Vibration Analysis of Composite Rectangular Plates Reinforced along Curved Lines*

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Abstract
In the past few decades, composite materials composed of straight fibers and polymer matrix have gained their status as the most promising material for light-weight structures. Technical merit of the composites as tailored material also provided practical advantages in the optimum design process. Recently, it is reported that the fabrication machine has been developed to make curved fibers embedded in the matrix material. Based on such technical advancement, this paper proposes an analytical method to study vibration of composite rectangular plates reinforced along curved lines. The approach is based on the Ritz method where variable fiber direction can be accommodated. For this purpose, the fibers continuously changing their direction are formulated as the variable bending stiffness in the total potential energy. A frequency equation is derived by the Ritz minimizing process, and frequency parameters are calculated as the eigenvalues in the eigenvalue problem. In numerical results, the accuracy of the method is presented by comparing present results with FEM results. The advantages of present plate are confirmed by comparing natural frequencies and mode shapes with those of conventional composite and isotropic plates, and the effectiveness of the new solution to the most recent problem is demonstrated.

Key words: Composite Plate, Free Vibration, Fiber Reinforced Plastic, Curvilinear Fiber, Local Anisotropy

1. Introduction

Conventional laminated composite plates are fabricated typically by stacking orthotropic plies, each of which is composed of matrix material and reinforcing fibers allocated in parallel. It is known that structural designers can make use of the fiber orientation angles in the plies to design the overall mechanical properties most effectively. For years, tailoring of such composite plates has been done by varying the orientation of parallel fibers or thickness of the plies.

Recent development in manufacturing techniques makes it possible to fabricate composite material with fiber orientations that vary continuously. This allows us to make the composite plate reinforced by curvilinear fibers and to distribute the stiffness property more effectively in the plate. Also noted is that the design space for fibrous composites can be dramatically expanded when it is compared to the conventional design problem for straight parallel fibers. Thus a new design technique is feasible to improve the plate property significantly. In recent literature, such composite plates with variable stiffness are technically named as variable stiffness plates[1] or locally anisotropic plates.
This study deals with vibration of such variable stiffness plates. There have been some related papers in the literature. Martin and Leissa\(^{(2)}\) presented a variable stiffness concept to improve the buckling performance of the plate by using the Ritz method. Hyer and Lee\(^{(3)}\) used the finite element method to analyze strength and buckling performance of variable stiffness plates considering curvilinear fibers. They varied the fiber angles from one element to another, and it turned out that such plates have higher failure load than that of the plates with straight fibers. Gürdal and Olmendo\(^{(1)}\) proposed an analysis method for the in-plane response of a variable stiffness plate reinforced by sinusoidal wave fibers. The improvement was confirmed in buckling performance of plate with fibers in curvilinear shape. Setoodeh and his co-workers\(^{(4)}\) first incorporated optimization concept into the area and combined a gradient method with the finite element method. They calculated the distribution of the optimum lamination parameters that yield lower compliance. But they did not obtain actual curvilinear fiber path from the obtained parameters. The present authors\(^{(5)}\) have determined the optimum short fiber distributions from optimally distributed lamination parameters and revealed that the laminated plates with optimally distributed short fibers have higher stiffness than the conventional plates. It was also found in the study that the optimally distributed fibers tend to be allocated with a certain directional tendency, and this suggests a possibility that an optimum continuous fiber distribution may exist.

The vibration analysis is proposed in the paper, as an initial approach for analyzing plate with curvilinear fibers, for single-ply plates reinforced along curved lines in the quadratic function. The Ritz energy method is used to determine the vibration characteristics of the plates, where a numerical parametric study may be carried out easily for arbitrary sets of typical boundary conditions\(^{(6)}\). The total potential energy is formulated in terms of the variable bending stiffness due to fibers continuously changing their direction. The minimizing process of a functional for the total potential energy yields the characteristic equations for eigenvalues (natural frequencies). In numerical results, the accuracy is confirmed by comparing the present results with those from the finite element method and it is shown that the present plates with parabolically shaped fibers have higher frequencies than the conventional straight-fiber composite plates. Moreover, the vibration mode shapes are compared among the present plates, diagonally reinforced plates, and isotropic plates to study the effects of variable stiffness.

### 2. Method of Analysis

#### 2.1 Stress – strain relations\(^{(7)}\)

For thin layers, the stress–strain relation involving the plane stress assumption in the coordinate system \(O – LT\) shown in Fig.1 is given by

\[
\begin{align*}
\{\sigma_L\} &= \begin{bmatrix} Q_{11} & Q_{12} & 0 \\ Q_{21} & Q_{22} & 0 \\ 0 & 0 & Q_{66} \end{bmatrix} \{\varepsilon_L\}, \\
\{\sigma_T\} &= \begin{bmatrix} 0 & 0 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & 0 \end{bmatrix} \{\varepsilon_T\},
\end{align*}
\]

(1)

where \(Q_{ij}(i, j = 1, 2, 6)\) are the reduced stiffnesses. These are defined as

\[
\begin{align*}
Q_{11} &= \frac{E_L}{1 - \nu_{LT} \nu_{LT}}, \quad Q_{22} = \frac{E_T}{1 - \nu_{LT} \nu_{LT}}, \\
Q_{12} &= \nu_{LT} Q_{11} = \nu_{LT} Q_{22}, \quad \text{and} \quad Q_{66} = G_{LT}
\end{align*}
\]

(2)

where \(E_L\) and \(E_T\) are moduli of elasticity in the L and T directions of an orthotropic layer, respectively, \(G_{LT}\) is the shear modulus, and \(\nu_{LT}\) and \(\nu_{TL}\) are the major and minor Poisson ratios.

The transformed stress–strain relations rotated with respect to a reference coordinate system \(O – xy\) shown in Fig.1 are
\[
\begin{bmatrix}
\sigma_x \\
\sigma_y \\
\tau_y \\
\end{bmatrix} = \begin{bmatrix}
\bar{Q}_{11} & \bar{Q}_{12} & \bar{Q}_{16} \\
\bar{Q}_{12} & \bar{Q}_{22} & \bar{Q}_{26} \\
\bar{Q}_{16} & \bar{Q}_{26} & \bar{Q}_{66} \\
\end{bmatrix} \begin{bmatrix}
\varepsilon_x \\
\varepsilon_y \\
\gamma_{xy} \\
\end{bmatrix}
\]

where transformed reduced stiffnesses \( \bar{Q}_{ij} \) \((i, j = 1, 2, 6)\), using fiber orientation angle \( \theta \) which is positive in the direction of counterclockwise with respect to \( x \) axis, are

\[
\begin{align*}
\bar{Q}_{11} &= Q_{11} \left( l^4 + 2Q_{22} + 2Q_{66} \right) l^2 m^2 + Q_{12} m^4 \\
\bar{Q}_{12} &= Q_{12} \left( l^4 + m^4 \right) + \left( Q_{11} + Q_{22} - 4Q_{66} \right) l^2 m^2 \\
\bar{Q}_{22} &= Q_{22} m^4 + 2\left( Q_{12} + 2Q_{66} \right) l^2 m^2 + Q_{12} m^4 \\
\bar{Q}_{16} &= \left( Q_{11} - Q_{12} - 2Q_{66} \right) m^3 - \left( Q_{22} - Q_{22} - 2Q_{66} \right) l^3 m \\
\bar{Q}_{26} &= \left( Q_{11} - Q_{12} - 2Q_{66} \right) l^3 m - \left( Q_{22} - Q_{22} - 2Q_{66} \right) m^3 \\
\bar{Q}_{66} &= \left( Q_{11} + Q_{22} - 2Q_{66} \right) l^2 m^2 + Q_{66} \left( l^4 + m^4 \right) \\
\end{align*}
\]

\( l = \sin \theta, \quad m = \cos \theta \).

The bending stiffnesses for symmetrically \( K \)-ply laminated plate are given by

\[
D_{ij} = \frac{2}{3} \sum_{k=1}^{K} \bar{Q}_{ij} \left( z_k - z_{k-1} \right) \quad (i, j = 1, 2, 6)
\]

where \( z_k \) is the distance to the upper surface of the \( k \)th layer, measured from the plate middle surface. The plates are limited to single layer plates in this study. Therefore, the bending stiffnesses used here are

\[
D_{ij} = \frac{h^3}{12} \bar{Q}_{ij} \quad (i, j = 1, 2, 6).
\]

2.2 Variable bending stiffnesses

To allow the plate stiffness to vary continuously, a fiber orientation angle \( \theta \) is defined as the function in terms of \( x \). As shown in Fig.2, the parabolically shaped fibers are assumed in this study as a representative case of composite plates with curvilinear fibers. The curvilinear fiber is defined in the form of a simple quadratic function

\[
f(x) = y = A \left( x + \frac{a}{2} \right) \left( x - \frac{a}{2} \right) = A \left( x^2 - \frac{a^2}{4} \right)
\]

where \( A \) is a proportion constant of a parabola and \( a \) is the width of the plate considered. Thus, the variable fiber orientation angle \( \theta(x) \) is given by the first derivative of \( f(x) \) at \( x \) as

\[
\frac{df}{dx} = 2Ax = \tan \theta
\]

\[
\theta(x) = \tan^{-1} \left( 2Ax \right)
\]

After substituting Eq.(8) into Eqs.(4) and (6), the variable bending stiffnesses for a single layer plate are obtained.
2.3 Free vibration analysis

The Ritz method, based on the energy method, is used for calculating vibration characteristics\(^6\), \(^8\). An advantage of the Ritz method is that it enables us to carry out a numerical parametric study with least computational efforts due to the simple input parameters. Moreover, one can determine the natural frequencies for arbitrary combinations of typical boundary conditions (i.e., free, simply supported and clamped edges) by using the special form of polynomials as displacement functions\(^6\). Thus, the Ritz method is appropriate for the present purpose where research is attempted for the first time for the fibrous composite plates with curvilinear fibers. In contrast, the finite element method (FEM) is not suitable for this purpose because of its complex input parameters, particularly in using a different fiber orientation angle in each element.

For the small amplitude (linear) free vibration of a thin plate, the deflection \(w\) may be written as

\[
w(x, y, t) = W(x, y) \sin \omega t \tag{9}\]

where \(W\) is the amplitude and \(\omega\) is an angular frequency. Then, the maximum strain energy due to the bending is expressed by

\[
U_{\text{max}} = \frac{1}{2} \int \int \kappa \left[ \begin{array}{ccc}
D_{11} & D_{12} & D_{16} \\
D_{21} & D_{22} & D_{26} \\
D_{61} & D_{62} & D_{66}
\end{array} \right] \kappa \ dA \tag{10}
\]

where \(\kappa\) is a curvature vector. The maximum kinetic energy is given by

\[
T_{\text{max}} = \frac{1}{2} \rho \omega^2 \int \int W^2 \ dA \tag{11}
\]

where \(\rho\) is an average mass per unit area.

In the Ritz method, the amplitude is assumed in the form

\[
W(\xi, \eta) = \sum_{m=0}^{M} \sum_{n=0}^{N} A_{mn} X_m(\xi) Y_n(\eta) \tag{12}
\]

where \(A_{mn}\) are unknown coefficients, and \(X_m(x)\) and \(Y_n(y)\) are the functions modified so that any kinematical boundary conditions are satisfied at the edges with “boundary indices”\(^6\).

After substituting Eq.(12) into the sum of energies (10) and (11), the stationary value is obtained by

\[
\frac{\partial (T_{\text{max}} - U_{\text{max}})}{\partial A_{mn}} = 0 \quad (m = 0, 1, ..., M; \ n = 0, 1, ...). \tag{13}
\]

The minimizing process gives a set of linear simultaneous equations in terms of the coefficients \(A_{mn}\), and the eigenvalues \(\Omega\) may be extracted by using existing computer subroutines. The eigenvalue \(\Omega\) is a frequency parameter defined as

\[
\Omega = \omega a^2 \frac{\rho h}{D_0} \tag{14}
\]

where \(D_0 = E_t h^3/12(1 - v_{LT}v_{TL})\) is a reference bending stiffness and \(h\) is plate thickness.
2.4 Numerical integration

To calculate the variable stiffness, a numerical integration technique is employed\(^{(9)}\). The plate is divided into \(n\) parts in the \(x\) direction and the orthotropic material constants, whose angle is a function of \(x\) defined by Eqs.(4), (6) and (8), are assigned for each minute interval of integration. Then, the integration in the Eq.(10) for determining the maximum strain energy is done by using numerical integration. In this study, the trapezoid formula is applied as an integration technique due to its simplicity and easiness for including local anisotropy.

The integration \(S\) of \(y = g(x)\) between a lower limit \(x_0\) and an upper limit \(x_n\) is given for equally spaced abscissas by

\[
S \approx \sum_{k=0}^{n} \Delta S = \sum_{k=0}^{n} \left( \Delta x \cdot \frac{y_k + y_{k+1}}{2} \right)
\]  

where \(\Delta S\) is the area of an incremented trapezoid with the orthotropic material constants defined at \(x = x_k\), and \(\Delta x\) is a constant step in the \(x\) direction defined by

\[
\Delta x = \frac{(x_n - x_0)}{n}.
\]

A comparison is made with the Simpson method to confirm the accuracy of the trapezoid formula. The Simpson method is a technique where whole integration interval is divided into \(2n\) parts, and the three consecutive points are interpolated by a quadratic function. An equation \(g(x) = x \cos(\tan^{-1}x)\) is used for a test function that is a typical function form in this study. Table 1 lists relative errors (%) between an exact integral value of the test function and the results of trapezoid and Simpson’s formulas with divisions from 10 to \(10^7\). The calculated errors are based on the exact value. The errors become almost the same order when the number of partition is \(10^3\). The trapezoid formula gives more accurate values for over \(10^3\) partitions than Simpson’s formula probably due to round-off errors. Therefore, the trapezoid formula with 100 (\(10^2\)) partitions is used for integration in this study by taking the computation time and accuracy into consideration.

<table>
<thead>
<tr>
<th>No. of partition</th>
<th>Trapezoid</th>
<th>Simpson</th>
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<td>10</td>
<td>-3.813E-02</td>
<td>1.521E-04</td>
</tr>
<tr>
<td>(10^2)</td>
<td>-3.821E-04</td>
<td>2.503E-06</td>
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<td>-8.130E-06</td>
</tr>
<tr>
<td>(10^7)</td>
<td>6.081E-07</td>
<td>1.275E-01</td>
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</table>

3. Numerical Results and Discussion

3.1 Comparison between the present plate and the conventional plate

Numerical results are given in this study for single layer square plates \((a / b = 1)\) with elastic constants of CFRP (graphite / epoxy):

\[E_L = 138 \text{ GPa}, \quad E_T = 8.96 \text{ GPa}, \quad G_{LT} = 7.1 \text{ GPa}, \quad \text{and} \quad \nu_{LT} = 0.30\]

The boundary conditions of the plates are identified by the four capitals in F, S and C written in counterclockwise starting from the plate left edge. The F, S and C denote free,
simply supported and clamped edges, respectively. For example, CCCC plate indicates a
totally clamped plate and CFFF plate is a cantilever plate with clamped left edge.

Instead of using a proportion constant \( A \) for parabola in Eq.(7), a non-dimensional
parameter \( \gamma \) is adopted in the calculation and defined by

\[
\gamma = \frac{B}{a/2} = \frac{A}{\frac{a}{2}}
\]

where \( B \) is a vertical distance in the \( y \) direction from the point at the intersection of parabola
and plate edges to the turning point (see Fig.2). The shape of parabola is controlled by \( \gamma \); for
instance, when \( \gamma = 0 \), the plate becomes unidirectional orthotropic plate reinforced by
straight fibers parallel to the \( x \) axis, and when \( \gamma = \infty \), it is a unidirectional reinforced plate
with straight fibers parallel to the \( y \) axis. For \( \gamma = 1 \), the plate is reinforced along curved
fibers parallel to a parabola passing at three points \((-a/2, 0), (0, -a/2)\) and \((a/2, 0)\).

Figure 3 shows variations of the lowest four frequency parameters \( \Omega \) of totally clamped
(CCCC) square plates with parabolically shaped fibers when the parameter \( \gamma \) is varied from
0.1 to 100. The frequency parameters are obtained in (a) by the present Ritz method, while
the parameters are calculated in (b) by the finite element method (FEM). The general
purpose FEM software (ANSYS) is employed in this study. As explained, 100 partitions are
used to numerically integrate the strain energy (see section 2.4) in (a). On the other hand,
the plate is divided into 20 \( \times \) 20 elements in the FEM. Thus the number of divisions in the
horizontal direction is 20 and in each element the straight fibers are assumed. So, the shape
of parabolically shaped fibers is approximated more roughly than those of the present plate
calculation. If the FEM plate model is divided into 100 partitions in one direction, the total
number of division has to be 100 \( \times \) 100 to keep the accuracy in the other direction, which
causes not only high calculation cost but also tedious input process. Moreover, if the shape
of fibers is changed, such lengthy input process has to be done again. Therefore, simple
input process is one of the advantages of the Ritz method when the geometry of the
parabolically shaped fibers is varied continuously.

Comparison of Figs.3(a) and (b) reveals that the frequency parameters from both
methods give similar variations although the FEM results have globally lower values than
the present ones. Only monotonous variations are seen for range of \( \gamma > 10 \), since the
parabolas become sharp enough to be regarded as the straight fibers parallel to the vertical
lines in the plate region. Therefore, the value of \( \gamma = 10 \) is taken as the upper limit of the
coefficient parameter in the following discussion. In addition, the same tendency is found
for other sets of boundary conditions, and the accuracy of the present calculation is well
confirmed.

![Fig.3 Frequency parameters of totally clamped (CCCC) square plates with parabolically shaped fibers obtained by the present method and FEM](image-url)
Fig. 4 Comparison of frequency parameters between the present plate with parabolically shaped fibers and the conventional one with straight fibers (SSSS, CCCC, CSFF, and CFCS square plate)
Figures 4(a)-(d) show comparisons of the lowest four frequency parameters between the present plate (with parabolically shaped fibers) and the conventional plate (with straight parallel fibers) for totally simply supported plate (SSSS), totally clamped plate (CCCC), and nonuniform boundary conditions (CSFF and CFCS). The left figures in Figs.4(a)-(d) present frequency parameters of the present plates when the coefficient parameter $\gamma$ is varied similarly as in Fig.3, except that $\gamma = 10$ is the upper limit. The right figures in Figs.4(a)-(d) present those of the conventional plates when the fiber orientation angles are varied from 0° to 90°. As expected, composite plates shown in the left and right figures have almost the same frequencies at $\gamma = 0.1$ and $\theta = 0^\circ$, because both plates become uni-directionally orthotropic plates in the $x$ direction. Similarly at $\gamma = 10$ and $\theta = 90^\circ$, both plates are considered to be uni-directionally stiffened in the $y$ direction. For $\gamma = 1$, all the fibers take parabola curves that are translated from a quadratic curve passing the points $(-a/2, a/2)$, $(0, 0)$ and $(a/2, a/2)$. This fiber structure may be similar to that of the conventional square plate having the fiber directions of -45° and 45° in the left and right halves, respectively.
It is commonly seen in Figs.4(a)-(d) that the higher order frequencies are influenced more by the parameters $\gamma$ and $\theta$, although the first frequencies are not so influenced (except in (d)) by the parameters. The frequency variations of the present plates are more sensitive with the parameter change than those of conventional plates, and they come close each other and veer away. These phenomena are called "curve veering".

More detailed observation is made in each figure. For simply supported plates (SSSS) in Fig.4(a), the second frequency parameter of the present plate is $\Omega = 96.21$ at $\gamma = 1.2$, which is higher than the maximum frequency of 83.99 at $\theta = 45^\circ$. Similarly for totally clamped plates (CCCC) in Fig.4(b), the second frequency $\Omega = 134.2$ of the present plate at $\gamma = 1.3$ is higher than the maximum frequency $\Omega = 130.9$ of the conventional plate at $\theta = 45^\circ$. In Figs.4(c) and (d) (CSFF and CFCS), the third frequencies of the present plate are $\Omega = 71.34$ and 121.1 at $\gamma = 0.7$ and 1.0, respectively. On the other hand, the conventional plates give the maximum third frequencies $\Omega = 63.34$ and 115.1 at $\theta = 30^\circ$ and 35$^\circ$, respectively. These results induce that the present plate gives higher frequencies than the conventional plate for a certain range of coefficient parameter $\gamma$ of the fibers, and this coefficient parameter is around $\gamma = 1.0$ when higher frequencies are obtained. This means that normal parabola (i.e., not extremely sharp or flat) fibers stiffen more than the conventional straight fiber structure. It is thus concluded that the fibrous composite plates with continuous curvilinear fibers may be designed more effectively than the conventional composite plates with straight parallel fibers.

3.2 Vibration mode

To study effects of curvilinear fibers on the vibration modes, the vibration mode shapes are presented in Fig.5(a)-(d) for the same boundary conditions with Fig.4. All the plates have the coefficient parameter $\gamma = 1.0$. The mark “×” represents the maximum displacement point and the thick lines represent the nodal lines (i.e., lines of zero displacement). The thin lines denote the displacement contour lines, where the interval between the maximum and nodal points is divided into ten equal increments.

In observing the vibration modes, the mode shapes of each plate are clearly affected by the parabolic fibers. The first and third modes of SSSS and CCCC (Fig.5(a) and (b)) have skewed peaks which are asymmetric with respect to $x$-axis, and the second and fourth modes have curved nodal lines, while an isotropic plate has symmetric peaks and straight nodal lines. Also the plates considered here are square and the boundary conditions are completely symmetric, but (1, 2) and (2, 1) modes give different mode shapes due to curved fibers, where $m$ and $n$ in $(m, n)$ are half wave numbers in the $x$ and $y$ directions, respectively. The CSFF and CFCS (Fig.5(c) and (d)) also have reasonably distorted shapes. The vibration modes of the present plates thus reflect the shape of fibers apparently and therefore the validity of the present analysis is well supported.

Figure 6 presents the mode shapes of the present plates and conventional plates with four sets of boundary conditions in the first and second rows, respectively, when the maximum frequencies are given for second or third modes. In the third row, modes of isotropic plate are presented. Since the first frequencies are not so significantly influenced by the each control parameters (coefficient parameter $\gamma$ for the present plate and fiber orientation angle $\theta$ for the conventional plate), the second modes for SSSS and CCCC and the third modes for CSFF and CFCS are presented when they have the maximum frequencies with respect to each parameter. The frequency values were already mentioned in the section 3.1.

There are no differences between the mode shapes of the present plates with maximum frequencies (Fig.6(a) and (b)) and the second mode with $\gamma = 1$ (Fig.5(a) and (b)) for CCCC and SSSS. For CSFF (Figs.5(c) and 6(c)), transition of the mode shape is observed from the horizontal line to the vertical one for $\gamma = 0.7 - 1.0$. For CFCS (Figs.5(d) and 6(d)), the same third mode is given for $\gamma = 1$. 
4. Conclusions

An analysis method was proposed to determine natural frequencies and mode shapes of the single-ply fibrous composite plates with continuous curvilinear fibers. As the first step in the analysis for such new types of composite plates, parabolically shaped fibers were assumed in the reinforced material. Since the bending stiffness varies continuously in the plate, it was approximated by assigning the orthotropic material constants to each minute interval of numerical integration, and the Ritz method was used due to its simplicity for deriving the eigenvalue problem.

The validity of the present calculation was shown by comparing the present results with the FEM results. When the frequency variations of the present plates with parabolically shaped fibers were compared to those of the conventional plates with straight parallel fibers, it was found that the present plates have higher frequencies for a certain range of the fiber geometric coefficients than the conventional ones. It is therefore concluded that there is a strong possibility to design more effectively the composite plates by optimizing continuous curvilinear fiber path. The vibration modes of the present plates reflected strong effect of the parabolically shaped fibers, and were compared with those of conventional plates and isotropic plates to study effects from the difference of the fiber structures.

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