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AN APPROACH TO DETERMINE THE FATIGUE LIFE OF ALUMINUIM WIND TURBINE BLADE

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Abstract

The importance of fatigue in any mechanical system cannot be over looked, due to the occasional cyclic load such systems may undergo. This is especially true for any Horizontal Wind Turbine system. Whose blade generates the desired energy, and thus are subjected to flexural, torsional and at times axial stresses (due to self-weight of the blade), that can cause micro-discontinuity in the material structure, that leads to eventual failure at stress levels much less than their (predictive) material yield and ultimate strength. For the purpose of this paper, which focuses on the structural aspect of the system, some assumptions/conditions will be made. One of such condition to be made is that the pre-process of obtaining the tangential and normal cyclic load acting on the blade are already obtained in advance, using the National Renewable Energy Laboratory (NREL) FAST integrated tool in QBlade. Please note that this is an open software and free for public download.

The fatigue life calculation on the other hand, will be done manually with some assumptions. The results from this project can be useful during validation process with other commercially available software such as ANSYS, NASTRAN, etc. or experimental verification. Hence this paper aims to create a different approach at computing the Fatigue life of any given wind turbine blade, at a more user-controlled level and less financial expense, although it is more time intensive depending on the control constraints in place during the fatigue strength determination. Please keep in mind that Fatigue calculation are deterministic as opposed to the actual stochastic nature of a real-world fatigue problem. Hence there are no perfect mathematical model for analyzing fatigue strength of a structure, since the material are usually assumed to be isotropic, which is not a true case, of material imperfections.

1. Introduction

The current theoretical approaches available for determining the fatigue life of aluminium structures is still a bit unrefined as compared to steel. Nevertheless, the flywheel rotating beam experimental S-N (Wöhler) curve, will be useful, when predicting the behaviour of the fatigue strength as the stress cycle increases.

2. Blade Design and Assumptions

The turbine blade under study is a 3-meter-long blade for micro horizontal wind turbine. This a three-blade rotor optimized for areas of low wind speed, and is capable of generating about 5 kilowatts of electricity at windspeed of 7-8m/s. Some of the Blade parameters which are utilized in the Qblade software to obtained the cyclic dynamic loads that are to be used for the fatigue life estimation, are shown in the table below,

Parameters	Values
Tip speed ratio (TSR)	7.5
Rotor rotational speed (in RPM)	189.5
Coefficient of pressure (Cp)	0.63
Average wind speed	8 m/s

The blade is divided into 17 finite sections (16 control volumes) along its length, which is adequate to account for changes in lift and drag forces due to change in geometry of the blades from the tip (airfoil section) to the circular root section.

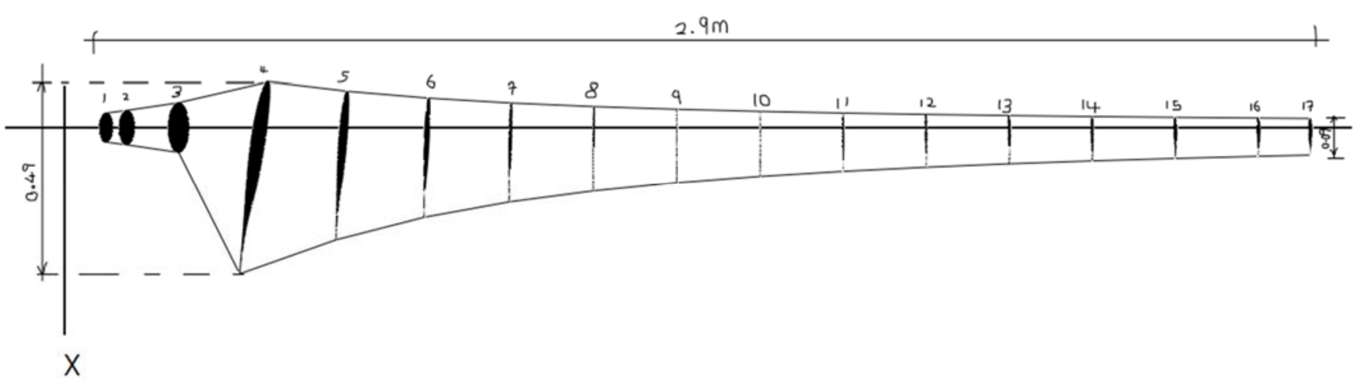


Figure 1 Blade Geometry

3. Material Selection and Properties

The 6000 Aluminum series will be selected for the blades. This material consideration is due to the fact that the conventional composite material has some environment challenges and are very difficult to recycle. Whereas aluminum is 100% recyclable and thus more environmentally friendly. And hence also have lower cost advantage than composite materials. Although the major draw backs of using this lightweight aluminum alloy, for a system under repeated dynamic loads is the much higher structural failure rate due to fatigue. But if proper prediction can be made for the number of cycles before fatigue failure, the blades can be changed and recycled on timely bases. Some important material properties that will be useful for this paper are:

Properties	Average Values
Tensile ultimate strength	333 MPa
Tensile yield strength	282 MPa
Poisson's ratio	0.327
Fatigue strength	170 MPa
Modulus of elasticity (E)	70 GPa
Material density	2740 kg/m3

4. Mathematical Model for Fatigue Strength Determination

a. Extracting The Maximal and Minimal Load Cycle from the Qblade FAST Analysis

Considering only one element from the 17 blade elements. The structural setup will look like that of a cantilever beam with cyclic loads acting at different sections of the blade. And a fixed support at the root of the blade is assumed since the focus of this paper is on the blade and not on the joint type. Hence, considering the element with the maximum von-misses stress ($\sigma' \approx \sigma'_m + \sigma'_a$), as shown below:

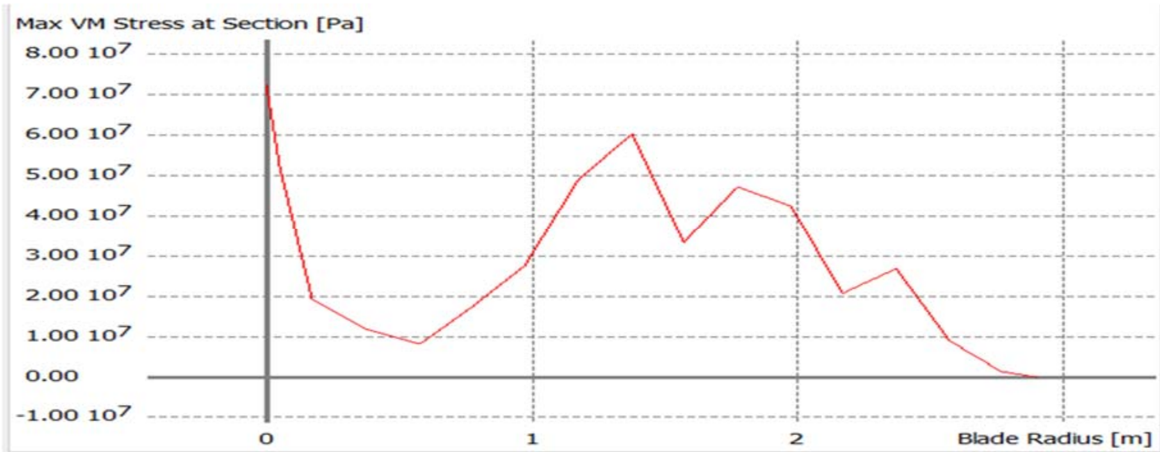


Figure 2 Von-misses stress distribution along Blade length

The FAST analysis in QBlade gives the following normal and tangential loading cycle with respect to time at a given section of the blade:

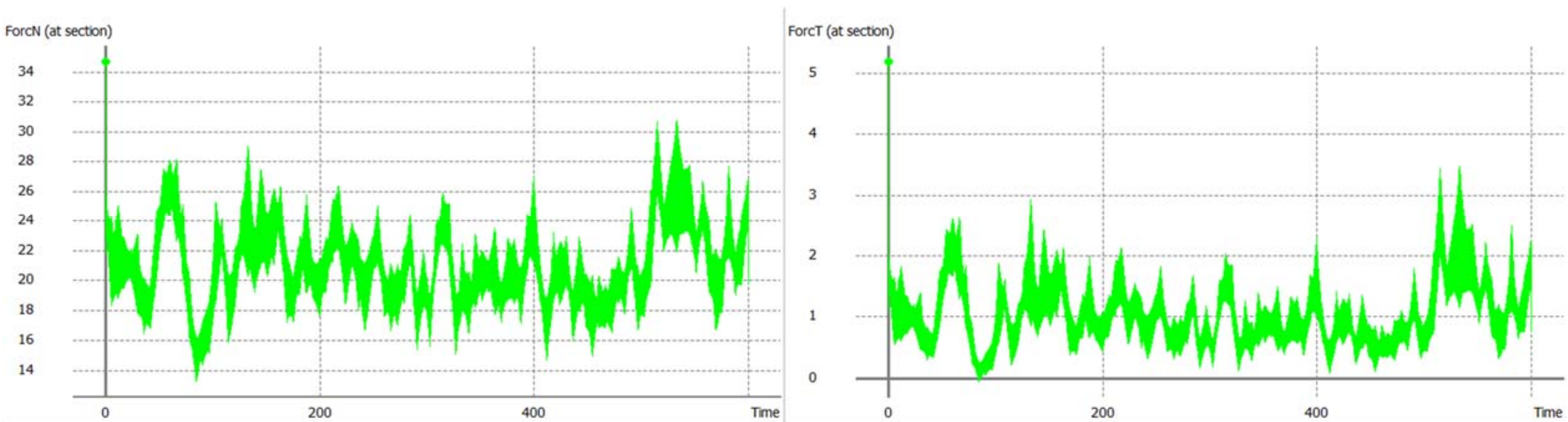


Figure 3 Normal and Tangential loading cycles

These readings are a bit noisy (i.e., just like how a real sensor would behave in its signal readings), also the generated dataset is about 600,000 points (equivalent to 10min). This will be difficult to compute, so firstly the results are filtered using any signal estimation method. For the purpose of this report, the complimentary filter method is used, which is in the following form;

the measurement for the k^{th} time step is,

$$\widetilde{\omega}_k = (1 - s)(\omega_k - b) + s\widetilde{\omega}_{k-1}$$

Where, b is the bias or offset value, that is assumed as, $b = 0$.

ω_k is the direct load reading from FAST analysis.

$\widetilde{\omega}_k$ is the estimated filtered measurement at k^{th} timestep.

$\widetilde{\omega}_{k-1}$ is the previous filtered measurement at $(k-1)^{th}$ timestep.

And s is the filtering variable that can be adjusted depending on how much noise that needs to be filtered out. For this study it is set to, $s = 0.9997$.

Applying the above formular is MS. Excel sheet3. This yields the following results shown in orange colour:

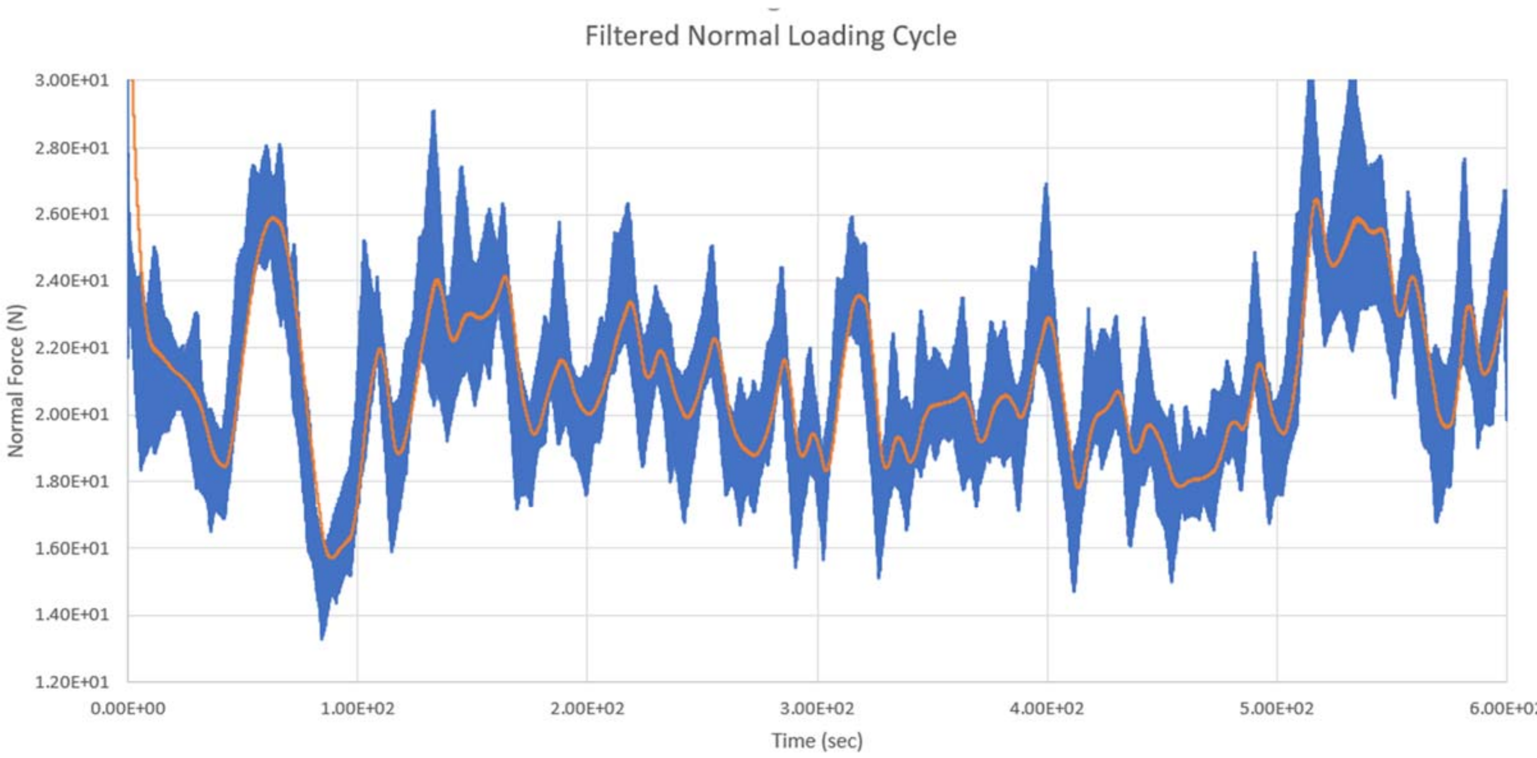


Figure 4 Filtering Normal-force cycle reading

Next, the maximum and minimum loading cycle will be extracted at every 60 seconds from the filtered result (please note that this is not the best practice, but is done for purpose of this paper). Also, the number of grid point is relaxed to just 7 sections (i.e., 16, 14, 12, 10, 8, 6 and 4, shown in fig.1). And the assumption that the circular cross section control volume will contribute negligible aerodynamic load to the system is true.

Observe that from the above plot that max. and min. loading for both normal and tangential force (shown in fig.3) have their frequency in phase, this means that they can be added using the Pythagoreans theorem to get a resultant force vector, responsible for the flexural loading;

$$F_r = \sqrt{(F_r{}^2 + F_r{}^2)}$$

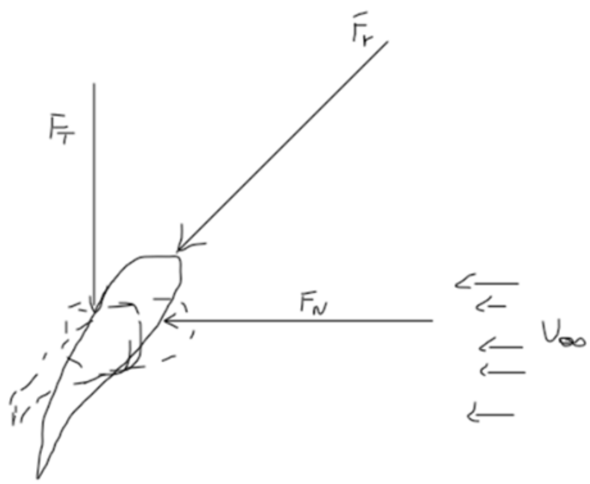
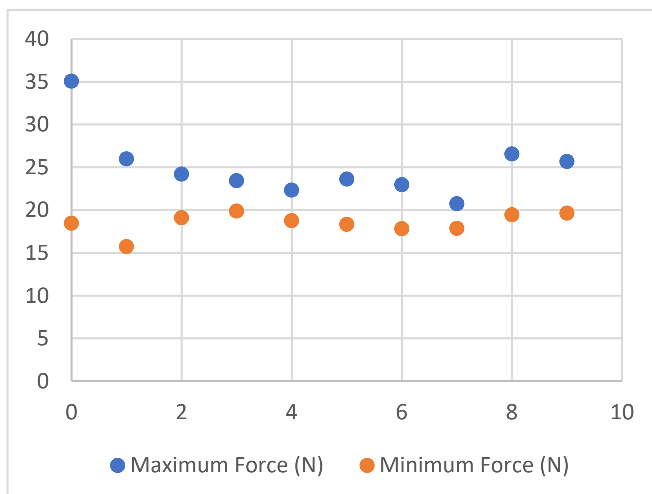


Figure 5 Freebody Diagram of the Resultant flexural Loading

Using Excel, a sample of the maximum and minimum force obtained at the 16th section of the blade is shown below as,



b. Calculating the Mid-range and Alternating Stress Components

These resultant max./min. forces can then be used to calculate the total extreme cases of the flexural bending stress on the blade structure at any given load cycle, using the following formular into the Excel sheet3;

$$\sigma = \sum_s \frac{F_s L_s C_q}{I_s}$$

An important assumption taken for the above formular is that, for each section the forces are all in same frequency and phase, but are of different amplitude. So that the sum of stresses in all the section can be calculated at a particular cycle/time.

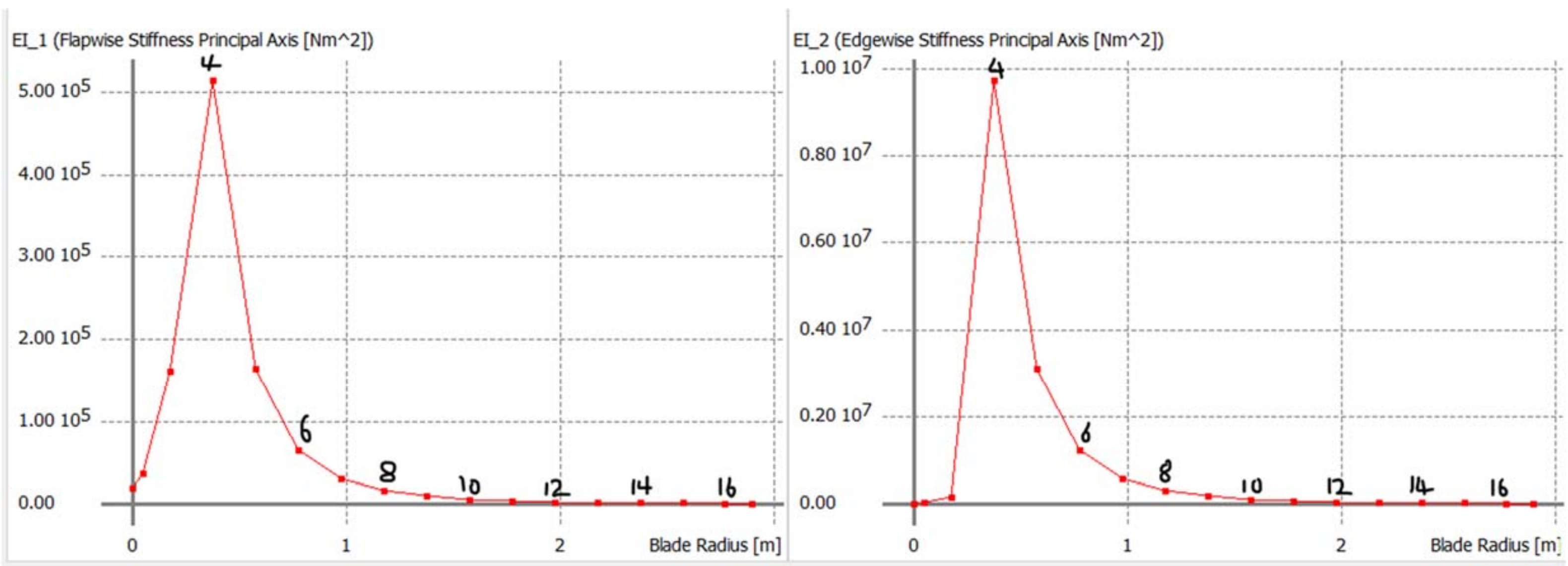
Where, s is the blade section, i.e., $s = 4, 6, 10, 12, 14$ & 16

L_s is the length(radius) from the root of the blade to a given nth section

C_q is the Quarter Chord length ($C_l/4$) of a given nth section airfoil.

F_s is the max./min. force at each given section.

I_s is the average moment of inertial at each section, that can be calculated from the stiffness (EI) obtained from the Qblade FEM analysis, as shown below;



So that $I_s = \frac{EI_1+EI_2}{2E}$, using this formular and $E = 70e^9\ Pa$, in the Excel sheet1, thus yields the following results, at each section; and σ_{max} & σ_{min} at each load cycle calculated in the excel sheet3; all shown in the table below:

Section	L_s (m)	C_q (m)	I_s (m^4)	Load Cycle	Max. Stress (Pa)	Min. Stress (Pa)
4	0.375	0.122362	7.31E-05	1	55819892	30883348.5
6	0.775	0.072868	9.2E-06	2	46355766.9	24592929.5
8	1.175	0.051171	2.24E-06	3	43280475.7	32531811.4
10	1.575	0.039299	7.78E-07	4	41704373.1	34225866.8

12	1.975	0.031861	3.36E-07
14	2.375	0.026776	1.68E-07
16	2.775	0.023085	9.26E-08

5	39760515	31830376.1
6	41868823.3	30477196
7	40632240.4	28948832.4
8	36266824.1	29732841.5
9	47326405	33082801.3
10	45971848.7	33511759.9

Now the midrange and alternating (amplitude) stress component can be calculated using,

$$\sigma_m = \frac{\sum_{cycle}(\sigma_{max}+\sigma_{min})}{2N_{cycle}} \text{ and } \sigma_a = \left|\frac{\sum_{cycle}(\sigma_{max}-\sigma_{min})}{2N_{cycle}}\right|$$

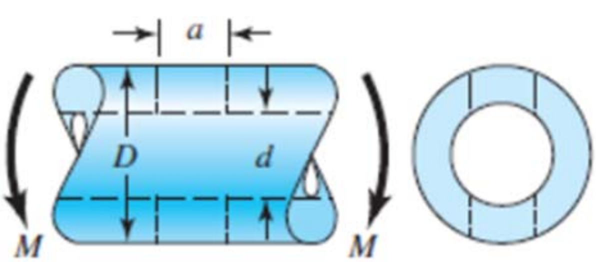
Where, $N_{cycle} = 10$, which is the number of load cycles being considered,

And thus gives the following values,

$$\sigma_m = 37.44024637\ Mpa \text{ and } \sigma_a = 6.458470036\ Mpa$$

c. Numerical Application of the ASME Elliptic Method for Fatigue Factor of Safety Estimation:

Next, the bending stress concentration factor, K_f should be estimated, using Table A-16 from "Shigley's Mechanical Engineering Design" Book, pg.1041.



An assumption that the blade root has circular hollow section, which should be thick enough to withstand the static/dynamic loads is true, so that,

$$\frac{d}{D} = 0.6$$

And that $D = 0.083m$ & $a = 0.05m$, which are obtained from the Qblade geometry. Using the above diameter ration and $\frac{a}{D} = 0.6$, but using a worst case of $\frac{a}{D} = 0.75$, which is the next available option. This gives the corresponding $K_f = 2.43$ from the aforementioned table.

Using the formular from the same Shigley's Book, pg326 (eq. 6-55 & 6-56), the mid-range and alternating von-mises stress can be calculated. And recall that only the flexural loading is being considered, and thus ignoring any tortional and axial loading, for the purpose of this paper. So that the formular for this case becomes,

$$\sigma'_m = \left(K_f\right)_{bending}(\sigma_m)_{bending}$$

$$\sigma'_a = \left(K_f\right)_{bending}(\sigma_a)_{bending}$$

And yields the estimated von-mises stress as,

$$\sigma'_m = \ 90.97979868\ Mpa\ \ \&\ \ \sigma'_a = 15.69408219\ Mpa$$

It is known that aluminum will fail at some point in time (i.e., it doesn't have infinite life). But for the purpose of the following estimations, assume that this aluminum has some fatigue limit S_e, which will be it fatigue strength, after the calculation. So that the fully corrected fatigue limit for this case study can be estimated using the marine equation as,

$$S_e = K_a K_b S'_e$$

This is calculated assuming hot-rolled surface finish for the aluminum structure, since it is not provided. For surface factor (from eq. (6-20) of the referenced book),$K_a = aS_{ult}^b$, where a & b are gotten from Table 6-2 of Shigley's Book, pg296, as $a = 57.7$ & $b = -0.718$. And the ultimate tensile strength of the material is given as, $S_{ut} = 333\ Mpa$. This gives, $K_a = 0.891375829$. Next the size factor can be calculated with, $k_b = 1.24d^{-0.107}$, from eq. (6-20) of the book; where $d = 0.6D = 49.8mm$. This gives $k_b = 0.816240511$. And because the von-mises is being used, that's why the loading factor, K_c was ignored and any other marine factor that was not given. And lastly the fatigue limit will be estimated as, $S'_e = 0.3S_{ult} = 99.9\ Mpa$ (note that 0.3 instead of 0.5 is used in order to match it with the ($90\ to\ 100\ Mpa$) fatigue strength of a 6000series aluminum,). This results in $S_e = 72.685\ Mpa$, and since the material yield tensile strength is given as, $S_y = 282\ Mpa$. Therefore, the fatigue factor of safety can be calculated using the ASME Elliptic method,

$$n_f = \sqrt{\frac{1}{\left(\frac{\sigma'_a}{S_e}\right)^2 + \left(\frac{\sigma'_m}{S_y}\right)^2}}$$

Solving the above gives, the fatigue factor of safety, $n_f = 2.576$

d. Numerical Application of the ASME Elliptic Method for the Determination of the Number of Cycle before Failure

In order to interpret the fatigue factor of safety value, first the factor of safety guarding against first cycle yield is calculated as, $n_y = \frac{S_y}{\sigma'_a + \sigma'_m} = 2.643571207$.

This means that fatigue failure will happen first (i.e., $n_f < n_y$), this is not good for this case, and yet other forms of loading were not considered. But because $n_f > 1$, it while lie within the safe region. In contest the ASME can be interpreted using the following plot

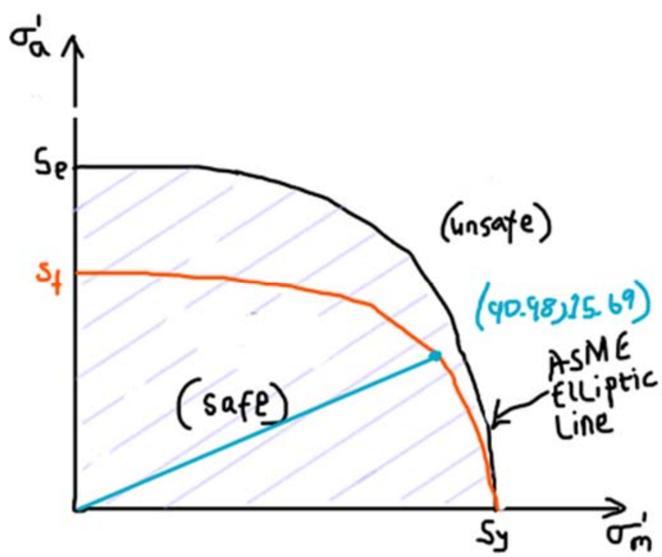


Figure 6 ASME Elliptic

Although the plot seems to suggest that the structure is within the safe region with infinite life. But because this is aluminum and not steel/titanium. Therefore, an equivalent fully reversed load can be calculated using,

$$S_f = \frac{\sigma'_a}{\sqrt{1 - \left(\frac{\sigma'_m}{S_y}\right)^2}}$$

This gives the fully reversed loading (which is the Fatigue stress or the damage impact of fatigue to the blade structure) as, $S_f = 16.581\ Mpa$. This in fact verifies that $S_f < S_e$, and thus is within the safe region. With the fatigue strength equal to the evaluated reverse stress, $S_f = \sigma_{rev}$, it can be deduced that the system should be in the high cycle region. The number of cycles to fatigue failure, N can be calculated using,

$$N = \left(\frac{S_f}{a}\right)^{\frac{1}{b}}$$

Where the constants, $a = \frac{(fS_{ut})^2}{S_e}$ and $b = -\frac{1}{3}\log\left(\frac{fS_{ut}}{S_e}\right)$, with the fraction of S_{ut} as, $f = 0.9$ (This is selected as the worst case just to be conservative for an aluminum material, as shown in fig.6-18 of the referenced book, which is for steel). With these, $a = 1235.7454$ and $b = -0.2051$

Therefore, the number of cycles to failure, $N = 1.348 \times 10^9\ cycles$

5. Conclusion

The fatigue life for an aluminum wind turbine blade is has been determined with the consideration of only flexural loading. The high cycles to failure result obtained is valid, because 6000 series aluminum material has an estimated fatigue strength of about 95 Mpa. But the estimated fatigue stress for the blade structure was just about 16.581 Mpa. This resulted in about one billion cycles before fatigue failure, as numerically proven above. Which essentially translates to about 43 years before fatigue failure.

6. Future Scope

There will definitely be some future work that can be done, especially in regards to improving the robustness of the numeric calculation to accommodate more blade elements, different loading criteria and frequencies. Also, this can be done using some programming languages, such as Fortran, MATLAB, etc., in order to streamline the analytical approach.

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