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Dynamic response of a horizontal axis wind turbine blade under aerodynamic, gravity and gyroscopic effects

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ABSTRACT

This work presents analytical and numerical dynamics studies of a horizontal axis wind turbine blade subjected to aerodynamic, centrifugal, gravity, and gyroscopic loads. The blade, assimilated to a long beam of variable cross section, is composed of homogeneous and isotropic material. It is discretized with blade elements of constant sections. Using Finite Element Method (FEM), the assembly of these elements constitutes an approximate model of the blade. The analytical study consists on defining the elementary matrices of rigidity, mass, and gyroscopic coupling between vibration and the blade rotation, as well as the elementary vector of the external loads. The numerical study deals with the resolution of the linear system of equations of the blade motion. Then, it will be possible to calculate its static and dynamic responses for a practical case. The numerical results show that the blade presents cyclic deformations under the considered loadings. These sustained vibrations directly affect the fatigue life of the blade, leading to a significant reduction in the operational efficiency of the wind turbine.

1. Introduction

The horizontal axis wind turbine blade is subjected to various loads. During its rotation, the blade is not only subjected to aerodynamic effect due to the wind, but also, to centrifugal effect due to its rotation, to gravity effect, to its own weight and to gyroscopic effect due to additional rotations. Particularly, the direction of the gravity force, which is variable relatively to the blade axis causing blade vibration during its rotation, will be studied and the gyroscopic effect will also be examined. In fact, many authors studied the dynamic behavior of blades, subjected to various types of external loads, to illustrate the negative effects on the blade structure and the environmental noise. The lengthened shape of the wind turbine blade is modeled by cantilever rotating beam of unsymmetrical and variable cross section. Generally, the Finite Element Method (FEM) is used, in the literature, to study the vibration behavior of the structure. The accuracy of the numerical results, obtained by the FEM, depends mainly on the used optimal number of elements and on the continuity of the results at the nodes.

To study the static and dynamic behavior of a wind turbine blade, more particularly modal analysis, El Ghazly [1] wrote a calculation program using shell triangular finite elements type. Also, Chung et al. [2] used the Beam Element Method (BEM) to study the dynamic response of a rotating cantilever beam. They do not consider the torsion freedom degree in their calculation. Younsi et al. [3] used 3D beam theory, taking into account all degree of freedom, to simulate linear dynamic response of a wind turbine blade with horizontal axis. Note that the gravity force is not considered. Maalawi et al. [4] carried out a mathematical approach to reduce the vibration of the blade structure. Thus, they discretized the blade with succession of adjacent elements with different length and section. Indeed, the variables retained to optimize the design of the blade are the length of each element and the inertial properties of the cross sections. Park et al. [5] proposed an analytical procedure, based on the BEM, to calculate the blade natural frequencies in relation with its constant rotation speed. Li et al. [6] studied the dynamic flapwise response of a blade under super-harmonic resonance. However, the blade is a part of a global system constituted by three blocs: blade, nacelle, and tower (Fig. 1). Murtagh et al. [7] analyzed a tower vibration of a wind turbine with horizontal axis and subjected to aerodynamic loading, induced by the blade rotation. Wang et al. [8] used the thin hulls theory to write the linear equations of the tower-blade motion.

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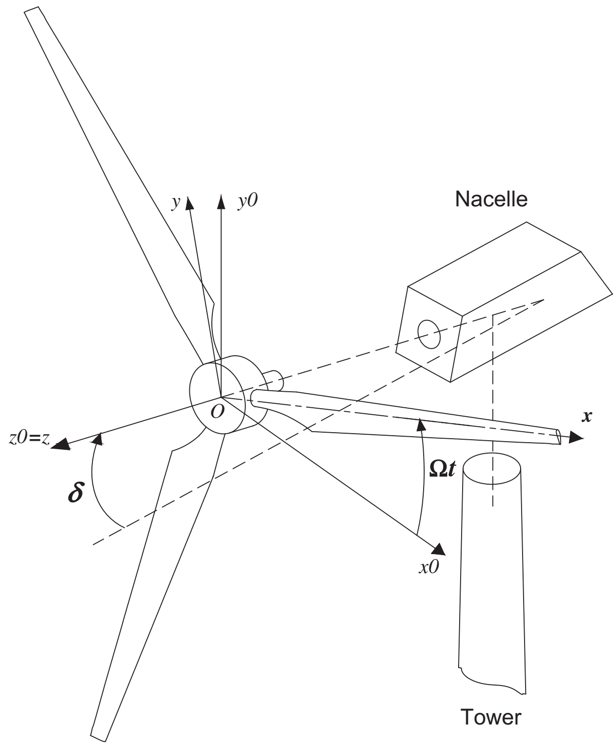

Fig. 1. Parameterization of the horizontal axis wind turbine.

Continuous vibration of the blade causes the fatigue of its structure. For that, El Assal et al. [9] realized fatigue tests on wind blades made of composite material (Glass Fiber Reinforced Polyester (GFRP)). These specimens are subjected to a combined bending and torsion cyclic loads. To reduce this effect, the authors studied technical solutions as done by Kong et al. [10] who proposed a design modification of the blade root to improve its fatigue life. Svendsen et al. [11] studied the resonance modes of rotating beams, on a wind turbine blade, to propose a method to reduce the vibration. Staino et al. [12] proposed a mechanical actuator integrated into the blade to control its edgewise vibration. They account in their dynamic study of three-bladed horizontal axis wind turbine the aerodynamic, centrifugal and gravity effect.

In our previous work Hamdi et al. [13], we studied the static and dynamic responses of a horizontal axis wind turbine blade, under free vibration regime, subjected to a sudden variation of the wind speed (the blade weight was taken negligible). In this work, a new procedure is presented to study the forced vibration regime of a wind turbine blade with horizontal axis rotating at a constant angular velocity Ω (Fig. 1). Thus, we consider the rotating gyroscopic effect of blade on its vibratory behavior under the gravity force. The analytical formulation, developed hereafter, is based on the FEM, the 3D Timoshenko beam theory taking into account the transverse shear, and the Euler configuration of the blade element degrees of freedom. This formulation is applied to study the dynamic response of a blade with 5 m-length that is designed and manufactured by Habali et al. [14]. Finally, we give the expression of the gyroscopic moment of torsion induced in the blade by its combined vibration and rotation.

To simplify our analysis, the blade material is supposed homogeneous isotropic and the mechanical behavior of the blade material is considered elastic linear. The turbine nacelle and the tower are supposed rigid and immobile and the connection between the blade and the hub is rigid. We note that, except indication, the matrices and vectors are expressed in the referential (O,x,y,z) related to the blade.

2. Analytical model

2.1. Blade discretization

The twisted blade is assimilated to a long beam with variable section. It is embedded through its root on the hub of the turbine rotor. The blade is discretized within several adjacent beam elements (Fig. 2). Each element has a length l, constant section S and n is the number of beam elements.

The displacement vector u of an arbitrary point $P(x,y,z)$, on the discretized blade, is deduced from the displacement vector U of point M, image of P through orthogonal projection on Ox axis, as follows:

$$u = \begin{bmatrix} 1 & 0 & 0 & 0 & z & -y \\ 0 & 1 & 0 & -z & 0 & 0 \\ 0 & 0 & 1 & y & 0 & 0 \end{bmatrix} U \tag{1}$$

The displacement vector U contains all the degrees of freedom: three translations u, v, w and three rotations α, β, γ relatively to x, y, z axes, respectively. Using centered linear interpolation, the displacement vector U is expressed according to the nodal displacement vector q_e of the blade element number e as:

$$U = (u, v, w, \alpha, \beta, \gamma)^{\mathrm{T}} = N q_e \tag{2}$$

$$q_e = (u_i, v_i, w_i, \alpha_i, \beta_i, \gamma_i, u_j, v_j, w_j, \alpha_j, \beta_j, \gamma_j)^{\mathrm{T}} \tag{3}$$

where i and $j = i + 1$ are the node numbers and N is the interpolation matrix that can be expressed as:

$$N = \frac{1}{2}\begin{bmatrix} (1-\xi) & 0 & 0 & 0 & 0 & 0 & (1+\xi) & 0 & 0 & 0 & 0 & 0 \\ 0 & (1-\xi) & 0 & 0 & 0 & 0 & 0 & (1+\xi) & 0 & 0 & 0 & 0 \\ 0 & 0 & (1-\xi) & 0 & 0 & 0 & 0 & 0 & (1+\xi) & 0 & 0 & 0 \\ 0 & 0 & 0 & (1-\xi) & 0 & 0 & 0 & 0 & 0 & (1+\xi) & 0 & 0 \\ 0 & 0 & 0 & 0 & (1-\xi) & 0 & 0 & 0 & 0 & 0 & (1+\xi) & 0 \\ 0 & 0 & 0 & 0 & 0 & (1-\xi) & 0 & 0 & 0 & 0 & 0 & (1+\xi) \end{bmatrix} \tag{4}$$

where ξ is the interpolation parameter varying between -1 and 1.

2.2. Static study

By taking into account the transverse shear, the linear part of the strain vector of the blade element ε_l is calculated according to the components of the displacement vector U as: