Failure Analysis of Two Wheeler Crankshaft Using Finite Element Analysis

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ABSTRACT

The crankshaft is also referred as crank. In a reciprocating engine, it translates reciprocating linear piston motion into rotational motion. In a reciprocating compressor, it converts the rotational motion into reciprocating motion. Here the failure of crankshafts for two wheelers mostly occurs in the crankpin. Thus the crankpin is an important component that mostly decides the life of the crankshaft. The crankshaft considered here is of Scooty ES. It is a petrol engine crankshaft. Abnormal sound was heard in crankshaft while it is in operation. It was identified as failure of crankshaft. Severe wear has been observed at crankpin bearing location where the oil hole is provided. Here the analysis of the two wheeler crankshaft is done. Its results are then compared and verified numerically, then by the use of ANSYS software. The results compared here are Von Mises Stresses and the strain occurring on the crankshaft.

Keywords: Crankshaft, Crankpin, Strain, Stress, Force, Moment

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INTRODUCTION

Crankshaft

Crankshaft failures may be resulted from by several causes which areoil absence, defective lubrication on journals, high operating oil temperature, misalignments, improper journal bearings or improper clearance between journals and bearings, vibration, high stress concentrations, improper grinding, high surface roughness, and straightening operations.

Reasons for failure of crankshaft assembly and crankpin may be

A)Shaft misalignment
B) Vibration cause by bearings application
C) Incorrect geometry (stress concentration)
D) Improper lubrication
E) High engine temperature
F) Overloading

G) Crankpin material & its chemical composition

H) Pressure acting on piston

The crankshaft comprises of the shafts, which rotate in the bearings, the crank pins to which the enormous finishes of the con rod are associated, the crank arms, which interface the crankpins, and the shaft parts.

FORCES ACTING ON THE CRANKSHAFT

A major source of forces imposed on a crankshaft, namely Piston Acceleration. The combined weight of the piston, ring package, wristpin, retainers, the connecting rod small end and a small amount of oil are being continuously accelerated from rest to very high velocity and back to rest twice each crankshaft revolution. Since the force it takes to accelerate an object is proportional to the weight of the object times the acceleration (as long as the mass of the object is constant), many of the significant forces exerted on those reciprocating components, as well as on the conrod beam and big-end, crankshaft, crankshaft, bearings, and engine block are directly related to piston acceleration. Combustion forces and piston acceleration are also the main source of external vibration produced by an engine. Here in this case Piston Force is considered.

STRESS CALCULATIONS OF CRANKSHAFT

Scooty ES

- Engine Displacement= 59.9 cc
- Bore × Stroke = (42.6 × 42.0) mm
- Maximum Power = 3.5 ps @ 5500 rpm
- Maximum Torque = 4.5 Nm @5500 rpm

Pressure Calculation:

Density of Petrol (C18 H18):
$\varrho = 750\ kg/m^3 = 750 \times 10^{-9}\ kg/mm^3$

Operating Temperature:
$T= 20^0C=20+273=293^0K$
As, Mass= Density × Volume

Molecular weight of petrol:

$M= 114.228 \times 10^{-3}$ kg/mole

Gas constant for petrol:

$R= \frac{8314.3}{114.228\times 10-3}$

$R= 72.7868 \times 10^3$ J/kg/mol.K

As, PV = mRT

$P\times 59.9\times 10^3 = 0.044925 \times 72.7868 \times 10^3 \times 293.15$

P= 16.0030 MPa / N/mm^2

Design Calculations

Gas Force (Fp):

Fp= Pressure (P) X Cross-section area of piston (A)

Fp= 16.003 x $[\frac{\pi}{4} \times (42.6)^2]$

Fp= 22.809 x 10^3 N

Moment of Crank-Pin (Mmax)

$M_{max} = \frac{Fp}{2} \times \frac{lc}{2}$

lc = Length of crank-pin = 34.9 mm

$M_{max} = \frac{22.809}{2} x10^3 \text{ x } \frac{34.9}{2}$

M_{max}= 199.008 x 10^3 N. mm

Section Modulus of Crank-pin (Z)

Z= $\frac{\pi}{32}$ x dc^3

dc= Diameter of Crank-pin = 13 mm

Z= $\frac{\pi}{32} x13^3$

Z= 215.68 mm^3

Torque obtained at Maximum Power of given Engine

P= $\frac{2\pi NT}{60}$

3.5 x 10^3 = $\frac{2\pi \times 5500 \times T}{60}$

T= 6.076 x 10^3 N. mm

Von Mises Stresses induced

Equivalent Bending Moment:

$M_{ev} = \sqrt{(\text{Kb} \times \text{ Mmax})^2 + 0.75 \times (\text{Kt} \times \text{ T })^2}$

$M_{ev} = \sqrt{(1 \times 199.008 \text{ x} 10^3)^{\,2} + 0.75 \times (1 \times 6.076 \times 10^3)^2}$

M_{ev}= 199.077 x 10^3 N. mm

Thus, $\sigma von = \frac{\text{Mev}}{Z}$

$$\sigma von = \frac{199.007 \text{ x } 10^3}{215.68}$$

$\sigma = \sigma von = 923.69$N/mm^2

Strain = $\frac{\sigma}{E}$

Strain = $\frac{923.69}{200 \times 10^3}$

Strain = $4.6134 \text{ x } 10^{-3}$

The material medium carbon steel property of the crankshaft is given below in table format.

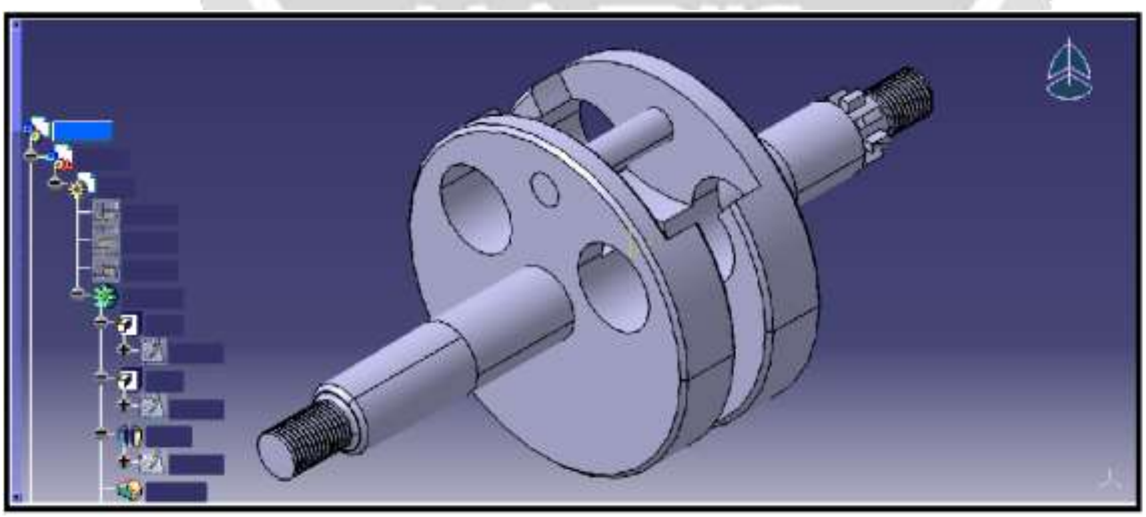
TABLE I : MATERIAL ATTRIBUTES

Density	2340 kg/m^3
Yield Tensile Strength	685 MPa
Ultimate Tensile Strength	885 MPa
Poisson's ratio	0.3

ANSYS RESULT

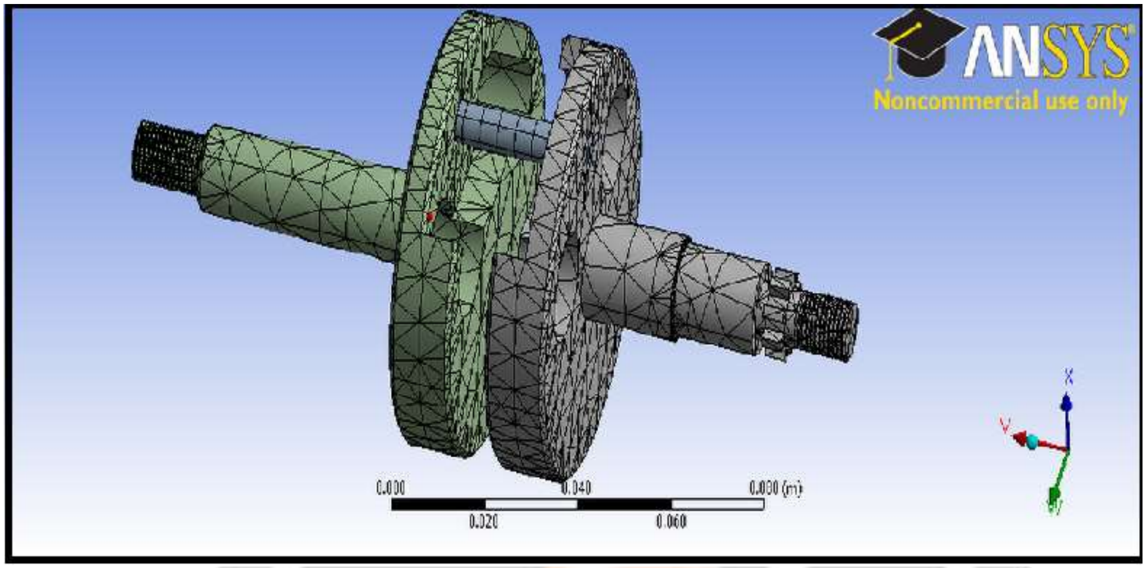
FEM Analysis of Existing Crankshaft

Figure shows the CAD model of shaft



CAD Model of Crankshaft

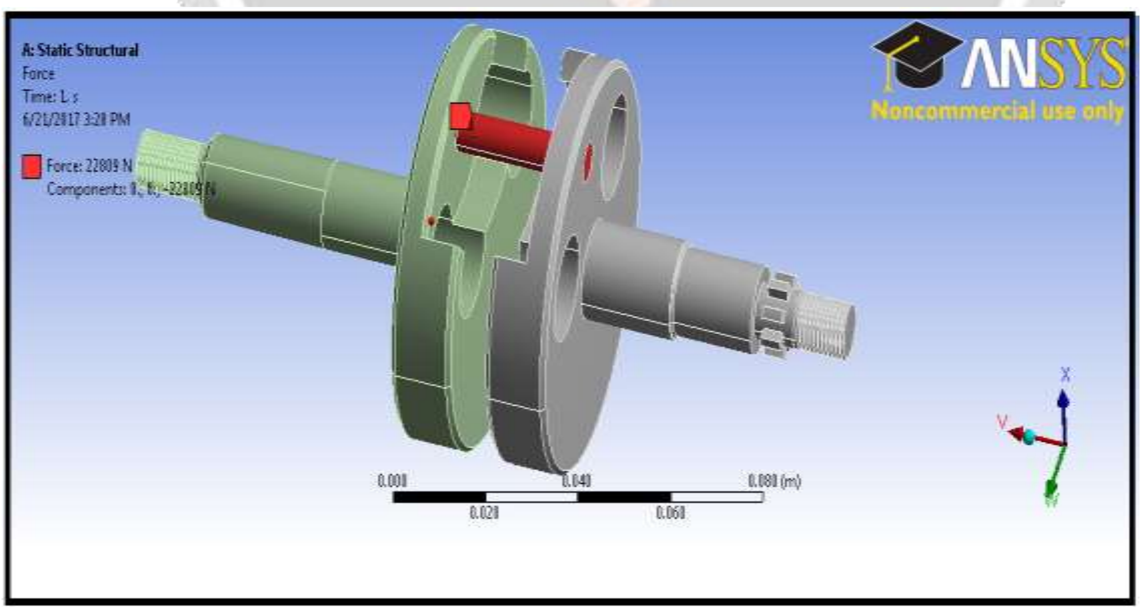
Figure shows the mesh modal of the crankshaft



Meshed model of Crankshaft

The Results of existing Crankshaft are described as below:

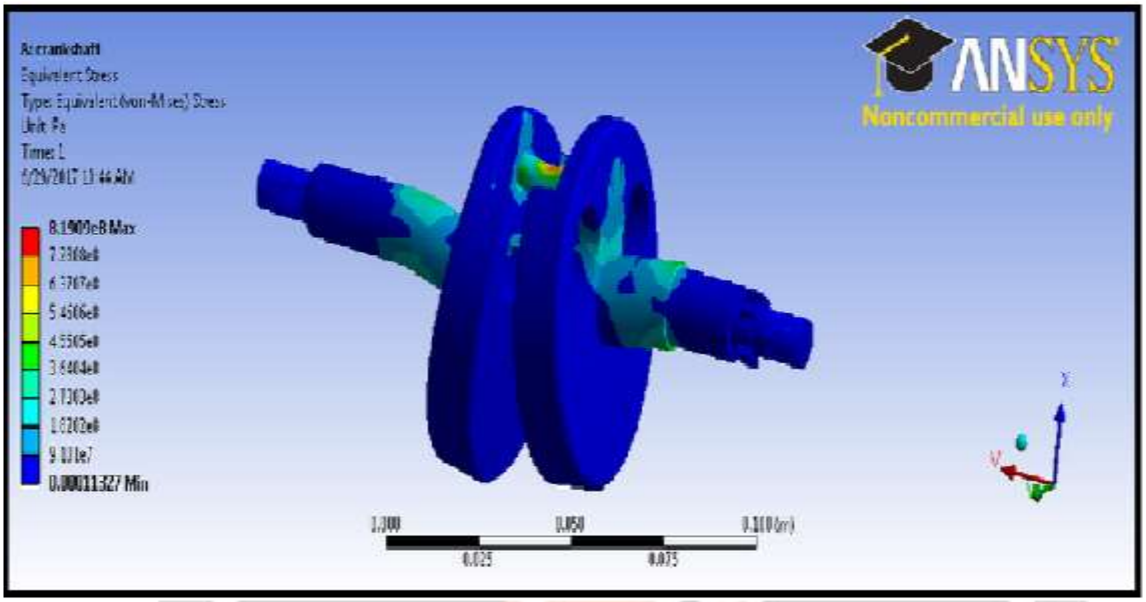
Figure Shows The force on the Crankshaft which is 22809 N



The force on the Crankshaft

Von-misses stress or Equivalent stress

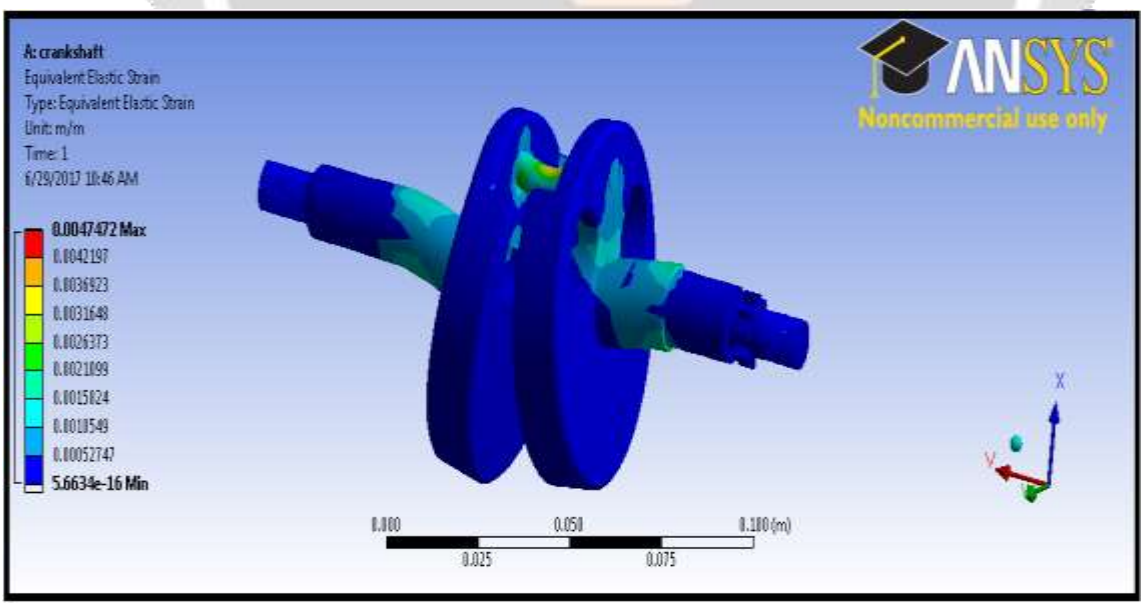
Figure Shows the maximum Equivalent (Von-Misses) stress on the crankshaft which is 819 MPa.



Maximum Equivalent (Von-Misses) stress

Equivalent Elastic Strain of Crankshaft

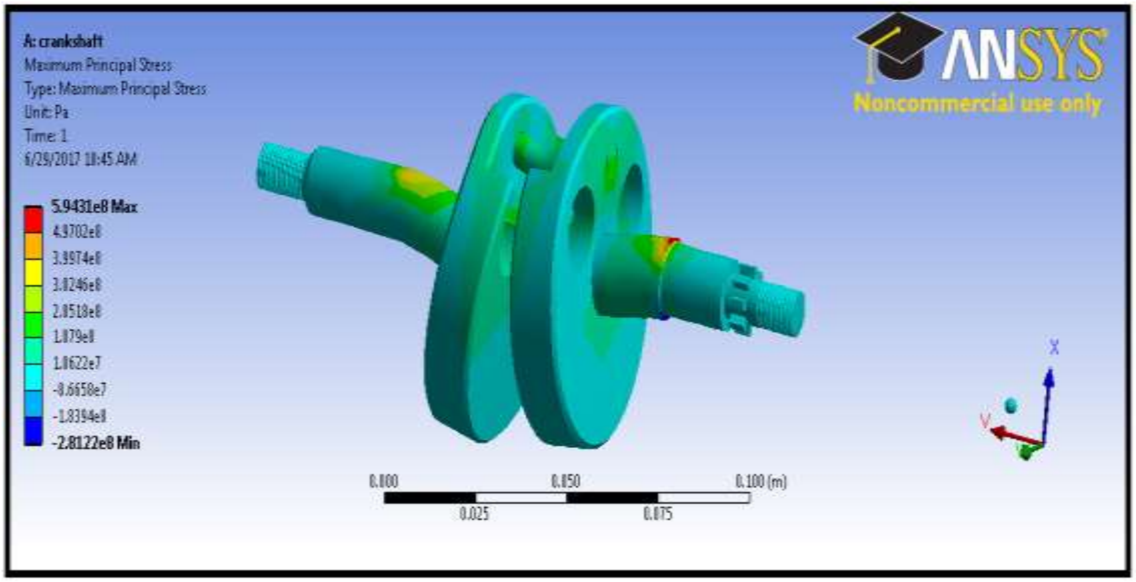
Figure Shows the Equivalent Elastic Strain of Crankshaft which is 0.0047472.



Equivalent Elastic Strain of Crankshaft

Maximum Principal stress of Crankshaft

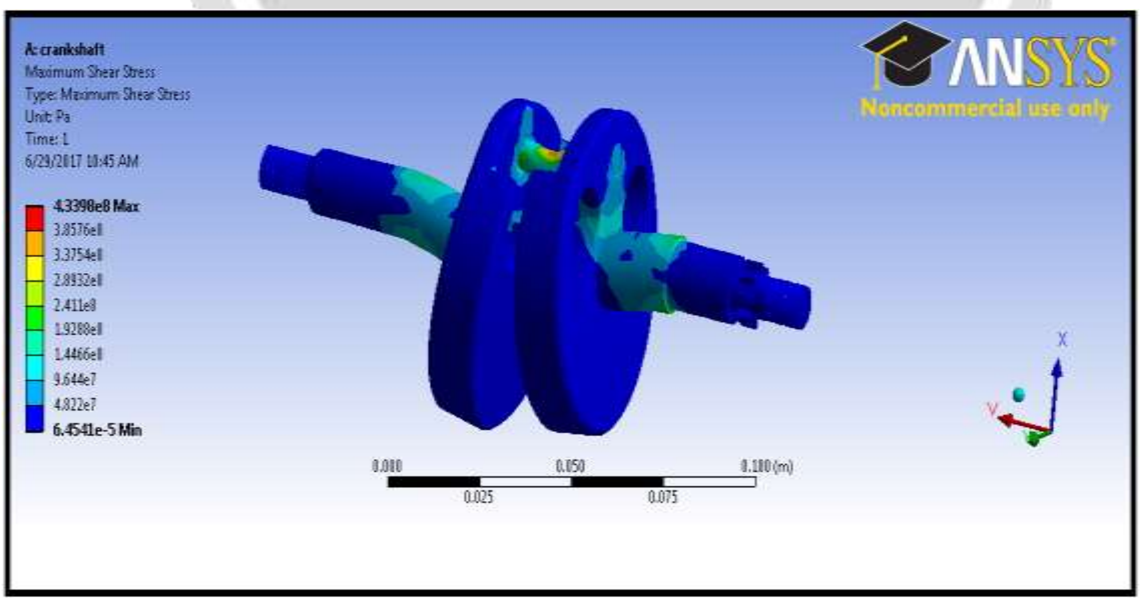
Figure shows the maximum principal stresses on shaft which are 594.31 MPa



Equivalent Elastic Strain of Crankshaft

Maximum shear stress on Crankshaft

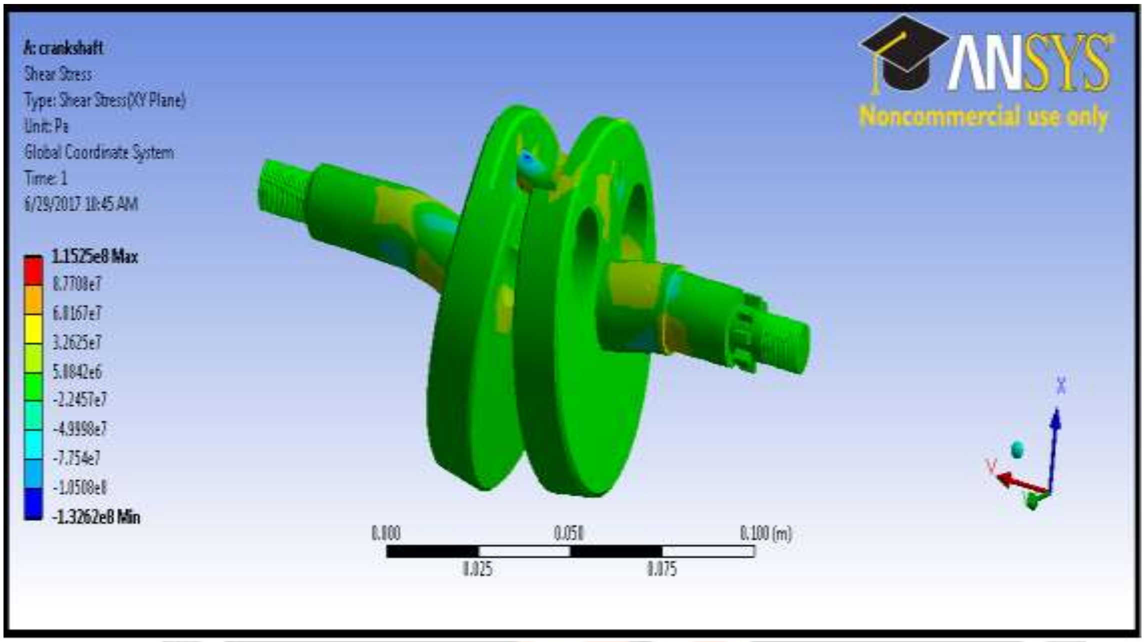
Figure showsthe maximum shear stress on the crankshaft which is 433.98 MPa



The maximumshear stress on the Crankshaft

The Shear stress on Crankshaft

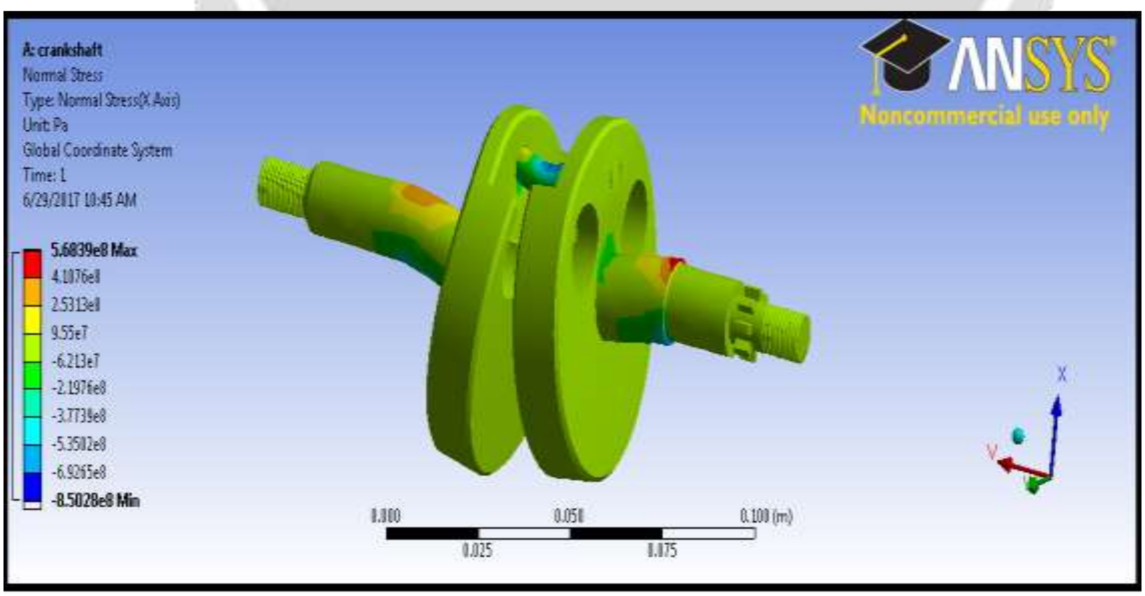
Figure showsthe shear stress on the shaft which is 115.25MPa



The shear stress on the Crankshaft

The normal stress on the Crankshaft

Figure shows The normal stress on the Crankshaft which is 568.39 MPa



The normal stress on the Crankshaft

The Factor of Safety of Crankshaft

Figure showsthe Factor of Safety of Crankshaft which is 0.10524

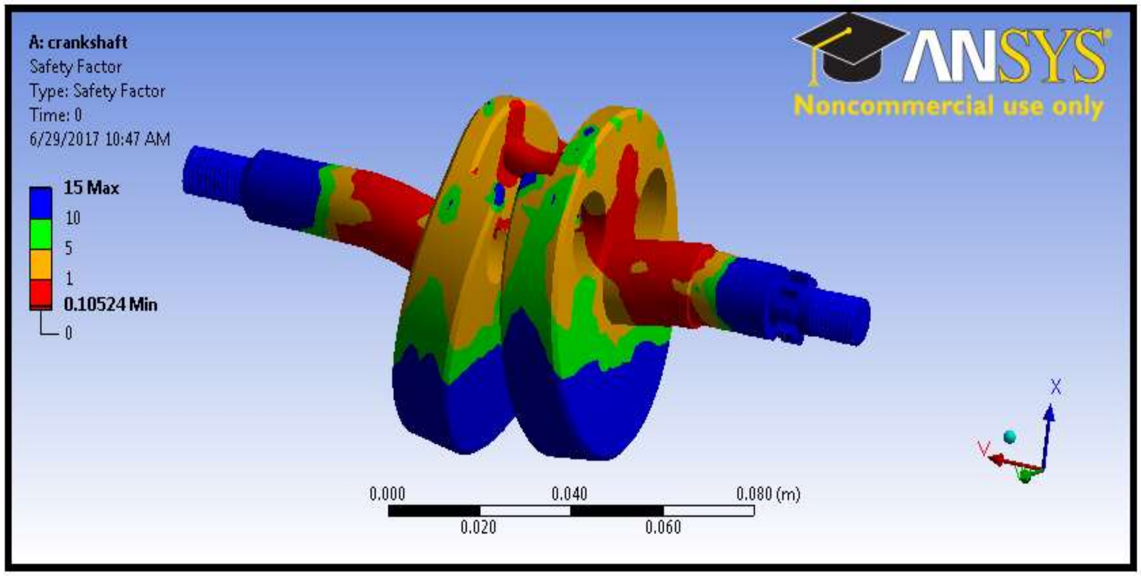
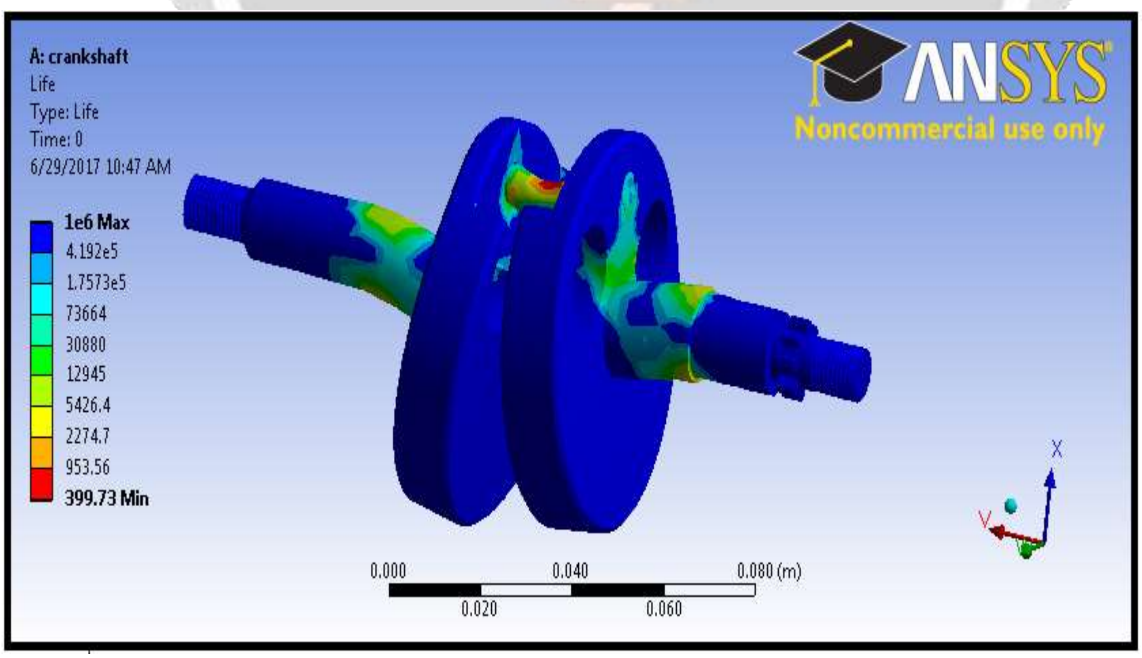


Fig.3.7 The Factor of Safety of Crankshaft

The Life of the Crankshaft

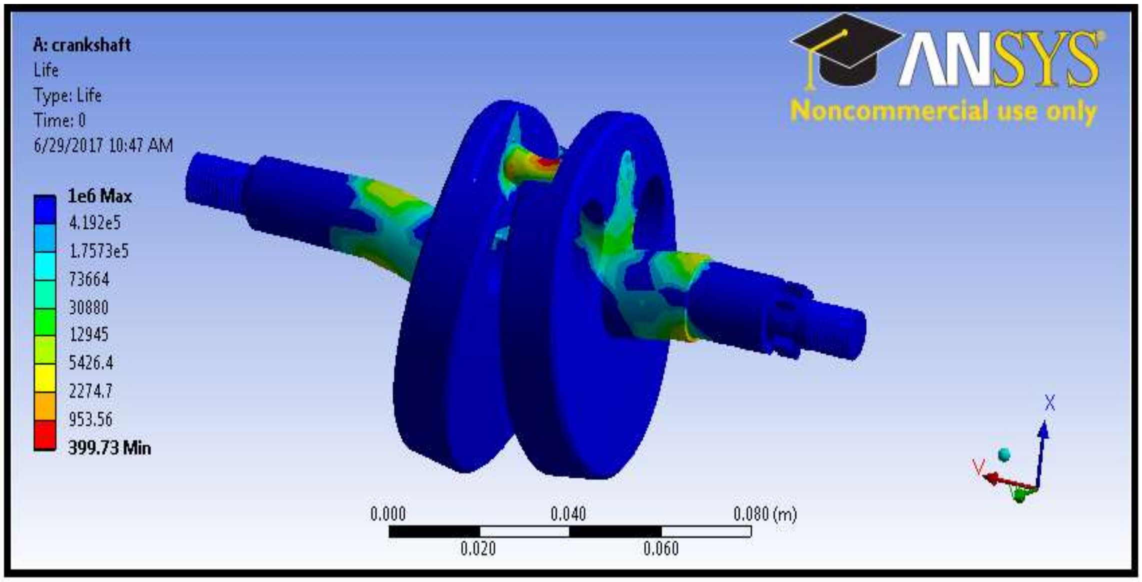
Figure shows life of the crankshaft which is 399.73.



The Life of the Crankshaft

The Damage of the Crankshaft

Figure Shows The Damage of the Crankshaft which is 2.5017



The Damage of the Crankshaft

Table II: Static Analysis of existing crankshaft

Parameters	Existing Crankshaft
Maximum principal stress	5.9431×10^{8} Pa
Maximum shear stress	4.3398×10^{8} Pa
Normal Stress	5.6839×10^{8} Pa
Damage	2.5017×10^{6}
Equivalent strain	0.004747mm
Equivalent stress	8.19×10^{8} Pa
Life	899.73×10^{6}
Factor of Safety	0.105

CONCLUSION

The crankshaft model was created in CATIA V5software and analysis of model done in Ansys workbench. Ansys results of equivalent stress, maximum shear stress and normal stress shows that existing material is below the limiting yield strength value of crankshaft. Also the ansys results are match with mathematical calculation and it was very closer to the existing material stress. The value of Von-misses stresses that comes out from the analysis is higher than material yield strength so material used for manufacturing of crankshaft is not safe. For better performance of crankshaft material can be changed which is having higher yield strength value.

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