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Fem Analysis of A Fatigue Life Prediction for Safe and Economical Machine Shaft

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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 behavior 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 Ansys workbench 15.**

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

I. INTRODUCTION

The finite element method (FEM), the solution region is regarded to be composed of many small, interconnected sub regions called the finite elements. Within each element, a feasible displacement interpolation function is assumed. Strain and stress computations at any point in that element are then performed following which the stiffness properties of the element are derived using elasticity theories.

Element stiffness's are then assembled to represent the stiffness of the entire solution region. Between solid modeling and the finite element analysis lays an important intermediate step of mesh generation. Mesh generation as a pre-processing step to FEM involves discretization of a solid model into a set of points called nodes on which the numerical solution is to be based. Finite elements are then formed by combining the nodes in a predetermined topology (linear, triangular, quadrilateral, tetrahedral or hexahedral).

Discretization is an essential step to help the finite element method solve the governing differential equations by approximating the solution within each finite element. The process is purely based on the geometry of the component and usually does not require the knowledge of the differential equations for which the solution is sought. The accuracy of an FEM solution depends on the fineness of discretization in that for a finer mesh, the solution accuracy will be better, that is, for the average finite element size approaching zero, the finite element solution approaches the classical (or analytical) solution, if it exists. We would always desire to seek the 'near to classical' solution. However, the extent of computational effort involved poses a limit on the number of finite elements (and thus their average size) to be employed. A relatively small number of finite elements in a coarse mesh would yield a solution at a much faster rate, though it will be less accurate compared to that obtained using a large number of elements in a fine mesh.

II. METHODOLOGY

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

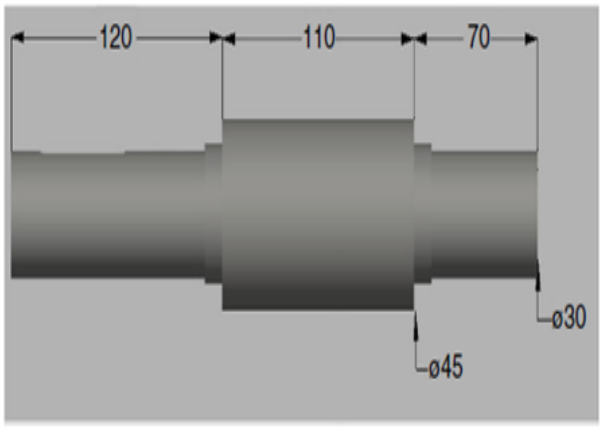

Fig.1. Design of shaft in 2D with the dimension in millimeter.

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1. Proposed Work

The fatigue analysis will be applied to uniaxial and biaxial loading conditions for geometrically stepped shaft of diameter 30 mm and 20 mm with notch radius 5 mm. The present work will study the numerically simulation analysis of the fatigue behavior 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 Ansys workbench 15.

2. Shaft Shear Stresses

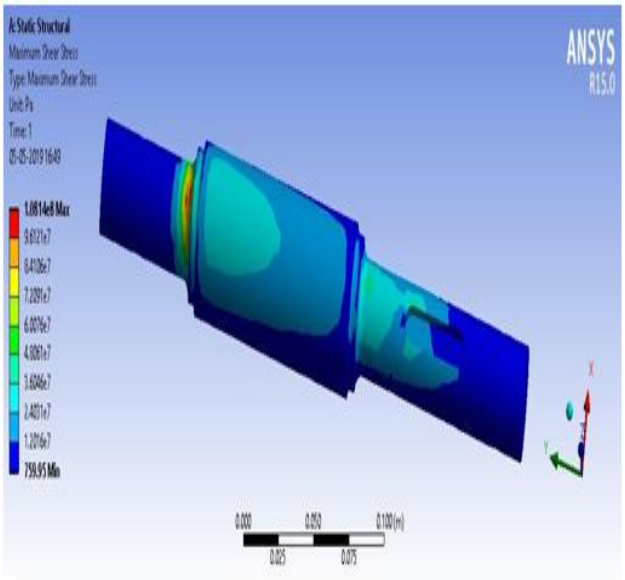

Fig.2. Results analysis of 30 mm diameter of shaft: Maximum shear stress at torque of 72,000 Nm.

Fig.2 shows the analysis of 30 mm diameter of shaft. This analysis shows the maximum value of maximum shear stress is 1.0814e8 and minimum value of maximum shear stress is 759.95.

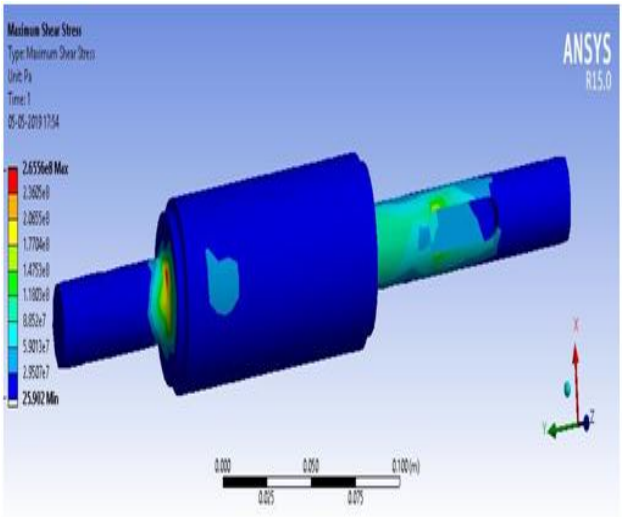

Fig.3.Results analysis of 20 mm diameter of shaft: Maximum shear stress at torque of 72,000 Nm.

Fig.4.2 shows the analysis of 20 mm diameter of shaft. This analysis shows the maximum value of maximum shear stress is 2.6556e8 and minimum value of maximum shear stress is 25.902.

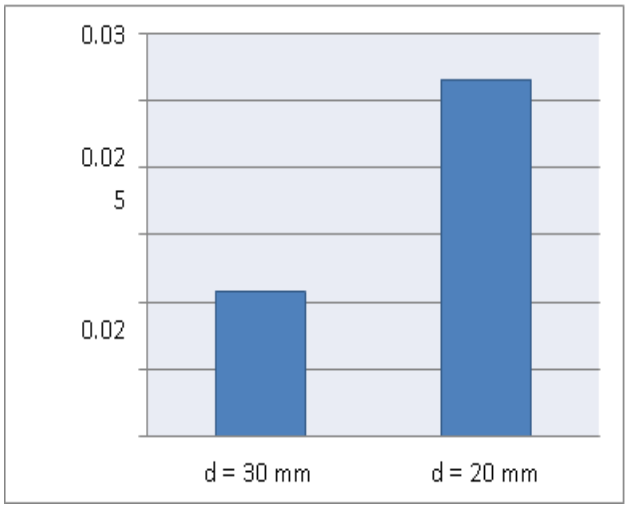

Fig.4. Comparison of Shaft shear stress at different diameter.

Fig.4 shows the comparison graph of shaft shear stress at different diameter. In this graph shows the 20mm diameter shaft shear stress is maximum.

3. Equivalent Von-Misses Stress

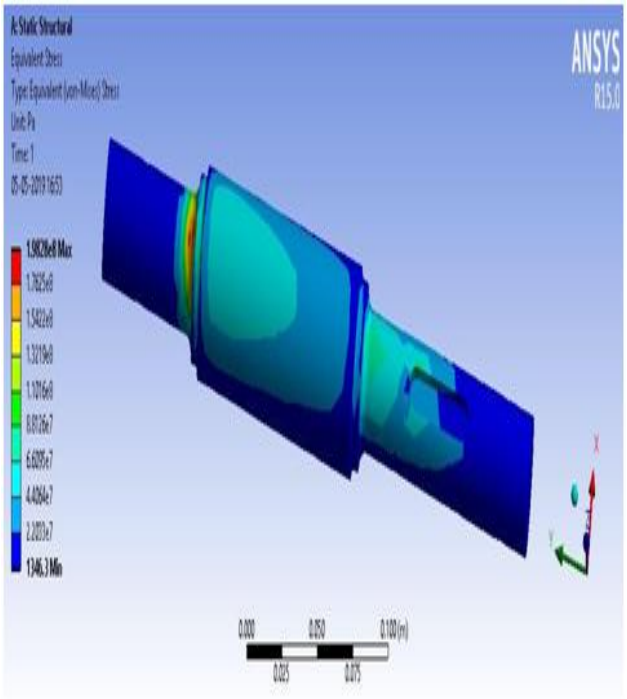

Fig.5. Results analysis of 30 mm diameter of shaft: Equivalent (Von-Mises) stress at torque of 72,000 Nm.

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Fig.5 shows the analysis of 30 mm diameter of shaft. This analysis shows the maximum value of Equivalent (Von-Mises) stress is 1.9828e8 and minimum value of Equivalent (Von-Mises) stress is 1346.3.

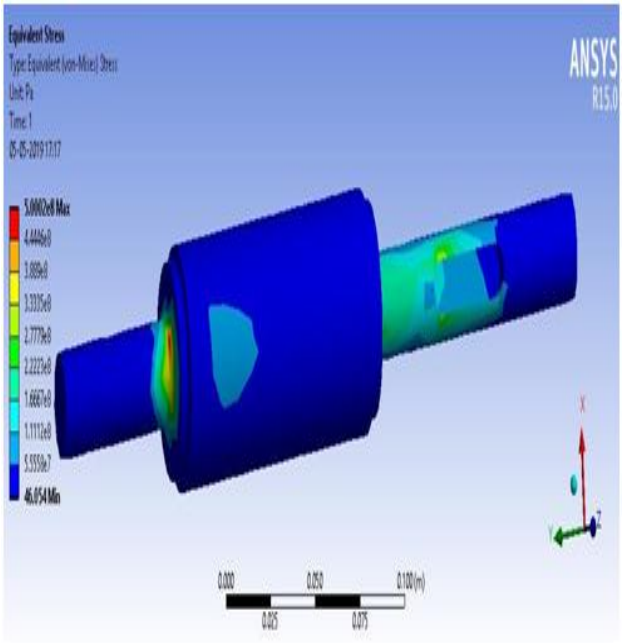

Fig.6. Results analysis of 20 mm diameter of shaft: Equivalent (Von-Mises) stress at torque of 72,000 Nm.

Fig.6 shows the analysis of 20 mm diameter of shaft. This analysis shows the maximum value of Equivalent (Von-Mises) stress is 5.0002e8 and minimum value of Equivalent (Von-Mises) stress is 46.054.

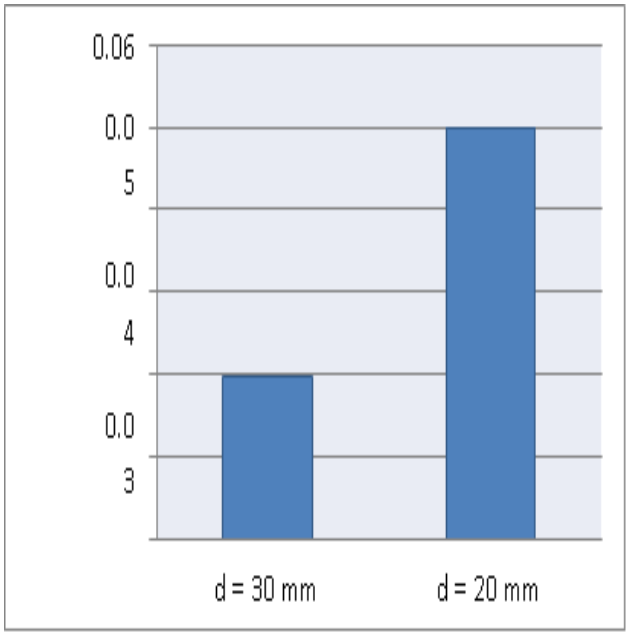

Fig.7. Comparison of Equivalent Von-Mises stress at different diameter.

4. Equivalent Strain

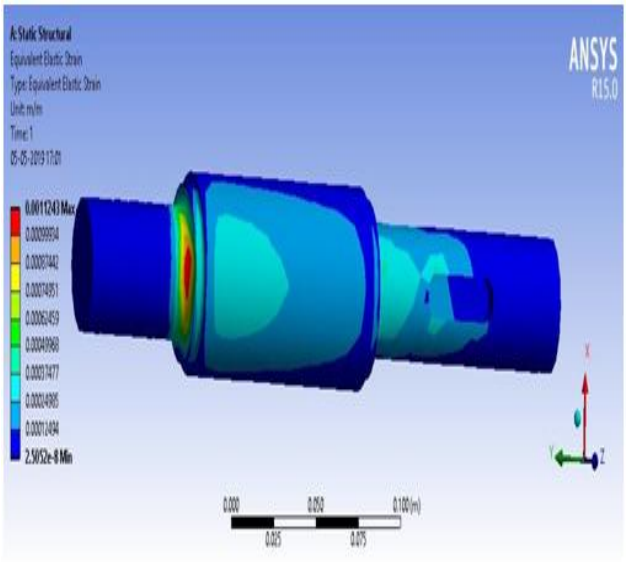

Fig.8. Results analysis of 30 mm diameter of shaft: Equivalent Elastic Strain at torque of 72,000 Nm.

Fig.8 shows the analysis of 30 mm diameter of shaft. This analysis shows the maximum value of Equivalent Elastic Strain is 0.0011243 and minimum value of Equivalent Elastic Strain is 2.5052e8.

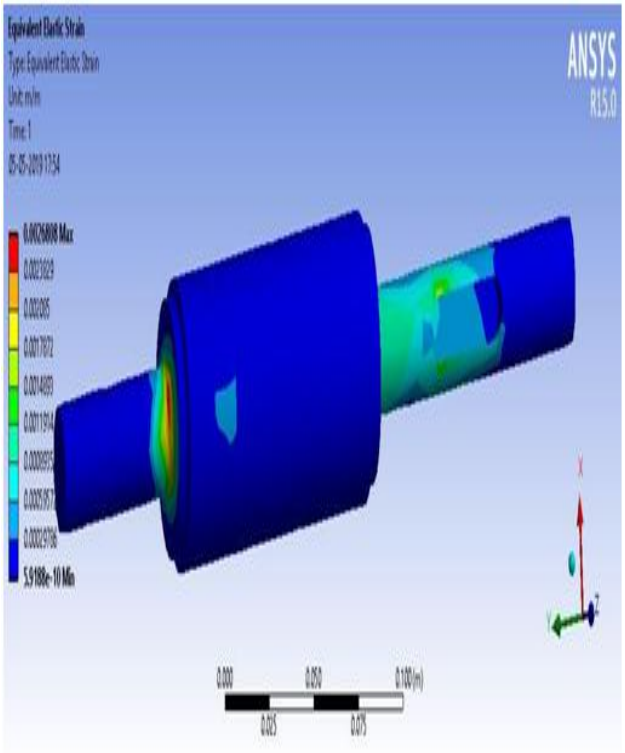

Fig.9. Results analysis of 20 mm diameter of shaft: Equivalent Elastic Strain at torque of 72,000 Nm.

Fig.9shows the analysis of 20 mm diameter of shaft. This analysis shows the maximum value of Equivalent Elastic

Strain is 0.002688 and minimum value of Equivalent Elastic Strain is 5.9188e-10.

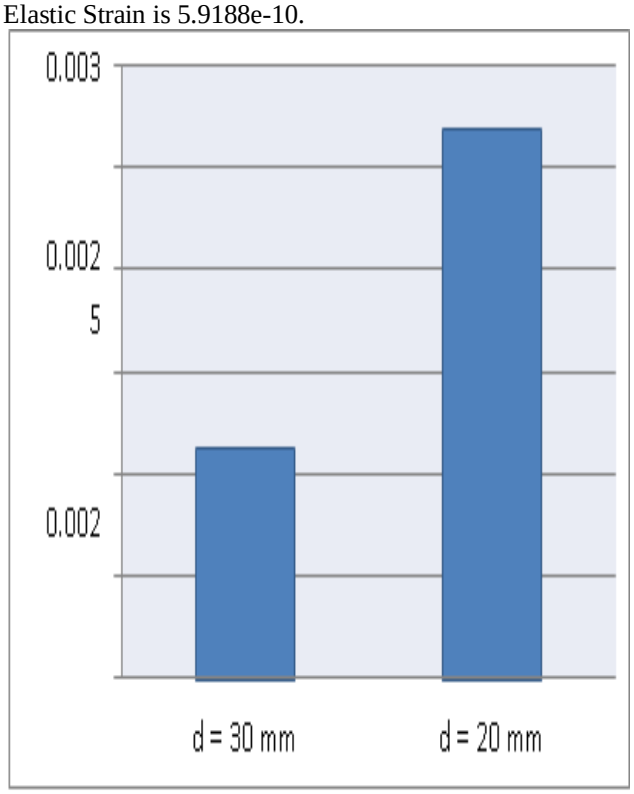

Fig.10. Comparison of Equivalent strain at different diameter.

5. Fatigue Damage

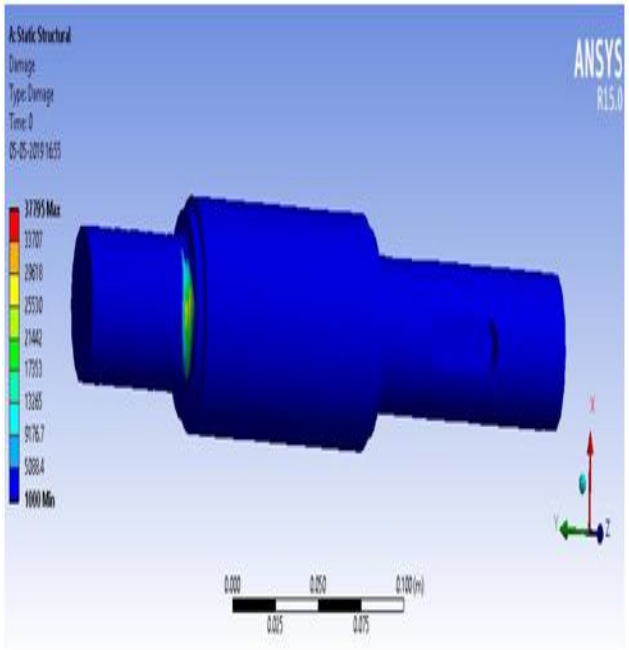

Fig.11. Results analysis of 30 mm diameter of shaft: Fatigue Damage at torque of 72,000 Nm.

Fig.11 shows the the analysis of 30 mm diameter of shaft. This analysis shows the maximum value of Fatigue Damage is 37795 and minimum value of Fatigue Damage is 1000.

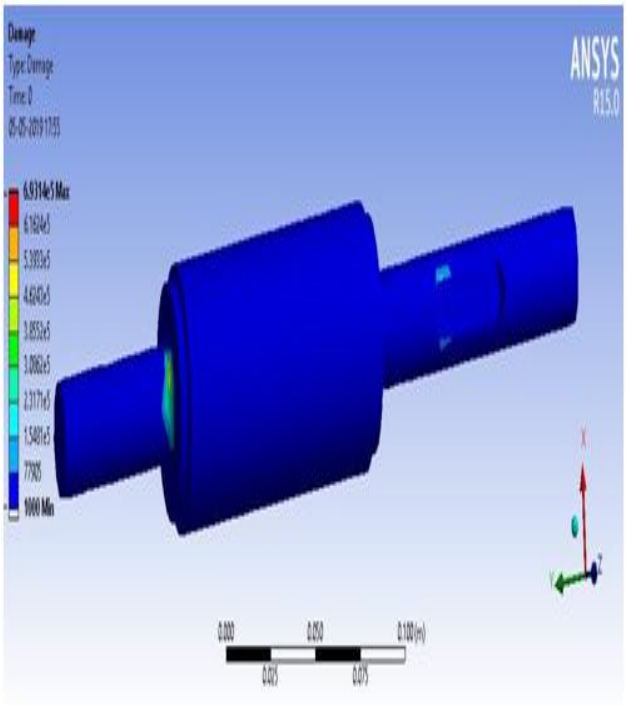

Fig.12. Results analysis of 30 mm diameter of shaft: Fatigue Damage at torque of 72,000 Nm.

Fig.12 shows the analysis of 30 mm diameter of shaft. This analysis shows the maximum value of Fatigue Damage is 6.9314e5 and minimum value of Fatigue Damage is 1000.

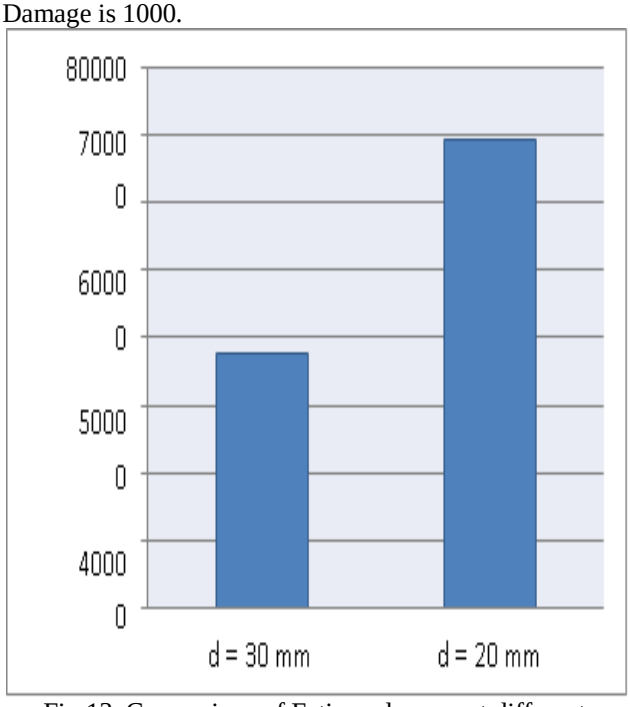

Fig.13. Comparison of Fatigue damage at different diameter.

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6. Factor of Safety of Fatigue Life

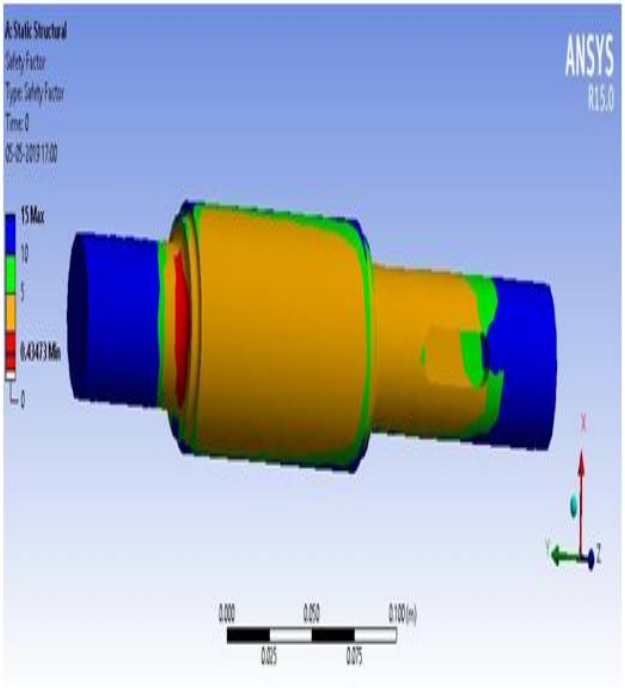

Fig.14. Results analysis of 30 mm diameter of shaft: Factor of safety at torque of 72,000 Nm.

Fig.14 shows the analysis of 30 mm diameter of shaft. This analysis shows the maximum value of Factor of safety is 15 and minimum value of Factor of safety is 0.43473.

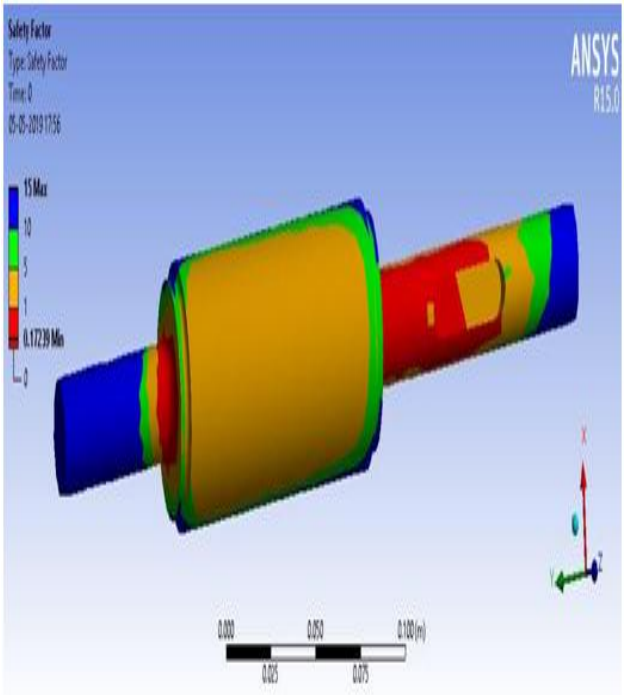

Fig.15. Results analysis of 20 mm diameter of shaft: Factor of safety at torque of 72,000 Nm.

Fig.15 shows the analysis of 30 mm diameter of shaft. This analysis shows the maximum value of Factor of safety is 15 and minimum value of Factor of safety is 0.17239.

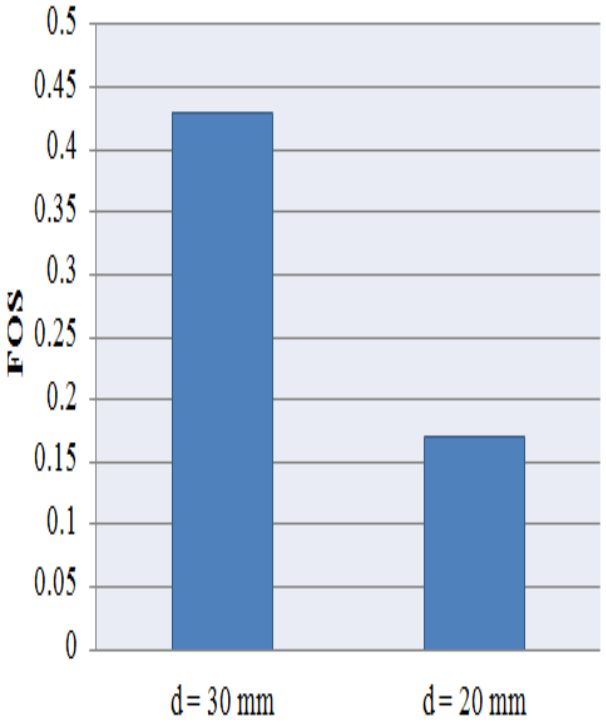

Fig.16. Comparison of FOS at different diameter.

7. Total Deformation

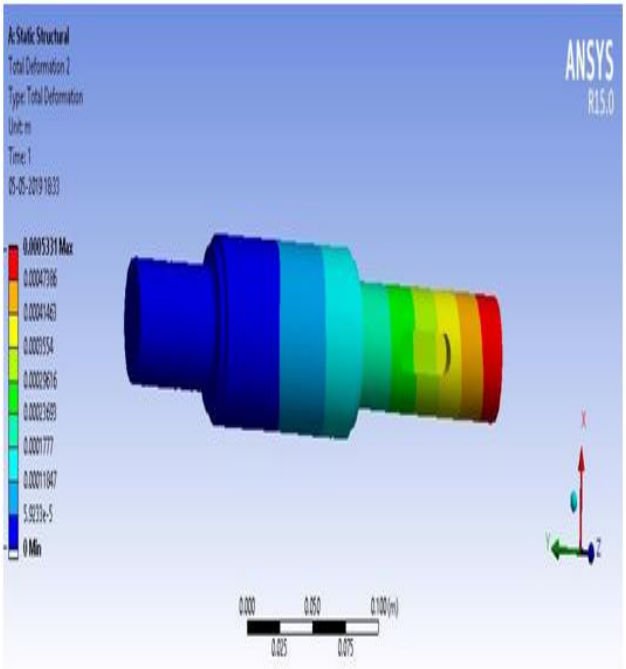

Fig.17. Results analysis of 30 mm diameter of shaft: Total Deformation at torque of 72,000 Nm.

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Fig.17 shows the analysis of 30 mm diameter of shaft. This analysis shows the maximum value of Total Deformation is 0.0005331 and minimum value of Total Deformation is 0.

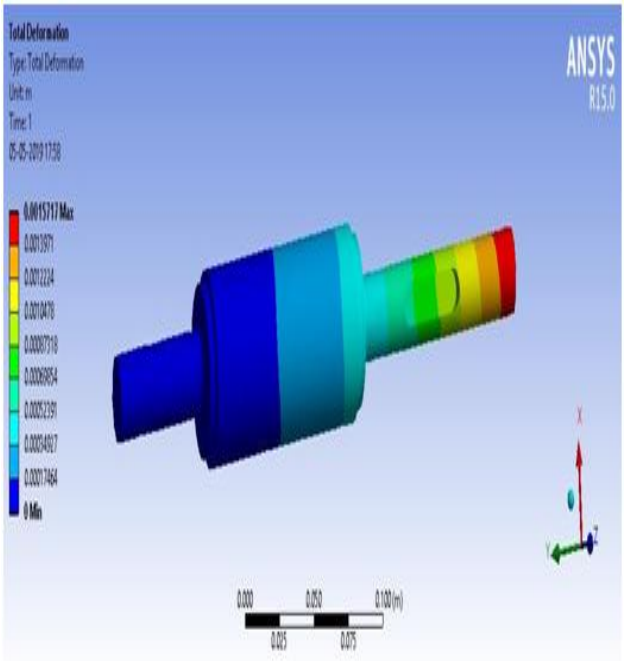

Fig.18. Results analysis of 20 mm diameter of shaft: Total Deformation at torque of 72,000 Nm.

Fig.18 shows the analysis of 20 mm diameter of shaft. This analysis shows the maximum value of Total Deformation is 0.0015717 and minimum value of Total Deformation is 0.

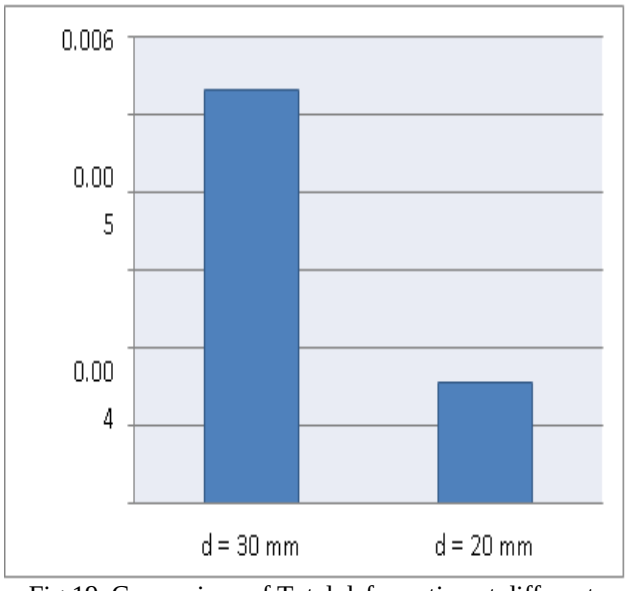

Fig.19. Comparison of Total deformation at different diameter.

III. CONCLUSION

The fatigue analysis will be applied to uni-axial and biaxial loading conditions for geometrically stepped shaft of diameter 30 mm and 20 mm with notch radius 5mm. This work will include the complete analysis of stepped shaft by using Solid works 11 and CAE analysis is performed using Ansys 15.0 and the fatigue analysis using nCode Design Life 14.5. Computer Aided engineering results obtained from this Analysis and many discussions have been concluded as given below:

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.Stepped shaft is at 30 mm diameter is observed in shaft shear stress is 0.010814 Mpa and when stepped shaft is at 20 mm diameter is observed in shaft shear stress is 0.026556 Mpa, which shows that stress taking capacity of 30 mm diameter shaft is more than 20 mm diameter.

Stepped shaft is at 30 mm diameter is observed in equivalent von mises stress is 0.019828 Mpa and when stepped shaft is at 20 mm diameter is observed in shaft shear stress is 0.050002 Mpa, which shows that von mises stress taking capacity of 30 mm diameter shaft is more than 20 mm diameter. Stepped shaft is at 30 mm diameter is observed in equivalent strain is 0.001124 Mpa and when stepped shaft is at 20 mm diameter is observed in equivalent strain is 0.002680 Mpa, which shows that equivalent strain taking capacity of 30 mm diameter shaft is more than 20 mm diameter.

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. 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. 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.

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 Misses stress distribution of the machine shaft structure under static loads.

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