Validation expérimentale d'un modèle numérique 3D pour la conception des éoliennes sous contraintes vibratoires : Plateforme *TREVISE*

Experimental Validation of a numerical 3-D Finite Model Applied to Wind Turbines Design under Vibration Constraints: TREVISE Platform

T. Sellami[1, 2], S. Jelassi[1], A.M Darcherif[1], H. Berriri[2] et M.F Mimouni[2]

1 ECAM-EPMI, Quartz-Lab, EA 9373, Cergy-Pontoise CEDEX, France, e-mail : takwa.sellami@etu.u-cergy.fr, m.darcherif@ecam-epmi.fr
2 ENIM, ESIER, 5000, Monastir CEDEX, Tunisie, e-mail : MFaouizi.Mimouni@enim.rnu.tn

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Keywords: *Wind turbine design; finite volumes method, vibration phenomena; experimental modal analysis; TREVISE Platform.*

Résumé

Ce papier s'intéresse à l'analyse dynamique d'éoliennes en étudiant les caractéristiques vibratoires des composants soumis aux vibrations (les pales, la nacelle et la tour). Un modèle numérique tridimensionnel (3-D) d'une éolienne a été créé avec le logiciel académique d'analyse par volumes-finis ANSYS en utilisant la méthode de volumes-finis (FVM). Pour vérifier la validité du modèle, des tests vibratoires expérimentaux ont été réalisés au sein de la plateforme vibratoire TREVISE à ECAM-EPMI. Les tests sont basés sur la technique d'analyse modale (MA), une des plus fines techniques d'identification de réponses dynamiques de structures. Afin de maximiser l'énergie d'entrée, des forces sur des bandes fréquentielles d'intérêt ont été appliquées pour exciter l'éolienne. Les vibrations sont mesurées à plusieurs endroits de l'éolienne en utilisant des accéléromètres connectés au système d'acquisition de donnés. En effet, les fonctions de réponses en fréquences (RFR) entre les spectres d'entrée et de sortie sont calculées pour extraire les modes et les fréquences propres de la structure et ainsi mettre à jour le modèle numérique conçu.

Abstract

In this paper, a dynamic analysis of wind turbines is investigated by studying the vibration characteristics of their components (the blades, the nacelle and the tower) under vibration constraints. A three-dimensional (3-D) numerical model of a wind turbine created by the finite volume method (FVM) is set up using the academic finite element analysis software ANSYS. To check up the validity of the model, experimental vibration tests are carried out using the vibration test system of TREVISE Platform at ECAM-EPMI. The tests are based on the modal analysis (MA) technique, which is one of the finest techniques for identifying structures dynamic response. To maximize the input energy, forces are applied in the frequency range of interest to excite the wind turbine. The vibrations are measured in several locations of the wind turbine using accelerometers connected to a dynamic data acquisition system. Indeed, frequency response functions (FRF), between input and output spectra,

are calculated to extract the mode shapes and the natural frequencies of the structure. Based on the obtained modal parameters, the numerical designed model is up-dated.

I. Introduction

Wind energy is still one of the first effective solutions for environmental degradation conditions, energy crisis and global warming. The wind energy capacities are expected to carry on being the essential sources of energy around the world. The Global Wind Energy Council declares that more than 35 GW of new wind power capacity was brought online in 2013 [1]. The windiest zones of countries are still capturing researchers to install hybrid renewable energy systems working in both grids connected and standalone mode [2]. Tunisia is one of the countries where wind and solar radiation are two abundantly available energy resources especially in the northeast of the Tunisian territory with wind speed levels of 10–15 m/s as shown in Fig. 1 [3]. The used wind power capacity would attain 1000 MW if clean energy potentials were fully exploited. Unfortunately, only 11.4% of it is explored in 2016 [4].

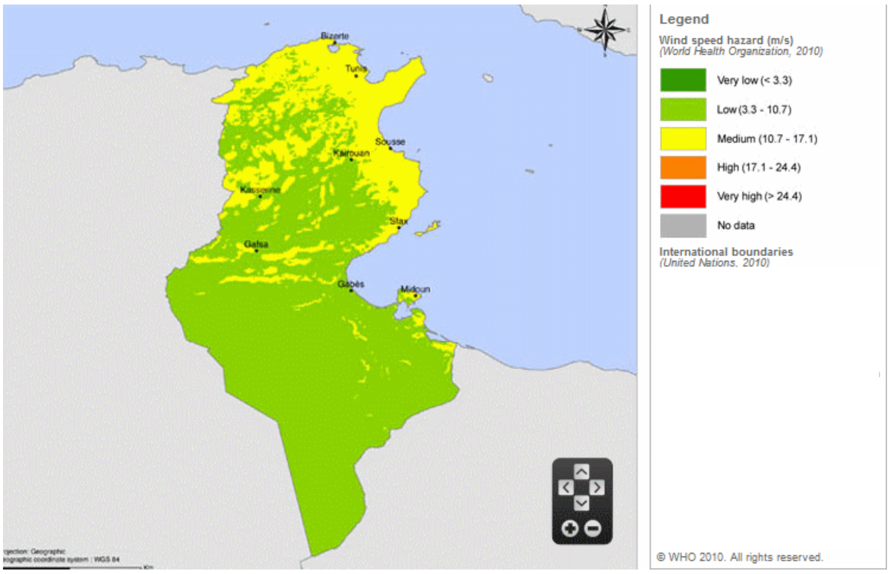

Fig. 1. Topographic map of average wind speed in Tunisia

The wind applies steady alternating and turbulent *forces specifically on the blades causing several stresses and deformations of the wind turbine* [5]. To operate under these complex conditions, dynamically complex wind turbine structures are designed [6]. Structure designing is based on dynamic analysis under multiple shocks and seismic events. The non-study of the structure dynamic can cause its damage citing for example the famous collapse on 7^{th} November 1940 of the Tacoma Narrows Bridge under steady wind loads during a 42-mph windstorm, after four months of its construction [7-8].

Sinusoidal vibrations are used for earthquakes and blasts tests [9]. Random vibrations are usually used for roads transports tests. However, before leading off studying the comportment of structures facing random, seismic or shock loads, a MA is required to ensure that the designed model responds with exactly the same manner as the real structure [10]. The technique gives engineers an idea of how the design will respond to different types of dynamic loads to avoide resonant vibrations. It helps in calculating solution controls for other dynamic analyses [11]. The principle of the technique is to evaluate the structure vibration characteristics. Hence, a numerical three-dimensional model of a real small wind turbine is set up using ANSYS to solve vibrations differential equations by FVM [12-13].

In this paper, an analytical free-vibration study of wind turbines is exposed. Then, the turbine vibrations are numerically analyzed with ANSYS. An experimental validation, by TREVISE vibration platform, of the numerical model is described in the fourth section basing on modal analysis tests.

II. Wind turbine analytical modal analysis

Wind turbines convert wind energy to electrical one. The turbine system can be composed to several parts, blades, a tower and a nacelle. Each part is including a mass *m*, a stiffness spring *k* and a viscous damping coefficient *c*. Many works have indicated that the dynamics of a blade vibration, as a complex system, could be considered as a multi-degree of freedom system [14]. However, the dynamics of the tower vibration can be considered as a single-degree of freedom system as its mass can only move along the vertical z-axis.
The wind turbine aerodynamic power P_a, captured from the wind, can be written by the following nonlinear expression

$$P_a = \frac{1}{2}\ \rho\ \pi \mathrm{R}^2\ V^3 C_p\left(\lambda, \beta\right) \qquad \text{(eq. 1)}$$

Where *R* is the rotor radius, *V* is the wind speed, *ρ* is the airflow density, C_p is the power coefficient depending on the pitch angle *β* and the tip speed ratio *λ* defined by $\lambda = R\omega_r / V$. ω_r is the rotor speed of the wind turbine.

The vectors *{u}*, *{u̇}* and *{ü}* present respectively the displacements vector, velocities vector and accelerations vector of the various masses. Under external forces action, the differential equation of motion of the wind turbine structure is given by

$$\left[M\right]\left\{\ddot{u}(t)\right\} + \left[C\right]\left\{\dot{u}(t)\right\} + \left[K\right]\left\{u(t)\right\} = \left\{F\left(t\right)\right\} \qquad \text{(eq. 2)}$$

Where [M] is the generalized mass matrix, [C] is the generalized damping matrix, [K] is the generalized stiffness matrix and *{F(t)}* is an external matrix. *u* presents the displacement in lateral direction. For the modal analysis study (free-vibrations conditions), eq. 2 becomes

$$\left[M\right]\left\{\ddot{u}(t)\right\} + \left[C\right]\left\{\dot{u}(t)\right\} + \left[K\right]\left\{u(t)\right\} = \vec{0} \qquad \text{(eq. 3)}$$

In the absence of damping eq. 3 becomes

$$\left[M\right]\left\{\ddot{u}(t)\right\} + \left[K\right]\left\{u(t)\right\} = \vec{0} \qquad \text{(eq. 4)}$$

Assuming a solution for eq. 4 given by

$$\left\{u(t)\right\} = \left\{U\right\} e^{j w_n t} \qquad \text{(eq. 5)}$$

Where {*U*} is the displacement amplitude vector. eq. 4 leads to

$$\left[K\right] - \omega_n^2 \left[M\right] = \vec{0} \qquad \text{(eq. 6)}$$

The equation of the eigenvalue is then expressed as follows

$$\left[M\right]\varphi_i \omega_i^2 = \left[K\right]\varphi_i \qquad \text{(eq. 7)}$$

Where ω_i is the eigenvalue and φ_i is the mode shape vectors.

III. Wind turbine numerical modal analysis

To solve vibrations analytical differential equations, a numerical 3-D model of a real small-scale wind turbine is set up by FVM using ANSYS. The characteristics of the turbine are presented in the appendix. The wind turbine structure is considered as a hybrid system combining mechanical parts (generator, shaft and the nacelle) and structural parts (the blades and the tower). A blade has a high stiffness and a low mass. The blade material is supposed elastic and linear. The structure material is considered homogeneous isotropic. The tower is modeled as Euler–Bernoulli beam systems having flexible and damping properties [15]. The nacelle and the tower are considered rigid. The connection between the different parts is rigid. A fine mesh is chosen in locations where the deformation results are of interest as presented in Fig. 2. Based on the modal analysis principle, the linear general equation of the structure motion (eq. 3) is solved to determine the structure natural frequencies and mode shapes.

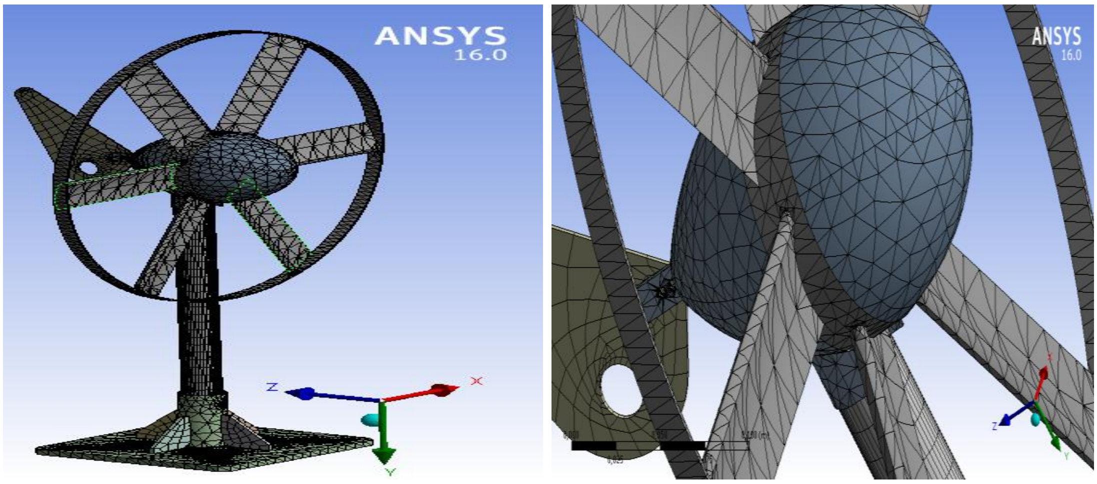

Fig. 2. Meshed wind turbine design by ANSYS (left) with zoom on the mesh elements size (right).

IV. Wind turbine experimental modal analysis

1. Experimental setup: vibration analysis platform (TREVISE)

The experimental modal analysis tests have been performed in the vibration analysis platform TREVISE. The platform ensures the understanding of vibration phenomena of different embedded systems. It is constituted of a collection of instruments, including a vibrator that can accommodate items up to 500 kg, shakers and two machines for vibro-climatic tests for vibrations endurance check under extreme temperatures (-100 / +200 ° C) as shown by Fig. 3. Fig .4 presents the composition of TREVISE vibrator supporting objects up to 500 kg for weight and 100 g for accelerations. The object under test can be excited with the help of an electro-dynamic vibration shaker in three directions (x, y and z-axis). The electromechanical muscle needs to provide a sufficient force to move the item under test at the acceleration level selected. An accelerometer is fixed on the shaking table to measure the vibratory motion during the random, swept sine or shock test. The shaker is controlled by an amplifier and a controller. The necessary selected requirements of each test are transmitted to the controller from a laptop. The controller compares continuously the measured control signal to the desired reference profile specified for the test. Then, it calculates the needed drive signal for the shaker to power the control. An adequate frequency bandwidth is also fixed to fit in the test's specification.

Afterward, signals are stored into one file by the data acquisition package. Thus, the package transforms the physical inputs (acceleration, force, current and strain) into legible digital signals. Then, the digital signals are analyzed, filtered and post-processed. A fast Fourier transform (FFT) analyzer is likewise used for signals spectral analysis with selectable resolution of the bandwidth. FRF are the transfer functions in the frequency domain given by the package to analyze systems. They correspond to the quotient of the responses and the excitations.

Fig. 3. Components of the vibration analysis platform TREVISE: the machine for vibro-climatic tests, TREVISE vibrator and the data acquisition package.

Fig. 4. Overview of the components of TREVISE vibrator.

2. Wind turbine experimental modal analysis tests

In the beginning, the MA tests were performed by exciting the wind turbine with a calibrated hammer at different points. Different pulse shocks were applied on the support of the turbine as shown in Fig. 5. The measured responses were collected by accelerometers measuring the vibratory motions, connected to the data acquisition system. Then, spectral contents of all the FRF were computed using the FFT analyzer. Each FRF is between an input signal and an output spectrum defined as an acceleration of the turbine over a force of the hammer. Afterward, the Peak Picking (PP) method was utilized to evaluate the modal parameters of the wind turbine [16]. The method is based on picking the common peaks of all the FRF in ascending order in every frequency band. The first picked peak corresponds to the first natural frequency of the wind turbine relative to the first mode shape. Afterwards, the MA tests of the wind turbine were achieved with TREVISE vibrator as presented in Fig. 5. Fifteen accelerometers were fixed in different locations of the wind turbine structure; on the tower, on every blade and on the nacelle. Swept sine input signals were chosen to excite the strucure in the frequency band of interest. Natural frequencies and mode shapes were evaluated from the FRF between accelerations of the turbine over the acceleration of the vibrating table. Equally, as presented in Fig. 6, the vibration characteristics of the wind turbine are identified by extracting the common peaks in every frequency band.

Fig. 5 . TREVISE plarform with the small-scale wind turbine and the test instrumentations: the calibrated hammer test (left) and TREVISE vibrator test (right).

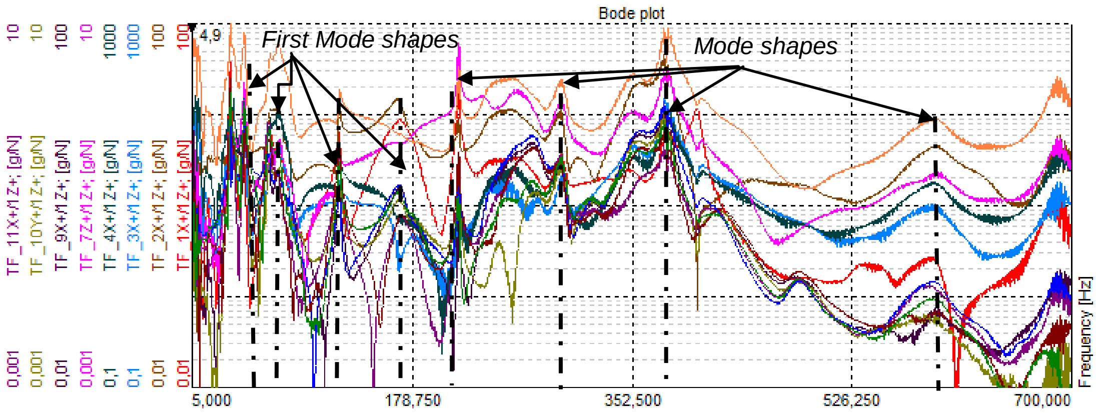

Fig. 6. Calculated FRF of the experimental vibration tests of the wind turbine structure presenting some mode shapes.

Modes obtained by the experimental MA tests are compared to those calculated by the numerical study. Results are gathered in tab. 1. The first found natural frequency is 17,09 *Hz*, then the frequencies increase to reach 73,11 *Hz* for the 10^{th} mode shape. It is deduced that the natural frequencies of the small-scale turbine are high compared to those of turbines producing electricity to grids. These high values are expected because the tested turbine has a dimension of 0.22% compared to a big turbine with 160 *m* tower-height. In addition, after adjusting the numerical model parameters of the wind turbine, relative errors between numerical and experimental tests are reduced. The maximum relative error saved is 8,982*%* corresponding to the eighth mode shape. For the first five natural frequencies, the errors do not reach the 2%. This result affirms that the VFM model designed by ANSYS can confidentially present the real Rutland 504 turbine used for the study.

A comparison between the equivalent elastic deformations of the first and the tenth mode shapes is presented in Fig. 7. The change of the natural frequency from 17,194 Hz to 73,113 Hz caused important deformations of the wind turbine. The tower and the nacelle are not affected. However, the blades are the components mostly suffering from the distortions because of their particular geometries and materials. But, even at the tenth natural frequency (73,113 Hz) the blades were not damaged as they were protected by the outer ring. That can explain the reason for the presence of the ring in this kind of wind turbines designs.

Mode shape	Frequency [Hz] by ANSYS	Frequency [Hz] by TREVISE platform	Relative Error (%)
1	17,194	17,09	0,60
2	28,872	28,33	1,87
3	31,33	31,01	1,02
4	36,2	36,62	1,16
5	39,238	39,06	0,45
6	47,687	47,65	0,18
7	51,32	52,79	2,86
8	51,383	56	8,982
9	67,843	63,78	5,98
10	73,113	71,1	2,75

Tab. 1. Numerical and experimental results: Mode shapes, natural frequencies and relative errors.

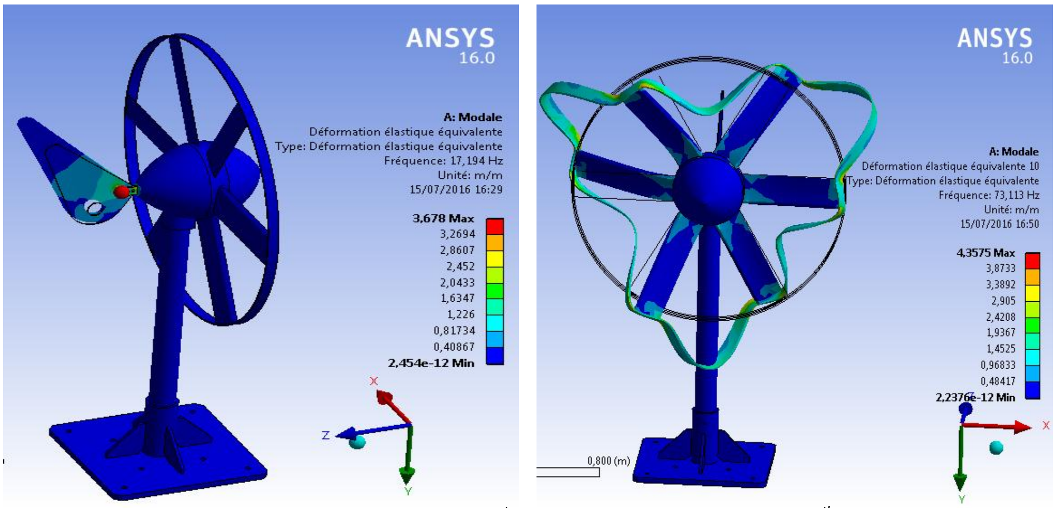

Fig. 7. Equivalent elastic deformation for the 1st mode shape (left) and the 10th mode shape (right).

V. Conclusion

The work presents a new method for creating original 3-D wind-turbine models by validating them by the experimental modal analysis tests. Hence, the vibration phenomena of wind-turbines were first studied by an analytical analysis. Equations of the structure motion were presented under free-vibration conditions. Then, a developed 3-D model of a small-scale wind turbine was designed by the finite volumes method. After specifying materials of each component of the turbine and well meshing the model, a free-vibration numerical study was realized to calculate the natural frequencies and mode shapes of the turbine for checking up the model. Afterwards, to validate the created model, experimental modal analysis tests were realized. A hammer was initially used for pulse shocks tests. Then, swept sine forces excited the turbine by TREVISE vibrator. The vibration characteristics of the turbine were evaluated by the peak picking method after calculating the frequency response functions. The vibration structure characteristics attained by ANSYS were compared to those obtained by the experimental modal tests. The model was updated until reaching the minimum errors. The work proves also that blades are the most sensitive parts of the wind turbine to vibrate. It is due to their particular geometries and materials. Based on this structural model, dynamic analyses under multiple shock and seismic events can now be investigated studying the temporal and frequency variation of the modal parameters of the components of the turbine and estimating their fatigue damage.

APPENDIX: Wind turbine characteristics

The tested wind turbine is a Rutland 504 wind turbine. The tower of the turbine is constructed from steel. The blades and the nacelle are constructed from the plastic material characterized by a modulus of elasticity E=2,410^9 *Pa*, Poisson ratio v=0,3 and a density ρ=*1200 kg/m*^{3}. The following table presents the dimensions of the designed turbine.

Nominal Power	***60W***
Radius R	***0,26m***
Tower attitude	***0,35m***
Number of blades	***6***

Tab. 2. Characteristics of the wind turbine.

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