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Vibration and Fatigue Analysis of Aero-generator Blades

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ABSTRACT: The world is running out of conventional energy sources and there is a pressing need of utilizing non-traditional energy sources to endure the ever escalating energy needs. Wind turbines provide an alternative way of generating energy from the power of wind. At windy places, wind speeds can achieve scintillating values of 10-12 m/s. Such high speeds of wind can be utilized to harness energy by installing a wind turbine usually having 3 blades. The geometry of the blades is made as such that it generates lift from the wind and thus rotates. The lift force generates a moment around the hub and thus the combined torque effort of 3 blades rotates the turbine and generates electricity. Rotational speed of the blades is usually 6 times that of wind speed. In this project, validation of a beam (a geometrical approximation of a blade) in vibration analysis is taken up first. The natural frequencies are matched with a published research paper and then an actual blade geometry is taken up to validate its 1st and 3rd natural frequencies with a published research paper and then a CFD analysis is taken up to find the lift and drag forces on the blade and subsequently these forces are used to calculate the fatigue life of the blade. Suitable materials for different parts of the blade are taken to see which combination of materials gives better results.

KEYWORDS: Wind turbine blade, Dynamic analysis, FEA software, Stresses & Strains, Total deformation, CFD, Fatigue Loading.

I. INTRODUCTION

A lot of literature in the form of research papers, journals, patents etc. is available where one can find that substantial work, theoretical and computational, has been done on the dynamic vibration analysis of beams [1-3]. But fatigue analysis of a proper bladed structure is a very rare occurrence and only a handful of literature is available on it. Vibration analysis is a very useful technique when it comes to analyzing the dynamics of a structure. It provides data about the natural frequencies, mode shapes and frequency response function of vibration. The natural frequencies are critical to a given structure. If a force with a similar frequency is applied to the structure, it can cause catastrophic failure by increasing the deflections many folds. Not just the frequency, but the mode shape of the structure also has to be replicated by the propagation of the applied force. Only when both, natural frequency and mode shape, match to applied frequency and propagation of force, then true resonance occurs and the material undergoes severe failure. Thus the structure must be designed to avoid the occurrence of resonance. And if resonance cannot be avoided, then it should have a facility of damping it away so as to cause minimum damage to the components. Many cases are present where avoiding resonance is not an option, e.g. a propeller blade has to pass through a number of resonant frequencies starting from 0 Hz as it rotates at a very high speed. Thus mechanisms are designed to wither away the effect of passing resonance in such machines components. A comprehensive stress analysis is also a very necessary tool to identify the hotspots in the geometry and carryout further reinforcing or design changes. Fatigue analysis is the most important analysis as far as mechanical devices subjected to varying service loads are concerned. Any component when subjected to repeated loading undergoes a series of molecular deformations adding up to a kind of failure termed as fatigue failure. In brief, it is the gradual ageing of the component due to forces and stresses and other environmental factors. A fatigue analysis is one of the most difficult and challenging analysis there is. It requires a lot of design data and a lot of experimental work to validate the results. A true design report is complete in useful sense only when it can co-relate the vibration characteristics to the applied forces, induced stresses and finally give as output- the service life of the component. Better yet if it can also give a cost analysis and optimization of the parameters to produce an economic design. Although these topics individually provide great insight into specific areas of necessary research work, yet they somehow seem incomplete if viewed separately. Therefore an attempt has been made in this research paper to bring

together these subtopics into a common heading and thus make the design report seem complete from a design point of view.

II. METHODOLOGY

The methodology of this work is divided into five parts namely;

- i. Geometric modelling,
- ii. Vibration analysis,
- iii. CFD analysis,
- iv. Stress based analysis and
- v. Fatigue analysis to predict service life of the component.

Computer Aided Three-dimensional Interactive Application (CATIA), which is a CAD software, is used to model the blade geometry of the wind turbine as specified in [8]. Modal analysis is a vital tool in identifying and eliminating the resonant frequencies that lead to severe fatigue damages in high/low cycle fatigue problems. Computational Fluid Dynamics gives us an idea about the various pressure forces like lift and drag acting on a component present in a fluid flow field. In this work, CFD tool of ANSYS i.e. FLUENT has been used to determine the lift and drag forces, their magnitude and directions. Also the center of pressure has been found out where these pressure forces act naturally. These values give us an idea about the pressure and force distribution on the blade. The output of the CFD analysis is used to conduct a stress analysis in ANSYS workbench to find out the hotspots in the designed Geometry. These stresses are taken up on a repetitive cycle for a fatigue analysis performed in the same workbench module.

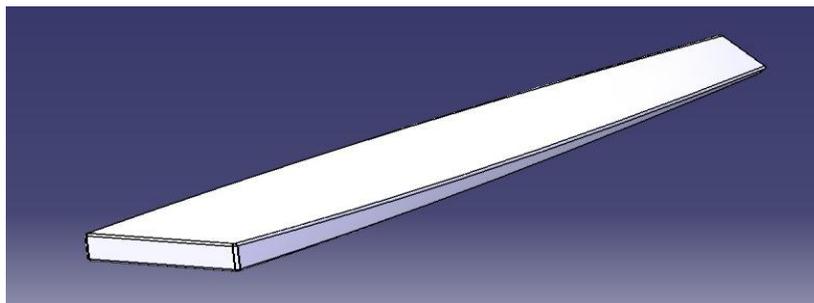


Fig 1: Model of a beam

III. RESULTS AND DISCUSSIONS

The modal analysis of the Timoshenko beam in ANSYS resulted in giving the values of the first 4 natural frequencies of the beam. These values were compared with those obtained from the reference [7]. The mode shapes of the beam are shown in Figures 21- 24 in order of their harmonics. A comparison Table of natural frequency values from present work and from [7] is given in Table no. 7.'

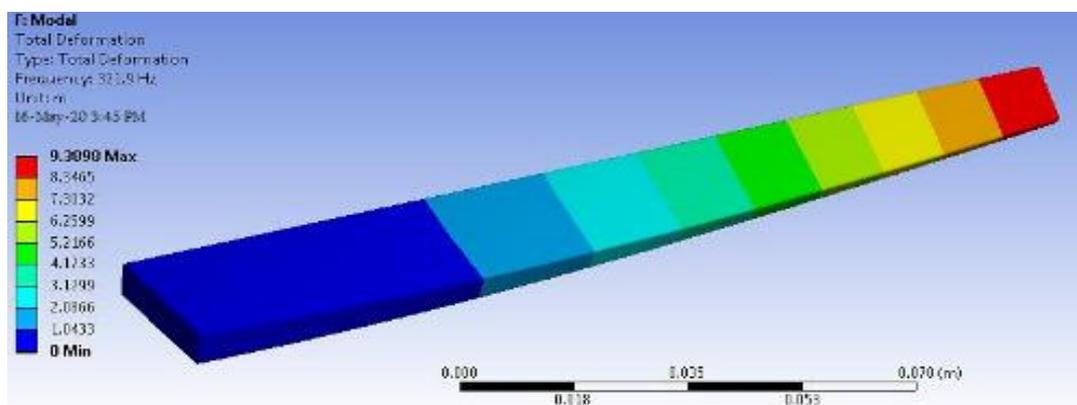


Fig 2: Mode shape of blade corresponding to 1st modal frequency

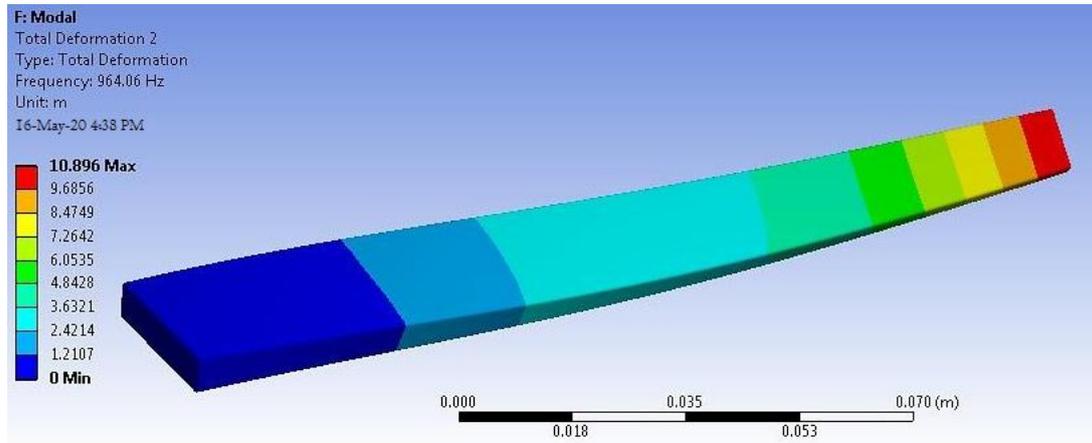


Fig 3: Mode shape of blade corresponding to 2nd modal frequency

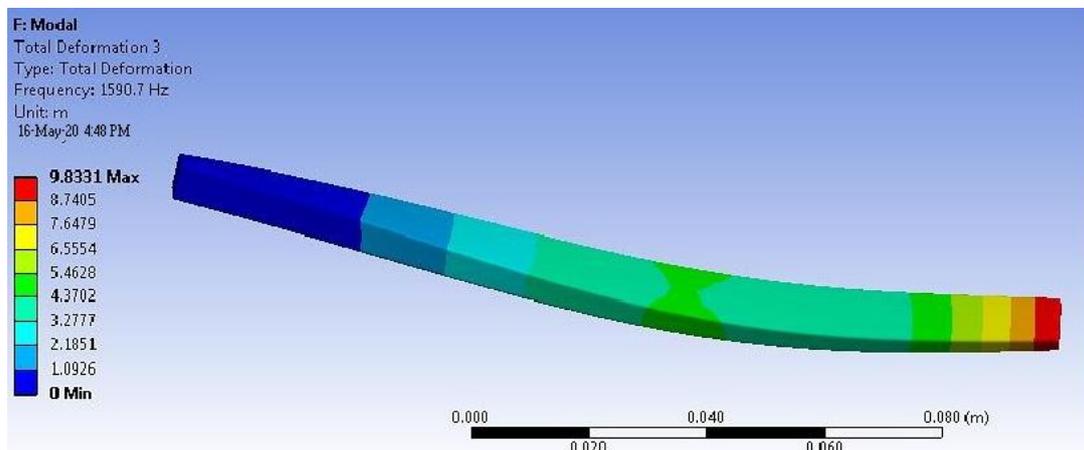


Fig 4: Mode shape of blade corresponding to 3rd modal frequency

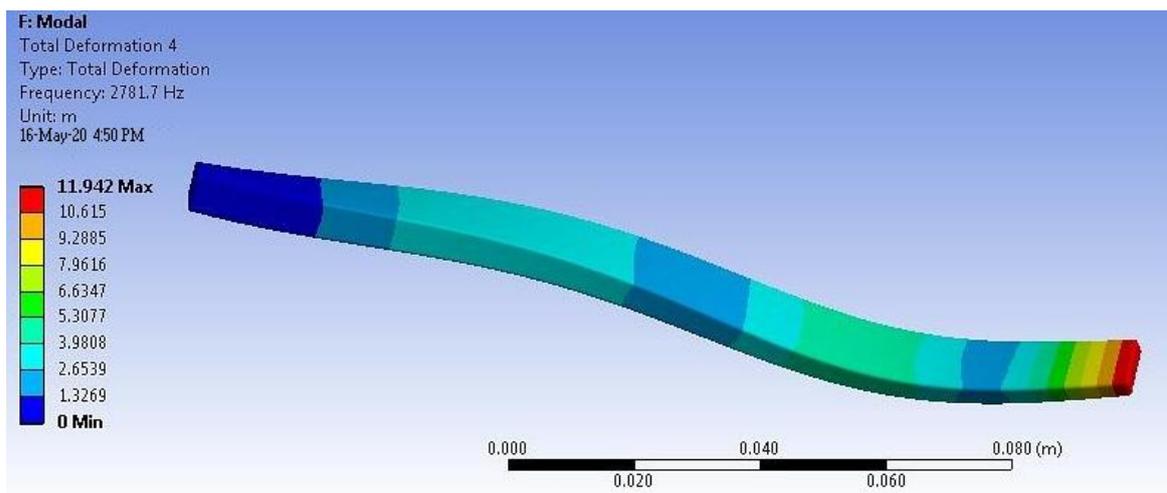


Fig 5: Mode shape of blade corresponding to 4th modal frequency

Table 1: Comparison of beam’s natural frequencies

Mode Shape	Natural frequency from ANSYS	Natural frequency ref. [7]
1	321.9 Hz	297.8 Hz
2	964.06 Hz	1137.3 Hz
3	1590.7 Hz	1645 Hz
4	2781.7 Hz	3578.3 Hz

From the above Table we can infer that obtained primary natural frequency is higher than its corresponding ref. [7] value. The subsequent frequency values are lower than their counterparts in ref. [7]. This is a desirable trend. Computational Fluid Dynamics analysis of the blade model was done in ANSYS FLUENT module. The pressure contours and velocity vector field around the skin of the blade were observed and analyzed.

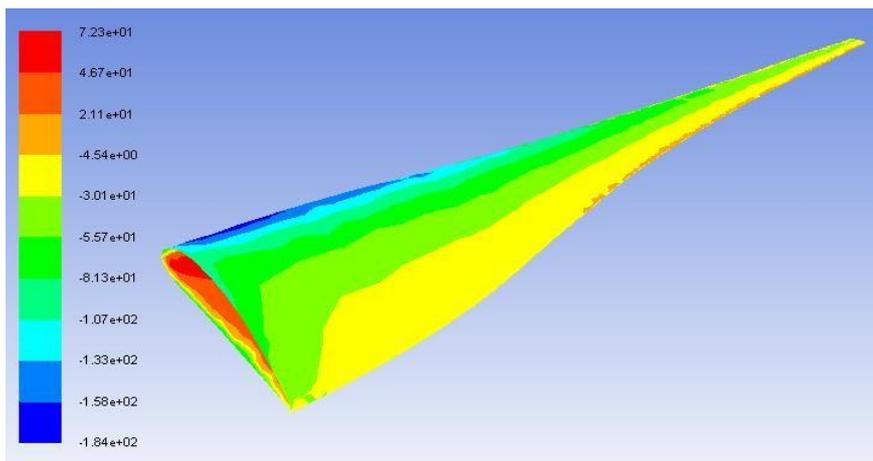


Fig 6: Pressure contour on upper skin of blade

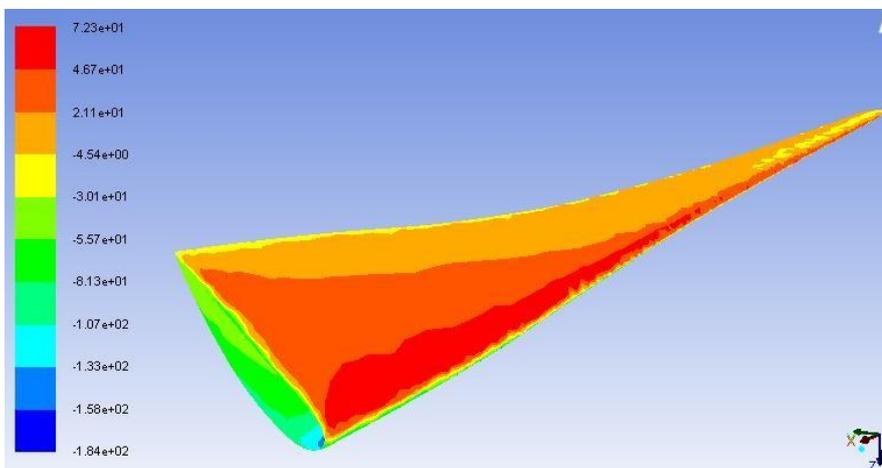


Fig 7: Pressure contour on lower skin of blade

Figure 6 shows the distribution of static pressure over the upper skin of blade surface. It can be observed that the pressure values are on the lower side of the spectrum. Similarly, Figure 7 shows distribution of static pressure on the lower surface of blade skin. Observation shows that pressure on the lower side is relatively higher than pressure on upper side, as expected. This creates a net upward force which produces a lift.

Figure 8 shows velocity vector field over the upper skin surface of the blade. It can be observed that the values of velocity are quite high and lie towards the red side of the spectrum. Figure 9 shows velocity vector field below the lower surface of blade skin. Observations reveal that these values are much lower than those corresponding to upper skin surface. Thus, by Bernoulli's principle, pressure below skin is greater than pressure above skin thereby leading to lifting the airfoil (blade) [15-17].

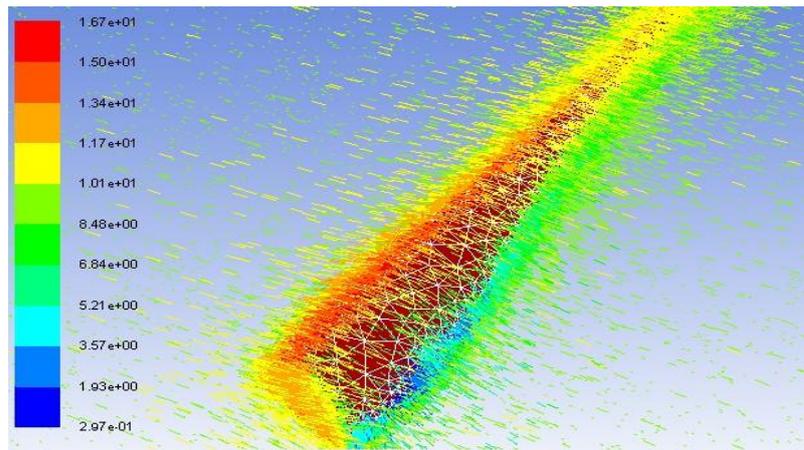


Fig 8: Pressure contour on lower skin of blade

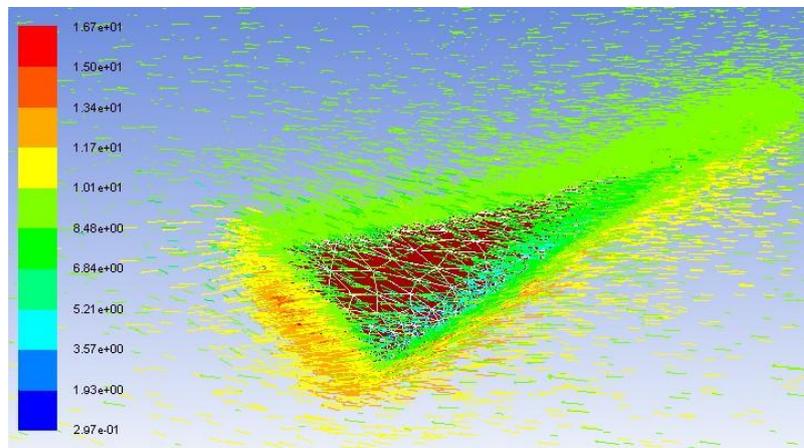


Fig 9: Velocity vector field on lower skin of blade.

Table 2 enlists the results from the fatigue analysis of the actual blade model. It gives results in terms of number of cycles, factor of safety, design life and equivalent alternating stress.

Table 2: Fatigue life prediction analysis of the blade model.

Type	Life	Damage	Safety Factor	Equivalent Alternating Stress
Minimum	1.57e+009 seconds		> 15	307.02 Pa
Minimum Occurs On	skin			skin
Maximum		1.		1.1573e+006 Pa
Maximum Occurs On		skin		skin

This fatigue analysis shows that the material structure of the windmill turbine won't fail due to repeated loading of magnitudes found from the CFO analysis. Thus it has a theoretical lifetime of infinity.

IV. CONCLUSIONS

A doubly tapered and twisted beam was modeled and its natural frequencies were computationally derived. The values of natural frequencies were validated with a published research paper [7] and good agreement was found between the values suggesting that the modeling and analysis of the beam is correct. Then a real world model of a windmill blade was modeled and its natural frequencies were computed computationally using CAE software. These results were validated against the work of Perkins and Cromack [8] from which geometric parameters of the blade model had been referred. Again, the modal frequency values showed good agreement with the reference and the geometric model was validated along with the CAE analysis. A new attempt was made to calculate the lift and drag forces on the blade surface using CFO tools. These force values were used to carry out a stress analysis of the blade model and thus regions of high stress concentration were observed from the CAE results. The deformation along the blade span was also studied. Finally a fatigue life prediction analysis was implemented on the blade model by referring to S-N curves of used materials from various references. The fatigue result showed no failure for the model. Possible reasons could be the small size of the mini-model. Due to smaller span and shorter chord length, sufficient lift and drag forces could not have been generated that would cause failure of the material [18]. Usually windmill blades are 40 m in diameter [5] and over a few tons in mass, while this mini-model is just 24.49kgs in mass. Therefore even the effect of self-weight is almost negligible as far as deflection and induced stresses goes.

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