, greatly take effect on the vibration and damping characteristics of laminated structures. Among various parameters, effects of the thickness of damping layer and constrained layer are investigated and discussed in view of natural frequencies and modal loss factors for various structural models.

(a) Unconstrained layer damping (UCLD)

The 12×12 meshes with nine-node elements and five sub-laminates through the thickness direction were used in the present layerwise finite element model. The viscoelastic damping layer was located on lower face of the original panel. The thickness of damping layer $h_C$ varies from 0 mm to 1 mm. Figure 6 shows the natural frequencies and loss factors of the lower five modes for the UCLD model. All of natural frequencies decrease and modal loss factors increase slowly as the thickness of damping layer $h_C$ increases.

(b) Constrained layer damping (CLD)

The (5+1+5) sub-layers of laminated hybrid panels were also reduced to six (4+1+1) sub-laminates in the layerwise thickness direction. As shown in Fig. 4, viscoelastic layer and constrained layer were located on the lower face of the original panel. The thickness of damping layer $h_C$ is fixed to 0.25 mm. Two types of constrained layer laminated with [0], and [90], were used to investigate the natural frequencies and modal loss factors of circumferential and longitudinal modes in view of damping efficiency of CLD model. The thickness of constrained layer $h_t$ varies from 0 mm to 1 mm, in other words, $n$ varies from 0 and 8.

Figure 7 shows the natural frequencies and loss factors of the lower five modes for CLD[0]. The values of CLD [0], are the same as those of UCLD with $h_C = 0.25$ mm. The natural frequencies of CLD [0], seem to be similar to those of UCLD except the first and second frequencies at $h_t = 1$ mm, where the mode change occurs between the first and second modes. The loss factors of CLD [0], increase as $h_t$ increases. It's noticeable that the loss factors of higher modes are relatively higher than those of lower modes. In mode shapes, there exists circumferential (torsional) mode shape in the lower frequency and longitudinal (bending) mode shape in higher frequency. The constrained layers with laminated [0],
greatly affect dynamic characteristics of higher modes.

Figure 8 shows the natural frequencies and loss factors of the lower five modes for CLD \([90]_s\). The natural frequencies of CLD \([90]_s\) are higher than UCLD, and the mode change also occurs between the first and second modes at \(h_2 = 0.125\) mm. The loss factors of CLD \([90]_s\) increase globally as \(h_2\) increases, except the first mode affected by mode change. Unlike CLD \([0]_s\), the loss factors of the lower mode are relatively higher than those of the higher modes. The constrained layers with laminated \([90]_s\) stiffen the original panel a little and influence the loss factor of the lower mode further.

\(( e )\) **Co-cured sandwich model**

The co-cured sandwich model has an embedded viscoelastic layer in the center of the original panel. In the finite elements of thickness direction, nine sub-layers were also reduced to five \((2 + 1 + 2)\) sub-laminates. The thickness of the upper laminated facesheet and the lower laminated facesheet, \(h_1\) and \(h_2\), is 0.5 mm with the same lamination type of \([0/90]_s\). The thickness of damping layer \(h_c\) varies from 0 mm to 1 mm. Figure 9 indicates that frequencies decrease rapidly when damping layer is embedded, but variations of frequencies become small as \(h_c\) increases especially in the co-cured sandwich model. Mode changes occur between first and second modes at \(h_c = 0.5\) mm and between the third and fourth modes at \(h_c = 0.1\) mm. The loss factors have higher value globally as compared with the others, but the variation of the loss factor is very little with the increase of \(h_c\).

\((3.4)\) **Effect of material degradation of viscoelastic layer**

In order to evaluate the effect of material degradation of viscoelastic layer on the dynamic characteristics, the numerical analyses of frequency response functions for the cylindrical hybrid panels with various viscoelastic damping treatments are performed in this section. The hybrid panel with viscoelastic layers is categorized as three models by means of the fabrication. One is the unconstrained layer damping model, and another is the constrained layer damping model. The other is the co-cured sandwich model with embedded viscoelastic layers. The frequency response functions are computed for the unconstrained layer damping (UCLD), constrained layer
damping (CLD) and co-cured sandwich model with the thickness of the damping layer $h_c=0.25$ mm.

(a) Results FRFs for various damping treatments

The FRF of original panel is compared with those of CLD, UCLD and co-cured sandwich model. Figure 10 shows the results of FRFs of hybrid panels with ISD 110 viscoelastic layers. All peak frequencies of UCLD move to left, compared with those of the original panel without changes of the peak values. The panel with embedded damping layer has the higher damping efficiency than any other cases. The CLD model has also a good damping efficiency relatively. Circumferential modes exist in the lower frequency, and longitudinal modes exist in the higher frequency according to previous vibration results of the original panel. The constrained layer laminated with $[0]_2$ makes longitudinal stiffness harder, and laminated with $[90]_2$ strengthens circumferential stiffness. Consequently, CLD $[0]_2$ has higher damping efficiency than CLD $[90]_2$ in the lower frequency region, and CLD $[90]_2$ has higher damping efficiency than CLD $[0]_2$ in the higher frequency region.

Figure 11 shows the results of FRFs of hybrid panels with ISD 112 viscoelastic layers exposed to a room temperature. The results using ISD112 have
higher damping efficiencies than ISD110 at the room temperature. The mode shapes of CLD\([90\]2 and co-cured sandwich model are different from the original panel model. Those results indicate that CLD\([90\]2 and co-cured sandwich models are affected by the damping material further than the others.

(b) Effects of frequency dependence of viscoelastic layer

It is well known that the material properties of viscoelastic layers are dependent on the exciting frequency and environmental temperature. In this section, the effects of frequency dependence of viscoelastic layers are investigated on the structural damping performance. The temperature is assumed to be 25°C. From Eqs. (29)−(32), the properties of ISD110 are more dependent on the exciting frequency than those of ISD112 at the room temperature in the interesting frequency region. The variation of loss factor isn’t negligible in the low frequency region and the stiffness of viscoelastic layers becomes stronger as the exciting frequency increases. From the previous results, it’s found that CLD\([90\]2 and co-cured model are efficient in view of structural damping. The assumed frequency of a frequency independent model is 200 Hz.

As shown in Fig. 12, the deviation of FRF is obviously observed below and above 200 Hz. Therefore, it’s necessary to consider the variations of the damping properties according to the exciting frequency for the accurate damping prediction.

(c) Effect of temperature on damping characteristics

The material properties of viscoelastic layers are mostly affected by environmental temperature. It is very complex and difficult to accurately analyze the damping characteristics due to temperature because of thermal stress and thermal deformations. In this study, only the change of damping properties is investigated neglecting the effects of thermal stresses and deformations. As shown in Fig. 13, natural frequencies decrease when the temperature increases, and damping characteristics are dependent on the kind of viscoelastic materials. The 3M-ISD110 has efficient damping properties at 50°C, and ISD112 has around 25°C. It’s necessary to select and design damping materials considering the operational temperature.
4. Conclusion

In this study, the damping characteristics of cylindrical hybrid composite panels with viscoelastic layers are investigated using the finite element method based on the layerwise shell theory, considering the effects of transversely shear deformation and material degradation of viscoelastic layers. Natural frequencies, loss factors and frequency response functions for various damping models such as unconstrained layer damping, constrained layer damping and co-cured sandwich model are compared with those of an original base panel in view of damping efficiency. The present results show that it is very important to consider the variation of the damping properties according to exciting frequency and environmental temperature for the accurate damping prediction of cylindrical hybrid composite panels with viscoelastic damping layers.

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References


