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Fatigue life analysis of a wind turbine blade subjected to Davenport loading using the computational Tool ANSYS

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Abstract. Most mechanical elements are subjected to periodic loadings so that throughout their lives some fibers are damaged due to bending cyclic. Even if the stress does not exceed the yield limit, the dynamic characteristic of loads can cause catastrophic failures. Finite Element programs are excellent tools to obtain important parameters of fatigue analysis. In this way, ANSYS uses its structural analysis potential to also analyze the phenomenon of fatigue through the ANSYS Workbench Fatigue Tool module. Firstly, in order to experiment the computational procedure of fadigue analysis a retangular cantilever beam was investigated. It was submitted to harmonic loading and modeled with a solid element of 20 nodes, each one with 3 degrees of freedom, Results obtained in numerical analysis at ANSYS were compared with those obtained in the analytical methodology. The good agreement between them is an important step to encourage the understanding of the quantification of fatigue life of the verne555 wind turbine blade submitted to a davenport wind load. The blade modelling was done using shell elements of 4-nodes and 6 degrees of freedom per node. The results obtained are presented as expected, which demonstrates the correct use of the chosen computational tool.

Keywords: Fatigue, Wind Blade, Computational Analysis

1 Introduction

The fatigue phenomenon is characterized by a process of local damage development as a consequence of cyclic stresses. According to Dowling [1], a fatigue fracture occurs when small cracks appear and develop until their final stage. These cracks arise due to alternating loads that act below the yield point of the material, which can thus cause changes in the microstructure of the component.

In his work, Norton [2] presents a brief summary of the history of reasearch about fatigue area. The German engineer August Wöhler stands out among the forerunners. He began his studies of fatigue failure in the 1850s. The results of his research showed the failures related to the number of stress cycles varying in time, besides a stress limit of resistance to fadigue for steels materials. These analyzes were recorded in an S-N diagram, also known as a Wöhler curve, and became a widely used method for analyzing dynamic loads in structural components.

Determining fatigue life in more complex structures is not an easy task, for this reason it is important to use computer tools that use the Finite Element Method (FEM) associated with tools developed to estimate when the component may fail. The works of Hansen [3] and Manwell et al [4], provide information about the importance of considering fatigue analysis in wind turbine projects, since they are structures subjected to cyclic loads. Filho [5] presents a small blade Verne555 and an experimental tests to analyze the fatigue parameters. Vieira [6] presents a numerical analysis of fatigue in wind turbine blades using configurations with and without extension in the blade length, observing the behavior of stress and strain via the Finite Element Method.

In this work, the primary objective is to experiment the fatigue analysis procedure using the computational tool Ansys Student 2022 R1/ fatigue tool module. For this, important parameters of the fatigue analysis are extracted from an ASTM-36 rectangular steel cantilever beam. This element is taken because it is easy to solve analytically. After, a wind turbine blade is evalueted at the same way. It is analyzed a reinforced and an unreiforced blade and presented some discussions.

2 Object of Study

For experimentation of the fatigue analysis module of Ansys Student 2022 R1, an ASTM A36 steel cantebeam was used. It is a rectangular 2mm x 100mm cross section and 500mm length. It is applied a harmonic load with amplitude of 1kN at the free end of the beam. The load is acting in the direction of the greatest cross section length. The steel used in the analysis has a young modulus and stress strength of 200 GPa and 460 MPa, respectively. It has, also, a Poisson ratio of 0.3 and a density of $7850 kg/m^3$. [7]

The wind turbine blade, here, presents the geometry of a Verne555 model developed by the company Enersud - Soluções Energéticas Ltda. It is adopted the same geometrical parameters by Morais [8]. According to the manufacturer's information, the blade is a composite with E-Glass fiber reinforced with polyester matrix, but, here, it was considered as an equivalent isotropic material. Two models were considered in this work, in order to analyze the behavior of the fatigue parameters. The first one, the blade is a hollow structure and the second, the blade has an internal aluminum reinforcement, as shown in Figure 1.

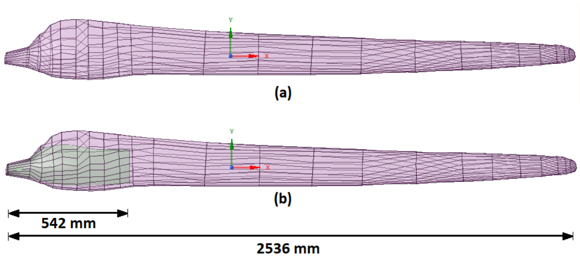

Figure 1. Schematic Verne555 blade (a) without reinforcement, (b) with reinforcement. Source: The author.

The mechanical properties of the outer surface of the blade were the same as those considered by Qiao et.al. [9]. For the reinforcement, an aluminum material was considered. All values are listed in Table 2.

Table 1. Mechanical properties of blade materials. * Source: Qiao et.al. [9].

Material	Epoxy/E-Glass*	Aluminum
Modulus of elasticity (GPa)	24.8	71.4
Poisson's coefficient	0.3	0.3
Density (kg/m^3)	2050	2770
Tensile Ultimate Strength (MPa)	455	310

3 Methodology

The blade fatigue analysis was conditioned to the experimentation of the computational numerical procedure that took the analytical results of a prismatic beam as comparative parameters. The fatigue limit of the material S'_e usually is obtained through tests with specimens, but it can also be estimated. Norton [2] presents the parameters used to determine the correction of this limit:

$$S_e = k_1 k_2 k_3 k_4 ... k_i S'_e \tag{1}$$

where k_i is the correction factor for the i-th effect and S_e the corrected fatigue limit.

In order to carry out an analysis of the fatigue life of a cantilever beam, a model is taken into account in which hypothetical factors are applied to calculate the estimate of the fatigue strength limit of the component subjected to a cyclic load. A loading factor $k_{load} = 1$ is considered as indicated by Norton [2] because it is a flexural load. The beam is larger than a specimen and taking into account its rectangular geometry, the equivalent diameter d_{eq} is found from Equation 2:

$$d_{eq} = \sqrt{\frac{A_{95}}{0.0766}} \tag{2}$$

where $A_{95} = 0.05bh$, where b is the base and h is the height of the rectangular beam. So, as $d_{eq} > 8mm$ according to Norton [2], the size factor will be $k_{size} = 1.189 d_{eq}^{-0.097} = 0.94$.

A surface finish factor was considered according to Equation 3:

$$k_{sur} = A\left(S_u\right)^b \tag{3}$$

where S_u is the ultimate stress of the steel and $A = 4.51$ and $b = -0.26$ are tabulated coefficients considering a cold-machined or cold-drawn component. From these values, we estimate the surface factor $k_{sur} = 0.89$.

The temperature loading factor, according to Norton [2], takes into account the treatment for steel, where, for $T \leq 450°C$ then $k_{temp} = 1$. For a reliability factor of 99.9999%, the respective tabulated value $k_{reab} = 0.62$, as presented by Norton [2], is considered. Assuming that the steel has a stress strength of 460 MPa and that the fatigue limit S_e' for steels is found from Equation 4:

$$S_e' = 0.5 S_u \tag{4}$$

According to Equation 1 and considering $S_e' = 230MPa$, the corrected fatigue limit will be $S_e = 119.3MPa$. According to Norton [2], the S-N curve is determined by the Equation 5:

$$S(N) = aN^b \tag{5}$$

To determine the number of cycles in the high cycling region, where the material strength at $N = 10^3$ is equal to $S(N) = S_m = 0.9S_u$ and $S(N) = S_e$ for $N = 10^6$ cycles, the parameters $b = -0.1801$ and $a = 1436.48$ are found. Thus, using the Equation 5 we arrive at the expression that determines the fatigue life for each stress level in the high cycling region, Eq.6:

$$S_n = 1436.48 N^{-0.1801} \tag{6}$$

The safety factor (SF) can be found from the expression $SF = S_e/\sigma'$, where S_e is the corrected fatigue strength limit and σ' the von Mises stress.

The damage analysis is obtained through the expression $D_i = n_i/N_i$, where n_i is the number of active cycles and N_i is the number of cycles that the material would endure when subjected to a stress of amplitude σ_i [8].

The computational numerical modeling of the beam has 12500 hexahedral solid elements with 2mm of lengh and three degrees of freedom at each node. For the fatigue life analysis, the S-N curve data were added to the material properties in the commercial software taking into account the corrected fatigue limit (S_e) for the high cycling region, between 10^3 and 10^6 cycles.

The method based on stress (Stress Life) was chosen for the analysis, then determined the loading as fully-reversed because it is a load with constant amplitude and totally reversed, that is, with mean stress equal to zero (R=-1). The analysis is designed for an infinite lifetime starting at 10^5 cycles.

After experimenting with the Ansys numerical procedure, it is possible to conduct, with more security, the fatigue analysis process of a more complex component. In this regard, two models of a wind blade made of isotropic material with properties similar to an epoxy/E-glass composite is taken for this analysis, one of them containing an aluminum reinforcement and the otherone don´t have any internal reinforcement. The blades were designed in AutoCAD and exported to Ansys Mechanical APDL. It was attributed shell elements, Shell181, with 5mm thickness. This element is suitable to analyze thin shell structures. It has four nodes and six degrees of freedom at each node. The blade reinforcement was modeled as a solid element, Solid185, with eight nodes and with three degrees of freedom at each node. [10] The mesh on the unreinforced blade has 11192 nodes and 11369 elements. The blade with aluminum reinforcement has totaling 12240 nodes and 14609 elements. In both cases, the finite element is 10mm lenght.

It is applied a pontual Davenport load varying in time in the transverse direction. The load acts at the point one third of the free end. The location of the load refers to the result of a load that would act in a distributed way along the blade, as happens in practice. It is estimated that the wind forces have a similar triangular distribution shape as illustrated in Figure 2.

The Stress Life analysis was defined in order to use the parameters of the S-N Curve of the materials. The point load was modeled as a Davenport loading. In the fatigue module of Ansys Student 2022 R1, a unit loading factor is used. Then the Davenport loading is applied (Figure 3) to an appropriate height for the tower, approximately 60 m, and the load parameter History Data was used. For loads of this nature, the Rainflow cycle counting method is used during the analysis in the Ansys Student 2022 R1/fatigue tool module software. The considered load varies in a time interval of approximately 35 seconds, discretized in 2048 points. Thus, according to Browell and Hancq[11] it became necessary to apply a scale factor of $f_{scale} = 1FEM/2048 = 0.0005$, where FEM is a unit into finite elements.

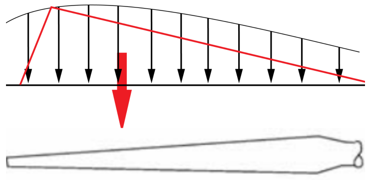

Figure 2. Load application location. Source: Hansen (Adapted).[3]

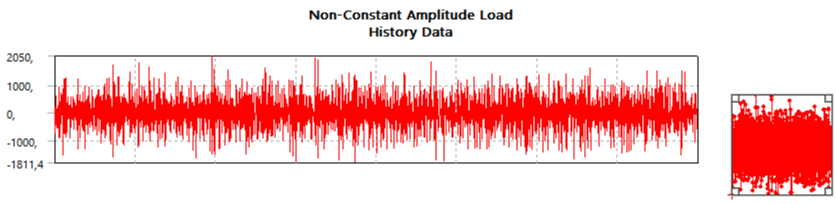

Figure 3. Davenport loading.

Goodman's theory was applied to correct the mean stress as it is a more conservative method. The multiaxial correction method was also defined as Signed Von-mises, because Goodman's theory treats positive and negative stresses differently, Browell, [12]. The S-N curve of the Epoxy-E-Glass composite is defined by the Equation 7, Qiao et. al.[9].

$$\frac{\sigma_a}{\sigma_u} = 0.934 - 0.0815 \log(N) \tag{7}$$

Figure 4 shows the S-N curves of the equivalent Epoxy/E-Glass composite and the Aluminum used in the blade reinforcement.

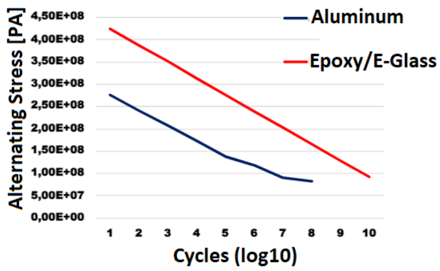

Figure 4. Aluminum and Epoxy/E-Glass S-N curves.

When the loading is of variable amplitude, there may be cycles with very small alternating stresses and may incorrectly predict damage if you have a high number of such cycles. The analysis was conducted with an infinite life of 10^9 cycles. This option defines which life will be used if the load has a stress amplitude smaller than the material's S-N curve.[13]

4 Results and Discussions

In the analyzed structures, the investigation point focused on the setting as it is a critical point of the project. With the use of non-reticulated finite elements, points of high stress concentration are observed, which can be neglected. For the beam, the analysis value was taken at a point approximately 0.006m away from the fixed point. The comparisons of numerical and analytical results are presented in Table 2 below.

According to this initial analysis, it is possible to have an understanding of the fatigue analysis mechanism using the functionalities of the fatigue module of the commercial software Ansys Workbench. This experimentation

Table 2. Comparison of beam results.

Parameter	Analytical	Numerical	Error (%)
Von-Mises Stress (MPa)	149.70	149.18	0.34
Life (cycles)	283,760	289,360	1.97
Damage	3561.8	3462.1	2.79
Safety Factor	0.7969	0.7996	0.34

encouraged to advance with the analysis of the wind turbine blade.

The wind blade was evaluated with and without reiforcement and Table 3 present the fadigue results. At first, the point of maximum stress value was taken on the blade. The results were consistent with expectations, as shown in the first two columns of results in Table 3. It is important to mention that the analysis was conducted with Design Life of 10^4, 10^5 and 10^6 cycles, where the expected life is related to the calculated life. So it was possible to obtain the damage and the safety factor, SF, for each specific case according to the blade life. As it was observed, the blade with the reinforcement will not present fatigue failure due to it presents damage less than 1 and safety factor greater than 1. However, the blade without reinforcement has a short life time. It is noteworthy that the large stress concentration observed in the beam is repeated for the blade without reinforcement. It is common, in practice, to consider artifices to relieve stress in these more critical points, since the perfect setting is an ideal and fictitious representation that corroborates for an increase in the safety factor foreseen in the design of structural elements. Because of this, it was decided to evaluate a point a little further away from the setting. Here, it was admitted a distance in the same proportion as was considered for the beam, 0.03 m. However, for the future study is recommended regarding a proposition that takes into account the geometry of the part, as well as aspects based on the theory.

Table 3. Analysis results using the two blade models.

Model/ Parameter	Without reinforcement	With reinforcement	Without reinforcement (0.03m)	With reinforcement (0.03m)
Life (Cycles)	7.37×10^4	1.33×10^6	1.25×10^6	1.33×10^6
Damage (For 10^4 cycles)	0.14	0.007	0.008	0.007
Damage (For 10^5 cycles)	1.36	0.075	0.08	0.075
Damage (For 10^6 cycles)	13.57	0.75	0.78	0.75
SF (For 10^4 cycles)	1.17	3.11	2.02	11.25
SF (For 10^5 cycles)	0.97	2.36	1.71	9.39
SF (For 10^6 cycles)	0.69	1.72	1.23	6.68

5 Conclusions

The objective of understanding fracture mechanisms is to seek solutions to avoid failures and/or structural collapse. This concern is part of the analysis process of elements subject to cyclic dynamic loads such as wind turbine blades.

Through the analysis of a cantilever beam, the agreement between the analytical and computational results was verified. With the experimentation process using an elementary structure, it was possible to understand the functionalities of tools in which they are made available by Ansys Student 2022 R1 software and fatigue tool module for fatigue life analysis.

The blade fatigue analysis process, despite being simplified, shows acceptable results when analyzing the blades with and without reinforcement, since the first will take more cycles to reach component failure because its greater stiffness due to the reinforcement.

It should be emphasized that the use of computational tools to solve engineering problems is an important apparatus because it allows accurate and fast results, reducing the analysis time.

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