Modes shape and harmonic analysis of different structures for helicopter blade

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Abstract
This study concerns the dynamic behavior of a helicopter blade. The objective is to simulate by the finite elements method, the behavior of a blade of different materials under an aerodynamic load. This study was conducted to evaluate the aerodynamic loads applied and evaluated by a numerical simulation the frequencies and eigenmodes and calculate the stresses acting on the structure for different modes. The study of the transient behavior has allowed the determination of the vibration responses due to unbalance and different excitation modes.

Keywords: blade, frequency, transient regimes behavior, finite elements, vibration.

I. Introduction

The main element of a helicopter is the rotor, itself consisting of a set of two pales around a substantially vertical axis, in a plane perpendicular to this axis. The vibrational behavior of the pales was the subject of many studies, in particular by R. Heffernan MS, Torok et all [1-2] who have studied the vibration problem of the helicopter blade rotating beams as well as the vibrations of rotors by the finite elements method (Timoshenko beam, Euler-Bernoulli) [3-7].

Friedmann and Straub [8] have formulated, for the discretization of linear equations of motion of the blade by the Galerkin method and treated the problem of aeroelasticity based on the finite elements method of variable order. Sivaneri and Chopra [9], Thakkar and Ganguli [10] treated Hamilton's principle as a variational principle for the rotor blade analysis. Bauchau and Hong [11] applied the finite elements method and analyzed the response time and stability based on the Floquet theory of beams undergoing large deformations at high rotational speeds.

Celi and Friedmann [12-13] described, respectively, the methods of formulation of the aeroelastic stability of the problem and the response to the helicopter rotor blades, using implicitly the aerodynamic and structural formulation based on a combination of a finite element model of the blade and a quasi-linearization technical solution. Crespo da Silva [14] established a partial derivative nonlinear rotor movement and vibration of the blade, taking into account the geometric nonlinearity, and analyzed the dynamic stability of the vibration movement.

To reduce the vibration of the helicopter rotor hub, Ganguli et al [15], Sinawi et al [17] used a rotating flexible beam and the aeroelasticity of the blade for the analysis of dynamic behavior based on the finite elements method. Optimal approaches to the profile of the rotor blade were highlighted. However, further analysis of the aeroelasticity of the of the helicopter rotor blade, the influence rotor blade vibration has not been taken into account. In reality, the motion of
the fuselage has an influence on the rotor hub [18-19]. The models of the associated rotor-fuselage are required, and therefore many researchers have studied and proposed several methods of rotor-fuselage interaction.

Rutkowski [20] studied the effect of the rotor / fuselage based on predictions of vibrations by using a simplified structural model of a helicopter in the plane of two-degree of freedom (DOF) beam finite elements. Hsu and Peter [21] have developed a new method of damping-impedance and the impedances of the fuselage to the rotor / fuselage associated with vibration analysis. Kunz [22] took into account the impedance method, to solve the equations of motion of a blade for constant elasticity, the equations for the charges of the hub and fuselage equations are associated with rotor-fuselage vibration analysis with a model of the blade and of the fuselage and the hub.

Stephens and Peter [23] presented an iterative method and an associated method for analyzing the response of a rotor-fuselage system. Cribbs et al. [24] derived a set of equations of motion for a dynamic fuselage system and the rotor-fuselage. The solution was obtained using the harmonic balance technique. Bauchau et al. [25] used an approached model to formulate the associated rotor-fuselage system and analyzed the response by the synthesis method of the corresponding mode.

2. Materials and profile of the blade

We choose three orthotropic composite materials (glass /epoxy, carbon /epoxy and boron /epoxy) and an isotropic material (steel), (Tab.1), [3].

Table. 1. a orthotropic composite materials.

<table>
<thead>
<tr>
<th>composite materials</th>
<th>E₁ (GPa)</th>
<th>E₂ (GPa)</th>
<th>E₃ (GPa)</th>
<th>G₁2 (GPa)</th>
<th>G₁3 (GPa)</th>
<th>G₂3 (GPa)</th>
<th>ν₁₂</th>
<th>ν₁₃</th>
<th>ν₂₃</th>
<th>ρ (Kg/m³)</th>
</tr>
</thead>
<tbody>
<tr>
<td>glass /epoxy</td>
<td>46</td>
<td>10</td>
<td>10</td>
<td>4.7</td>
<td>4.7</td>
<td>4</td>
<td>0.3</td>
<td>0.13</td>
<td>0.13</td>
<td>1850</td>
</tr>
<tr>
<td>carbon /epoxy</td>
<td>159</td>
<td>14</td>
<td>14</td>
<td>4.8</td>
<td>4.8</td>
<td>4.3</td>
<td>0.32</td>
<td>0.14</td>
<td>0.14</td>
<td>1550</td>
</tr>
<tr>
<td>boron /epoxy</td>
<td>224.6</td>
<td>12.7</td>
<td>12.7</td>
<td>4.4</td>
<td>4.4</td>
<td>2.4</td>
<td>0.25</td>
<td>0.01</td>
<td>0.01</td>
<td>2440</td>
</tr>
</tbody>
</table>

Table. 1. b isotropic material.

<table>
<thead>
<tr>
<th>material</th>
<th>E(GPa)</th>
<th>G(GPa)</th>
<th>ν</th>
<th>ρ(Kg/m³)</th>
</tr>
</thead>
<tbody>
<tr>
<td>steel</td>
<td>207</td>
<td>83</td>
<td>0.247</td>
<td>7849</td>
</tr>
</tbody>
</table>

2.1 Blade Characteristics

The geometry of the blade is determined using CATIA software. Data is processed by the ANSYS software to mesh [4]. The NACA 23012 wing airfoil is studied (Fig. 1) 5-digit serial. It is composed of 26 static pressure taken, numbered from 1 to 12 extrados side, and 13 to 24 from intrados side. The wing span for b = 6 m and chord c = 0.4 m [5].
3. Aerodynamic study

3.1 Mobility of the blade

The three mobilities of the blade are (fig. 2.):
- The beat $\beta$, blade lifted perpendicular to the rotor plane;
- The drag $T \delta$, blade motion in the rotor plane;
- The pitch $P \theta$, blade rotation of the blade along its span. [6]

4. Modeling of the blade by finite elements method

4.1 Presentation of the problem

The local coordinates and the global system of the blade are shown in Figure 3.
4.2. Working hypotheses

To describe the variation of the second degree of the centrifugal inertial force in the axial direction, it has an element of 8 nodes (I.J.K.L.M.N.O.P) has five degrees of freedom to each node: three translations in $(U_x, U_y, U_z)$ following x, y, z, and two rotations $(\theta_x, \theta_z)$ along x, z. (Fig. 4), [7]

\[ u_y^i \]

Where $u_{0x}$, $u_{0y}$ and $u_{0z}$ are the displacements along the axis x, y, z, and $\theta_x$, $\theta_z$ is the rotations around the X and Z axis respectively. [8]

4.2.1 Shape function

The nodal displacement vector is:

\[ U = [u_{0x} \ u_{0z} \ u_{0yi} \ \theta_{xi} \ \theta_{zi} \ \ldots \ u_{0xp} \ u_{0zp} \ u_{0yp} \ \theta_{xp} \ \theta_{zp}]^T \]

4.2.2 Displacement field

\[ \theta_x(x, z) = -\frac{\partial u_{0y}}{\partial x} \quad (1) \]
\[ \theta_z(x, z) = -\frac{\partial u_0y}{\partial z} \]  

(2) 

The displacement field is written as: [9]

\[ u_x(x, z, y) = u_{0x}(x, z) - y \frac{\partial u_{0y}}{\partial x}(x, z) \]  

(3) 

\[ u_z(x, z, y) = u_{0z}(x, z) - y \frac{\partial u_{0y}}{\partial z}(x, z) \]  

(4) 

\[ u_y(x, z, y) = u_{0y}(x, z) \]  

(5) 

4.2.3 Strain

Le champ des déformations s’écrit

The field strain is written as:

\[ \varepsilon_{xx} = \frac{\partial u_{0x}}{\partial x} - y \frac{\partial^2 u_{0y}}{\partial x^2} \]  

(6) 

\[ \varepsilon_{zz} = \frac{\partial u_{0z}}{\partial z} - y \frac{\partial^2 u_{0y}}{\partial z^2} \]  

(6) 

\[ \varepsilon_{yy} = 0 \]  

(7) 

\[ \kappa_{yz} = 0 \quad \kappa_{yx} = 0 \]  

(8) 

\[ \kappa_{xz} = \left( \frac{\partial^2 u_{0x}}{\partial z} + \frac{\partial^2 u_{0z}}{\partial x} \right) - 2y \frac{\partial^2 u_{0y}}{\partial x \partial z} \]  

(9) 

The strain tensor as:

\[ \varepsilon = \begin{bmatrix} \varepsilon_{xx} & \varepsilon_{xz} & 0 \\ \varepsilon_{xz} & \varepsilon_{zz} & 0 \\ 0 & 0 & 0 \end{bmatrix} \]  

(10) 

The strain matrix is reduced to three nonzero components:

\[ \varepsilon = \begin{bmatrix} \varepsilon_{xx} \\ \varepsilon_{zz} \\ \kappa_{xz} \end{bmatrix} \]  

4.2.4 Stress field

The stress field of a laminate layer is given by the following relationship:

\[ \begin{bmatrix} \sigma_{xx} \\ \sigma_{zz} \\ \sigma_{xz} \end{bmatrix}_k = \begin{bmatrix} l_{11} & l_{12} & l_{13} \\ l_{21} & l_{22} & l_{23} \\ l_{31} & l_{32} & l_{33} \end{bmatrix} \begin{bmatrix} \varepsilon_{xx}^m \\ \varepsilon_{zz}^m \\ \varepsilon_{xz}^m \end{bmatrix}_k + y \begin{bmatrix} l_{11} & l_{12} & l_{13} \\ l_{21} & l_{22} & l_{23} \\ l_{31} & l_{32} & l_{33} \end{bmatrix}_k \begin{bmatrix} K_x \\ K_z \\ K_{xz} \end{bmatrix}, \]  

(11) 

\[ \bar{q}_{ij} = c_{ij} - \frac{c_{i3} c_{j3}}{c_{33}} \]  

Or by integrating into the thickness

\[ N(x, z) = [A_{ij}] e^m(x, z) + [B_{ij}] \kappa(x, z), \]  

(12) 

\[ A_{ij} = \sum_{k=1}^{n} (h_k - h_{k-1}) (\bar{q}_{ij})_k' \]  

and

\[ B = \sum_{k=1}^{n} \frac{1}{2} (k_k^2 - k_{k-1}) (\bar{q}_{ij})_k' \]  

(13) 

The developed expression of the results in membrane is thus:
\[
\begin{bmatrix}
N_x \\
N_{xz} \\
N_{zz}
\end{bmatrix} = \begin{bmatrix}
A_{11} & A_{12} & A_{13} \\
A_{12} & A_{22} & A_{23} \\
A_{13} & A_{23} & A_{33}
\end{bmatrix} \begin{bmatrix}
\epsilon_{xx} \\
\epsilon_{xx} \\
\gamma_{xx}
\end{bmatrix} + \begin{bmatrix}
B_{11} & B_{12} & B_{13} \\
B_{12} & B_{22} & B_{23} \\
B_{13} & B_{23} & B_{33}
\end{bmatrix} \begin{kappa}_x \\
\kappa_z \\
\kappa_{xz}
\end{bmatrix}
\] (14)

Bending moments and torsion are:
\[
M(x, z) = [B_{ij}] \epsilon_m(x, z) + [D_{ij}] \kappa(x, z),
\]
\[
D = \sum_{k=1}^{n} \frac{1}{3} (h_{k}^3 - h_{k-1}^3) \theta_k
\] (15)

The developed expression of the torque can be written as:
\[
\begin{bmatrix}
M_{xx} \\
M_{xz} \\
M_{zz}
\end{bmatrix} = \begin{bmatrix}
B_{11} & B_{12} & B_{13} \\
B_{12} & B_{22} & B_{23} \\
B_{13} & B_{23} & B_{33}
\end{bmatrix} \begin{bmatrix}
\epsilon_{xx} \\
\epsilon_{xx} \\
\gamma_{xx}
\end{bmatrix} + \begin{bmatrix}
D_{11} & D_{12} & D_{13} \\
D_{12} & D_{22} & D_{23} \\
D_{13} & D_{23} & D_{33}
\end{bmatrix} \begin{kappa}_x \\
\kappa_z \\
\kappa_{xz}
\end{bmatrix}
\] (16)

The constitutive equation of a laminated plate is:
\[
\begin{bmatrix}
N \\
M
\end{bmatrix} = \begin{bmatrix}
A & B \\
B & D
\end{bmatrix} \begin{bmatrix}
\epsilon \\
\kappa
\end{bmatrix}
\] (17)

4.2.5 The strain energy

The total strain energy of a solid can be written as follows:
\[
U_e = \iiint \left( \frac{1}{2} C_{ijkl} e_{ij} e_{kl} - \sigma_{ij}^m e_{ij} \right) dV - \iiint f_i^v u_i dV - \iint f_i^s u_i dS
\]

Where V and S are respectively the volume and the external surface of the element. This expression can be written in matrix form [10] as:
\[
U_e = \iiint \left( \frac{1}{2} \begin{bmatrix}
\epsilon_m \\
\kappa
\end{bmatrix}^T \begin{bmatrix}
A & B \\
B & D
\end{bmatrix} \begin{bmatrix}
\epsilon_m \\
\kappa
\end{bmatrix} \right) dV - \iiint \{u\}^T \{f_v\} dV - \iint \{u\}^T \{f_s\} dS
\] (18)

The virtual variation of the displacement field \( \{\delta u\} \) gives:
\[
\delta U_e = \iiint \left( \frac{1}{2} \begin{bmatrix}
\delta \epsilon_m \\
\delta \kappa
\end{bmatrix}^T \begin{bmatrix}
A & B \\
B & D
\end{bmatrix} \begin{bmatrix}
\delta \epsilon_m \\
\delta \kappa
\end{bmatrix} \right) dV - \iiint \{\delta u\}^T \{f_v\} dV - \iint \{\delta u\}^T \{f_s\} dS
\] (19)

The above expression defines completely the behavior of an element by a linear equations system in the form of:
\[
[K_e] \{q_e\} = \{F_e\}
\] (20)

Where \([K_e]\) the stiffness matrix of the element is, \(\{q_e\}\) is the vector of nodal displacements.
So the elementary rigidity matrix is:
\[
[K]_e = \iiint [B_m]^T [A] [B_m] + [B_m]^T [B] [B_f] + [B_f]^T [B] [B_m] + [B_f]^T [D] [B_f] dV
\] (21)

4.2.6 The kinetic energy

The kinetic energy of the system can be written as follows:
\[
T = \frac{1}{2} \iiint \rho V^2 dV
\] (22)
\[ E_c = \frac{1}{2} \iiint \rho \left[ \left( \frac{\partial u_x}{\partial t} \right)^2 + \left( \frac{\partial u_y}{\partial t} \right)^2 + \left( \frac{\partial u_z}{\partial t} \right)^2 \right] \, dxdydz, \]

\[ T = \frac{1}{2} \iiint \rho [\dot{\varphi}][\dot{\varphi}]^T [N]^T [N] \, dV \tag{23} \]

\[ [M]_e = \iiint \rho [N]^T [N] \, dV \tag{24} \]

Where \([M]_e\) is the elementary mass matrix.

### 4.2.7 Aerodynamic force

The aerodynamic force and torque at the rotor blade center of gravity are calculated by

\[
\begin{bmatrix}
F_x \\
F_y \\
F_z \\
F_C
\end{bmatrix} =
\begin{bmatrix}
T \\
F_N
\end{bmatrix} \tag{25}
\]

### 4.2.8 System of equations

The equation system is obtained by applying the Hamilton's Principle on the basis of the energetic method of the dynamic systems. The total energy of the system is the sum of the potential energy and kinetic energy [11].

\[ \Pi = U + T \tag{26} \]

If \(\delta_U\) is the virtual field kinematically admissible as \(\delta_U(t_1) = 0\) and \(\delta_U(t_2) = 0\), the total energy of the system is stationary, that is to say:

\[ \delta \Pi(t) = 0 \tag{27} \]

or

\[ \delta \int_{t_1}^{t_2} \Pi(t) \, dt = 0 \tag{28} \]

### 4.2.9 Discretization by finite elements method

An assumption of separation of variable spatial and temporal evolution of the fields

\[ u(x, y, z) = \sum_{i=1}^{n_e} N_i(x, y) u_i(t) \tag{29} \]

it comes to

\[ \{\dot{u}\} = [N]\{\dot{q}_e\} \]

\[ \{\ddot{u}\} = [N]\{\ddot{q}_e\} \]

The discretization of Hamilton's principle gives:

\[
\{\delta q_e\}^T \int \left( [B_m]^T [A] [B_m] + [B_m]^T [B] [B_f] + [B_f]^T [B] [B_m] + [B_f]^T [D] [B_f] + \rho [N]^T [N] [\dot{q}_e] \\
- [N]^T \{f^u\} \right) \, dv - \{\delta q_e\} \int_s [N]^T \{f^u\} \, ds = 0
\]

The general system of equations is:

\[ [M_e]\{\ddot{q}_e\} + [k_e]\{q_e\} = \{f_e\} \tag{30} \]
The transition matrix is written as:

\[
[T] = \begin{bmatrix}
[t^m] & 0 & 0 \\
0 & [t^m] & 0 \\
0 & 0 & [t^m]
\end{bmatrix}
\]

with

\[
[t^m] = \begin{bmatrix}
\cos(x,X) & \cos(x,Y) & \cos(x,Z) \\
\cos(z,X) & \cos(z,Y) & \cos(z,Z)
\end{bmatrix}
\]

4.2.10 Resolution Procedure

1. Evaluate the matrices \([M]\) and \([K]\)
2. Evaluate the frequencies of the system: \(|[K] - \omega^2[M]| = 0\), we deduce \(\omega_i\) et \(\{\Phi_i\}\) for each mode \(i\):
   a - calculate \(\hat{m}_i\) and \(f_i\)
   b - solve the equation of mode: \(\ddot{Y}_i + \omega_i^2Y_i = f_i/\hat{m}_i\)
   c - determine the response of the structure: \(\{q(t)\} = [\Phi][Y(t)]\)

5. Numerical results

5.1 Presentation of the blade

We developed a finite elements model which has 2584 nodes and 862 elements, distributed over the blade (Fig. 5). The numerical values used are:

- \(b = 6\)m for the length of the blade
- \(c = 0.4\)m for chord

![Fig. 5. Finite elements model of blade: 2584 nodes, 862 elements](image)

5.2 Element type of meshing:

Blade meshing:

The element used for in this work is SHELL99, it is an element used for the structural applications in stratified. The element 8th nodes (I,J,K,L,M,N,O,P) has five degrees of freedom to each node: three translations in the x, y, z axis and two rotations around x, z. We consider a blade with ten plies of glass / epoxy with the following orientation \([0^\circ/90^\circ/0^\circ/90^\circ/0^\circ/90^\circ/0^\circ/90^\circ/0^\circ/0^\circ]\) (fig. 6).
5.3 Results and Interpretation

The objective of this study is to determine the natural frequencies and mode shapes of the blade.

5.3.1 Modal analysis

5.3.1.1 Natural frequencies of the blade

The frequencies of the blade are shown in Table 2:

Table 2. Natural frequencies of the blade.

<table>
<thead>
<tr>
<th>Mode number</th>
<th>mode type</th>
<th>frequency[Hz]</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Glass epoxy</td>
</tr>
<tr>
<td>1</td>
<td>first flap</td>
<td>1.093</td>
</tr>
<tr>
<td>2</td>
<td>2nd flap</td>
<td>6.624</td>
</tr>
<tr>
<td>3</td>
<td>first lag</td>
<td>6.980</td>
</tr>
<tr>
<td>4</td>
<td>3rd flap</td>
<td>17.654</td>
</tr>
<tr>
<td>5</td>
<td>first torsion</td>
<td>19.328</td>
</tr>
</tbody>
</table>

The frequencies of isotropic material are higher compared to the orthotropic material.

4. 3.1.2 modal shape
5.3.2 Static Analysis

5.3.2.1 Stresses of the blade

Glass epoxy

 Carbon epoxy
Fig. 8. Constraint according to x axis.

Table. 3. Max stress according to x

<table>
<thead>
<tr>
<th>Materials</th>
<th>Glass epoxy</th>
<th>Carbon epoxy</th>
<th>Boron epoxy</th>
<th>Steel</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\sigma_{xx\text{max}}$ (MPa)</td>
<td>115</td>
<td>184.55</td>
<td>208.86</td>
<td>90.49</td>
</tr>
</tbody>
</table>

The stress of isotropic material is greater compared to the orthotropic material. The material becomes more rigid in the loading direction.

5.3.2.2 Evolution of the nodal stress

SX stress according to x axis
SY stress according to Y axis
SZ stress according to Z axis
EV: glass epoxy A: steel
EC: carbon epoxy BE: Boron epoxy

Fig.9.a Evolution of the nodal stress/x.

Fig. 9.b Evolution of nodal stress/xz.
We notice an increase in stress in the middle of the blade, from this point the stress remains constant for this mode; the maximum value of the Von Mises stress is the largest of the blade. For other modes, the equivalent stress increases until the maximum value, which is located in the middle of the blade, then decreases to a minimum value.

5.3.2.3 Displacement of the blade

Fig. 9. c Evolution of the nodal stress/z.

Fig. 10. Displacement according to the x axis.
Table 4. Maximum value of displacement according to the x axis.

<table>
<thead>
<tr>
<th>Materials</th>
<th>Glass epoxy</th>
<th>Carbon epoxy</th>
<th>Boron epoxy</th>
<th>Steel</th>
</tr>
</thead>
<tbody>
<tr>
<td>$D_{\text{max}}(m)$</td>
<td>0.0037</td>
<td>0.005</td>
<td>0.0012</td>
<td>0.00045</td>
</tr>
</tbody>
</table>

The isotropic material undergoes a lower displacement compared to orthotropic material and becomes more rigid in the loading direction.

5.2.3 Harmonic Analysis

5.2.3.1 Analysis of the aerodynamic response to the forces and the centrifugal force of the blade

This study evaluates the response to aerodynamic forces and the centrifugal force in the form of graphs illustrating the signal amplitude as a function of frequency according to the axis (X,Y) (Fig. 11). We suppose that the vector of excitation forces of drag = TX 28.26N, rotor thrust FN N = 6059, the centrifugal force FC = 475.23 KN, and the speed varied in the range of 0-100 Hz.

Fig. 11. Response to the aerodynamic and centrifugal forces in frequency according to the X (---), Y (-----) axis.
The above figures represent the curve of frequency response for rotating speeds from 0 to 100 rpm (the plotted amplitude of vibration is defined as the mean between the maximum and the minimum displacements according to the x, and y axis). These spectra show the existence of dominant amplitude components in the rotational frequency. This is the answer to the aerodynamic forces and centrifugal forces.

It is observed that when the load level increases, the resonance peaks are shifted to lower frequencies, this is due to the loss of rigidity of the blade. The frequency shift is larger for modes of higher range, due to loss of rigidity.

5.2.3.1 Numerical simulation in transient mode at the resonance

The results of the blade response are in the form of graphs showing the signal amplitude versus time (s).

**Fig. 12.** Temporal signal response of the displacement according the X (—), Y (—) axis.

Figures 12 shows the curve of displacement (vibration amplitude is defined as the mean between the minimum and the maximum displacements according the x and y axis). It can be
clearly seen in all graphs representing the spectrum of the various movements spread over the entire range of time, it means our blade works by the three modes (flapping, lagging, and twisting), ie the blade no longer works following the one of the modes mentioned above.

**Conclusion**

We can conclude from these analytical and numerical modeling approaches that the dynamic behavior of the helicopter blade of different materials, as frequencies of isotropic material are higher compared to the orthotropic material. The visualization of modes shows that the end of the blade is the area most highly stressed whatever the lag, flapping or twisting. Also, the results show the difference between an orthotropic composite material and an isotropic material. This difference appears at the displacements level.

For the displacement along the blade, we notice symmetry of distribution of the relative displacement according to the inflection point of the isotropic and orthotropic material. The displacements of an isotropic material are less important than those of the orthotropic material, so it becomes more rigid in the loading direction by cons, it becomes more ductile according the x, y and z axis.

The stresses of an isotropic material is larger compared to the orthotropic material and becomes more rigid in the loading direction and is more ductile in other directions. There is also the location of maximum stresses at the subjected areas to the maximum bending for the principal stresses, the stress intensity and the equivalent stress of Von Mises.

The results of numerical simulation for transient behavior, at the resonance, show clearly that the graphs representing the spectrum of various displacements are distributed over the entire range of time, which means that our blade works by the three modes (flapping, lagging, and twisting).

**References**