World Academy of Science, Engineering and Technology
International Journal of Aerospace and Mechanical Engineering
Vol:8, No:12, 2014

Open Science Index, Aerospace and Mechanical Engineering Vol:8, No:12, 2014 publications.waset.org/10000582.pdf

Behavior Fatigue Life of Wind Turbine Rotor with Longitudinal Crack Growth

S. Lecheb, A. Nour, A. Chellil, H. Mechakra, N. Hamad, H. Kebir

***Abstract*—**This study concerned the dynamic behavior of the wind turbine rotor. Before all we have studied the loads applied to the rotor, which allows the knowledge their effect on the fatigue, also studied the rotor with longitudinal crack in order to determine stress, strain and displacement. Firstly we compared the first six modes shapes between cracking and uncracking of HAWT rotor. Secondly we show show evolution of first six natural frequencies with longitudinal crack propagation. Finally we conclude that the residual change in the natural frequencies can be used as in shaft crack diagnosis predictive maintenance.

***Keywords*—**Wind turbine rotor, natural frequencies, longitudinal crack growth, life time.

I. INTRODUCTION

THE Horizontal Axis Wind Turbines (HAWT) have the main rotor shaft and electrical generator at the top of a tube and must be functioned in the wind. All wind turbines have a gearbox which converts the slow rotation of the rotor and rotates rapidly, which is necessary for the electrical generator is operational. The rotors are the main part in rotating machines; the failure is due to the initiation and propagation of cracks in materials under cyclic loading. The effect of cracks on the dynamic behavior of structural elements is very important. Cracks in a component will change its dynamic behavior and change its stiffness and damping properties. A finite element analysis for the cracked shaft for structural component is developed by [1] and showed significant changes in the resonance frequencies and amplitudes of vibration by the existence of moderate cracks. Reference [2] analyzed the stability and the degree of stability of a cracked flexible rotor supported on plain bearing and proposed tree stability decreases with the presence of the crack. Reference [3] made modeling and analysis of vibration of a single rotor, with a crack in the respiration. Reference [4] studied experimentally the response of a cracked rotor to periodic axial excitation and therefore proposed that the rotor response to axial impulse excitation could be used for the reliable diagnosis of rotor cracks. One of the issues is the renewable systems out of the rotor due to the propagation of undetected crack. The shaft breakage and fatigue failure occurs after the following three stages: crack initiation, crack propagation, the final rupture. Reference [5] studied the vibration control of wind turbine incorporating a passive control. The method of passive control using a tuned mass damper to reduce vibration of the blades and the tower of a wind turbine was introduced. Reference [6] talked about the estimation of aeroelastic damping of operational wind modes located on experimentation; the analysis also provided vibrations in a turbine with three blades. [7] directed extend the life of a box smoothing transient generator couple for monitor vibration. The dynamic behavior is exchange between the cracked and uncracked components for exactly the same material and size. Odd if the crack is small, the dynamic behavior change. This occurs because the natural frequencies are decrease. It is shown that cracks in a structure will change the flexibility and stiffness of material. The aim of this study is to determine the stress, strain, displacement, and frequency modes shapes also the fatigue failure of wind turbine rotor due to deferent crack size in its shaft under cyclic loading, by using ABAQUS software. So in this study we will focus the rotor in general and its main shaft specially. The conversion of wind power into mechanical energy to the rotor of a wind turbine is influenced by different forces acting on the blades and the tower of the wind turbine: Coriolis force gravity, aerodynamic forces, gyroscopic forces, with mechanical effects affecting the energy transformation [8].

II. MODEL OF ROTOR

In this early work on the prediction of critical speeds of rotors were presented by Jeffcott [9], the model has been improved by the introduction of the gyroscopic effects and critical speeds of simple rotor systems. The Rayleigh -Ritz method is widely used because it provides a simple model of rotor with two degrees of freedom [10], [11], but it is not very accurate in the case of real systems study. Horkildsen [12] included in the rotational inertia and gyroscopic moment for the first time. Items can therefore be added or removed according to processes that want to be highlighted. Many results on the dynamics of rotors whose support is fixed patterns for Rayleigh -Ritz and finite elements are presented in [11]. Samali [13] studied a random rotor subjected to earthquake vibrations, his model takes into account possible components of movement 6 to the support (3 translations and 3 rotations), but considered as a rigid rotor shaft placed on flexible bearings. Suarez [14] and Singh [15] took into account the flexibility of to develop the tree equations of motion using the finite element method for a rotor whose support 6 is subjected to motion components. The dynamic behavior of rotors has been extensively studied by [11], [16], [17].

S. Lecheb is with the Laboratory of Engines Dynamics and Vibroacoustics, University of Boumerdes, Algeria (e-mail: samir_lecheb@yahoo.fr).

A. Nour and A. Chellil are with the Laboratory of Engines Dynamics and Vibroacoustics, University of Boumerdes, Algeria.

H. Kebir is with the Laboratory Roberval, University of Technology Compiègne, UTC, France.

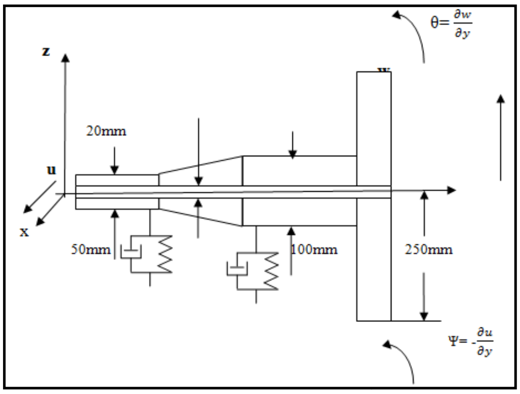


Fig. 1 Model of rotor

For a test square specimen having an edge crack on the ridge, the stress intensity factor SIF is determined by [18] for a homogeneous material subjected to a uniform loading:

$$K = Y\sigma\sqrt{\frac{\pi a}{Q}} \tag{1}$$

σ is the applied stress, Q is the shape factor. For a semi-elliptic notch,

$$Q = 1 + 1,464\ (a/c)^{1.65} \tag{2}$$

Y is a geometric correction factor,

$$Y = B_0 + B_1(a/t) + B_2(a/t)^2 + B_3(a/t)^3 + B_4(a/t)^4 \tag{3}$$

B is function as a, c and t (a and c are the dimensions of the notch and t is the thickness of the specimen).

III. Numerical Simulation of Rotor

In this paper we determine the stresses distribution, the strain and the displacement of the rotor for two cases which are rotor without crack and rotor with crack. Also deducing the Mode deformation as a function of the frequencies .we introduces ABAQUS software for simulates this rotor.

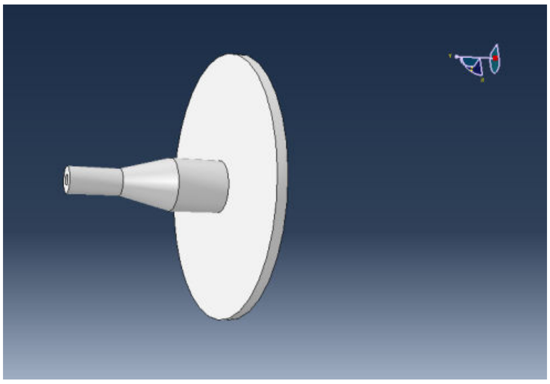


Fig. 2 Rotor model

In our work we introduce two forces, the first is aerodynamic and the second is centrifugal. We take the following forces and boundary condition (Fig. 3).

Mesh is a decomposition or arrangement the model to finite element. We choose the tetrahedral type of element. In our rotor the number of elements is 43903 elements.

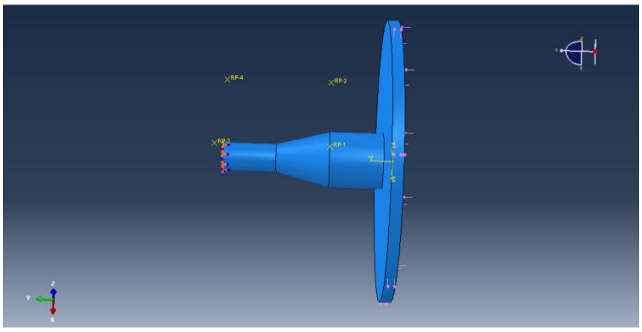


Fig. 3 Boundary condition in rotor

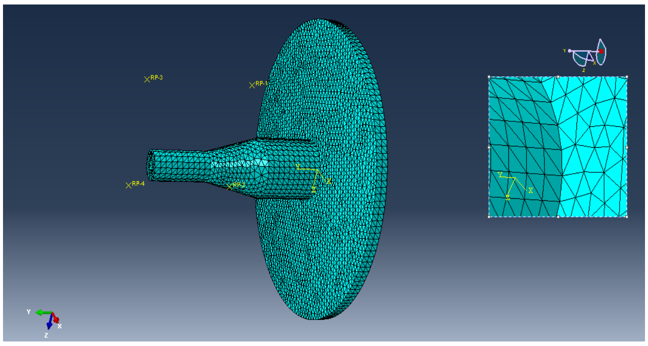


Fig. 4 Rotor mesh

A. Stress

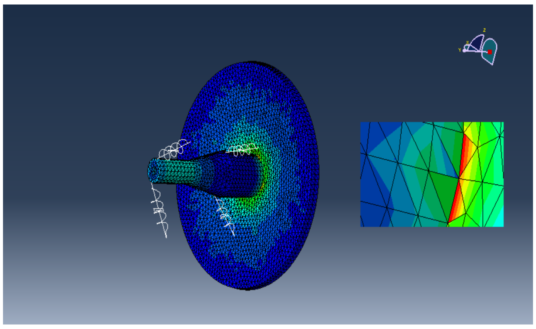


Fig. 5 Maximum stress of Von Misses

The stress is concentrate between shaft and disc which is the maximum value of $\sigma_{v.Mises} = 1.008e+02$Mpa.

B. Strain

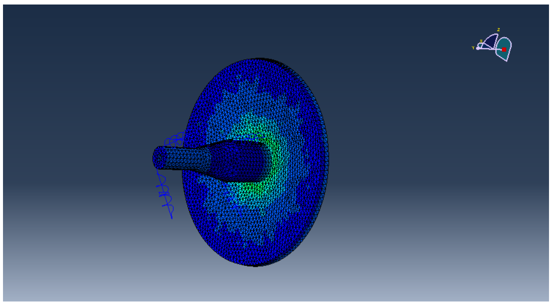


Fig. 6 Maximum strain

We notice that the maximum value of strain is equal to 3.076e-4.

C. Displacement

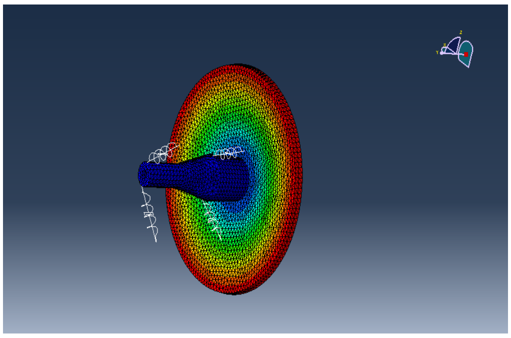


Fig. 7 Maximum displacement

The displacement is very higher at the disc which the maximum value equal to 3.168e-1 mm because the centrifugal force.

D. Modes Shapes

Table I shows the first six natural frequencies with displacements:

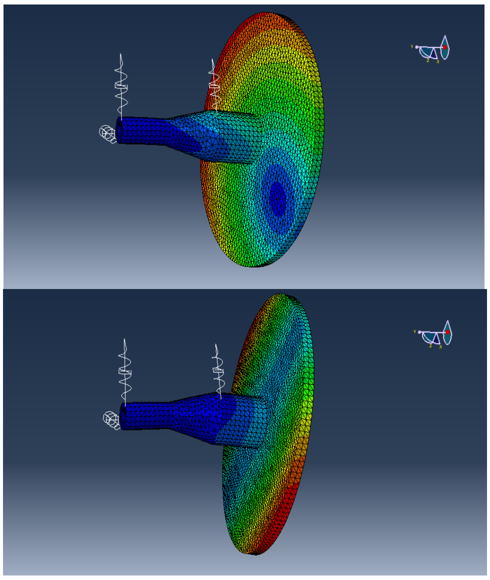
TABLE I
SIX FIRST NATURAL FREQUENCIES OF UNCRACKED ROTOR

F1 (Hz)	F2 (Hz)	F3 (Hz)	F4 (Hz)	F5 (Hz)	F6 (Hz)
2.65	7.71	8.73	13.23	15.11	18.21

IV. ROTOR MODEL WITH CRACK

We simulate the same model with deferent crack size, we begin with 50 mm and we finish with 100 mm (see Fig. 9).

In this case we take the crack into account, so we will mesh it with small size of the mesh element which equal to 10 element with tetrahedral element around the crack finally we get 45006 element .and we use the tetrahedral element as the mesh element for all the part. With based to the crack which was meshed by very small element as showing in Fig. 10.



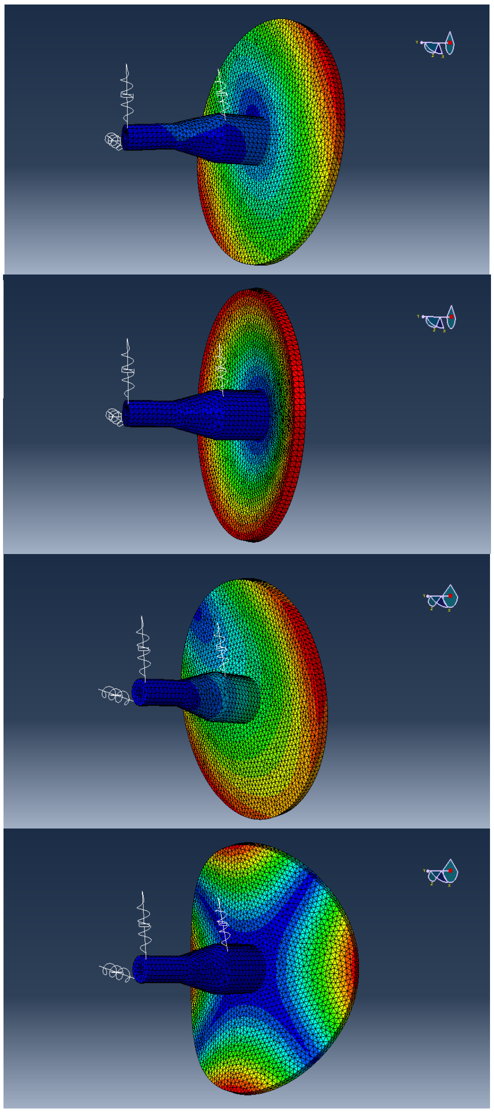


Fig. 8 First six modes shape

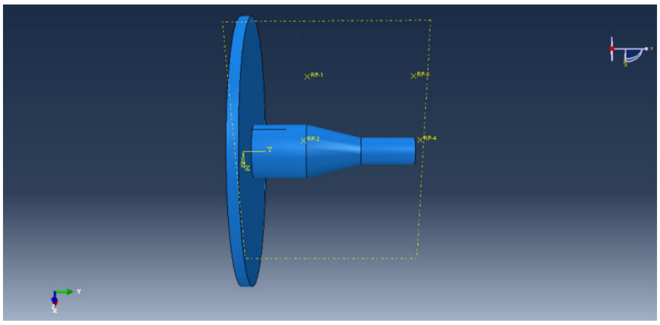


Fig. 9 Rotor with crack

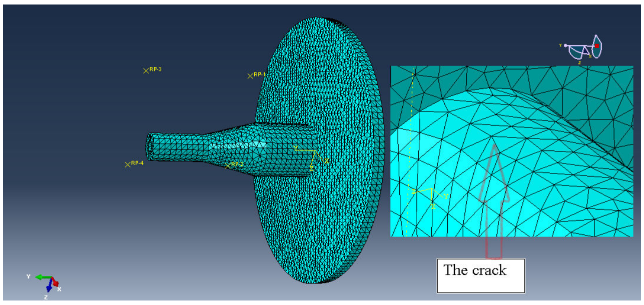


Fig. 10 Mesh element size

V. ROTOR MODEL WITH CRACK

We use in this case 50 mm the length of crack size, and we notice that the stresses are concentrated in the crack tip as showing in Fig. 11.

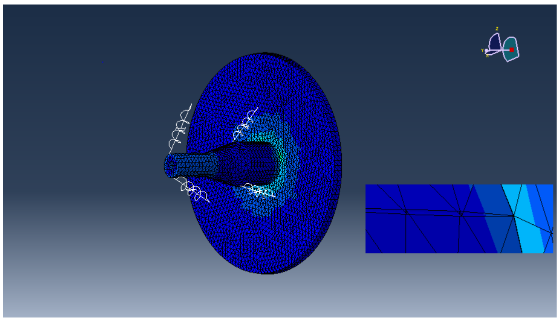


Fig. 11 Von Mises stress for cracked rotor

The stress is concentrate between shaft and disc which is the maximum value of $\sigma_{V.Mises} = 2.035e+02$ Mpa.

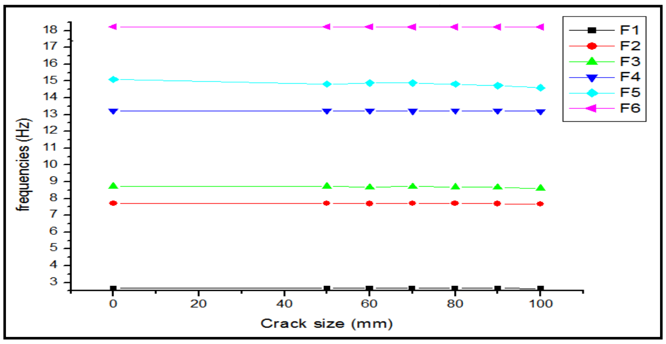


Fig. 12 Six first frequencies as function a crack size

We notice that the frequencies with crack are smaller than without crack and the displacement with crack are higher than without crack and that the frequencies before crack is higher than after crack.

The line black represent the Von-Mises stress(s) and we notice that when crack size increase the Von-Mises stress increase, the line green represent the displacement (U_{max}), we notice that there is small deformation the line red represent the strain (E_{max}), we notice that nearly constant, but there is the small increasing in strain. In this part we simulate the rotor of the wind turbine which subjected by two forces, aerodynamic force and centrifugal force. Also we determine the mode shapes, stresses, displacements, and the strain. And we see that the stresses are concentrated around the crack. The influence of crack size in the stress, displacement, strain and frequency is provided.

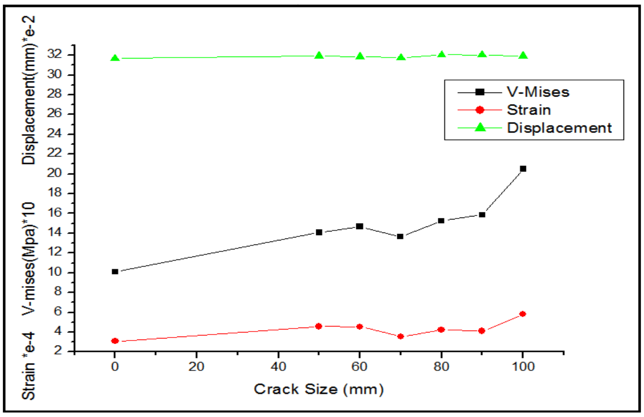


Fig. 13 Stress, strain, and displacement as function a crack size

VI. CONCLUSION

From results we have conclude that when we have a cracks. In statically parte, after cracking the maximum strain and displacement increased, the maximal stress are concentrated at crack-tip, also we can deduce that the dangerous mode shape when we have a maximum displacement. In dynamical part the crack initiation is related to the decrease in natural frequencies modes shapes. About longitudinal crack propagation (from 0 to 100 mm), the displacement, strain and stress increase with crack size increases, and the six first naturals frequencies decrease with crack size increase. The nonlinear change of displacement, strain, stress and natural frequencies can be used as a residual indicator to crack diagnosis predictive maintenance.

REFERENCES

[1] Nelson, H.D. Nataraj C. The dynamics of a rotor system with a cracked shaft, Journal of Vibration, Acoustics, Stress, and Reliability in Design, 108, pp189–196, 1986.

[2] Meng, G., Garsh W. Stability and stability degree of a cracked flexible rotor supported on journal bearing, Journal of Vibration and Acoustics, 122, pp 116–125, 2000.

[3] Jun O.S.; Eun H.J. Modeling and vibration analysis of a simple rotor with a breathing crack, Journal of Sound and Vibration, 155, pp273–290, 1992.

[4] Darpe, A.K., Chawla, A., Gupta, K. Analysis of the response of a cracked Jeffcott rotor to axial excitation, Journal of Sound and Vibration., vol.249, 3, pp 429-445, 2002.

[5] Murtagh, P.J., Ghosh, A., Basu, B.; Broderick, B.M. Passive control of wind turbine vibrations including blade/tower interaction and rotationally sampled turbulence. 2008, vol. 11, issue 4, pp 305-317. (Wind Energy)

[6] Hansen, M.H., Thomsen, K., Fuglsang, P., Knudsen, T., Two methods for estimating aero-elastic damping of operational wind turbine modes from experiments. Wind Energy, , pp 91-179, 2006.

[7] Molinas, M.; Suul, J.A.; Undeland, T. Extending the life of gear box in wind generators by smoothing transient torque with STATCOM. IEEE, Trans. Ind. Electron, vol. 57, no. 2, pp 476–484, 2010.

[8] Z. X. Xing, Q. L. Zheng, X. J. Yao, and Y. J. Jing, Integration of large doubly-fed wind power generator system into grid, 8th Int. Conf. Electrical Machines and Systems, pp.1000-1004, 2005.

[9] H. H. JEFFCOTT 1919 Phil Mag., Series 6, 37, 304. The Lateral Vibration of Loaded shafts in the Neighborhood of a Whirling Speed The Effect of Want of Balance.
[10] Tondi A. Some problems of rotor dynamics, London, Chapman and Hall, 1965, 433 p.
[11] Lalanne M., Ferraris G. Rotordynamics prediction in engineering, 2nd Edition, Chichester, John Wiley, 254 p, 1998.
[12] Horkildsen. T., Solution of a Distrbuted Mass and Unbalanced Rotor System Using a consistent Mass Matrix. MSE Engineering Report, Arizona University, 1972.
[13] Samali B., Kim K. B., Yang J. N. Random vibration of rotating machines under earthquake excitations, Journal of Engineering Mechanics, June 1986,Vol. 112, n° 6, pp. 550-565.
[14] Suarez L. E., Rohanimanesh M. S., Singh M. P. Seismic response of rotating machines, Earthquake Engineering and Structural Dynamics, 21, pp. 2 1-36, 1992.
[15] Singh M. P., Chang T. S., Suarez L. E. A response spectrum method for seismic design evaluation of rotating machines, ASME, 114, pp. 454-460, 1992.
[16] Childs D., Turbomachinery rotordynamics, Wiley and Sons, 1992.
[17] Rao J.S., Rotor Dynamics, 2nd Edition, John Wiley, 1991.
[18] Zheng X.J., Glinka G., and Dubey R.N., Stress intensity factors and weight functions for a corner crack in a finite thickness plate, Engineering Fracture Mechanics, vol. 54, 1 , pp. 49–61, 1996.