Fatigue Life And Strength Analysis of A Main Shaft Using Solid Works And CAE Analysis

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Abstract- *High-strength bolts are widely used in wind turbines and play a significant role in their operation. In this paper, in order to study the ultimate strength of high-strength bolts in the connection between the hub and main shaft in wind turbine with pretension effects, two kinds of finite element models are presented: a solid bolt model and a simplified bolt model. In this proposed study the numerically simulation analysis of the fatigue behaviour of stepped shaft made of Carbon steel SAE 1045_390_QT under a biaxial loading (torsion-bending moment). This work will include the complete analysis of stepped shaft by using Ansysworkbench 15*.

Keywords- wind turbine; finite element analysis; high-strength bolts; fatigue life

I. INTRODUCTION

The durability of a machine structure can be defined as the skill of the structure in order to keep up its mechanical performance through its service life. Therefore, there are a close relationship between durability and safety. Structural failure is primarily due to static and fatigue lots. Hence, the machine can be analyzed according to uncommon types of loading which is static and fatigue, in order to design safe and depend-able structures. Many literatures, research on experimental or numerical studies related to machine based structures [1–3].Element treatment is not limited to the engineering field, but furthermore, extends to health check and geospatial an application which is defined as a method of applying statistical analysis to data which has a geographical feature. Rapid advancement in a finite element is due to powerful computer processors and unremitting software development. In con-temporary years the aid of finite element in engineering was enormously increased Key factors in finite element analysis (FEA) are numerical computations with the intention of estimate all parameters and boundaries agreed.

II. COMPUTER AIDED DESIGN SYSTEM

Developing CAD software is an arduous and challenging task. However, it is theback end wherein the core of Computer Aided Design rests. The concepts emerge as anamalgamation of geometry, mathematics and engineering that renders the software the capability of free-form or generic design of a product, its analysis, obtaining its optimized form, if desired, and eventually its manufacture. By late 1960s, the term Computer Aided Design (CAD) was coined in literature. By 1980s and 1990s, CAD/CAM had penetrated virtually every industry including Aerospace, Automotive, Construction, Consumer products, Textiles and others. Software has been developed over the past two decades for interactive drawing and drafting, analysis, visualization and animation. A few widely used products in Computer Aided Design and drafting are Pro-Engineer, AutoCAD, CATIA,IDEAS, Solid Works and in analysis are NASTRAN, ABAQUS, ALGOR and ANSYS. Many of these software's are being planned to be upgraded for potential integration of design, analysis, and optimization.

III. FINITE ELEMENT METHOD

The Finite Element Method is essentially a product of electronic digital computer age. Though the approach shares many features common to the numerical approximations, it possesses some advantages with the special facilities offered by the high speed computers. In particular, the method can be systematically programmed to accommodate such complex and difficult problems as non homogeneous materials, non linear stress-strain behaviour and complicated boundary conditions. It is difficult to accommodate these difficulties in the Least Square Method or Ritz Method etc. an advantage of Finite Element Method is the variety of levels at which we may develop an understanding of technique. The Finite Element Method is applicable to wide range of boundary value problems in engineering. In a boundary value problem, a solution is sought in the region of body, while the boundaries (or edges) of the region the values of the dependant variables (or their derivatives) are prescribed.

IV. INTRODUCTION TO THE SOLIDWORKS

SolidWorks Corporation was founded in December 1993 by Jon Hirschtick with headquarters in Waltham, Massachusetts, USA, who recruited a team of engineers to build a company that developed 3D CAD software that was easy-to-use, affordable and available on the Windows desktop, with its headquarters at Concord, Massachusetts, and released its first product, Solid Works 95, in 1995. SolidWorks currently markets several versions of the Solid Works CAD software in addition to eDrawings, a collaboration tool, and Draft Sight, a 2D CAD product. In 1997 Dassault Systems, best known for its CATIA CAD software, acquired the company and currently owns 100% of its shares. SolidWorks was headed by John McEleney from 2001 to July 2007 and Jeff Ray from 2007 to January 2011. The current CEO is Bertrand Sicot.

Used in Many Industries: Such as Automotive & Aerospace, Consumer Goods, Electronics, Machinery, Medical, Mould, Tool & Die and Telecom etc.

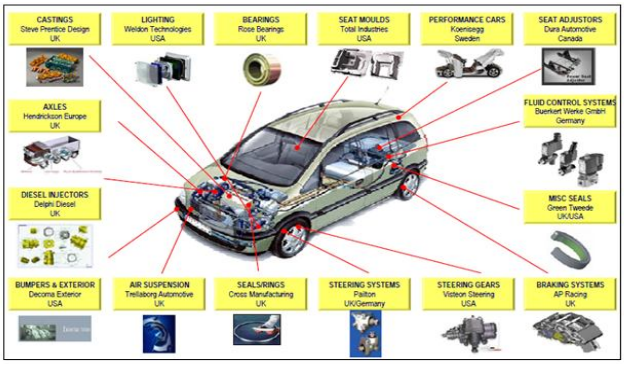

Figure 1: SolidWorks Everywhere in Automotive

V. DESIGN CONSIDERATIONS FOR SHAFT

For the design of shaft following two methods are adopted,

Design based on Strength

In this method, design is carried out so that stress at any location of the shaft should not exceed the material yield stress. However, no consideration for shaft deflection and shaft twist is included.

Design based on Stiffness

Basic idea of design in such case depends on the allowable deflection and twist of the shaft.

1.5.1 Design based on Strength

The stress at any point on the shaft depends on the nature of load acting on it. The stresses which may be present are as follows.

Basic stress equations:

Bending stress

$$\sigma_b = \frac{32M}{\pi d_0^3(1-k^4)}$$

Where,

M: Bending moment at the point of interest
d : Outer diameter of the shaft
k: Ratio of inner to outer diameters of the shaft (k = 0 for a solid shaft because inner diameter is zero)

Axial Stress

$$\sigma_a = \frac{4\alpha F}{\pi d_0^2(1-k^2)}$$

Where,
F: Axial force (tensile or compressive)
α: Column-action factor(= 1.0 for tensile load)

The term α has been introduced in the equation. This is known as column action factor. This arises due the phenomenon of buckling of long slender members which are acted upon by axial compressive loads.
Here, α is defined as

$$\alpha = \frac{1}{1-0.0044(L/K)} \qquad \text{for } L/K < 115$$

$$\alpha = \frac{\sigma_{yc}}{\pi^2 nE}\left(\frac{L}{K}\right)^2 \qquad \text{for } L/K > 115$$

Where,
n = 1.0 for hinged end
n = 2.25 for fixed end
n = 1.6 for ends partly restrained, as in bearing
K = least radius of gyration,
L = shaft length
σ_{yc} = yield stress in compression
Stress due to torsion

$$\tau_{xy} = \frac{16T}{\pi d_0^3(1-k^4)}$$

Where,
T: Torque on the shaft
τ_{xy}: Shear stress due to torsion

Combined Bending and Axial stress

Both bending and axial stresses are normal stresses, hence the net normal stress is given by

$$\sigma_x = [\frac{32M}{\pi d_0^3(1-k^4)} \pm \frac{4\alpha F}{\pi d_0^2(1-k^2)}]$$

The net normal stress can be either positive or negative. Normally, shear stress due to torsion is only considered in a shaft and shear stress due to load on the shaft is neglected.

Maximum shear stress theory

Design of the shaft mostly uses maximum shear stress theory. It states that a machine member fails when the maximum shear stress at a point exceeds the maximum allowable shear stress for the shaft material. Therefore,

$$\tau_{max} = \tau_{allowable} = \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + \tau_{xy}^2}$$

Substituting the values of σ_x and τ_{xy} in the above equation, the final form is,

$$\tau_{allowable} = \frac{16}{\pi d_0^3(1-k^4)}\sqrt{\left\{M + \frac{\alpha F d_0(1+k^2)}{8}\right\}^2 + T^2}$$

Therefore, the shaft diameter can be calculated in terms of external loads and material properties. However, the above equation is further standardized for steel shafting in terms of allowable design stress and load factors in ASME design code for shaft.

VI. METHODOLOGY

The shaft is a principal component part that transmits circular motion from the pulley derived from the motor to thehammer mill of the machine, which may lead to the cracking of the hammer mill and the nut. The shaft diameter ofan existing palm kernel cracking machine was measured as30 mm using an electronic digital calliper. The finite element computation will be performed using CATIA. Thefinite-element meshes of these models will be generated in ANSYS 15.

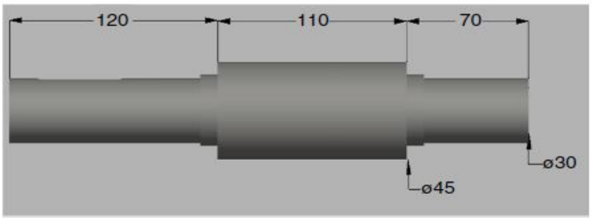

Fig.2 Design of shaft in 2D with the dimension in millimeter

VII. RESULT

SHAFT SHEAR STRESSES

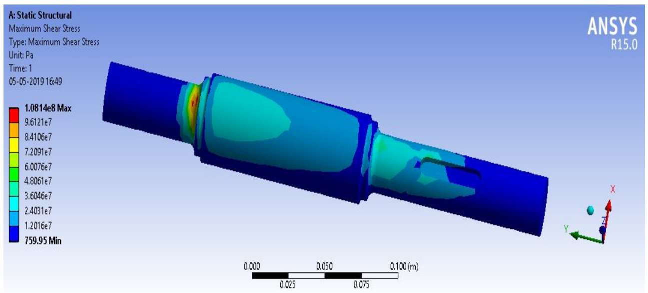

Fig. 3 Results analysis of 30 mm diameter of shaft: Maximum shear stress at torque of 72,000 Nm

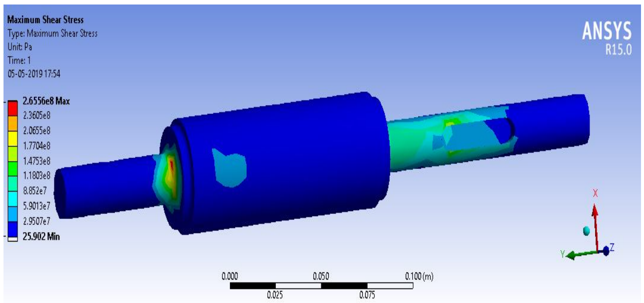

Fig.4 Results analysis of 20 mm diameter of shaft: Maximum shear stress at torque of 72,000 Nm

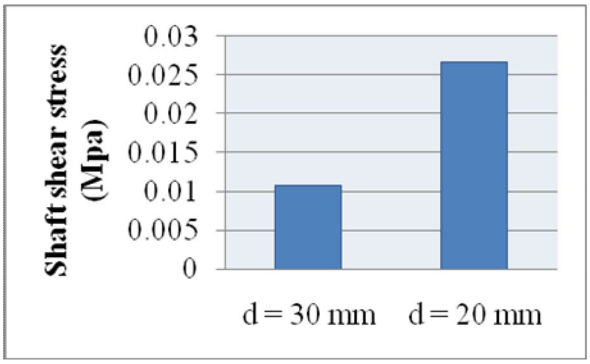

Fig.5 Comparison of Shaft shear stress at different diameter

EQUIVALENT VON-MISES STRESS

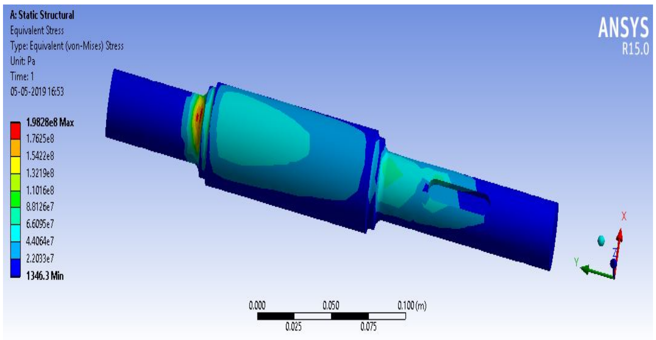

Fig.6 Results analysis of 30 mm diameter of shaft: Equivalent (Von-Mises) stress at torque of 72,000 Nm

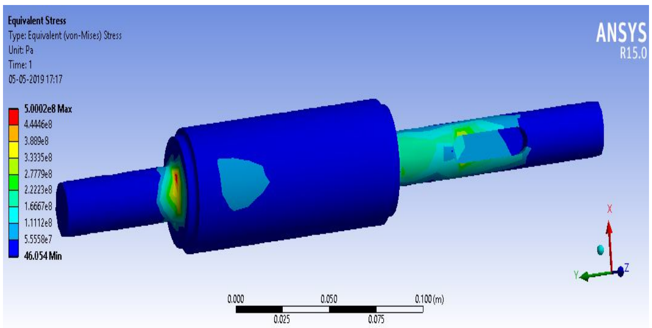

Fig.7 Results analysis of 20 mm diameter of shaft: Equivalent (Von-Mises) stress at torque of 72,000 Nm

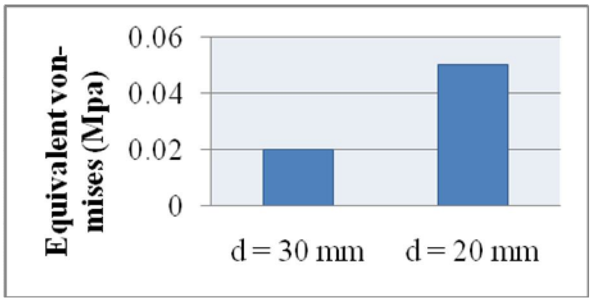

Fig.8 Comparison of Equivalent Von-Mises stress at different diameter

EQUIVALENT STRAIN

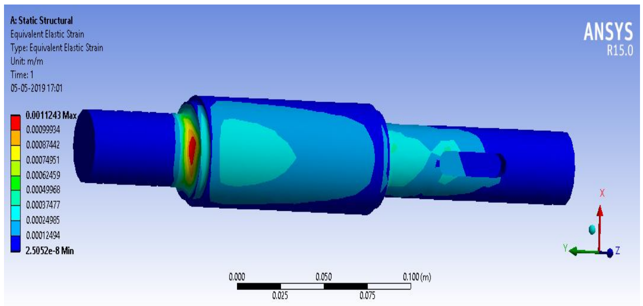

Fig.9 Results analysis of 30 mm diameter of shaft: Equivalent Elastic Strain at torque of 72,000 Nm

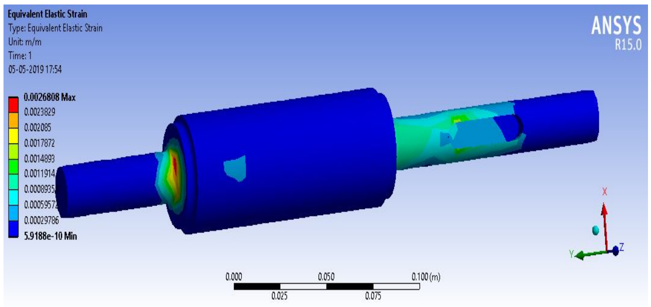

Fig.10. Results analysis of 20 mm diameter of shaft: Equivalent Elastic Strain at torque of 72,000

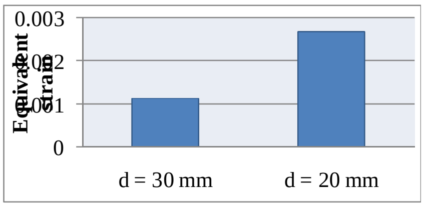

Fig.11 Comparison of Equivalent strain at different diameter

FATIGUE DAMAGE

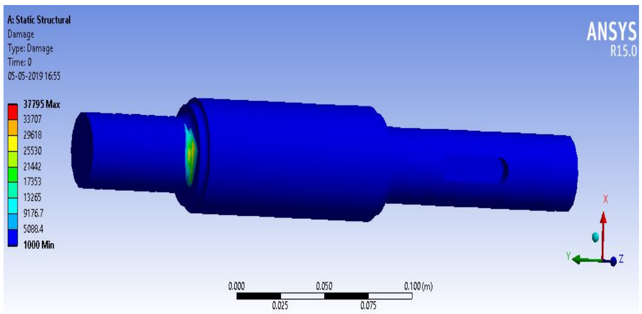

Fig.12. Results analysis of 30 mm diameter of shaft: Fatigue Damage at torque of 72,000 Nm

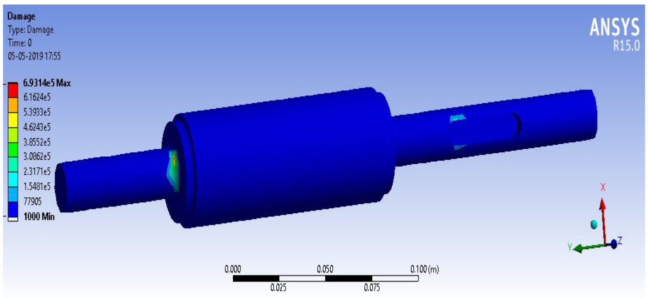

Fig.13Results analysis of 30 mm diameter of shaft: Fatigue Damage at torque of 72,000 Nm

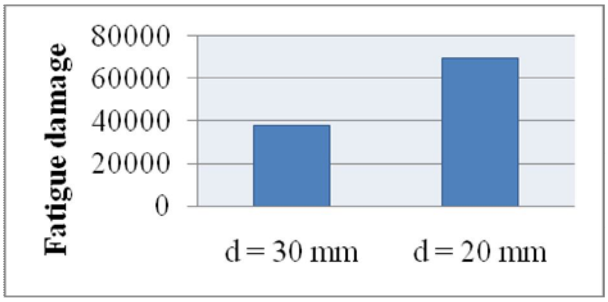

Fig. 14: Comparison of Fatigue damage at different diameter

FACTOR OF SAFETY OF FATIGUE LIFE

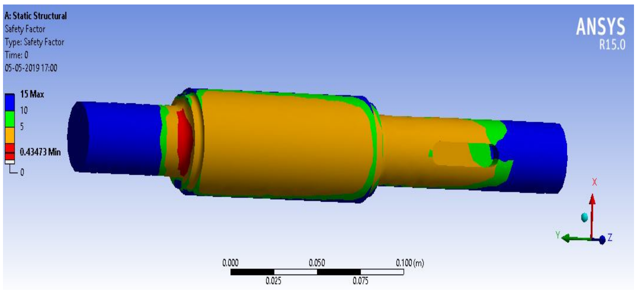

Fig.15. Results analysis of 30 mm diameter of shaft: Factor of safety at torque of 72,000 Nm

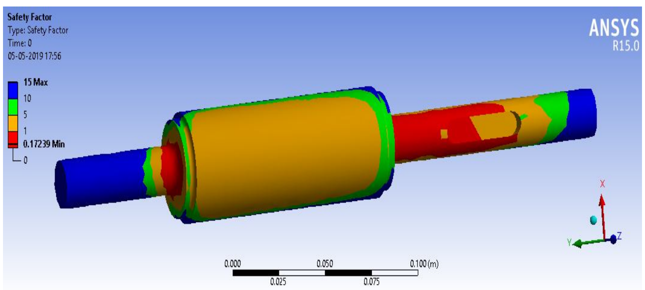

Fig.16 Results analysis of 20 mm diameter of shaft: Factor of safety at torque of 72,000 Nm

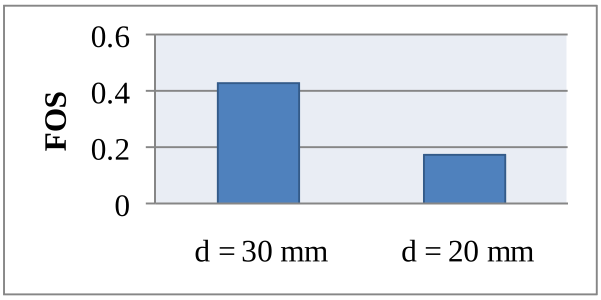

Fig.17 Comparison of FOS at different diameter

TOTAL DEFORMATION

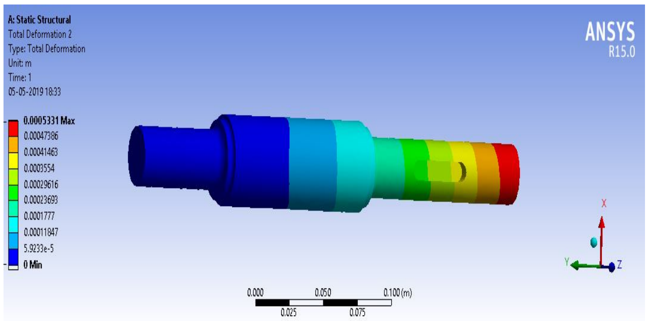

Fig.18 Results analysis of 30 mm diameter of shaft: Total Deformation at torque of 72,000 Nm

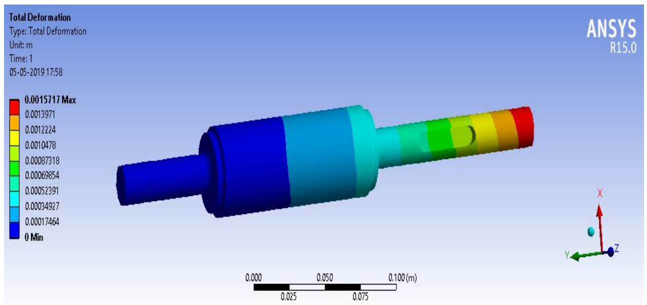

Fig.19 Results analysis of 20 mm diameter of shaft: Total Deformation at torque of 72,000 Nm

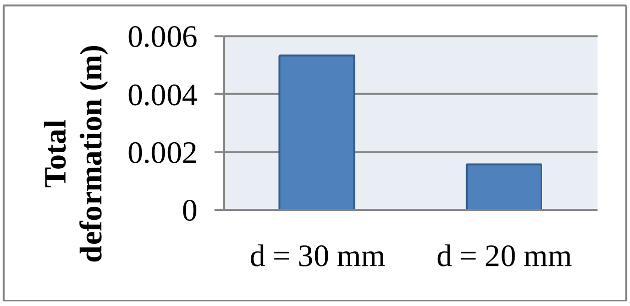

Fig. 20 Comparison of Total deformation at different diameter

VIII. CONCLUSION

Computer Aided engineering results obtained from this Analysis and many discussions have been concluded as given below:

1. The 3D design and analysis of the shaft of 30 mm diameter and 20 mm diameter palm kernel-cracking machine have been analyzed and simulated.
2. The result for fatigue damage for 20 mm diameter is 69314 and for a 30 mm diameter is 37795, which shows that diameter 30 mm shaft is better than 20 mm diameter.
3. The result for a factor of safety for 20 mm diameter is 0.17 and for a factor of safety for 30 mm diameter is 0.43, which indicates the shaft can carry two times the load.
4. The result for a total deformation for 20 mm diameter is 0.00157 and total deformation for 30 mm diameter is 0.00533, which indicates the shaft can carry two times the load.
5. The working conditions of a machine shaft structure were investigated. The FEA was carried out taking into account linear constitutive relations of a plastic deformation and residual stresses to determine the stress and strain states, Von Mises stress distribution of the machine shaft structure under static loads.

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