Vibration and Buckling of Cross-Ply Composite Beams using Refined Shear Deformation Theory

Thuc Vo
Glyndwr University, t.vo@glyndwr.ac.uk

Fawad Inam
Glyndwr University, f.inam@glyndwr.ac.uk

Follow this and additional works at: http://epubs.glyndwr.ac.uk/aer_eng

Part of the Manufacturing Commons, Mechanics of Materials Commons, Polymer and Organic Materials Commons, Structural Materials Commons, and the Structures and Materials Commons

Recommended Citation
Vibration and Buckling of Cross-Ply Composite Beams using Refined Shear Deformation Theory

Abstract
Vibration and buckling analysis of cross-ply composite beams using refined shear deformation theory is presented. The theory accounts for the parabolical variation of shear strains through the depth of beam. Three governing equations of motion are derived from the Hamilton's principle. The resulting coupling is referred to as triply coupled vibration and buckling. A two-noded C1 beam element with five degree-of-freedom per node is developed to solve the problem. Numerical results are obtained for composite beams to investigate modulus ratio on the natural frequencies, critical buckling loads and load-frequency interaction curves.

Keywords
Composite beams, Refined shear deformation theory, Triply coupled vibration and buckling

Disciplines

Comments
Copyright © 2012 Glyndŵr University and the authors, all rights reserved. This article was first presented at the 2nd International Conference on Advanced Composite Materials and Technologies for Aerospace Applications, June 11-13, 2012, Wrexham, UK and published in the conference proceedings by Glyndŵr University.

Permission to copy, reprint/republish this material for advertising or promotional purposes or for creating new collective works for resale or redistribution must be obtained from Glyndŵr University. By choosing to view this document, you agree to all provisions of the copyright laws protecting it. It is published here with the permission of the Authors, and the full proceedings are available to purchase at http://www.lulu.com/shop/richard-day-and-sergey-reznik/advanced-composite-materials-and-technologies-for-aerospace-applications/paperback/product-20214156.html;jsessionid=566556BF08A9459FC8807B9BF9878A8D#productDetails

This conference paper is available at Glyndŵr University Research Online: http://epubs.glyndwr.ac.uk/aer_eng/7
Vibration and Buckling of Cross-Ply Composite Beams using Refined Shear Deformation Theory

Thuc Vo, Fawad Inam

School of Mechanical, Aeronautical and Electrical Engineering, Glyndŵr University, Plas Coch, Mold Road, Wrexham, LL11 2AW, UK

Abstract: Vibration and buckling analysis of cross-ply composite beams using refined shear deformation theory is presented. The theory accounts for the parabolical variation of shear strains through the depth of beam. Three governing equations of motion are derived from the Hamilton’s principle. The resulting coupling is referred to as triply coupled vibration and buckling. A two-noded C^1 beam element with five degree-of-freedom per node is developed to solve the problem. Numerical results are obtained for composite beams to investigate modulus ratio on the natural frequencies, critical buckling loads and load-frequency interaction curves.

Key Words: Composite beams, Refined shear deformation theory, Triply coupled vibration and buckling.

1. Introduction

Structural components made with composite materials are increasingly being used in various engineering applications due to their attractive properties in strength, stiffness, and lightness. Understanding their dynamic and buckling behaviour is of increasing importance. The classical beam theory (CBT) known as Euler-Bernoulli beam theory is the simplest one and is applicable to slender beams only. For moderately deep beams, it overestimates buckling loads and natural frequencies due to ignoring the transverse shear effects. The first-order beam theory (FOBT) known as Timoshenko beam theory is proposed to overcome the limitations of the CBT by accounting for the transverse shear effects. Since the FOBT violates the zero shear stress conditions on the top and bottom surfaces of the beam, a shear correction factor is required to account for the discrepancy between the actual stress state and the assumed constant stress state. To remove the discrepancies in the CBT and FOBT, the higher-order beam theory (HOBT) is developed to avoid the use of shear correction factor and have a better prediction of response of laminated beams. The HOBTs can be developed based on the assumption of higher-order variations of in-plane displacement or both in-plane and transverse displacements through the depth of the beam. Many numerical techniques have been used to solve the dynamic and/or buckling analysis of composite beams using HOBTs. Some researchers introduced the dynamic stiffness matrix method to solve exactly the free vibration and buckling problems of axially loaded composite beams with arbitrary lay-ups. Although the HOBTs offer a slight improvement in accuracy compared to the FOBT, they are computationally more demanding due to higher-order terms included in the theory. Hence, there is a scope to develop accurate refined shear deformation theory which is simple to use.

In this paper, which is extended from previous research (Vo and Thai, 2012), vibration and buckling analysis of composite beams using refined shear deformation theory is presented. The displacement field of the present theory is chosen based on the following assumptions: (1) the axial and transverse displacements consist of bending and shear components in which the bending components do not contribute toward shear forces and, likewise, the shear components do not contribute toward bending moments; (2) the bending component of axial displacement is similar to that given by the CBT; and (3) the shear component of axial displacement gives rise to the higher-order variation of shear strain and hence to shear stress through the depth of the beam in such a way that shear stress vanishes on the top and bottom surfaces. The most interesting feature of this theory is that it satisfies the zero traction boundary conditions on the top and bottom surfaces of the beam without using shear correction factors. The three governing equations of motion are derived from the Hamilton’s principle. The resulting coupling is referred to as triply coupled vibration and buckling. A two-noded C^1 beam element with five degree-of-freedom per node which accounts for shear deformation effects and all coupling coming from the material anisotropy is developed to solve the problem. Numerical results are obtained for
composite beams to investigate effects of fiber orientation and modulus ratio on the natural frequencies, critical buckling loads and load-frequency interaction curves as well as corresponding mode shapes.

2. Kinematics

A laminated composite beam made of many plies of orthotropic materials in different orientations with respect to the x-axis, as shown in Fig. 1, is considered.

Based on the assumptions made in the preceding section, the displacement field of the present theory can be obtained as:

\[ U(x, z, t) = u(x, t) - z \frac{\partial w_b(x, t)}{\partial x} + \frac{z}{2} \left[ \frac{1}{4} \frac{5}{3} \left( \frac{z}{h} \right)^2 \right] \frac{\partial w_t(x, t)}{\partial x} \]

\[ V(x, z, t) = 0 \]

\[ W(x, z, t) = w_b(x, t) + w_t(x, t) \]

where \( u \) is the axial displacement along the mid-plane of the beam, \( w_b \) and \( w_t \) are the bending and shear components of transverse displacement along the mid-plane of the beam, respectively. The non-zero strains are given by:

\[ \varepsilon_x = \frac{\partial u}{\partial x} = \varepsilon_x^b + z \kappa_x^b + f \kappa_x^s \]

\[ \gamma_{xz} = \frac{\partial w_b}{\partial x} + \frac{\partial w_t}{\partial z} = (1 - f) \gamma_{xz}^b + g \gamma_{xz}^s \]

where

\[ f = z \left[ - \frac{1}{4} + \frac{5}{3} \left( \frac{z}{h} \right)^2 \right] \]

\[ g = 1 - f' \left[ \frac{5}{4} - \frac{1}{4} - \frac{5}{3} \left( \frac{z}{h} \right)^2 \right] \]

and \( \varepsilon_x^b, \gamma_{xz}^b, \kappa_x^b, \kappa_x^s \) and \( \kappa_{xy} \) are the axial strain, shear strains and curvatures in the beam, respectively, defined as:

\[ \varepsilon_x^b = u'' \]

\[ \gamma_{xz}^b = w_x'' \]

\[ \kappa_x^b = -w_x''' \]

\[ \kappa_x^s = -w_t''' \]

where differentiation with respect to the x-axis is denoted by primes (').

3. Variational Formulation

In order to derive the equations of motion, Hamilton's principle is used:

\[ \delta \left[ \int (K - U - V) dt \right] = 0 \]

where \( U \) is the variation of the strain energy, \( V \) is the variation of the potential energy, and \( K \) is the variation of the kinetic energy.

The variation of the strain energy can be stated as:

\[ \delta U = \int \left( \sigma_x \delta \varepsilon_x + \sigma_{xz} \delta \gamma_{xz} \right) dv = \int \left( N_x \delta w_x' + M_x^b \delta k_x^b + M_x^s \delta k_x^s + Q_{xz} \delta \gamma_{xz} \right) dx \]

where \( N_x, M_x^b, M_x^s \) and \( Q_{xz} \) are the axial force, bending moments and shear force, respectively, defined by integrating over the cross-sectional area \( A \) as:

\[ N_x = \int \sigma_x dv \]

\[ M_x^b = \int \sigma_x zdA \]

\[ M_x^s = \int \sigma_x fdA \]

\[ Q_{xz} = \int \sigma_{xz} gdA \]

The variation of the kinetic energy can be expressed as:

\[ \delta V = - \int \left[ \int \delta \left( \rho_b \left( \dot{u} \dot{u} - m_x \ddot{u} - m_x \dddot{u} \right) \right) + \delta \left( \rho_b \left( \dot{w}_x + \dot{w}_t \right)^2 \right) \right] dx \]

The variation of the potential energy is obtained as:

\[ \delta K = \int \left[ \rho_b \left( \dot{u} \dot{u} + \dot{w}_x + \dot{w}_t \right) \right] dv = \int \left[ \delta \left( m_x \dot{u} - m_x \ddot{u} \right) + \delta \left( m_x \dot{w}_x + m_x \ddot{w}_x \right) + \delta \left( m_x \dot{w}_t + m_x \ddot{w}_t \right) + \delta \left( m_x \dot{u} + m_x \ddot{u} \right) + \delta \left( m_x \dot{w}_x + m_x \ddot{w}_x \right) + \delta \left( m_x \dot{w}_t + m_x \ddot{w}_t \right) \right] dx \]

where the differentiation with respect to the time \( t \) is denoted by dot-superscript convention and \( \rho_b \) is the density of a \( k \)th layer and \( m_x, m_b, m_t, m_p, m_y \) and \( m_z \) are the inertia coefficients, defined by:

\[ m_x = -m_x + \frac{5}{4} \frac{m_x}{3h} \]

\[ m_b = -m_b + \frac{5}{4} \frac{m_b}{3h^2} \]

\[ m_t = -m_t + \frac{5}{4} \frac{m_t}{3h^2} \]

\[ m_y = -m_y + \frac{5}{4} \frac{m_y}{3h^2} \]

\[ m_z = -m_z + \frac{5}{4} \frac{m_z}{3h^2} \]

where

\[ (m_x, m_b, m_t, m_p, m_y) = \int \rho_b \left( 1, z, z^2, z^3, z^4 \right) dA \]

By substituting Eqs. (6), (8) and (9) into Eq. (5), the following weak statement is obtained:

\[ 0 = \int_0^T \left[ \int \left( \delta \dot{u} \left( m_x \dot{u} = m_x \ddot{u} \right) + \delta \dot{w}_x \left( m_x \dot{w}_x + m_x \ddot{w}_x \right) + \delta \dot{w}_t \left( m_x \dot{w}_t + m_x \ddot{w}_t \right) + \delta \left( m_x \dot{u} + m_x \ddot{u} \right) + \delta \left( m_x \dot{w}_x + m_x \ddot{w}_x \right) + \delta \left( m_x \dot{w}_t + m_x \ddot{w}_t \right) \right) dx dt \]
4. Constitutive equations

The stress-strain relations for the \(k^{th}\) lamina are given by:

\[
\sigma_x = \bar{Q}_{ij} y_x, \quad \sigma_y = \bar{Q}_{ij} y_y, \quad (13)
\]

where \(\bar{Q}_{ij}\) and \(\bar{Q}_{ij}\) are the elastic stiffnesses transformed to the x-direction. More detailed explanation can be found in (Jones, 1999).

The constitutive equations for bar forces and bar strains are obtained by using Eqs. (2), (7) and (13):

\[
\begin{align*}
N_x & = R_{i1} y_x \\
M_{ij} & = R_{i2} y_x \\
Q_{ij} & = \text{sym.} R_{i4} y_x
\end{align*}
\]

where \(R_{ij}\) are the laminate stiffnesses of general composite beams and given by:

\[
\begin{align*}
R_{i1} & = \int_j A_i dy \\
R_{i2} & = \int_j B_i dy \\
R_{i3} & = \int_j \left( -\frac{B_i}{4} + \frac{5}{3h^2} E_{i1} \right) dy \\
R_{i4} & = \int_j \left( \frac{D_{i2}}{4} + \frac{5}{3h^2} F_{i1} \right) dy \\
R_{i5} & = \int_j \left( \frac{D_{i1}}{16} + \frac{5}{6h^2} F_{i1} + \frac{25}{9h^4} H_{i1} \right) dy \\
R_{i6} & = \int_j \left( \frac{25}{16} A_{i5} - \frac{25}{2h^2} D_{i5} + \frac{25}{h^4} F_{i5} \right) dy
\end{align*}
\]

where \(A_i, B_i\) and \(D_i\) matrices are the extensional, coupling and bending stiffness and \(E_{i1}, F_{i1}, H_{i1}\) matrices are the higher-order stiffnesses, respectively, defined by:

\[
\left( A_i, B_i, D_i, E_i, F_i, H_i \right) = \int_0^1 \bar{O}_{ij} \left( 1, 1, z, z, z, z \right) dz
\]

5. Governing equations of motion

The equilibrium equations of the present study can be obtained by integrating the derivatives of the varied quantities by parts and collecting the coefficients of \(\delta u\), \(\delta u_b\) and \(\delta w\):

\[
\begin{align*}
N_i' = m_i \ddot{u} - m_i \ddot{w}_0 - m_i \ddot{w}_s, \\
M_{ij}' = P_i \left( w_{ij} + w_{ij} \right) = m_i \left( \ddot{w}_0 + \ddot{w}_s \right) = m_i \left( \ddot{w}_0 + \ddot{w}_s \right), \\
Q_{ij}' = P_i \left( w_{ij} + w_{ij} \right) = m_i \left( \ddot{w}_0 + \ddot{w}_s \right) = m_i \left( \ddot{w}_0 + \ddot{w}_s \right). \quad (17)
\end{align*}
\]

The natural boundary conditions are of the form:

\[
\begin{align*}
\delta u : N_i, \\
\delta w_x : M_{ij} + Q_{ij} - P_i \left( w_{ij} + w_{ij} \right) - m_i \ddot{u} - m_j \ddot{w}_0 - m_j \ddot{w}_s, \\
\delta w_y : M_{ij} + Q_{ij} - P_i \left( w_{ij} + w_{ij} \right) - m_i \ddot{w}_0 - m_j \ddot{w}_s
\end{align*}
\]

6. Finite element formulation

The present theory for composite beams described in the previous section was implemented via a displacement based finite element method. The variational statement in Eq. (12) requires that the bending and shear components of transverse displacement \(w_b\) and \(w_s\) be twice differentiable and \(C^1\)-continuous, whereas the axial displacement \(u\) must be only once differentiable and \(C^0\)-continuous. The generalized displacements are expressed over each element as a combination of the linear interpolation function \(\Psi_i\) for \(u\) and Hermite-cubic interpolation function \(\psi_j\) for \(w_b\) and \(w_s\) associated with node \(j\) and the nodal values:

\[
\begin{align*}
u & = \sum_{j=1}^n u_j \Psi_j, \\
w_b & = \sum_{j=1}^n w_{bj} \psi_j, \\
w_s & = \sum_{j=1}^n w_{sj} \psi_j
\end{align*}
\]

Substituting these expressions in Eq. (19) into the corresponding weak statement in Eq. (12), the explicit form of the governing equations of motion can be expressed with respect to the laminate stiffnesses \(\bar{R}_{ij}\). Eq. (17) is the most general form for axial-flexural coupled vibration of axially loaded of composite beams, and the dependent variables, \(u, w_b\) and \(w_s\) are fully coupled.

7. Numerical examples

In this section, a number of numerical examples are presented and analysed for verification the accuracy of the present theory in predicting the natural frequencies, critical buckling loads and corresponding mode shapes of composite beams with arbitrary lay-ups. All laminate are of equal thickness and made of the same orthotropic material, whose properties are as follows:

\[
\begin{align*}
\delta u : N_i, \\
\delta w_x : M_{ij} + Q_{ij} - P_i \left( w_{ij} + w_{ij} \right) - m_i \ddot{u} - m_j \ddot{w}_0 - m_j \ddot{w}_s, \\
\delta w_y : M_{ij} + Q_{ij} - P_i \left( w_{ij} + w_{ij} \right) - m_j \ddot{w}_0 - m_j \ddot{w}_s
\end{align*}
\]
Material I:

\[
\begin{align*}
E_1 / E_2 &= \text{open}, \\
G_{12} &= G_{13} = 0.6E_2 \\
G_{23} &= 0.5E_2 \\
\nu_{12} &= 0.25
\end{align*}
\]  
(22)

Material II:

\[
\begin{align*}
E_1 / E_2 &= \text{open} \\
G_{12} &= G_{13} = 0.5E_2 \\
G_{23} &= 0.2E_2 \\
\nu_{12} &= 0.25
\end{align*}
\]  
(23)

For convenience, the following non-dimensional terms are used in presenting the numerical results:

\[
\frac{P_{cr}}{\omega} = \frac{pL^4}{Ebh^3} \\
\frac{\omega}{\omega_c} = \frac{\omega L^2}{\sqrt{\rho}}
\]  
(24)

As the first example, vibration and buckling analysis of a symmetric and an anti-symmetric cross-ply composite beam with simply-supported boundary condition is performed. Material I and II with \( E_1/E_2 = 10 \) and 40 are used. The fundamental natural frequencies and critical buckling loads for different span-to-height \( L/h \) ratios are compared with exact solutions (Khdeir and Reddy, 1994; Khdeir and Reddy, 1997) and the finite elements results (Murthy et al., 2005; Aydogdu, 2005; Aydogdu, 2006a) in Tables 1 and 2. An excellent agreement between the predictions of the present model and the results of the other models mentioned can be observed.

Material I with \( E_1/E_2 = 40 \) is chosen to show the effect of the axial force on the fundamental natural frequencies of beam with various \( L/h \) ratios (Fig. 2). It can be seen that the change of the natural frequency due to axial force is noticeable. The natural frequency diminishes when the axial force changes from tensile to compressive, as expected. It is obvious that the natural frequency decreases with the increase of axial force, and the decrease becomes more quickly when the axial force is close to critical buckling load. For an anti-symmetric cross-ply lay-up, with \( L/h = 5, 10 \) and 20, at about \( P \approx 3.903, 4.936 \) and 5.290, respectively, the natural frequencies become zero which implies that at these loads, bucklings occur as a degenerate case of natural vibration at zero frequency. It also means that the buckling loads of composite beams under axial force can be also obtained indirectly through vibration problem by increasing the axial force until the corresponding natural frequency vanishes. Besides, Fig. 2 explains the duality between the buckling load and natural frequency. In order to show the effect of material anisotropy, the loads are calculated using the following formulas:

Table 1.

<table>
<thead>
<tr>
<th>Lay-ups</th>
<th>Reference</th>
<th>( L/h )</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>5</td>
</tr>
<tr>
<td>([0^\circ/90^\circ/0^\circ])</td>
<td>Khdeir and Reddy (1994)</td>
<td>9.208</td>
</tr>
<tr>
<td></td>
<td>Aydogdu (2005)</td>
<td>9.207</td>
</tr>
<tr>
<td></td>
<td>Present</td>
<td>9.206</td>
</tr>
<tr>
<td>([0^\circ/90^\circ/0^\circ])</td>
<td>Murthy et al. (2005)</td>
<td>6.045</td>
</tr>
<tr>
<td></td>
<td>Khdeir and Reddy (1994)</td>
<td>6.128</td>
</tr>
<tr>
<td></td>
<td>Aydogdu (2005)</td>
<td>6.144</td>
</tr>
<tr>
<td></td>
<td>Present</td>
<td>6.058</td>
</tr>
</tbody>
</table>

Table 2.

<table>
<thead>
<tr>
<th>Material I</th>
<th>( L/h )</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>5</td>
</tr>
<tr>
<td>([0^\circ/90^\circ/0^\circ])</td>
<td>Khdeir and Reddy (1997)</td>
</tr>
<tr>
<td></td>
<td>Aydogdu (2006a)</td>
</tr>
<tr>
<td></td>
<td>Present</td>
</tr>
<tr>
<td>([0^\circ/90^\circ/0^\circ])</td>
<td>Aydogdu (2005)</td>
</tr>
<tr>
<td></td>
<td>Present</td>
</tr>
</tbody>
</table>

Figure 2. Load-frequency curves of a simply-supported cross-ply composite beam \( (L/h = 5, 10 \) and 20).


8. Conclusions

A two-noded $C^1$ beam element of five degree-of-freedom per node is developed to study the vibration and buckling behaviour of cross-ply composite beams using refined shear deformation theory. This model is capable of predicting accurately the natural frequencies, critical buckling loads and load-frequency interaction curves. It accounts for the parabolical variation of shear strains through the depth of the beam, and satisfies the zero traction boundary conditions on the top and bottom surfaces of the beam without using shear correction factor. The present model is found to be appropriate and efficient in analyzing vibration and buckling problem of cross-ply composite beams.

Reference


