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Fatigue Life Prediction of Crankshaft of Pulsar 180 Engine

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***Abstract:* -- The crankshaft of the engine is a most critical part in the engine as it is a heavy structure with complicated geometry. During working due to repetitive bending and shear stresses are usual stresses induced in the crankshaft, which are solely responsible for the crankshaft failure by fatigue. Thus, the assessment of the fatigue strength and life evaluation of the crankshaft assumes a vital part in the design and development of the crankshaft, taking into account of its safety and reliability during the operation. The Pulsar 180 DTS-i engine is considered for analysis which is made up of forged alloy steel 41Cr4Mo[1]. However, the analysis is also carried out by changing the material grey cast iron SAE J431 G2500 to compare the results. The static strength and fatigue failure criteria of the crankshaft are predicted analytically using S-N approach and modified Goodman theory. The 3D geometric model of crankshaft is created by reverse engineering. The FE model of the crankshaft is then developed using ANSYS Workbench. The investigation of the static structural strength and fatigue life is done utilizing ANSYS Workbench desktop software code and are validated with analytical solutions. Fatigue lives of the crankshaft for both forged alloy steel and grey cast iron are predicted based Von-Mises theory (Distortion Energy theory) and Maximum Principal Stress theory and are compared. Also the variation of fatigue lives of both forged alloy steel and grey cast iron crankshafts with different values of mean stress are predicted and compared. The evaluated results are graphically presented and discussed.**

There was a close agreement between the results in the Von-Mises stress obtained by analytical and FEA, which was 258 MPa by the analytical calculations and 248.11 MPa from the FE analysis and maximum stress obtained by both analytical and FE analysis were less than the ultimate tensile strength 1020 MPa of the material AISI 4140 alloy steel. The fatigue life of the forged alloy steel Crankshaft is 66% higher than the grey cast iron Crankshaft. And also the fatigue life increases with increase in the value of mean stress.

***Keywords:*—Fatigue life, LEFM, Mean Stress, Crankshaft, Crankpin, FE modeling, ANSYS.**

I. INTRODUCTION

The Engine is the heart of the automobile, as it provides blood and vitamin to the automobile to work. The reciprocating and rotating system of the internal combustion engines comprises of the piston, connecting rod and crankshaft. The engine represents a four-bar mechanism. The first bar is the connecting rod, the second bar is the crankshaft from the crank pin to the key journal, the third bar is the engine block, and the fourth bar is the piston. Crankshaft is a part of this mechanism which rotates as the piston moves up and down.

The crankshaft, generally also known as the crank[1], is usually responsible for the conversion of the motion from reciprocating into rotational. In a reciprocating engine; the crankshaft converts reciprocating linear piston movement into rotation of the drive shaft, while in a reciprocating compressor; the crankshaft interprets the rotation of the drive shaft into the linear reciprocation of the piston. The crankshaft is supported over pair of bearings, generally termed them as the main bearings. The main journals help in the crankpin bearing to rotate in a circular path around the main journal centers. The diameter of this circular path is twice that of the offset of the crankpin axis or the offset rod journal center from the crankshaft axis and this diameter of the circular path is generally the engine's 'stroke'.

Fig 1: Pulsar 180 DTSi Engine Crankshaft

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Forces acting on the Crankshaft: The gas forces generating from the combustion of the fuel in the cylinder will act on the piston[1], which is transmitted to the crankshaft through the connecting rod. The net force acting on the crankshaft will be the combined force effect of gas force, centrifugal force and the inertia force. Hence shear and bending stresses are induced in the crankshaft due to the effect of torsional moment of the shaft. Many crankshafts are failing by the progression of fracture caused by the repetitive bending and/or reversed torsional stresses.
Failure modes in Crankshaft[2]:

Fatigue failure-The fatigue failure is initiated due to a crack in the material and the regions of occurrence of this crack are at the stress concentration areas and also internal cracks due to materials defect. This crack starts to grow due to the cyclic loading and fluctuating stresses caused by these cyclic loading and in some point when the cross section is reduced, a sudden fracture will occur.

Failure due to Bending-The crankshaft failure may occur in the region of extreme bending, which will be at either ends of the crankpin or in the crankpin center. Failure may also occur due to the twisting of the crankpin at the extreme twisting region of crankpin, which is caused by shear stresses.

Fig 2: Failure due to bending & crack at the crankpin [9]

Industrial engines are generally made up of carbon steels with 0.35% Carbon. Heavy duty Nickel Cast iron is also being used in relatively low speed engines. In transport engines manganese steel alloys are adopted. Nickel chromium steel alloy crankshafts are mostly adopted in aero engines. Two major kinds of processes generally being adopted in the production of crankshafts for the engines, they are either by Casting or Forging.

H. Bayrakceken et. al.[3] studied the crankshaft failure of diesel engine with single cylinder, which has been deployed in the agricultural vehicles. In this study two different crankshafts made of same material, one is surface hardened and the other is annealed are used. Both are analyzed for the chemical and also the difference between the microstructure of materials used. The study showed the inclusions of carbides in the material by improper heat treatment as well as production processes adopted in the production of crankshafts. These regions of the carbide inclusion are the regions of crack initiation. The study also gave note on the failure of the crankshafts due to fracture which has been started in the regions of sharp fillet; the crack growth and direction have been influenced by the lubrication holes present in the crankshafts.

Bhumesh J Bagde et. al.[4] studied the problem of failure of crankshaft due to fatigue. In this study, first the 3D geometric model of the crankshaft of single cylinder engine was generated by using Pro/E WF 4.0 and the FE model of the same is created with load and boundary conditions in ANSYS s/w. This FE model was used for the static structural analysis of total deformation, max. Stress and fatigue life was also evaluated for different materials like EN-9, SAE1045, SAE1137, SAE3140 & Ni-Cast iron. The results were analyzed for the critical locations where stress induced was max and the possible areas of crack initiation. Results also showed that the fatigue life obtained for EN-9 & SAE1137 were far better than other materials.
Mayuresh Nikam et. al.[1] carried out a comparative analysis of the Bajaj Pulsar 180 DTS-i engine crankshaft by analytically and numerically by using ANSYS desktop commercial software code. The static structural simulation for the total deformation and the Von-Mises stress are calculated analytically and compared with the numerical results of ANSYS and Von-Mises stress was within a range of 13% and deformation was within a range of 31%. The fatigue life was obtained from Goodman line and S-N curves and the same from ANSYS and results are compared, it was observed the values are within an agreeable range.

II. OBJECTIVES

Aim and Objectives:

The core focus of this project is FE modeling of the Crankshaft and static structural analysis and prediction fatigue life based on LEFM theories using ANSYS Workbench.

- To develop refined enough FE model of the Crankshaft using the 3D solid elements in the ANSYS Workbench.
- To perform the linear static structural analysis and post process the results to predict the

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stress distribution, total deformation and fatigue life of the Crankshaft.

- To estimate the structural strength of forged alloy steel and grey cast iron Crankshaft based on Von-Mises theory and Maximum Principal Stress theory.
- To predict the fatigue life of both forged alloy steel and grey cast iron Crankshaft by considering effect of mean stress.

III. METHODOLOGY

The static strength of the both forged alloy steel and grey cast iron crankshaft is calculated numerically. The fatigue life of the forged alloy steel crankshaft is determined by analytically using Stress-Life approach and Goodman theory.

The 3D geometric model of the crankshaft required to generate the finite element model for analysis is created by reverse engineering process with the help of PTC Creo 2.0.

The FE model is created in ANSYS by applying boundary and loading conditions. Static strength and fatigue life of both forged alloy steel and grey cast iron crankshaft are predicted based on the Von-Mises and Maximum Principal Stress theories. And also fatigue life for various values of mean stress are predicted by varying the load ratio in the fatigue tool of ANSYS. The evaluated results are graphically represented and discussed.

IV. STRESS ANALYSIS AND FATIGUE LIFE PREDICTIONS OF FORGED ALLOY STEEL

A. Analytical Solution:

To calculate the stresses induced on the single throw crankshaft, the maximum gas forces acting on the piston, when it is at top dead center (TDC) condition is considered.

Engine Specifications:

i) Engine type: Single cylinder, 4-stroke, air-cooled.
ii) Displacement: 178.6 cc
iii) Bore*Stroke: 63.5*56.4 mm
iv) Maximum Power:
17.02 @ 8500 (ps @ RPM)
12.518 @ 8500 (kW @ RPM)
v) Maximum Torque: 14.22 @ 6500 (Nm @ RPM)

Crankshaft Material Properties:

The crankshaft is made up of alloy steel 41Cr4Mo (AISI4140 alloy steel)[5].

Table 1: AISI 4140 material monotonic properties

Monotonic Properties	Forged Steel AISI 4140
Young's Modulus (E), GPa	205
Yield Strength (S_y), MPa	675
Ultimate Tensile Strength (S_{ut}),MPa	1020
Density (ρ), kg/m3	7850
Poisson's Ratio	0.29
Brinell Hardness Number (HB)	302

Pressure Calculations:

Density of Petrol (C8H18), ρ = 750x10-9 kg/mm3
Operating temperature, T = 20oC = 293.15 oK
Mass = 0.13395 kilograms
Petrol - Molecular weight, M = 114.228×10-3 kg/mol
Petrol - Gas constant, R = 8314.45 J/mol-K
R=72.7882x103 J/kg-K
Also, pV=mRT .... (1)
p=16.0034 N/mm2

Design Calculations:

Gas Force, Fp=Pressure × C/S area of piston .... (2)
Fp=50681.5 N
Moment on the crankpin:
$Mmax=Fp2 \times lc2$ .... (3)
Mmax=684200.25 N-mm
Section modulus of crankpin:
$Z=\pi 32\times dc3$ .... (4)
Z=2650.72 mm3
Torque at maximum power: [12.518 @ 8500 (kW @ RPM)]
$P=2\pi NT60\ W$ .... (5)
T=14.0633×103 N-mm
Equivalent Bending moment:
$Meq=\sqrt{(Kb\times Mmax)2+34(Kt\times T)2}\ N-m$ .... (6)
Kb: Combine shock, bending fatigue factor = 1
Kt: Combine shock, torsion fatigue factor = 1
$\boldsymbol{Meq=684308.64\ Nmm}$
Von-Mises Stress:
$\sigma von=MeqZ\ N/mm2$ .... (7)
$\boldsymbol{\sigma von=258\ N/mm2}$
Deformation:
Strain, $\epsilon=\sigma E/$, Stress, $\sigma=FpAc$ .... (8)
Surface area of crankshaft, $Ac=dc\times lc$

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Deformation, $\delta=\epsilon\times lc$

$\boldsymbol{\delta=0.00824\ mm}$

Fatigue Analysis:

Endurance Strength:

$Se=kakbkckdkekfSe'$ .... (9)

Endurance limit, Se'=0.5Sut= 510 MPa

Where[6],

Surface Factor ka=0.7193 for Machined, Size Factor kb=0.858, Loading Factor kc=1, Temperature Factor kd=1,

Reliability Factor ke=0.868, Miscellaneous Effects Factor kf=1

Endurance Strength, $\boldsymbol{Se}$=$\mathbf{300}\ \boldsymbol{MPa}$

From Modified Goodman theory,

Goodman Line:

$$\frac{S_a}{S_e}+\frac{S_m}{S_{ut}}=1$$

$$S_f=\frac{\sigma_a * S_u}{S_u-\sigma_m},\qquad \text{Since, } \sigma_a=\sigma_{von},\ \sigma_m=0$$

$$\boldsymbol{S_f=\sigma_a=\sigma_{von}=258\ MPa}$$

Goodman Line:

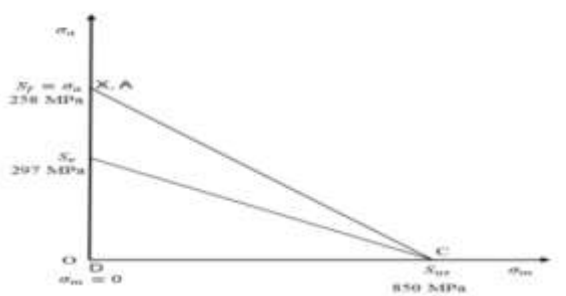

Fig 3: Goodman diagram

From diagram,

$$S_f=AO=\frac{XD\times OC}{DC},\qquad S_f=\sigma_a$$

S-N Curve:

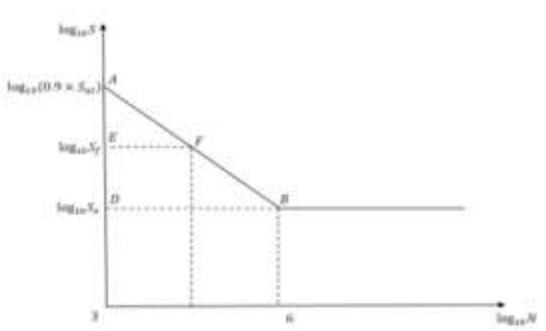

Fig 4: S-N curve

From Diagram,

$$EF=DB\times\frac{AE}{AD}$$

$$\log_{10}(N)=(6-3)\times\frac{[\log_{10}(0.9S_{ut})-\log_{10}(S_f)]}{[\log_{10}(0.9S_{ut})-\log_{10}(S_e)]}$$

N = 16017 Cycles

B. Numerical Solution:

Finite Element Modeling:

Geometric Modeling of Crankshaft - The 3D geometric model is generated in Creo Parametric 2.0 geometric modeling software developed by PTC Inc.

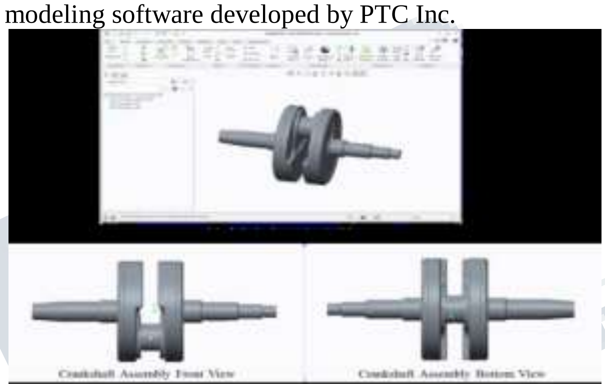

Fig 5: 3D Geometric assembly model of crankshaft

Material Properties - The properties tabulated in table 1 are applied to the imported geometric 3D model in the ANSYS Workbench.

Meshing - The 10-node Tetrahedral SOLID 187 element is used to divide the crankshaft assembly into 211001 elements having 336328 nodes[7][8].

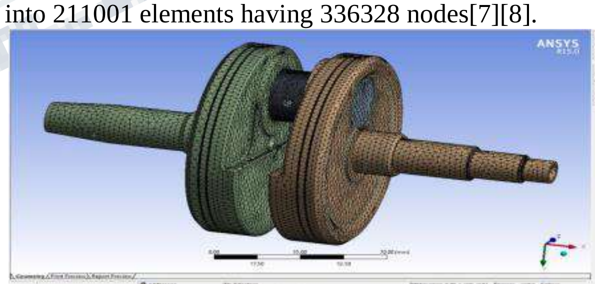

Fig 6: Meshed model of the crankshaft in ANSYS

Boundary Conditions - The Flywheels are rested on the bearings and hence for static analysis, the two flywheels are fixed by cylindrical support at the bearing and end displacements are constrained[7][8].

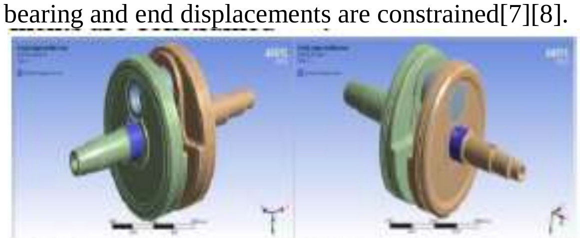

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Fig 7: Cylindrical support on Flywheel & Flywheel Drive

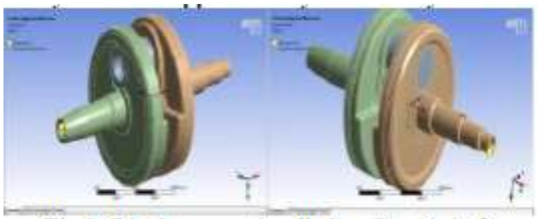

Fig 8: Displacement implied on Crankshaft

Loading Conditions - The load acting on the crankpin through the connecting rod is taken at the TDC condition, a downward pressure of 21.43 MPa is applied on crankpin top surface[9].

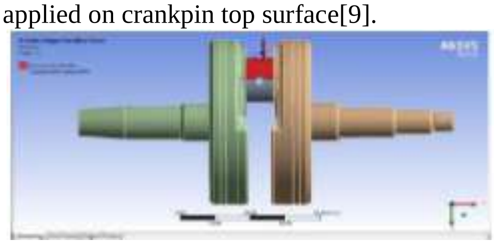

Fig 9: Load applied on the Crankpin top surface

Fatigue Tool:

- ♣ Considering Fully Reversed Load Cycle

Fig 10: Fully reversed constant amplitude load diagram

Fig 11: Goodman mean stress correction theory diagram

- ♣ Considering Effect of Mean Stress

The various mean stress effect on the fatigue life are predicted by varying the load ratio in the fatigue tool. For load ratio of R=-0.5

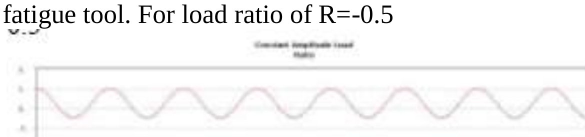

Fig 12: Constant amplitude with load ratio -0.5 load diagram

***C. Results and Discussion**:*

Von-Mises Stress:

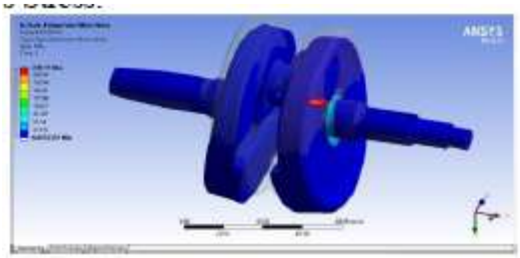

Fig 13: Contour plots of Von-Mises stress induced in Crankshaft

Maximum Von-Mises stress induced at the neck of the Flywheel_Drive shaft is 248.11 MPa. Comparing induced stress with materials ultimate strength, we could say that the design is safe.

Total Deformation:

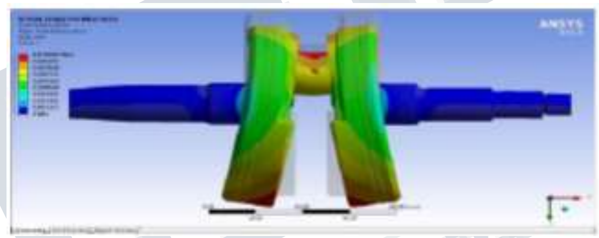

Fig 14: Contour plots of total deformation

The occurrence of the max deformation is on the crankpin at its center and the value of max deformation is 0.010099 mm.

Fatigue Life:

- ♣ Fully Reversed Load Cycle

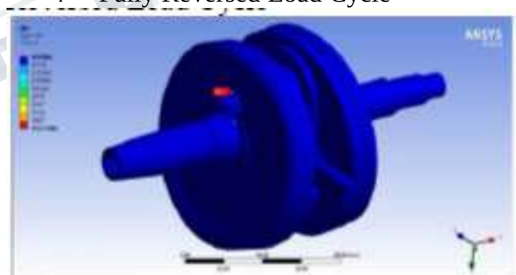

Fig 15: Contour plots of fatigue life under fully reversed load cycle

The fatigue life of Crankshaft predicted Von-Mises stress of 248.11 MPa and using Goodman mean stress correction is of 12221 cycles. The critical location of fatigue is at the inner side of crankpin

- ♣ Mean Stress Effect

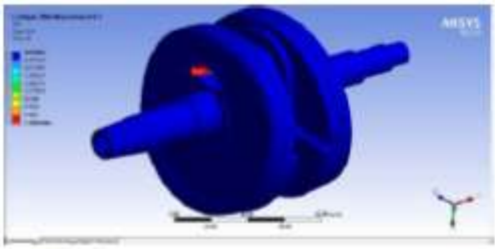

Fig 16: Contour plots of fatigue life under load ratio of -0.5 load cycle

The fatigue life of Crankshaft predicted from ANSYS based on mean stress of 62.03 MPa and using Goodman mean stress correction is of 27989 cycles. The critical location of fatigue is at the inner side of crankpin.

Fatigue Safety Factor:

The fatigue safety factor Sf, is equal to the ratio of limiting value of fatigue amplitude strength σA of amplitude stress σa and the amplitude stress σa itself[10].

- Fully Reversed Load Cycle

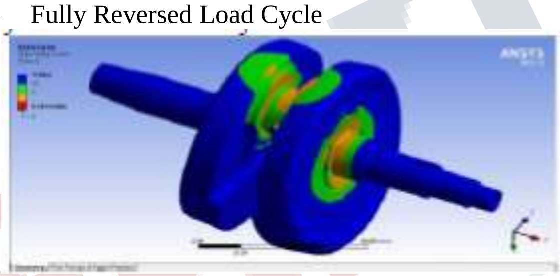

Fig 17: Contour plots of fatigue safety factor for constant amplitude fully reversed load cycle

The fatigue safety factor estimated is 0.34743 based on the Goodman line for the fatigue estimation of the Crankshaft.

- Mean Stress Effect

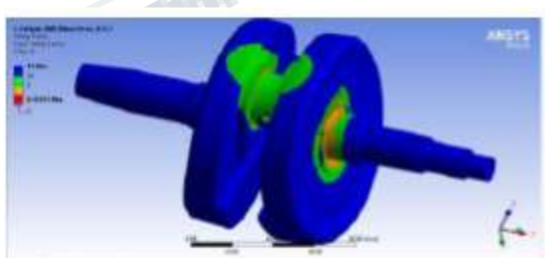

Fig 18: Contour plots of fatigue safety factor for constant amplitude load ratio of -0.5 load cycle

The fatigue safety factor estimated is 0.45054 based on the Goodman line for the fatigue estimation of the Crankshaft.

Comparison of Analytical & Numerical Results:

Table 2: Analytical & numerical results tabulation

Parameter	Analytical	Numerical
Von-Mises Stress (MPa)	258	248.11
Total Deformation (mm)	0.00824	0.010099
Fatigue Life (Cycles)	16017	12221

It is drawn that there is a 3.83% of considerable difference observed between the Von-Mises stress values, a 20.56% in total deformation and a 23.7% in fatigue life obtained in both analytical and numerical methods due to the error in 3D geometric modeling, load application, boundary conditions applications in numerical method and assumptions considered in analytical method.

Fatigue Life for Different Values of Mean Stress:

Table 3: Fatigue life for various values of mean stress

Load Ratio 'R'	Mean Stress 'σ_m' (MPa)	Fatigue Life 'N' (Cycles)	Fatigue Safety Factor 'S_f'
-1.50	-62.03	5415	0.27
-1.25	-31.02	7924	0.30
-1.00	0	12221	0.34
-0.75	31.02	17335	0.39
-0.50	62.03	27989	0.45
-0.25	93.04	50104	0.53
0	124.05	93385	0.64
0.25	155.07	2.5621×10^5	0.81
0.50	186.08	1×10^6	1.11

It is predicted from the fatigue analysis that Crankshaft will have an infinite fatigue life of 1x106 cycles and fatigue safety factor of 1.11 for the load ratio of 0.50 that means when the minimum load of cyclic loading is half of the max loading point, which gives a maximum mean stress, that is for a min stress of 124.05 MPa and mean stress of 186.08 MPa.

V. STRESS ANALYSIS AND FATIGUE LIFE PREDICTIONS OF GREY CAST IRON

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A. Finite Element Modeling

For the simulation of static structural strength and fatigue life of the Crankshaft made up of grey cast iron, the geometric model & contacts used in design module, meshing, boundary and loading conditions in mechanical module are used same as in the previous case of forged alloy steel simulation except changing the material properties.

Material Properties:

Table 3: SAE J431 G2500 material monotonic properties

Monotonic Properties	Grey Cast Iron SAE J431 G2500
Young's Modulus (E), GPa	114
Ultimate Tensile Strength (Sut), MPa	173
Density (ρ), kg/m3	7150
Poisson's Ratio	0.29
Brinell Hardness Number (HB)	170-229

B. Results & Discussion:

Maximum Principal Stress:

Fig 19: Contour plots of maximum principal stress induced in Crankshaft

Maximum Principal Stress induced at the neck of the Flywheel shaft is 105.37 MPa. Comparing induced stress with materials ultimate strength, we could say that the design is safe.

Total Deformation:

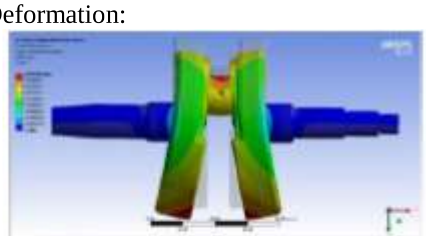

Fig 20: Contour plots of total deformation

The occurrence of the max deformation is on the crankpin at its center and the value of max deformation is 0.01816 mm.

Fatigue Life:

- Fully Reversed Load Cycle

The fatigue life of Crankshaft predicted Maximum Principal Stress of 105.37 MPa and using Goodman mean stress correction is of 4044 cycles. The critical location of fatigue is at the inner side of crankpin

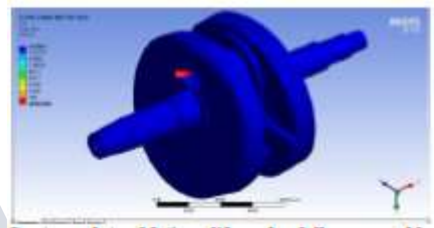

Fig 21: Contour plots of fatigue life under fully reversed load cycle

- Mean Stress Effect

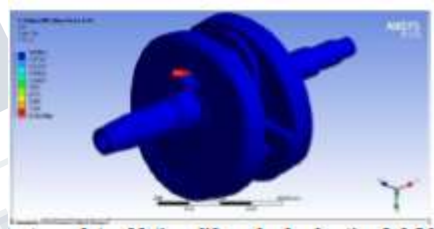

Fig 22: Contour plots of fatigue life under load ratio of -0.5 load cycle

The fatigue life of Crankshaft predicted from ANSYS based on mean stress of 26.34 MPa and using Goodman mean stress correction is of 9139 cycles. The critical location of fatigue is at the inner side of crankpin.

Fatigue Safety Factor:

- Fully Reversed Load Cycle

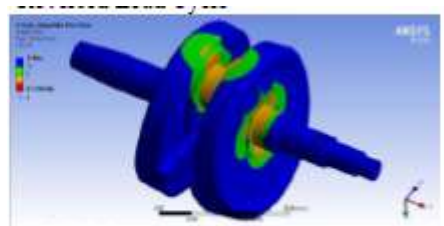

Fig 23: Contour plots of fatigue life under fully reversed load cycle

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The fatigue safety factor estimated is 0.23768 based on the Goodman line for the fatigue estimation of the Crankshaft

- Mean Stress Effect

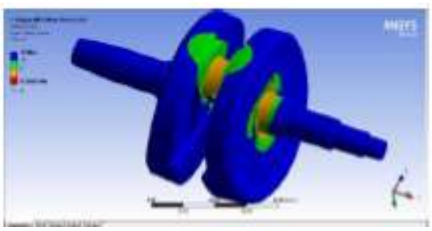

Fig 24: Contour plots of fatigue safety factor for constant amplitude load ratio of -0.5 load cycle

The fatigue safety factor estimated is 0.31691 based on the Goodman line for the fatigue estimation of the Crankshaft.

Fatigue Life for Different Values of Mean Stress:

Table 4: Fatigue life for various values of mean stress

Load Ratio 'R'	Mean Stress 'σ_m' (MPa)	Fatigue Life 'N' (Cycles)	Fatigue Safety Factor 'S_f'
-1.50	-26.34	1625	0.18
-1.25	-13.17	2268	0.21
-1.00	0	4044	0.23
-0.75	13.17	6080	0.27
-0.50	26.34	9139	0.31
-0.25	39.52	16656	0.38
0	52.68	39944	0.47
0.25	65.85	$1.059\text{x}10^5$	0.63
0.50	79.03	$7.72\text{x}10^5$	0.95
0.75	92.2	$1\text{x}10^6$	1.46

It is predicted from the fatigue analysis that Crankshaft will have an infinite fatigue life of 1x106 cycles and fatigue safety factor of 1.46 for the load ratio of 0.75 that means when the minimum load of cyclic loading is half of the max loading point, which gives a maximum mean stress, that is for a min stress of 79.03 MPa and mean stress of 92.2 MPa.

VI. CONCLUSIONS

There was a close agreement between the results in the Von-Mises stress obtained by analytical and FEA, which was 258 MPa by the analytical calculations and 248.11 MPa from the FE analysis by ANSYS Workbench 15.0 and maximum stress obtained by both analytical and FE analysis were less than the ultimate tensile strength 1020 MPa of the material AISI 4140 alloy steel.

From the fatigue analysis of Crankshaft of both forged alloy steel AISI 4140 and grey cast iron SAE J431 G2500 in ANSYS, it is drawn that the fatigue life of the forged alloy steel Crankshaft is 66% higher than the grey cast iron Crankshaft. And also the fatigue life increases with increase in the value of mean stress.

REFERENCES

[1] Mayuresh Nikam, Santosh Kumar and Avinash G. Patil, 2015, "Comparative analysis of crankshaft of Pulsar 180 DTS-i", International Journal on Innovations in Mechanical and Automotive Research", Vol. 1, Issue 2, Pg. 109-115.

[2] K. Pandiyan and Dr. Ashesh Tiwari, 2015, "Crankshaft design methodology for diesel engines", International Journal of Innovative Research in Science, Engineering and Technology, Vol. 4, Issue 8, Pg. 7580-7586.

[3] H. Bayrakceken, S. Tasgetiren & F. Aksoy, 2007, "Failures of single cylinder diesel engines crank shafts", Engineering Failure Analysis 14, Pg. 725-730.

[4] Bhumesh J. Bagde & Laukik P. Raut, 2013, "Finite element structural and fatigue analysis of single cylinder engine crankshaft", International Journal of Engineering Research and Technology, Vol. 2, Issue 7, Pg. 1540-1544.

[5] Mahesh L. Raotole, D. B. Sadaphale and J. R. Chaudhari, 2013, "Prediction of fatigue life of crankshaft using S-N approach", International Journal of Emerging Technology and Advanced Engineering, Vol. 3, Issue 2, Pg. 447-454.

[6] Joseph E. Shigley, Charles R. Mischke, Richard G. Budynas & Keith J. Nisbett, "Mechanical Engineering Design", 8th Edition, 2008, Tata McGraw-Hill Publishing Company Limited, New Delhi, India.

[7] Nitin S. Gokhale, Sanjay S. Deshpande, Sanjeev V. Bedekar & Anand N. Thite, "Practical Finite Element Analysis", 1st Edition, 2008, Finite to Infinite, Pune, India.

[8] Anant Prakash Agarwal & Dr. S. K. Srivastava, 2012, "Fatigue life prediction of crankshaft based on strain life theories", International Journal of Engineering Research and Technology, Vol. 1, Issue 8, Pg. 1-5.

[9] Rinkle Garg & Sunil Baghla, 2012, "Finite element analysis and optimization of crankshaft design", International Journal of Engineering and Management Research, Vol. 2, Issue 6, Pg. 26-31.

[10] Jalska D, 2011, "On the Goodman's Fatigue Safety Factor", Advanced Engineering.

[11] Ralph I. Stephens, Ali Fatami, Robert R. Stephens & Henry O. Fuchs, "Metal Fatigue in Engineering", 2nd Edition, 2001, John Wiley & Sons Inc., New York, USA.

[12] Singiresu S. Rao, "The Finite Element Method in Engineering", 5th Edition, 2014, Butterworth-Heinemann, Burlington, USA.

[13] "Provisional Training Notes – Pulsar DTSi", Service Training Center, Bajaj auto ltd.

[14] T. L. Anderson, "Fracture Mechanics Fundamentals and Applications", 3rd Edition, 2005, CRC Press, Boca Raton, Florida.

[15] Rahul Sapkal & Shailesh Pimpale, 2014, "Evaluating design of the automotive crankshaft for fatigue life using finite element method", International Journal of Engineering Research and Technology, Vol. 3, Issue 9, Pg. 296-299.

[16] R. M. Metkar, V. K. Sunnapwar and H. D. Hiwase, 2011, "A fatigue analysis and life estimation of crankshaft – a review", International Journal of Mechanical and Materials Engineering", Vol. 6, Issue 3, Pg. 425-430.