Journal of Mechanical Design and Vibration, 2014, Vol. 2, No. 4, 94-101
Available online at http://pubs.sciepub.com/jmdv/2/4/4

DOI:10.12691/jmdv-2-4-4

Determination of Optimal Stacking Sequence for Modal Characteristics Evaluation of Composite Marine Propeller Blade

M.L. PavanKishore*, R.K. Behera

Department of Mechanical engineering, National Institute of Technology, Rourkela, India
*Corresponding author: kishoremamdoor9 @gmail.com

Received November 15, 2014; Revised November 21, 2014; Accepted November 25, 2014

Abstract The design of optimum marine propeller is one of the most important aspects of naval architecture. With the increase in demands for high operating efficiency, power and low level of noise, vibration reduction the design of propellers became extremely complex. This paper describes the numerical prediction of free vibration characteristics of a B-series propeller using finite element approach as a base line method. The propeller analysis is performed as a single objective function subjected to the constraints imposed by cavitation, material strength and propeller thrust. An important aspect of autonomous underwater vehicle is to evaluate its modal characteristics in terms of its mode shapes and natural frequencies. The effect of stacking sequences, fibre orientation angles are studied and finally an optimum stacking sequence has been determined for optimum characteristics of B-series (B4-0.7) marine propellers.

Keywords: *Ansys, finite elements, mesh, mode shapes, pre twist*

Cite This Article: M.L. PavanKishore, and R.K. Behera, “Determination of Optimal Stacking Sequence for Modal Characteristics Evaluation of Composite Marine Propeller Blade.” *Journal of Mechanical Design and Vibration*, vol. 2, no. 4 (2014): 94-101. doi: 10.12691/jmdv-2-4-4.

1. Introduction

Since the advent of propellers blades in the application of marine engineering cause an important role. The blade failure is predominantly vibration related it may be either marine propeller blade or steam turbine blade or wind turbine blade. The blades being very flexible structural members in the sense that a significant number of their natural frequencies can be in the region of possible nozzle excitation frequency. From manufacturers point of view it is important to know the natural frequencies of the blades, in order to make sure that the boss on which the blade should not have some of the frequencies otherwise a resonance may occur in the whole structure leading to un damped vibrations which may eventually wreck the ship.

The movement of a ship through the water is achieved by the power so developed in the engine via the propeller shaft to propel in water. The propeller produce thrust through the production of lift by their rotating blades. The blades are attached to boss or hub. The boss is fitted to the propeller shaft through which the power of the propulsion machinery of the ship is transmitted to the propeller. The strength requirements of propellers indicate that not only the blades be sufficiently strong to withstand long periods of arduous service without suffering failure. J.E.Collony [1] address the problem of the wide blades, tried to combine both theoretical and experimental investigations. A single free standing blade can be considered as cantilever beam with a rectangular cross section. The vibration characteristics of such blade are always coupled between the two bending modes in the flap wise and chord wise directions and the torsion mode. Leissa and Jacob [2] were the first ones to investigate the free vibration characteristics of cantilevered twisted beams and plates as a three dimensional vibrational problem. Kielb [3] presented a comprehensive review of structural dynamics aspects of pre twisted beams following on the same path of lines presented very comprehensive summary of the analytical work performed in the area of twisted plates. The equation relating to twisting plates are much more complex than corresponding Euler and Timoshenko beam formulations. An early analytical model to calculate the natural frequencies of a cantilever beam was suggested by South well [4] based on Rayleigh energy theorem which defines a simple equation relating the fundamental frequency to that of rotating frequency of a beam was suggested. Fried man & Straub [5] have formulated the discretization of linear equations of motion of blade by the Galerkin method and considered the problem of aero elasticity based on the finite element method of variable order. Sivaneri& Chopra [6], Thakkar & Ganguli [7] treated Hamilton’s principle as a variation principle for the rotor blades analysis. Crespo da silva [8] established a partial derivative nonlinear rotor movement and vibration of the blade taking into account of the geometric non linearity and analyzed the dynamic stability of the vibration movement. Comparing the beam model, plate and shell are more approximate models of blades.

Ramamurti and Kielb [9] studied rotating twisted plates by using finite element method with two different shape functions. Sreenivasa murthi and Ramamurti [10] studied rotating pre twisted and tapered plates by finite element method with triangular shell elements having 3 nodes with 18 degrees of freedom respectively. Leissa et al [11] studied cylindrical shells with chamber and twist on the shallow shell theory by Ritz method. Recently Tsuji et al [12] studied rotating thin twisted plates by the Rayleigh Ritz method.Hu and Tsuji [13] studied the rotating thin cylindrical panels with twist on general shell theory. Lo [14] in his analysis simplified the nonlinear problem by assuming the blade to be rigid everywhere except at the root presented the solution in phase plane. Boyce, Diprima& Handle man [15] used Rayleigh Ritz and South well methods to determine the upper and lower bounds of natural frequencies of turbine blade vibrating perpendicular to plane of rotation. Schihans [16] investigated the stiffening effect of the centrifugal forces on the first mode frequency of a rotating cantilever blade by the use of successive approximation. Hirsch [17] considered the effect of non-rigid supports. Horway [18] considered hinged ends and Nodorson [19] considered the effect of loose hinged supports on the natural frequencies of rotating blades. Garland [20] considered the case of a cantilever blade and used the Rayleigh Ritz method to determine the frequencies and amplitudes.

1.1. Composite

The implementation of fiber reinforced composite materials to the application of blades continues to increase, since it is important to explore the potential benefits that can be designed into the physics of these materials. Vibration is often critical to the successful operation of engineering structures which are composed of composite materials for example propeller blades, helicopter rotor, wind turbine blades, automatic and aerospace panels. Most of these structural components can be approximated as laminated composite beams [21].Free vibration analysis of laminated composite beams has been conducted by significant amount of research. Yddorom & Koral [22] studied the out of plane free vibration problems of symmetric ply laminated beams using the transformation cross method Khedir & Reddy [23] have been studied free vibrations of cross ply laminated beams with arbitrary boundary conditions. Investigations on the natural frequency modes of graphite epoxy cantilever plate and shells was carried out by crawley [24] and the free vibrations of rotating composite plates was analyzed by Wang [25] and Shaw [26].The first established work on pre twisted composite plates was carried out Qatu et al [27] to determine the natural frequencies of stationary plates using laminated shallow theory using Ritz method. Brig and Migliore [28] provided preliminary design method based on theory of Euler Bernoulli theory on layup structures and it was found that layup sequence which was obtained by this method did not meet the requirements of actual strength at the root area through analyzing the finite element model. Alejandro [29] proposed to analyze basic vibration mode of composite layup structure of wind turbine blades with the normal operating conditions by transforming complex geometric blade model into equivalent beam model using the VABS calculating program Wangyu Liu [30] studied the influence of fiber angle of the layup blade on strength through the finite element model of the blade and combined the response surface methodology and results showed that when layup angle is near 45 it can get higher stability strength. Stacking sequence of a laminated cylindrical shell is optimized based on natural frequency by Shakeri et al [31].

2. Materials of Propeller Blade

The materials from which the propellers are made in present can be broadly classified as members of bronze or stainless steels. These materials are better known materials that are in use for manufacture of all types of propellers ranging from the large commercial vessels and warships through to pleasure run about and modeled propellers. The use of high tensile Brass accounted for 64% of all propellers produced with Manganese Aluminum Bronze and Nickel Aluminum Bronze accounting for small proportions in 1960’s. In 1980’s NAB has completely gained dominance over the other materials accounting for 82% of the propellers. In present trend high tensile brass accounts for less than 7% of the propellers used. The stainless steel gained comparatively popular usage in the period from 1960 to 1970’s but progressively lost favour to the copper based materials to account for 3% of the materials used for propeller manufacture and are mostly applied to ice class propellers. properties of both materials are given in the following tables.

2.1. Carbon Based Composites

In recent years carbon based composites as propeller material have made an entry into the commercial craft and yacht market. Since they have been used in specialized naval vehicles such as submarines for many years. In addition to the use of acoustic properties for the materials there is a further advantage in their light weight when compared with conventional marine propellers.

Table 1. Material properties for metallic propeller

Material used	MAB
Young’s modulus	125.5[Gpa]
Rigidity Modulus	44[Gpa]
Poisson’s ratio	0.326
Density	7.53[g/cc]

Table 2. Material properties of composite propeller

Material	S-glass fabric/epoxy
Ex	22.925[Gpa]
Ey	22.925[Gpa]
Ez	12.4[Gpa]
υ_{XY}	0.12
υ_{YZ}	0.2
υ_{ZX}	0.2
Gxy	4.7[Gpa]
Gyz	4.2[Gpa]
Gzx	4.2[Gpa]
density	1.8 g/cc

3. Solid Modleing of Propeller Blade

Propellers more particularly propeller blades are more complex shapes which require the right hydrodynamic surfaces for modeling. The important parameters that characterize the modeling of a propeller are thus diameter, number of blades, the type of series for blade section, pitch distribution, the skew angle, the rake angle, the area of the blades. In this work a four bladed propeller was chosen for analysis. The global parameters that define the modeling of propeller blade are given as follows

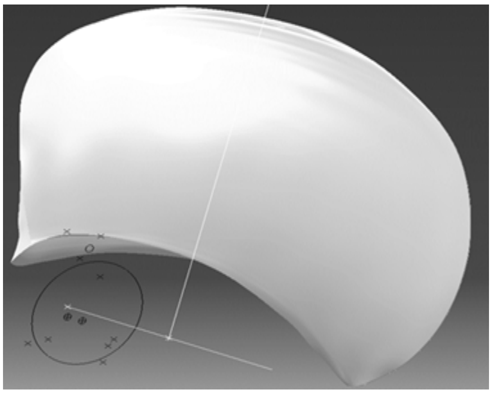

Figure 1. Single surface Model Propeller Blade

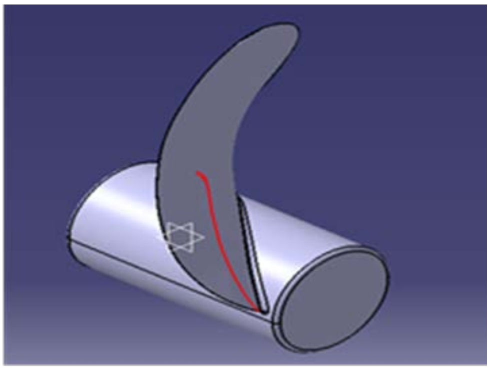

Figure 2. Solid Model of Single blade Propeller

For modelling of such complicated shapes of blades CatiaV5R20 was selected as base line modeller tool for analysis. The surfaces being important parameter for modelling the blade these surfaces are joined by a series of air foil points at various radii. The air foil points generated individually are rotated at respective pitch angles. All the rotated sections are projected onto the cylinder. These projection points are joined by means of a multi spline option to generate the surface of the blade. The solid model of the blade was generated using surface edit options. The air foil points and corresponding blade surface are shown in the following figures.

3.1. Propeller Blade Specifications

Table 3. Propeller Blade Characteristics

Type of series	wageningen B screw series propeller
Delivered power (P_D)	648KW
Advance speed (V_A)	4.372m/s
Propeller rate of rotation(N)	380 rpm
Propeller diameter (D)	2.12m
Number of blades (Z)	4
Blade area ratio (A_E/A_0)	0.70
P/D Ratio	0.9

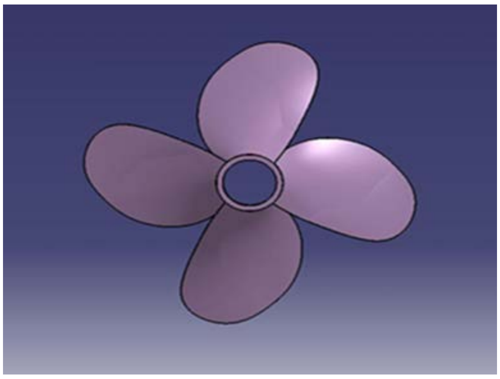

Figure 3. Solid Model of 4 Bladed Propeller

3.2. Meshing of Propeller Blade

The solver used for developing fine mesh over the blade surface is Hyper mesh 11.0. Intially two dimensional meshes was generated for the blade surface using quadrilateral elements. Later the 2Dmesh was converted to three dimensional mesh using solid map options. The total numbers of finite elements created for the model are given below.

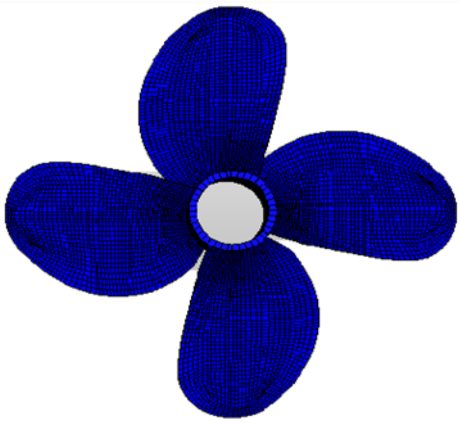

Figure 4. Meshed Model of Metallic Propeller Blade

Figure 5. Composite Propeller Blade

4. Mathematical modeling of Twisted Blade

Assumptions

The equation of motion is derived based on the following assumptions.

The rate of pre twist along the longitudinal axis of the blade is uniform.

The shear and rotary inertia effects of the blade are negligible due to slender shape of the blade.

The neutral and centroidal axis of the blade coincides with each other in the cross section.

Gyroscopic coupling between the stretching and bending motions are negligible and

No external forces act on the blade.

4.1. Geometry and Deformation of Shell Element

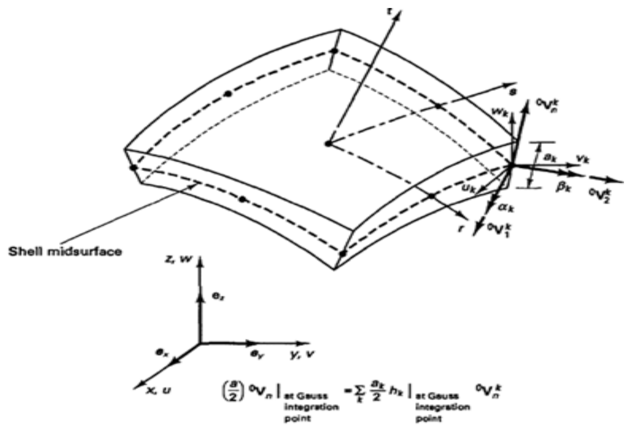

Figure 6. General shell element

$$\ell_x(r,s,t)=\sum_{k=1}^{q} h_k \ell_{xk}+\frac{t}{2}\sum_{k=1}^{q} a_k h_k\, {}^{e}V_{nk}^{k}$$
$$\ell_y(r,s,t)=\sum_{k=1}^{q} h_k \ell_{yk}+\frac{t}{2}\sum_{k=1}^{q} a_k h_k\, {}^{e}V_{ny}^{k} \quad (1)$$
$$\ell_z(r,s,t)=\sum_{k=1}^{q} h_k \ell_{zk}+\frac{t}{2}\sum_{k=1}^{q} a_k h_k\, {}^{e}V_{nz}^{k}$$

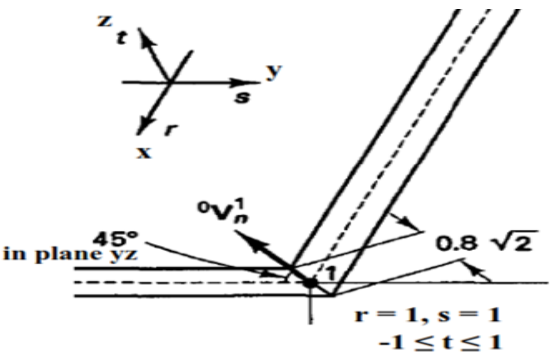

Figure 7. Coordinate Interpolation

$$\ell_x(r,s,t)=\sum_{k=1}^{q} h_k \ell_{xk}+\frac{t}{2}\sum_{k=1}^{q} a_k h_k\, {}^{e}V_{nk}^{k}$$
$$\ell_y(r,s,t)=\sum_{k=1}^{q} h_k \ell_{yk}+\frac{t}{2}\sum_{k=1}^{q} a_k h_k\, {}^{e}V_{ny}^{k} \quad (2)$$
$$\ell_z(r,s,t)=\sum_{k=1}^{q} h_k \ell_{zk}+\frac{t}{2}\sum_{k=1}^{q} a_k h_k\, {}^{e}V_{nz}^{k}$$

$${}^{0}V_n^{1}=\begin{bmatrix} 0 \\ \frac{-1}{\sqrt{2}} \\ \frac{1}{\sqrt{2}} \end{bmatrix} \quad (3)$$

For t=1

$$x=x_1$$
$$y=y_1+0.5*0.8\sqrt{0.2*(\frac{-1}{\sqrt{2}})} \quad (4)$$
$$z=z_1+0.5*0.8\sqrt{0.2*(\frac{-1}{\sqrt{2}})}$$

For t= -1

$$x=x_1$$
$$y=y_1-0.5*0.8\sqrt{0.2*(\frac{-1}{\sqrt{2}})} \quad (5)$$
$$z=z_1-0.5*0.8\sqrt{0.2*(\frac{1}{\sqrt{2}})}$$

Displacement Interpolation

$$u_x(r,s,t)=\sum_{k=1}^{q} h_k u_k+\frac{t}{2}\sum_{k=1}^{q} a_k h_k\, {}^{e}V_{nk}^{k}$$
$$v_y(r,s,t)=\sum_{k=1}^{q} h_k v_k+\frac{t}{2}\sum_{k=1}^{q} a_k h_k\, {}^{e}V_{ny}^{k} \quad (6)$$
$$w_z(r,s,t)=\sum_{k=1}^{q} h_k w_k+\frac{t}{2}\sum_{k=1}^{q} a_k h_k\, {}^{e}V_{nz}^{k}$$

Strain displacement matrix B(r,s,t)

$$\frac{\partial}{\partial x}=\frac{\partial}{\partial r}\frac{\partial r}{\partial x}+\frac{\partial}{\partial s}\frac{\partial s}{\partial x} \quad (7)$$

$$J_{ij}^{-1} \downarrow$$

$$\begin{bmatrix} \frac{\partial u}{\partial r} \\ \frac{\partial u}{\partial s} \\ \frac{\partial u}{\partial t} \end{bmatrix}=\begin{bmatrix} \frac{\partial h_k}{\partial r}\begin{bmatrix}1 & tg_{1x}^{k} & tg_{2x}^{k}\end{bmatrix} \\ \frac{\partial h_k}{\partial s}\begin{bmatrix}1 & tg_{1x}^{k} & tg_{2x}^{k}\end{bmatrix} \\ h_k\begin{bmatrix}0 & g_{1x}^{k} & g_{2x}^{k}\end{bmatrix} \end{bmatrix}\begin{bmatrix} u_k \\ \alpha_k \\ \beta_k \end{bmatrix} \quad (8)$$

General Shell element

Stress strain law

$$\tau=C_{sh}\varepsilon \quad (9)$$

$$\tau^T=[\tau_{xx}\ \tau_{yy}\ \tau_{zz}\ \tau_{xy}\ \tau_{yz}\tau_{zx}] \quad (10)$$

$$\varepsilon^T=[\varepsilon_{xx}\ \varepsilon_{yy}\ \varepsilon_{zz}\ \varepsilon_{xy}\ \varepsilon_{yz}\ \varepsilon_{zx}] \quad (11)$$

$$C_{sh} = Q_{sh}^T \left(\frac{E}{1-\nu^2}\begin{bmatrix} 1 & \nu & 0 & 0 & 0 & 0 \\ & 1 & 0 & 0 & 0 & 0 \\ & & 0 & 0 & 0 & 0 \\ & symmetric & & \frac{1-\nu}{2} & 0 & 0 \\ & & & & k\frac{1-\nu}{2} & 0 \\ & & & & & k\frac{1-\nu}{2} \end{bmatrix}\right) Q_{sh} \quad (12)$$

In which E and υ are the elastic and poisons ratio respectively. The material property matrix

$$[D] = [T_\varepsilon]^T [D'][T_\varepsilon] \quad (13)$$

where the transformation matrix $[T_\varepsilon]$ is

$$[T_\varepsilon] = \begin{bmatrix} l_{11}^2 & l_{12}^2 & l_{13}^2 & l_{11}l_{12} & l_{12}l_{13} & l_{13}l_{11} \\ l_{21}^2 & l_{22}^2 & l_{23}^2 & l_{21}l_{22} & l_{22}l_{23} & l_{23}l_{21} \\ l_{31}^2 & l_{32}^2 & l_{33}^2 & l_{31}l_{32} & l_{32}l_{33} & l_{33}l_{31} \\ 2l_{11}l_{21} & 2l_{12}l_{22} & 2l_{13}l_{23} & \begin{pmatrix} l_{11}l_{22} \\ +l_{21}l_{12} \end{pmatrix} & \begin{pmatrix} l_{12}l_{23} \\ +l_{22}l_{132} \end{pmatrix} & \begin{pmatrix} l_{13}l_{21} \\ +l_{23}l_{11} \end{pmatrix} \\ 2l_{21}l_{31} & 2l_{22}l_{32} & 2l_{23}l_{33} & \begin{pmatrix} l_{21}l_{32} \\ +l_{31}l_{22} \end{pmatrix} & \begin{pmatrix} l_{22}l_{33} \\ +l_{32}l_{23} \end{pmatrix} & \begin{pmatrix} l_{23}l_{31} \\ +l_{33}l_{21} \end{pmatrix} \\ 2l_{31}l_{11} & 2l_{32}l_{12} & 2l_{33}l_{13} & \begin{pmatrix} l_{31}l_{12} \\ +l_{11}l_{32} \end{pmatrix} & \begin{pmatrix} l_{32}l_{13} \\ +l_{12}l_{33} \end{pmatrix} & \begin{pmatrix} l_{33}l_{11} \\ +l_{13}l_{31} \end{pmatrix} \end{bmatrix} \quad (14)$$

Where l_{ij} are the directional cosines of the unit vector

$$\begin{aligned} l_1 &= \cos(e_x, e_{\bar{r}}); \\ l_2 &= \cos(e_x, e_{\bar{s}}); \\ l_3 &= \cos(e_x, e_t); \end{aligned} \quad (15)$$

$$\begin{aligned} m_1 &= \cos(e_x, e_{\bar{r}}); \\ m_2 &= \cos(e_x, e_{\bar{s}}); \\ m_3 &= \cos(e_x, e_t); \end{aligned} \quad (16)$$

$$\begin{aligned} n_1 &= \cos(e_x, e_{\bar{r}}); \\ n_2 &= \cos(e_x, e_{\bar{s}}); \\ n_3 &= \cos(e_x, e_t); \end{aligned} \quad (17)$$

Shear Locking

$$\begin{aligned} &\varepsilon = \bar{\varepsilon}_{rr} g^r g^r + \bar{\varepsilon}_{ss} g^s g^s + \bar{\varepsilon}_{rt}(g^r g^t + g^t g^r) \\ &+\bar{\varepsilon}_{rs}(g^r g^s + g^s g^r) + \bar{\varepsilon}_{st}(g^s g^t + g^t g^s) \\ &\qquad \downarrow \\ &\bar{\varepsilon}_{rr} g^r g^r + \bar{\varepsilon}_{ss} g^s g^s + \bar{\varepsilon}_{rt}(g^r g^t + g^t g^r) \\ &\qquad \text{In – layer Strains} \\ &\bar{\varepsilon}_{rs}(g^r g^s + g^s g^r) + \bar{\varepsilon}_{st}(g^s g^t + g^t g^s) \end{aligned} \quad (18)$$

Transverse Shear strains

The element stiffness matrix is computed from

$$[K] = \int_{\Omega_e} [B]^T [D][B] d\Omega \quad (19)$$

And the element mass matrix is given as

$$[M] = \int_{\Omega_e} [N]^T [\rho][N] d\Omega \quad (20)$$

In general the equation of motion can be written

$$M\ddot{q} + C\dot{q} + kq = r \quad (21)$$

Where M, C, k mass, damping, stiffness matrix
$\ddot{q}, \dot{q}, q$ acceleration, velocity, displacement
r external force vector

5. Application of Boundary conditions

For determining the natural frequencies Ansys14.0 is used as a postprocessor for metallic blade. Solid 45 & solid 46 element type are used for metallic and composite blades. The hub is constrained with all degrees of freedom arrested, assuming the propeller blade as a cantilever beam fixed at one end. The mode shapes and corresponding natural frequencies are determined. The composite material with various layup sequences are used to evaluate the best one that match up with isotropic material.

6. Modal Analysis of Propeller Blade

The modes of vibration of a propeller blade beyond the fundamental and first torsional/flexural modes are extremely complex. This complexity arises from the non-symmetrical outline of the blade. The variable thickness distribution both chordally and radially and the twist of the blade caused by changes in the radial distribution of pitch angle. Specifications for such rotating components with substantial dynamic loads often require the structural modes occurring in specified frequency range. The dynamic analysis of complex systems is often substantially complicated for such cases computer finite element methods are called for. A powerful Eigen extraction method for natural vibration analysis for rotating components is used. This method is based on a Block lancoz algorithm. Lancoz method has been developed for finding the some or all of the eigen values [12,13] and eigen vectors of large symmetric sparse matrix. For dynamic analysis an unsymmetrical Block Lancoz algorithm may be employed to handle matrices with repeated eigen values, permit dynamic response analysis with multiple simultaneously applied loads/accommodate unsymmetrical damping matrices [14,15] Lancoz method may also be used to reduce order structural models for component synthesis and control applications[16]. By applying the necessary boundary conditions and respective mechanical and material properties the natural frequencies and corresponding mode shapes of both metallic and composite finite element model are extracted. This analysis represents the un damped free vibrations of the propeller blade in absence of damping and applied loads.

7. Results

In this study the free vibration characteristics of propeller blade with different ply ups and number of layers were determined. The comparison of natural frequencies and corresponding mode shapes were plotted. The numerical results from Table 4 and fig predict that, the first natural frequency of propeller blade when replaced with MAB has high operational frequencies and composite with first layup sequence produced the lowest. The percentage variation in terms of natural frequencies between MAB & composite with first layup is varied between 0.260% to 0.3684%. From Table 5 and fig shows that with change in number of layers from 4 to 8 enhances the composite material to match up with the frequencies of metallic one. From results it can be predicted that with further increase in number of layers and change of layup sequences the composite propeller blade is made to operate higher frequencies than that of metallic one.

Table 4. Composite with various stacking sequences

Stacking Sequence
$(45_2/0_4/90/45/0_2/-45/0/90)_{sym}$
$(0/45/-45)_7(0/90)_2$
$(45/-45/0)_7(0/90)_2$
$(45/0/-45)_7(0/90)_2$
$(-45/0/45)_7(0/90)_2$

Table 5. Natural Frequencies of composite for 4-Layers

MAB	SQ1	SQ2	SQ3	SQ4	SQ5
459.11	329.53	289.96	329.53	339.54	339.54
1511.3	1106.8	1064.5	1106.8	1120.0	1120.0
1725.5	1355.6	1217.0	1355.6	1355.6	1355.6
3166.7	2325.5	2390.2	2325.5	2348.2	2348.2
4217.7	3306.0	3014.7	3306.0	3335.3	3335.3
5846.5	4191.6	3978.9	4191.6	4325.4	4325.4
6382.8	4694.8	4600.6	4694.8	4888.5	4888.5
6888.8	4953.3	4962.2	4953.3	5091.8	5091.8
7755.2	5914.3	5489.1	5914.3	5988.2	5988.2
8752.4	6403.3	6240.7	6403.3	6472.4	6472.4

Table 6. Natural Frequencies of composite for 8-Layers

SQ1	SQ2	SQ3	SQ4	SQ5
333.96	338.22	336.02	319.40	326.38
1111.6	1148.3	1134.2	1119.0	1125.4
1362.4	1331.7	1397.4	1322.8	1351.0
2341.3	2497.8	2380.0	2414.6	2404.1
3337.4	3250.6	3365.7	3246.9	3304.0
4290.1	4400.1	4336.3	4250.8	4303.9
4856.5	4880.7	4926.4	4823.6	4879.4
5061.8	5281.0	5172.4	5129.7	5153.9
5980.9	5890.2	6026.3	5840.7	5935.8
6454.6	6636.4	6593.8	6538.3	6550.0

Table 7. Natural Frequencies of composite for 16-Layers

SQ1	SQ2	SQ3	SQ4	SQ5
324.12	318.77	331.15	325.94	325.45
1121.5	1119.4	1130.9	1126.8	1125.0
1334.3	1319.3	1362.8	1344.1	1345.7
2404.8	2418.0	2394.7	2406.9	2406.9
3276.2	3245.7	3333.3	3296.7	3296.2
4292.7	4246.2	4320.0	4292.8	4295.5
4872.2	4820.3	4902.5	4870.4	4870.9
5153.6	5126.3	5157.3	5142.2	5147.9
5896.2	5846.6	5979.3	5924.9	5924.0
6542.5	6536.5	6581.7	6567.6	6552.7

Table 8. Natural Frequencies of composite for 25-Layers

SQ1	SQ2	SQ3	SQ4	SQ5
322.84	320.60	328.90	325.94	325.52
1116.0	1115.7	1123.8	1126.8	1120.1
1313.4	1319.3	1346.4	1344.1	1335.1
2410.8	2406.4	2391.7	2406.9	2399.3
3232.1	3247.8	3305.0	3296.7	3279.8
4275.7	4246.5	4294.2	4292.8	4278.0
4839.7	4819.9	4872.4	4870.4	4250.5
5138.7	5080.2	5101.3	5142.2	5093.4
5831.1	5852.1	5940.8	5924.9	5901.8
6517.8	6497.5	6529.3	6567.6	6512.3

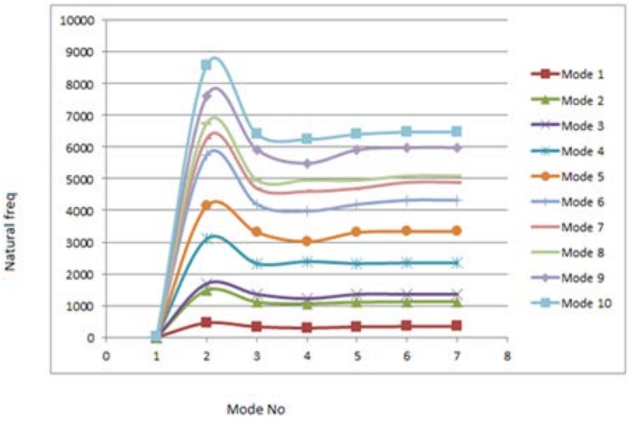

Figure 8. NF s Mode No (4-Layers)

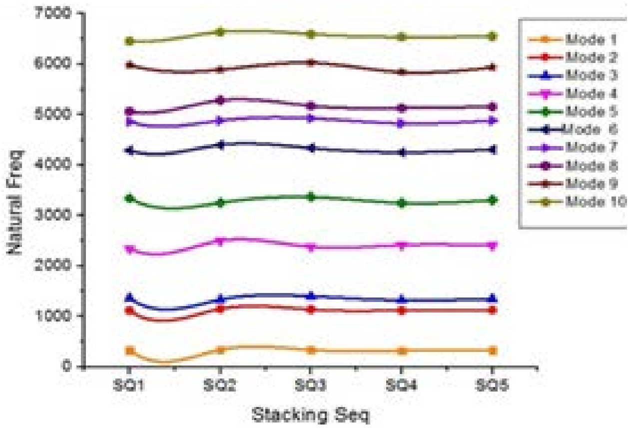

Figure 9. NF s Mode No (8-Layers)

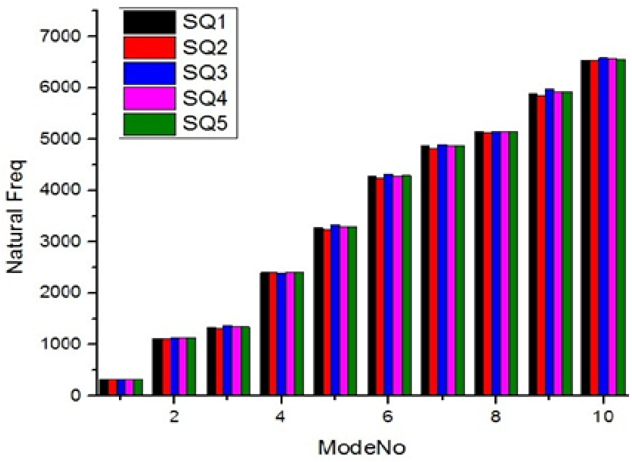

Figure 10. NF s Mode No (16-Layers)

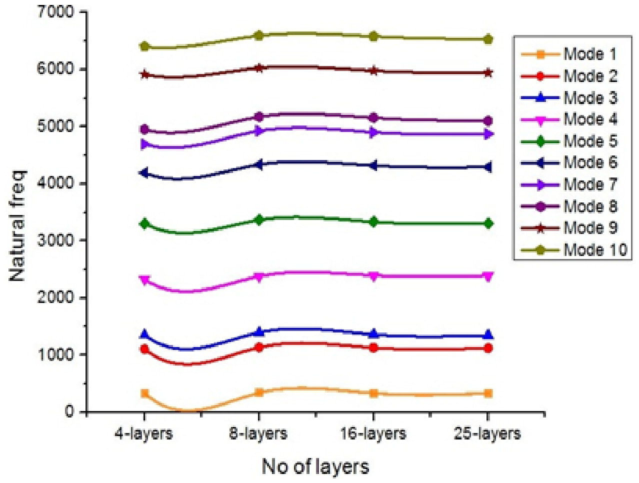

Figure 11. NF s Mode No (25-Layers)

8. Conclusions

1. From the results presented it is clear that the change in fibre orientation, number of layers and laminate stacking sequence yield to different dynamic behaviour of the component. Different layers apply different contributions to the overall stiffness of the blade depending upon the location from mid plane.

2. From Table 4 & Table 5 with the increase in number of layers from 4 to 8 an increase in the flexural frequencies and torsion frequencies for stacking sequence of $(\pm 45/0)_{11}$, $(45/-45/0)_{11,}$ $(0/\pm 45)$ are observed. For stacking sequence of $(45/0/-45)_{11}$& $(-45/0/45)_{11}$ decrease in torsion modes and flexural are observed.

3. From Table 5 & Table 6 the increase in number of layers from 8 to 16 varied in steps increase in the first bending frequency and torsion frequencies for stacking sequence of $(\pm 45/0/90)$ are observed. For stacking sequence of $(\pm 45/0)_{11}(0/90)_7$, $(0/\pm 45)_{11}(0/90)_7$ decrease in torsion and flexural modes are observed, while for $(+45/0/-45)_7(0/90)_2$, $(-45/0/+45)_7(0/90)_2$ increase in both the flexural and torsion are observed.

4. From Table 6 & Table 7 it is seen that with further increase in number of layers from 16 to 25 layers decrease in flexural and torsional frequencies are observed in all the stacking sequences.

5 From Table 4, Table 5, Table 6 the stacking sequence $(\pm 45/0)_7$ $(0/90)_2$for composite has got highest bending frequencies and torsional frequency. From the results it is possible to verify the influence of stacking sequence of the laminate on torsional vibration the laminate with $(\pm 45°)$ has larger torsional natural frequencies. The natural frequencies related to stiffness of the structure (± 45) are much stiffer in torsion. The laminate (0/45) of fibres has lowest torsional frequency than the other lamination schemes and most of the fibres are oriented at 0 ° direction and thus appropriate for flexural modes (bending).In fact this can be applied that the fibres oriented at 0 ° are appropriate for flexural loads and fibres oriented at 45 ° area more appropriate for torsion loads. In order to attain increase in bending & torsion modes an optimum numbers of layers are selected in between 8 and 16 layers. From the results it can be predicted that the stacking sequence number 3 has got highest operating frequencies for 16 layers close to the metallic one. The desired vibration characteristics can be attained for the blade without modifying the shape. Hence the ply orientation, material selection should be optimized to attain maximum operating frequencies.

References

[1] J.E.Conolly, “Strength Of Propellers”, reads in London at a meeting of the royal intuition of naval architects on Dec 1,1960,pp 139-160.

[2] Leissa, K.I. Jacob, Three-dimensional vibrations of twistedcantilevered parallelepipeds, Journal of Applied Mechanics-Transactions of the ASME 53, (1986), 614-618.

[3] R.E. Kielb, A.W. Leissa, J.C. MacBain, Vibrations of twistedcantilever plates-a comparison of theoretical results, International Journal for Numerical Methods in Engineering 21, (1985), 1365-1380.

[4] South well. R and F. Gough, The free transverse vibration of airscrew blade, British A.R.C. Reports and Memoranda, 1921, No. 766.

[5] P.P. Fried mann, F. Straub, Application of the finite element methodto rotary-wing aero elasticity, Journal of American Helicopter Society 25 (1980) 36-44.

[6] N.T. Sivaneri, I. Chopra, Dynamic stability of a rotor blade usingfinite element analysis, AIAA Journal 20 (1982) 716-723.

[7] Decolon.Estimation a posteriori et adaptation de malliages. coordinateur Michel Fortin, Hermes, 2000.

[8] M.R.M. Crespo Da Silva, A comprehensive analysis of thedynamics of a helicopter rotor blade, International Journal of Solids and Structures 35 (7–8) (1998) 619-635.

[9] V. Ramamurti, R. Kielb, Natural frequencies of twisted rotating plates, Journal of Sound and Vibration 97 (1984) 429-449.

[10] S. Sreenivasamurthy, V. Ramamurti, A parametric study ofvibration of rotating pre-twisted and tapered low aspect ratiocantilever plates, Journal of Sound and Vibration 76 (1981) 311-328.

[11] A.W. Leissa, J.K. Lee, A.J. Wang, Rotating blade vibration analysisusing shells, Journal of Engineering for Power, Transactions of American Society of Mechanical Engineers 104 (1982) 296-302.

[12] T. Tsuiji, M. Shugyo, T. Yamashita, Free vibration analysis ofrotating thin twisted plates, Transactions of Japan Society of Mechanical Engineers 61 (C) (1995) 4575-4586 (in Japanese).

[13] X.X. Hu, T. Tsuiji, Free vibration analysis of rotating twistedcylindrical thin panels, Journal of Sound and Vibration 222. 1999 209-224.

[14] Lo,H.,A nonlinear problem in the bending vibration of a rotatingbeam, J.Appl.Mech., Trans.ASME, 1952, 19, p.461.

[15] Boyce, W.e. Diprima, R.C. and andleman, G.H., Vibrations of rotating beams of constant sections,proc.and Us Natl.cong.Appl,Mech.,ASME, 1954,p.165.

[16] Schilhansl, M.J., Bending frequency of arotatingcantileverbeam, J.Appl.Mech., ASME, 25, 1958, p. 28.

[17] Hirsch,G.,Investigation of vibration in bending of rotating turbineblades on non-rigid support, 1958, Jahrbuck Wissens chart., GessellschLuftahrt p. 174.

[18] HorwayG.,chord wise and beam wise bending of hinged rotorblades, J Aero. Sci., 1948, 15, p.497.

[19] Niordson, F., Vibration of turbine blades with loose hingesupports, 1954, Engr.Digest, 15, p. 359.

[20] Garland C.F.,The normal modes and vibrations of beams havingnoncollinearelasticandmassaxes,J.appl.Mech.,Trans.ASME, 1940,p.a-97.

[21] Kapuria S, Alam N., Efficient layerwise finite elementmodel for dynamic analysis of laminated piezoelectric beams, Computer Methods in Applied Mechanics and engineering, 2006, Vol. 195, pp. 2742-2760.

[22] Yõldõrõm V, Kõral E., Investigation of the rotary inertia and sheardeformation effects on the out-of-plane bending and torsionalnatural frequencies of laminated beams, Composite Structures, Vol. 49, 2000, pp. 313-320.

[23] Khdeir A.A., Reddy J.N., Free vibration of cross-ply laminatedbeams with arbitrary boundary conditions, International Journal of Engineering Science, 1994, Vol. 32, pp. 19711980.

[24] Crawley, E. F. The Natural Modes of Graphite/Epoxy CantileverPlates and Shells, Journal of Composite Materials, 1979, Vol. 13, pp. 195-205.

[25] Wang, J. T. S., Shaw, D. and Mahrenholtz, O. Vibration of RotatingRectangular Plates, Journal of Sound and Vibration, Vol. 112, No. 3, 1987, pp. 455-468.

[26] Shaw, D., Shen, K. Y. and Wang, J. T. S. Flexural Vibration of Rotating Rectangular Plates of Variable Thickness, Journal of Sound and Vibration, 1988 Vol. 126, No. 3, pp. 373-385.

[27] Qatu M. S. and Leissa A. W. Vibration studies for Laminated Composite Twisted Cantilever Plates, International Journal of Mechanical Sciences, 1991, Vol.33, pp. 927-940.

[28] BirG.S, and Migliore P.G. Computerized method for preliminary structural design of composite wind turbine blades [J]. Journal of solar energy engineering, 2001, vol. 123, iss. 4, pp. 372-381.

[29] Alejandro D. Otero, and Fernando L. Ponta. Structural analysis of wind turbine blades by a generalized Timoshenko beam model [J]. Journal of Solar Energy Engineering, 2010, vol. 132, pp. 151-158.

[30] Liu Wang-yu, Zhang Hai-quan, and Zeng lin. Reliability analysis of bionic wind turbine blades based on the response surface methodology [J]. Journal of Solar Energy Engineering., 2010, vol. 31, iss. 9, pp. 1204-1208.

[31] Shakeri M, Yas MH, Gol MG, Optimal stacking sequence of laminated cylindrical shells using genetic algorithm, MechAdv Mater Struct, 2005, 305-312.