Vibration analysis of stiffened plates using Finite Element Method

Abstract
This paper presents the vibration analysis of stiffened plates, using both conventional and super finite element methods. Mindlin plate and Timoshenko beam theories are utilized so as to formulate the plate and stiffeners, respectively. Eccentricity of the stiffeners is considered and they are not limited to be placed on nodal lines. Therefore, any configuration of plate and stiffeners can be modeled. Numerical examples are proposed to study the accuracy and convergence characteristics of the super elements. Effects of various parameters such as the boundary conditions of the plate, along with orientation, eccentricity, dimensions and number of the stiffeners on free vibration characteristics of stiffened panels are studied.

Keywords
vibration, stiffened plates, Finite Element Method, super element.

1 INTRODUCTION
Noise and vibration control is an increasingly important area in the most fields of engineering. There are many vibrating parts in structures of ships, aircrafts and offshore platforms. The amplitude of their motions can be large due to the inherently low damping characteristics of these structures. Such noise is commonly eradicated by use of heavy viscoelastic damping materials which lead to increase in cost and weight. Vibration isolators between pieces of equipment and their supporting structures can be another solution. Clearly, isolating large structures can be difficult, expensive and in some cases, such as the wings of an aircraft, almost impossible. In recent years, much attention has been focused on active noise control of structures. However, their installation and maintenance can be expensive, so possible passive solutions would be preferable [20].

In the case of plates/shells, one common and cost effective approach in order to improve their NVH\(^1\) performance is to add stiffeners. Stiffened plates are lightweight, high-strength

\(^{1}\)Noise Vibration Harshness
structural elements, commonly used in ships, aircrafts, submarines, offshore drilling rigs, pressure vessels, bridges, and roofing units [19, 21]. Most of these structures are required to operate in dynamic environments. Therefore, a thorough study of their dynamic behavior and characteristics is essential in order to develop a perfect strategy for modal vibration control [8]. The stiffeners enhance the rigidity of base structures by increasing their cross sectional second moment of inertia. The configuration of the stiffeners should be consistent with the natural modes likely to be excited by the service loads, so as to arrive at a design with a high strength-to-weight ratio [4]. In general, the stiffening of the structures is applied, because of two main reasons: Increasing load carrying capacity and preventing buckling, especially in the case of in-plane loading [6].

Different geometries of stiffened shells have been studied in the literature which can be categorized into three groups including plates, single curved shells, and double curved shells. The Superimposition of the stiffeners with respect to the plate mid-plane, i.e. eccentric or concentric is also a matter of concern for the structural analysts. The stiffener of which, centroid is coincident with the plate/shell mid-surface, is called concentric, otherwise eccentric stiffener [16].

The analysis of stiffened plate vibration has been the purpose of numerous investigations. Among the known solution techniques, the finite element method is certainly the most favourable. Olson and Hazell [14] predicted and measured the first 24 eigenfrequencies of stiffened plates using FEM and real-time laser holography. Mustafa and Ali [12] developed an eight-noded orthogonally stiffened super finite element to study the free vibration of a stiffened cylindrical shell with diaphragm ends. Experimental measurements of natural frequencies and mode shapes of an orthogonally stiffened shell were also carried out to substantiate theoretical predictions. A plate with centrally placed stiffener has been studied by Mukherjee and Mukhopadhyay [11]. An isoparametric stiffened plate element has been utilized in their analysis considering shear deformation in order to analyse thick as well as thin plates. In the proposed formulation, the stiffeners can be placed anywhere within the plate element and they are not required to necessarily follow the nodal lines. Koko and Olson [9] have developed another super element to model the free vibration of stiffened plates. This super element allows a coarser mesh (at the expense of more complex interpolation functions), so that only a single element per bay or span is needed. Sinha and Mukhopadhyay [18] investigated stiffened shells utilizing a high-precision triangular shallow shell element in which stiffeners can be anywhere within the plate element. The vibration analysis of stiffened plates has been investigated by Barrette and Beslin [3] using hierarchical finite elements with a set of local trigonometric interpolation functions. Nayak and Bandyopadhyay [13] presented a finite element analysis for free vibration behaviour of doubly curved stiffened shallow shells. The eight-/nine-node doubly curved isoparametric thin shallow shell elements along with the three-node curved isoparametric beam element has been used in this study. Their Formulation suffers from the limitation, that stiffeners can only be placed along nodal lines in x or y directions. Two years later Samanta and Mukhopadhyay [17] developed a new 3 noded stiffened shell element and applied it in determining natural frequencies and mode shapes of the different stiffened structures.
Another stiffened element with seven degrees of freedom per node has been presented by Voros [21]. Torsion-flexural coupling, torsional warping effect and the second-order terms of finite rotations have been considered in this investigation.

In the present work, four different elements including two conventional and two super elements [2] have been used so as to predict the dynamic characteristics of stiffened panels. The formulation of the plate and stiffeners are both based on the first order shear deformation theories so it can be applicable to both thin and thick plates. In the proposed formulation, the stiffeners are modeled in such a way that they can be placed anywhere within the plate element. This may be considered as a prominent advantage over most of approximate analyses, since this method can be applied to any plate and stiffeners configuration. It is worth to mention that, the shape function of the plate is also used to express the displacement of the stiffener at any generic point along it. In this way, displacement compatibility between the plate and the stiffeners is ensured automatically in the whole continuum and no additional node is utilized for the stiffeners.

After studying the accuracy and convergence of super elements, they have been utilized to investigate different problems. Because of significantly less time and fewer global DOFs needed for super elements, they are useful for preliminary designs and parametric studies; where, repeated calculations are often required. Effects of various parameters such as the orientation, eccentricity and number of stiffeners on free vibration characteristics have been studied. These examples demonstrate the strength of the developed formulation, and it is hoped that the results presented will prove useful to other researchers. Up to authors’ knowledge, such results have not been published before.

2 FORMULATION

The equation of motion for free vibration of elastic bodies, with infinitesimal displacements is:

$$[M]\{\ddot{d}\} + [K]\{d\} = \{0\},$$  

Where $[M]$ and $[K]$ is overall mass and stiffness matrix, respectively. $\{d\}$ is the displacement vector and dots denote derivatives with respect to time. Overall matrices in equation (1) are obtained by assembling matrices corresponding to each element, and applying appropriate boundary condition. In this paper, the following hypotheses are made:

- The material of the plate and the stiffeners is isotropic, linear elastic and Hookian.
- In-plane displacements are neglected in order to reduce the computational time. If the plate edges are immovable in the plane, the in-plane displacements will be much smaller than out-of-plane ones. Therefore, in such cases this can be a rational assumption.
- Stresses in the direction normal to the plate middle surface are negligible.
- Normal to the undeformed mid-plane remains straight and unstretched in length, but not necessarily normal to the deformed mid-plane. This assumption implies the consideration
of shear deformation, but it also leads to the nonzero shear stresses at the free surface, because of constant shear stress through the plate thickness.

- Rotary inertia effect is included.
- The magnitude of transverse deflection \( w \) is small in comparison to the plate thickness \( h \).

Stiffness and mass matrices corresponding to each element are the summation of the plate stiffness and mass matrices, and the contributions of stiffeners to this element as

\[
[K] = [K_p] + [K_s],
\]

\[
[M] = [M_p] + [M_s],
\]

in which plate and stiffener are denoted by subscripts \( p \) and \( s \), respectively \([5, 7, 10]\).

### 2.1 Plate element

A flat, thin/thick plate of uniform thickness is considered. As it was assumed, constitutive material is homogeneous, linear elastic and Hookian. For the purpose of finite element modeling 4 types of element are used, including eight-/nine-node conventional elements and eight-/twelve-node super elements. Each node has 3 degrees of freedom. They consist of one displacement in transverse direction, and two rotations about x-axis and y-axis. Displacement at each point within the element is related to nodal values by

\[
\begin{bmatrix}
w \\
\theta_x \\
\theta_y \\
\end{bmatrix} = \sum_{i=1}^{n} N_i \begin{bmatrix} w_i \\ \theta_{xi} \\ \theta_{yi} \end{bmatrix},
\]

where \( i \) and \( n \) are the corresponding node and total number of nodes in a the plate element, and \( N_i \) is the shape function of the \( i^{th} \) node. In the isoparametric formulation the above functions are used for defining the displacement as well as the location of any point within the element in terms of nodal coordinates.

\[
\begin{bmatrix}
x \\
y \\
\end{bmatrix} = \sum_{i=1}^{n} N_i \begin{bmatrix} x_i \\ y_i \end{bmatrix},
\]

Implementing Mindlin plate theory, displacement field can be expressed as follow (figure 1)

\[
\begin{bmatrix}
U \\
V \\
W \\
\end{bmatrix} = \begin{bmatrix}
-z\theta_y \\
-z\theta_x \\
w \\
\end{bmatrix}.
\]
in which $z$ is measured from the neutral surface of whole structure consists of the plate and one or more stiffeners. Based on linear elasticity the strain component are given by

$$
\begin{bmatrix}
\varepsilon_x \\
\varepsilon_y \\
\varepsilon_{xy} \\
\varepsilon_{xz} \\
\varepsilon_{yz}
\end{bmatrix} =
\begin{bmatrix}
-z \frac{\partial \theta_y}{\partial x} \\
-z \frac{\partial \theta_x}{\partial y} \\
-z \left( \frac{\partial \theta_x}{\partial x} + \frac{\partial \theta_y}{\partial y} \right) \\
\frac{\partial ^2 w}{\partial x^2} - \theta_y \\
\frac{\partial ^2 w}{\partial y^2} - \theta_x
\end{bmatrix},
$$

(7)

$$\{ \varepsilon_p \} = \{ \varepsilon_x \ \varepsilon_y \ \varepsilon_{xy} \ \varepsilon_{xz} \ \varepsilon_{yz} \},
$$

(8)

Where $\varepsilon_x$ and $\varepsilon_y$ are normal strains, and $\varepsilon_{xy}$, $\varepsilon_{xz}$ and $\varepsilon_{yz}$ are shear strains. For isotropic, linear elastic, Hookian Materials

$$\{ \sigma_p \} = [D_p] \{ \varepsilon_p \},
$$

(9)

$$D_p =
\begin{bmatrix}
\frac{E}{1-\nu^2} & \frac{\nu E}{1-\nu^2} & 0 & 0 & 0 \\
\frac{\nu E}{1-\nu^2} & \frac{E}{1-\nu^2} & 0 & 0 & 0 \\
0 & 0 & G & 0 & 0 \\
0 & 0 & 0 & \kappa G & 0 \\
0 & 0 & 0 & 0 & \kappa G
\end{bmatrix},
$$

(10)

$$\{ \sigma_p \} = \{ \sigma_x \ \sigma_y \ \tau_{xy} \ \tau_{xz} \ \tau_{yz} \},
$$

(11)

Where $\sigma_x$ and $\sigma_y$ are normal strains, $\tau_{xy}$, $\tau_{xz}$ and $\tau_{yz}$ are shear strains, $E$ is the elasticity modulus, $G$ is the shear modulus, $\nu$ is the poison’s ratio and $\kappa$ is the shear correction factor to compensate the error due to the assumption of constant shear strains within the plate thickness. In this stage, strain energy functional of the plate element can be obtained

$$U_p = \frac{1}{2} \int \{ \varepsilon_p \}^T \{ \sigma_p \} \ dV,
$$

(12)
In which $V$ is the volume of the plate element. The kinetic energy of vibrating plate element is also given by

$$T_p = \frac{1}{2} \int \rho \left[ \left( \frac{\partial U}{\partial t} \right)^2 + \left( \frac{\partial V}{\partial t} \right)^2 + \left( \frac{\partial W}{\partial t} \right)^2 \right] dV,$$

(13)

In which $t$ denotes time and $\rho$ is the plate mass density. The integrations are calculated using gauss quadrature scheme. Special considerations are applied so as to avoid shear locking effect [5, 7, 10, 15, 22].

2.2 Stiffener

In this section, the matrices corresponding to the stiffener which can be placed everywhere within the plate element are extracted. The proposed method releases the formulation from the limitation of stiffeners to be lied along nodal lines. Hence, oblique stiffeners can be analysed. For this purpose, by use of same shape function for both the plate and the stiffeners, displacement compatibility is guaranteed and no additional node is introduced for the stiffeners. The stiffener specifications are calculated at the Gauss points along it. Therefore a transformed coordinate system is implemented (Figure 2). Based on Timoshenko beam theory, displacement field of the stiffener is given by

$$\begin{bmatrix} U' \\ V' \\ W' \end{bmatrix} = \begin{bmatrix} -z\theta_y' \\ -z\theta_x' \\ w \end{bmatrix},$$

(14)

$$\begin{bmatrix} \theta_x' \\ \theta_y' \end{bmatrix} = \begin{bmatrix} \cos \phi & \sin \phi \\ -\sin \phi & \cos \phi \end{bmatrix} \begin{bmatrix} \theta_x \\ \theta_y \end{bmatrix},$$

(15)

where $\theta_x'$ and $\theta_y'$ are rotations about $x'$ and $y'$, respectively. As mentioned before, $z$ is measured from the reference surface of whole structure. Displacement functions can be expressed in terms of nodal values of plate element using a transformation matrix (Eq. 15) and aforementioned shape functions.
Again, by applying linear elasticity

\[
\begin{bmatrix}
\varepsilon_{x'} \\
\varepsilon_{x'z'} \\
\varepsilon_{x'y'}
\end{bmatrix} =
\begin{bmatrix}
-z \frac{\partial \theta_{y'}}{\partial x'} \\
\frac{\partial w}{\partial x} - \theta_{y'} \\
-z \frac{\partial \theta_{x'}}{\partial x'}
\end{bmatrix},
\]

(16)

\[
\{\varepsilon_s\} = \{\varepsilon_{x'}, \varepsilon_{x'z'}, \varepsilon_{x'y'}\},
\]

(17)

Similar to the plate element, for an isotropic material

\[
\{\sigma_s\} = [D_s] \{\varepsilon_s\},
\]

(18)

\[
D_s = \begin{bmatrix}
\frac{E}{1-\nu^2} & 0 & 0 \\
0 & \kappa G & 0 \\
0 & 0 & G
\end{bmatrix},
\]

(19)

\[
\{\sigma_s\} = \{\sigma_{x'}, \tau_{x'z'}, \tau_{x'y'}\},
\]

(20)

In which \(\sigma_{x'}\) is the normal stress along the longitudinal axis of the stiffener, \(\tau_{x'z'}\) is shear strain and \(\tau_{x'y'}\) is torsional strain. So energy functional of the stiffener are obtained as follow

\[
U_s = \frac{1}{2} \int \{\varepsilon_s\}^T \{\sigma_s\} dV,
\]

(21)

\[
T_s = \frac{1}{2} \int \rho \left[ \left( \frac{\partial U'}{\partial t} \right)^2 + \left( \frac{\partial V'}{\partial t} \right)^2 + \left( \frac{\partial W'}{\partial t} \right)^2 \right] dV',
\]

(22)

in which \(V'\) is the volume of the part stiffener, confined within the plate element. The governing equation of vibration of stiffened plates is derived by use of Hamilton’s principle which requires

\[
d \int_{t_i}^{t_f} [(T_p + T_s) - (U_p + U_s)] dt = 0,
\]

(23)

where \(d\) is the variational operator. Now stiffness and mass matrices of the stiffened element can be calculated. By assembling matrices corresponding to each element in a suitable manner, overall stiffness and mass matrices are obtained. Finally, boundary conditions are applied and governing eigenvalue equation for un-damped free vibration of stiffened plates takes the form

\[
[K] \{d\} = \omega^2 [M] \{d\},
\]

(24)

where \(\omega\) is the natural frequency in radian per second, and \(\{d\}\) represents eigenvector [5, 7, 10, 15, 22].
2.3 Super element

The super element is a compound one, which is composed of a number of conventional finite elements. In this regard, each super element is divided into a number of conventional elements and corresponding matrices to each subdivided element are constructed (Figure 3).

![Figure 3 Super elements.](image)

Subdivided elements’ matrices are assembled, as in general finite element procedure. Degrees of freedom related to servant nodes, defined later, are then condensed out while the effect of them on super element matrices is taken into account. This procedure is named as dynamic condensation, which is described in the following. The dynamic equilibrium equation neglecting damping effect is given by,

\[
\begin{align*}
\begin{bmatrix}
M_{ss} & M_{sc} \\
M_{cs} & M_{cc}
\end{bmatrix}
\begin{bmatrix}
\ddot{d}_s \\
\ddot{d}_c
\end{bmatrix}
+ \begin{bmatrix}
K_{ss} & K_{sc} \\
K_{cs} & K_{cc}
\end{bmatrix}
\begin{bmatrix}
\dot{d}_s \\
\dot{d}_c
\end{bmatrix}
= \begin{bmatrix}
F_s \\
F_c
\end{bmatrix},
\end{align*}
\]  
(25)

Where index s denotes the terms related to the DOFs of super nodes that will remain after condensation, and c represents the terms related to other DOFs which will condensed out through this procedure. They are sometimes called as master and servant nodes, respectively. Using appropriate transformation matrix the following equation is obtained,
In this way, the effect of servant node is also embedded in the condensed super element. Aforementioned element has remarkable ability to reduce the size of problem.

3 NUMERICAL EXAMPLES

3.1 Vibration of a Mindlin plate with different boundary conditions

In order to assess the vibration behaviour of the super plate elements, free vibration of a square Mindlin plate with different boundary conditions is first investigated. This problem has been studied by several investigators using exact or other approximate methods. The natural frequencies are presented in terms of the non-dimensional eigenvalue \( \lambda \), given by

\[
\lambda = \frac{\rho h \omega^2 a^4}{D \pi^4},
\]

(28)

where \( D \) is the plate flexural rigidity, \( h \) is the plate thickness, and \( a \) the side length of the plate. The eigenvalues for the various combinations of edge conditions are shown in Table 1.

The comparison between present and other numerical results shows a good agreement in all ten cases considered. It should be noted that, unlike super element developed by Koko and Olson [9] the present super elements result in accurate values for second and higher modes. It is also indicated that, implementing super elements leads to reduction in computational time with no significant change in results accuracy, which is expected because of the significantly fewer number of global DOFs, used in the super elements.

3.2 Clamped square plate with a central stiffener

To validate the present formulations and study the characteristics of super elements, a stiffened clamped square plate with central stiffener (Figure 4) is analyzed. \( (E = 6.87 \times 10^{10} \ N/m^2, \nu = 0.3, \rho = 2823 \ kg/m^3) \)

![Figure 4 Clamped square stiffened plate with one central stiffener.](image)
The first ten natural frequencies from the super elements are compared with the finite element and experimental results in Table 2. A fundamental modeling difference between the super element developed by Koko and Olson [9] and present elements, is that they allow for in-plane displacements of the plate, whereas the present theory assumes pure bending deformation of the plate. However, such effects are expected to be small for thin plates.

All frequencies from the present approach are very close to the experimental and conventional finite element results. The agreement between the super elements and the experimental results reported by Olson and Hazell is wonderful. Although deviation of the eigenfrequencies computed employing S12 from mentioned results is negligible, based on authors’ point of view results obtained by S8 are more accurate. Because Due to the finite stiffness of any clamped structure, exact edge conditions are not possible in practice. Consequently numerical results must be on the stiff side of experimental ones.

Moreover, it can be seen from the table that there is a significant reduction in the global DOFs and consequently in computational time by utilizing super elements, compared to the conventional finite elements. As another conclusion, effect of neglecting in-plane deflections on natural frequencies isn’t considerable in this case.

<table>
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<tr>
<th>Boundary condition</th>
<th>Mode Shape</th>
<th>Durvasla et al. [9]</th>
<th>Koko and Olson [9]</th>
<th>Present Analysis</th>
<th>Q8</th>
<th>Q9</th>
<th>S8</th>
<th>S12</th>
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Global DOF 1443 1323 288 192
Run Time [S] 59.4288 46.8052 2.434 3.0184
Table 2  Natural frequencies of Clamped square stiffened plate with one central stiffener.

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<td>10</td>
<td>2069</td>
<td>2025</td>
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</tr>
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| Global DOF | 1443 | 1323 | 1023 | 651 |
| Run Time [S] | 52.6695 | 41.4733 | 18.0866 | 9.7595 |

3.3 Clamped square plate with two parallel stiffeners

This configuration has first been studied by Olson and Hazell [14] using experimental and conventional finite element method (Figure 5). Boundary conditions and material properties are same as the previous example.

Eigenfrequencies obtained from the present analysis are presented in Table 3 along with the previous numerical and experimental results.

Again, there are significantly fewer global DOFs used in the super element models than the conventional finite element models, which leads to considerable reduction in computational time. Present super elements predictions are almost in agreement with the results obtained from previous super element results [14]. But there is relatively remarkable difference with the experimental results. The reason may be

1. Perfectly clamped edges are impossible in practice, so experimental results are lower than the numerical ones, as said.
2. In-plane deflections are neglected in this investigation. With increase in number of stiffeners these displacements play more important role. Although this simplification leads to less computational time, can affect results when number of stiffeners increases. Because in this case deviation of reference surface of whole structure from plate midplane and consequently first moment of inertia gets more considerable.

Eigenfrequencies from present conventional elements are always on the stiff side of experimental values except for two modes, contrary to what is expected of a displacement based theory. It is possible that the experimental procedure have not measured the frequencies of those modes accurately. In the case of super elements, again S8 leads to more acceptable results, especially for the first three modes. Therefore, the viability of the super element formulation is clearly exhibited in that most of the super element solutions are close to the experimental and conventional finite element results, even though this method uses significantly fewer DOFS.

3.4 Convergence study

The accuracy of numerical calculations depends on the number of the divisions. Convergence characteristics of the elements used with respect to number of divisions is studied in this section. The Analysis is performed on a clamped square plate with central stiffener (Figure 4). Results are shown in Figure 6.

It can be seen obviously from the Figure 6 that the conventional elements have faster rate of convergence compared to super elements. In fact, super elements do not converge necessarily. Although these elements lead to acceptable results with low number of elements, recedes from the exact eigenfrequencies with an increase in number of divisions. The reason may be eliminating internal nodes and making the super element softer. As another conclusion, 2 and 3 elements per bay is needed for analyzing a stiffened plate using the S12 and S8 elements,
respectively, but in other cases 5 elements should be employed in each bay. The rest of present calculations are based on mentioned division numbers.

3.5 Effect of eccentricity

To investigate the effect of eccentricity on the free vibration of stiffened plates, a clamped square plate with one, two and three stiffeners has been analysed for both concentric and eccentric types (Figure 7). \( E = 2.06 \times 10^{11} \text{ N/m}^2, \nu = 0.3, \rho = 7650 \text{ kg/m}^3 \)

The results obtained from such case are presented in Table 4. The effect of eccentricity on eigenfrequencies has been observed.

It is interesting to note that the inclusion of eccentricity does not affect the values of natural frequencies of the clamped plate with merely one stiffener. As the number of stiffeners increases, effect of eccentricity is more insignificant. Deviation of neutral surface of whole structure, with respect to which second moments of inertia, should be calculated from mid
Table 4 Natural frequencies of a clamped square plate with concentric and eccentric stiffeners.

<table>
<thead>
<tr>
<th>Stiffener No.</th>
<th>Mode No.</th>
<th>Natural Frequency [Hz]</th>
<th>Eccentric</th>
<th>Concentric</th>
<th>Eccentric</th>
<th>Concentric</th>
<th>Eccentric</th>
<th>Concentric</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>1</td>
<td>131.8507</td>
<td>132.2772</td>
<td>203.8146</td>
<td>202.7164</td>
<td>270.4664</td>
<td>263.0167</td>
<td></td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>155.8791</td>
<td>154.4151</td>
<td>254.2835</td>
<td>251.3998</td>
<td>279.7375</td>
<td>277.2788</td>
<td></td>
</tr>
<tr>
<td></td>
<td>3</td>
<td>196.5682</td>
<td>197.046</td>
<td>255.8007</td>
<td>253.8769</td>
<td>362.572</td>
<td>358.7522</td>
<td></td>
</tr>
<tr>
<td></td>
<td>4</td>
<td>219.5702</td>
<td>218.9409</td>
<td>288.6985</td>
<td>291.2963</td>
<td>380.6867</td>
<td>386.0687</td>
<td></td>
</tr>
<tr>
<td></td>
<td>5</td>
<td>296.9567</td>
<td>297.4925</td>
<td>342.6987</td>
<td>342.8742</td>
<td>406.3232</td>
<td>414.3255</td>
<td></td>
</tr>
<tr>
<td></td>
<td>6</td>
<td>312.0192</td>
<td>310.6612</td>
<td>362.2659</td>
<td>358.1533</td>
<td>449.9921</td>
<td>447.2937</td>
<td></td>
</tr>
<tr>
<td></td>
<td>7</td>
<td>320.108</td>
<td>315.2892</td>
<td>394.7799</td>
<td>398.0341</td>
<td>511.1413</td>
<td>507.8233</td>
<td></td>
</tr>
<tr>
<td></td>
<td>8</td>
<td>364.9126</td>
<td>368.3815</td>
<td>430.851</td>
<td>427.1797</td>
<td>512.9898</td>
<td>508.1754</td>
<td></td>
</tr>
<tr>
<td></td>
<td>9</td>
<td>428.1735</td>
<td>430.8037</td>
<td>437.438</td>
<td>438.3834</td>
<td>526.6003</td>
<td>537.712</td>
<td></td>
</tr>
<tr>
<td></td>
<td>10</td>
<td>431.9031</td>
<td>433.4756</td>
<td>458.5447</td>
<td>455.2726</td>
<td>575.1297</td>
<td>578.5875</td>
<td></td>
</tr>
</tbody>
</table>

plane of the plate may be the reason. It is worth to mention that in the case of concentric stiffeners the neutral surface is same as the mid plane of the plate.

From the preceding discussion it can be concluded that the consideration of eccentricity affects the natural frequencies of stiffened clamped plates. But when there is just one stiffener, the effect of eccentricity can be neglected without any significant loss of accuracy. Whether this conclusion is true for other boundary conditions or not, needs more investigations.

### 3.6 Number of stiffeners

The objective of the present example is to study the influence of the number of stiffeners on natural frequencies of square clamped plates. Dimensions and material properties are the same as previous section. Number of stiffeners is changed from 0 to 11 and results are summarized in Figure 8 for the first five modes.

Figure 8 clearly shows that the fundamental frequency is increasing with the increase in the number of stiffeners, which was expected before. However this increase gradually becomes insignificant after a critical value of the number of stiffeners, as generally observed for all 5 modes. This critical value is 4 or 5 stiffeners. Based on authors’ point of view the reason is that, utilizing stiffeners leads to eliminating some mode shapes, so the fundamental frequency increases. But after this critical value, no mode shape elimination occurs and stiffened plate acts as a plate with larger thickness. Therefore providing more than four stiffeners is not recommended based on economical point of view.

### 3.7 Inclination angle

Improving the vibration or noise characteristics of structures by changing its configuration has been a subject that has fascinated the minds of engineers and scientists during last decades [1]. In this section, the orientation angle of the stiffener arranged over a clamped square plate is selected to optimize the dynamic characteristics of these plates/stiffener assemblies (Figure 9).
The objective is to find the inclination angle ($\phi$) for the stiffener arrangement that maximizes the fundamental frequency of the stiffened plate structure. The first natural frequency as a function of the stiffener inclination angle is calculated and plotted in Figure 10.

With due attention to Figure 10 an optimum value for inclination angle of $80^\circ$ was found to maximize the fundamental frequency. The presented approach can be invaluable in the design of stiffened plates for various vibration and noise control applications.

### 3.8 Practical configurations in ship

Two practical configurations which are used practically in the body of ships are analysed. These two cases are square clamped plates with one stiffener (Case 1) and with two orthogonal stiffeners (Case 2) shown in figure 11. The dimensions of the plate and the stiffeners are the same in both cases. The material properties are similar to the previous example.

Calculations have been performed for different thicknesses. Results are presented in Tables 5 and 6.
Table 5: Natural frequencies of a clamped square plate with one stiffener.

<table>
<thead>
<tr>
<th>Mode No.</th>
<th>Natural Frequency [Hz]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>86.4467 84.9931 97.4543 96.5923 108.403 108.305 119.275 120.07 130.063 131.851 62.9993 63.1424</td>
</tr>
<tr>
<td>2</td>
<td>107.066 103.144 121.843 117.709 135.566 131.352 148.248 144.07 159.915 155.879 127.921 128.1</td>
</tr>
<tr>
<td>3</td>
<td>129.431 127.264 146.207 144.824 162.516 162.151 178.555 179.38 194.422 196.568 127.921 128.1</td>
</tr>
<tr>
<td>4</td>
<td>140.835 137.783 161.622 159.078 181.613 179.812 200.857 199.98 219.406 219.57 189.19 189.242</td>
</tr>
<tr>
<td>5</td>
<td>192.135 188.67 218.779 216.512 244.402 243.719 269.304 270.49 293.698 296.957 227.883 227.937</td>
</tr>
<tr>
<td>7</td>
<td>235.767 229.005 268.148 260.942 299.898 282.119 327.159 301.56 342.069 320.108 287.433 287.241</td>
</tr>
<tr>
<td>8</td>
<td>262.72 236.939 288.852 263.251 309.871 297.411 333.857 331.33 361.462 364.913 287.433 287.241</td>
</tr>
<tr>
<td>10</td>
<td>280.729 274.283 320.07 314.748 356.307 353.896 392.332 393.08 427.82 431.903 362.964 362.435</td>
</tr>
</tbody>
</table>

Table 5: Natural frequencies of a clamped square plate with one stiffener.
Table 6  Natural frequencies of a clamped square plate with two orthogonal stiffeners.

<table>
<thead>
<tr>
<th>Mode No.</th>
<th>Thickness</th>
<th>12 mm</th>
<th>14 mm</th>
<th>16 mm</th>
<th>18 mm</th>
<th>20 mm</th>
<th>20 mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>136.508</td>
<td>135.436</td>
<td>151.87</td>
<td>151.789</td>
<td>166.767</td>
<td>167.585</td>
<td>181.527</td>
</tr>
<tr>
<td>2</td>
<td>145.206</td>
<td>143.964</td>
<td>163.12</td>
<td>162.829</td>
<td>179.772</td>
<td>180.543</td>
<td>195.239</td>
</tr>
<tr>
<td>3</td>
<td>147.093</td>
<td>145.637</td>
<td>166.304</td>
<td>165.753</td>
<td>184.786</td>
<td>182.654</td>
<td>201.193</td>
</tr>
<tr>
<td>4</td>
<td>152.084</td>
<td>148.545</td>
<td>171.095</td>
<td>167.503</td>
<td>185.318</td>
<td>202.719</td>
<td>204.454</td>
</tr>
<tr>
<td>5</td>
<td>277.964</td>
<td>257.374</td>
<td>297.104</td>
<td>276.159</td>
<td>312.346</td>
<td>294.133</td>
<td>326.711</td>
</tr>
<tr>
<td>7</td>
<td>287.006</td>
<td>283.365</td>
<td>322.463</td>
<td>321.417</td>
<td>357.026</td>
<td>358.029</td>
<td>388.049</td>
</tr>
</tbody>
</table>
Figure 10 Optimum inclination angle of the stiffener.

Figure 11 Square clamped plate with two orthogonal stiffeners.

Figure 6 and 7 show that adding stiffener to the plate can increase its natural frequency significantly, which was predictable. Moreover, from obtained results, it seems that natural frequencies vary linearly with the thickness of the plate.

4 CONCLUSION

The vibration analysis of stiffened plates using both conventional and super elements has been presented. The capability of placing stiffeners anywhere within the plate element has enabled the proposed formulation to encounter any configuration of stiffened plates. The efficiency of the super elements has been examined with different types of problems. The comparison of the present approach with the existing numerical and experimental results shows remarkable agreement. Although super elements yield acceptable results in significantly short time, conventional elements are superior to them according to their convergence characteristics. As a result, these elements are attractive for preliminary designs and parametric studies, where repeated calculations are often needed. It is also observed that the fundamental frequency of
stiffened plates is increasing with the increase in the number of stiffeners up to a specific number after which there is no appreciable increase in frequency. Moreover, the effect of neglecting the eccentricity of the stiffeners has been studied in details. It is understood that for a clamped plate with only one stiffener eccentricity can be neglected with no considerable change in results. However, effect of eccentricity for more stiffeners should be included. This paper has also presented a rational design approach to optimize the dynamic characteristics of stiffened plates. In order to maximize the fundamental frequency, the optimal orientation angle is found to be equal to 80°. Further works can be undertaken to study hydroelastic analysis of stiffened panels, which may be important in practical point of view for marine structures.

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References


