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MODELING &ANALYSIS OF ALUMINUM ALLOY CRANKSHAFT FOR OPTIMIZATION OF WEIGHT USING FEA

Eknath B. Pore[1], Prof. D.R. Kotkar[2]
ME Student,Department of Mechanical Engineering[1], Assistant Professor, Department of Mechanical Engineering[2]
Dhole Patil College of engineering, wagholi, Pune ,India
[1]eknathpore@gmail.com

Abstract— **In this project we have performed static & fatigue analysis of crankshaft made of forged steel and Aluminum Alloy (Al 5083) Reinforced with Silicon carbide (Sic material for the purpose of optimization of weight. In this project the material (forged steel) of crankshaft is replaced with developed Aluminum alloy from similar single cylinder four stroke engines. The model of crankshaft is created in CATIA V5 and imported in ANSYS 14 work bench for Static and Fatigue analysis. Analyzed crankshafts are compared in terms of Von misses stress, equivalent strain, total deformation, Fatigue life , Safety factor. Finite element analysis is performed to obtain the variation of stress at critical locations. Mathematical Calculations for various factors are verified by simulations in ANSYS. Results achieved from the analysis are used in optimization of weight of the forged steel crankshaft. The optimization process results in increased fatigue strength, reduction in weight and cost of the crankshaft.**

Keywords— ANSYS14. Workbench, Aluminum and Sic alloy, Connecting rod, CATIA V5, Finite Element analysis, von misses stress.

I.INTRODUCTION

The basic function of the crankshaft is to convert reciprocating motion of a piston into rotary motion.Crankshaft continuously undergoes cyclic loading during working conditions. This gives rise to fatigue. Thus factors' affecting its service life and durability needs to be considered at the designing phase itself. Design & development of a crankshaft which can sustain cyclic loading without undergoing failure is an important issue in the manufacturing industry, target is always to manufacture a crankshaft having less weight and high fatigue strength. Lighter crankshaft gives higher efficiency & higher power output.

This Project highlights a study related to single cylinder four stroke engines. Crankshafts made up of forged steel & Aluminum Alloy (Al 5083) are modeled & analyzed to optimize the weight. Finite element analysis is carried out in stages for each crankshaft. Stresses obtained from the analysis are used for super positioning of dynamic loading conditions of crankshaft. Static & Fatigue analysis carried out for finding the possibilities of optimization of weight & hence reduction in cost.

II.SPECIFICATION OF THE PROBLEM

A. Problem Definition

Fatigue failure is a common problem experienced in cyclic loading of crankshaft. Fillet areas of crankshaft undergo stress concentration which may lead to initiation of crack and finally failure of the crankshaft

To manufacture a crankshaft of suitable material which can have adequate fatigue strength, service life and durability is a challenge to all engineers. Crankshaft having less weight increases efficiency and power output and also cost reduction is possible with the changes in the materials.

B.ObjectivesOf The Work

To perform static and fatigue analysis of a crankshaft made of Aluminum Alloy (Al 5083) Reinforced with Silicon carbide (SiC) which was fabricated by ultrasonic assisted stir casting process to compare the stress distribution, deformation and fatigue life and safety factor with structural steel. This work checks possibility of whether a forged steel crankshaft can be replaced with a developed Al alloy crankshaft.

Following objectives are proposed to follow during Project work

a. Preparing model of crankshaft of single cylinder engine single cylinder engine using CATIA
b. Analysis of model of crankshaft using ANSYS software.
c. Identification of opportunities of weight optimization.

III.LITERRAURE REVIEW

Yenetti Srinivasa Rao et al. [1] conducted study to investigate weight and cost reduction opportunities for a Transport Refrigeration Compressor Crankshaft. They carried out dynamic load analysis, static & cyclic stress analysis, fatigues analysis, torsional analysis & topology.

They concluded that weight reduction up to 12% and cost reduction up to 23% is achieved without changing life and vibration characteristics.

Jonathan Williams and Ali Fatemi [2] carried out study to compare characteristics of cast iron and forged steel crankshaft.

They concluded with the findings

1. Forged steel has higher ductility than cast iron.

2. Fatigue strength for forged steel is much higher than cast iron hence its life is also longer as compared to cast iron crankshaft.
3. Forged steel has six times longer life than cast iron crankshaft.

C.M.Balamurugan et al.[3] Carried out study to compare and evaluate fatigue performance of cast iron and forged steel crankshaft. Dynamic simulation and finite element analysis was performed to obtain values of stress at critical areas.
They concluded that forged steel crankshaft can be replaced with cast iron crankshaft for batter performance and life.

Amit Patil,et al.[4]in their paper focused on the failure of crankshaft due to fatigue which are put into service in several applications. The motivation behind their paper was to study how fatigue phenomenon leads to the failure of the crankshaft.

They summarized that Fatigue failure is the cause associated with material and hence while investigating all these different cases of crankshaft failure different metallographic tests were conducted of the failure regions through which various mechanical properties of material are evaluated and which helped to find the failure mode of the crankshaft.

K. Thriveni, et al.[5] in their study preformed static and fatigue analysis of crankshaft using ANSYS.

They compared theoretical calculations with the ANSYS result and found that maximum deformation appears at the centre of the crankpin neck surface. Maximum stress is concentrated at the fillet. Von mises stress was less thus design was safe.

C. Azoury et al. [6] carried out experimental & analytical modal analysis of crankshaft. Dynamic behavior was found using impact testing.

They concluded that experimental values and FEA values are almost same.

Ram.R.Wayzode,et al. [7] prepared model of crankshaft using software and analysed it through ANSYS. Static & fatigue analysis was carried out for obtaining results.

They concluded that forged steel is more suitable material for crankshaft as compared with cast iron.

IV.THEORETICAL CALCULATION OF CRANKSHAFT

Material type: Forged steel
Carbon: 0.35-0.45
Manganese: 0.60-0.90
Young's Modulus: 2.21 x10^5 N/mm^2
Poisson's Ratio: 0.30
Density: 7833 kg / m^3
Yield strength: 680 N/mm^2
Ultimate Tensile Strength: 850 N/mm^2

Material type I: Al- SiC (10% SiC)
Young's Modulus: 2 x 10^{5}MPa
Poisson's Ratio: 0.3
Density: 2900 kg / m^3
Yield Strength: 430 N/mm^2

After doing the calculation the load on various points of crankshaft is obtained.
The stresses obtained from static analysis would be utilized for the further fatigue analysis using S-N curve to obtain Fatigue life and fatigue Safety factor.

3.2 Theoretical Calculation of Crankshaft

Crank Pin radius	22.6
Shaft diameter	34.925
Thickness of the crank web	21.336
Bore Diameter	53.73
Length of the crank pin	43.6
Maximum Pressure	35 bar

Table 1.Specifications of Crankshaft

Design of crankshaft when the crank is at an angle of maximum bending Moment

At this position of crankshaft bending moment on the shaft is maximum and the twisting moment is zero.

Let
D = piston diameter or cylinder bore in mm.
p = Maximum intensity of pressure on the piston in N/mm^2.

The thrust in the connecting rod will be equal to the gas load on the piston (Fp). We know that gas load on the piston,

$Fp=\{(\pi/4) \times D^2 \times p \} = 7.93KN$

Distance between two bearings is given by,

$B = 2D = 2 \times 53.73 = 107.46$ mm
$B1 = b2 = b = 53.73$ mm
Due to this piston gas load (FP) acting horizontally, there will be two horizontal reactions H1 and H2 at bearings 1 and 2 respectively, such that

$H1 = H2 = Fp/2 = 3.965$ KN

Design of Crankpin
Let
d_c = Diameter of crankpin in mm,
lc = Length of the crankpin in mm,
σ_b = Allowable bending stress for the crankpin in N/mm^2.

Bending moment at the centre of the crankpin,

$Mc = H1.\ b2$ ……………1

We also know that

$Mc = \{(\pi/32) \times (d_c{}^3) \times \sigma_b\}$ ……..2

from equations 1 & 2

Diameter of crankpin
$d_c = 45.2$ mm.$\sigma_b = 23.4$ N/mm^2.

The length of the crankpin is given by
$lc = \{Fp/(d_{c\,x} P_b)\} = 43.6$ mm

Where $P_b = 4.026$ N/mm^2 *= permissible bearing pressure in* N/mm^2.

Design of left hand Crank web

The crank web is designed for eccentric loading. There will be two stresses acting on the crank web, one is direct compressive stress and the other is bending stress due to piston gas load (Fp).

Let
w = Width of crank web
t = thickness of the crank web

The width of crank web (w) is taken as

$w = 1.125d_c + 12.7$ mm $= 63.55$ mm

The thickness (t) of the crank web is given empirically as,
$t = 0.39 \times D = 21.336$

We know that maximum bending moment on the crank web,

$M = H1\{b2-(lc/2)-(t/2)\} = 168.86 \times 10^3$Nmm

And Section Modulus is,

$Z = [(w \times t^2)/6] = 4821.59$ mm^3

Bending stress bending stress induced in the crank web is,

$\sigma_{b\,=} M/Z = 35.02$ N/mm^2

Here, induced bending stress is less than the allowable bending stress which is (143**N/mm^2). Hence the design is safe***. *Considering factor of safety 3.*

Design of right hand crank web:
The dimensions of the right hand crank web (i.e. thickness and width) are made equal to left hand crank web from the balancing point of view.

Design of shaft
Let
ds = Diameter of shaft in mm.

We know that bending moment on shaft is,

$BM = Fp \times c = 238.05 \times 10^3$Nmm

Where, c =clearance = 30 mm (assuming)
BM =

And twisting moment on shaft is,
$TM = Fp \times r$
Where, r = Offset of Crankpin
= stroke /2 =33.58 assuming stroke length to be 25 % more than bore diameter.

$TM = 266.45 \times 10^3$Nmm

Equivalent moment on shaft is given by,
$Ms = (BM^2 + TM^2)^{½} = 357300.16$ Nmm

Now, we know that

$Ms = \{(\pi/32) \times (d_s{}^3) \times \sigma_b\}$

Hence, shaft diameter ds = given 34.925 mm
$\sigma_b = 85.43$ N/mm^2.

Design of crank pin against fatigue loading

According to distortion energy theory, the Von-Misses stress induced in the crank-pin is,

$Mev=((Kb+Mc)^2)+ ¾(KtxTc)^2)^{1/2}$

Where
Kb = combined fatigue and shock factor for bending = 2 (Assume)
Kt = combined fatigue and shock factor for torsion. = 1.5 (Assume)

Putting the values in above equation we get

$Mev = ((Kb+Mc)^2)+ ¾(Kt \times Tc)^2)^{1/2} = 159.592 \times 10^3$Nmm.

Also we know that,

$Mev = \{(\pi/32) \times (d_s{}^3) \times \sigma_v\}$

Von mises stress

$\sigma v = 38.15$ N/mm²
Now
Twisting moment

$Tev = ((Kb+Mc)^2)+ (Kt \times Tc)^2)^{1/2} = 357 \times 10^3$ N mm.

Shear stress:
$\tau e = (\pi/16) \times dc^3 \times \tau$
$\tau = 19.69$N/mm^2.

A.FEA of Steel and composite material.

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A 3D model of a crankshaft is used for analysis in ANSYS14 workbench. The loading conditions are assumed to be static. Analysis is done with pressure loads applied at the piston end and at the fixed crank end..

B. Material Properties

Property	Steel C45	Al-SiCp(10% SiCp)
Young's modulus, MPa	$2.05x10^5$	$2 x10^5$
Poisson ratio	0.29	0.3
Yield Strength, MPa	360	430
Density, kg/m^3	7850	2900

C. Solid Modelling of Steel and composite material

Catia V5 is used for Modelling of crankshaft.

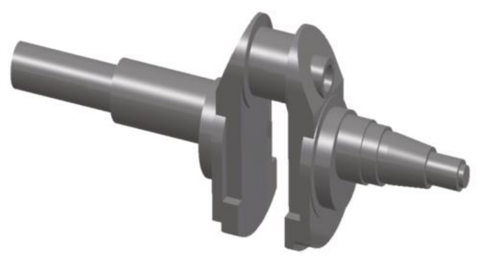

Fig 1-Solid Modelling of Crankshaft

D.FEM Analysis

The element selected is 10 tetrahedral. Finite element analysis is carried out on carbon steel crankshaft as well as on aluminum alloy reinforced with SiC particles. The material properties for Al alloy composite were taken from the reference papers. From the analysis the equivalent stress (Von-mises stress), equivalent strain and total deformation are determined.No.of nodes and elements generated are 88912 and 41058 Respectively

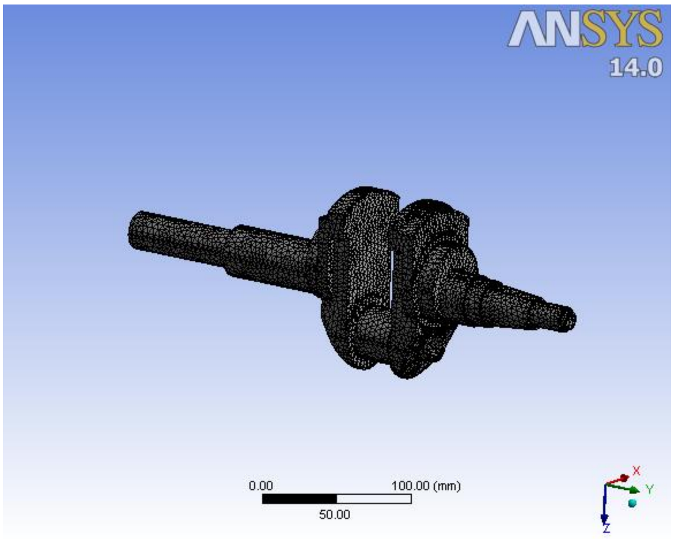

Fig.2Meshed Model of Crankshaft

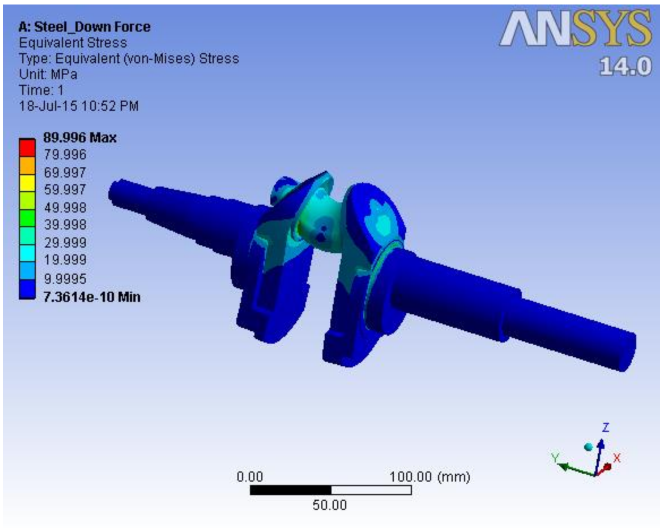

Fig. 3.VonMises Stress of Structural steel crankshaft

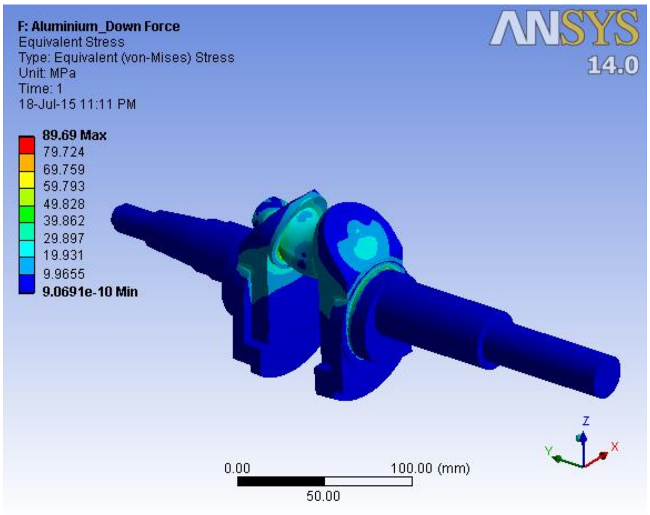

Fig. .4Von Mise's Stress of Al 5083 alloy Composite Crankshaft

Fig. 3 and Fig.4 shows Min Equivalent stress as .00017052 MPa and maximum 143.84MPa and minimum equivalent stress as .00001799Pa and maximum 144.02MPa for a crankshaft made of Structural steel and Al alloy composite respectively.

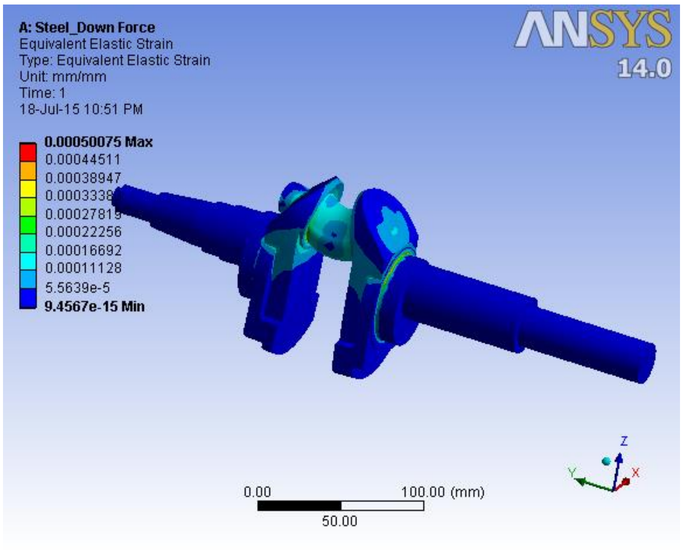

Fig. 5 Equivalent elastic strain of Structural steel crankshaft

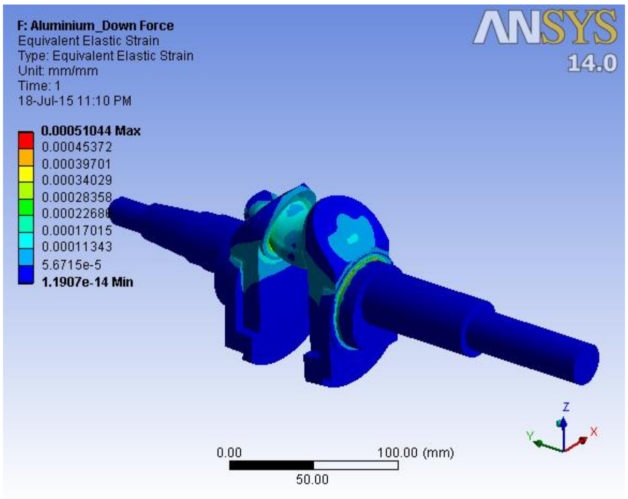

Fig. 6 Equivalent elastic strain of Al 5083 alloy Composite Crankshaft

Fig. 6 and Fig. 7 shows Min Equivalent elastic strain as 1.137 8X 10^{-9} mm/mm and1.156 X 10^{-9} mm/mm and max Equivalent elastic strain as .00079mm/mm and 0.00076 mm/mm for a cranksaft made of Structural steel and Al alloy composite respectively.

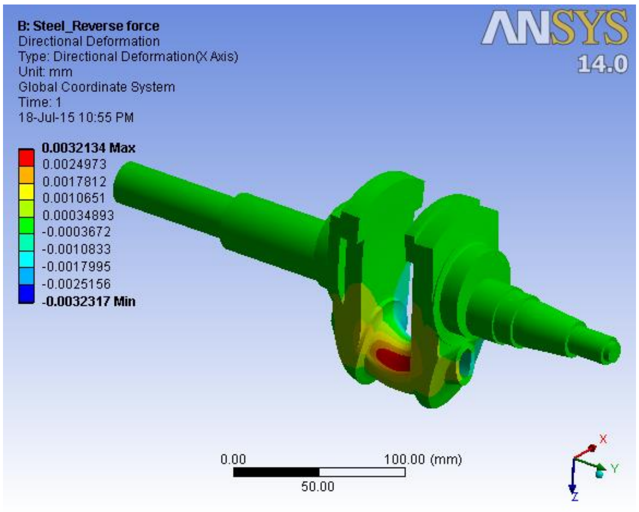

Fig.7 Total Deformation of Structure steel Crankshaft

Fig. 7 and Fig. 8 shows Min Total deformation as 0 for both and max Total deformation as 0.032185 mm and 0.032199 mm for a crankshaft made of Structural steel and Al alloy composite respectively.

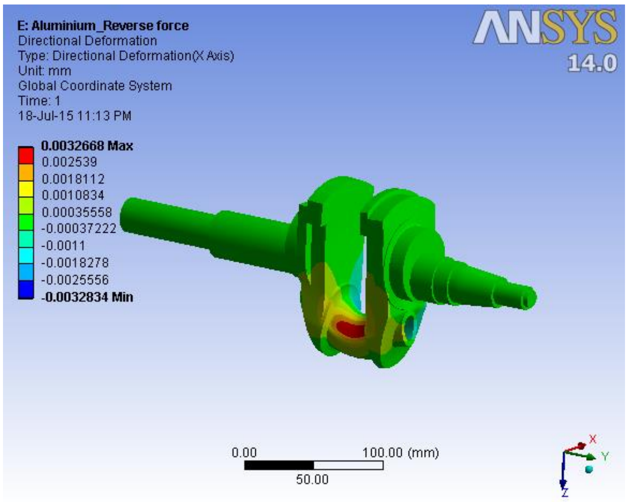

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Fig.8 Total Deformation of Al 5083 Composite Crankshaft

V.FATIGUE LIFE PREDICTION

The Stress Life (SxN) theory was employed to evaluate the crankshaft fatigue life.

Calculation for Factor of Safety, Weight, Stiffness, Life for forged steel Crankshaft

As yield stress is considered as a criteria of design , calculations are done based on Soderbeg's equation.

f.s = factor of safety
σm = mean stress
σy = yield stress
σv = variable stress
σe = endurance stress

$1/f.s = \sigma m/\sigma y + \sigma v/\sigma e$

i) Safety Factor For Steel C 45

$\sigma max = 143.84$ $\sigma min = 0.00017052$

$\sigma m = \sigma max + \sigma min/2 = 71.92$

$\sigma y = 360$ Mpa

$\sigma v = \sigma max - \sigma min/2 = 71.73$

$\sigma e = 0.6\times 360 = 216$

$1/f.s = 0.531 = 1.88$

Factor of safety [F.S] = 1.88

ii) Calculation for Weight and Stiffness For carbon Steel (c45):

Density of steel = 7.850×10^{-6} kg/mm^3

Volume = Area x length=378.6x 97.6=37829.8mm^3

Deformation = 0.032985 mm

Weight of forged steel = volume ×density

= 37829.8x 7.85 x10^{-6}

= 0.29 kg

= 0.29×9.81 = **2.91 N**

Stiffness = weight/deformation

= 0.29/0.032985=8.79 kg/mm=87.9 N/mm

iii) Fatigue Calculation of life For Carbon Steel

Result for fatigue of connecting rod:

$N=1000(sf/0.9\sigma u)^{3/\log}(\sigma e'/0.9\times\sigma u)$

Where,

N = No. of cycles

σe = Endurance Limit

σu = Ultimate Tensile Stress

$\sigma e'$ = Endurance limit for variable axial stress

ka = Load correction factor for reversed axial load = 0.8

ksr = Surface finish factor = 1.2

ksz = Size factor = 1

$\sigma e' = \sigma e\times ka\times ksr\times ksz$

$sf = f.s.\sigma v/(1-f.s.\sigma m/)$

σu=750 Mpa

$\sigma e' = \sigma e\times ka\times ksr\times ksz$

= 216×0.8×1.2×1

= 207.36Mpa

$sf = f.s.\sigma v/(1-f.s.\sigma m/\sigma u)$

= 1.88x 71.73/(1-1.88x 71.92/750)

= 164.5 MPa

$N=1000(sf/0.9\sigma u)^{3/\log}(\sigma e'/0.9\times\sigma u)$

$= 1000(164.5/0.9\times 750)^{3/\log}(207.36/0.9\times 750)$

= 1.3 x 10^{6}cycles

3.5 Calculation for Factor of Safety, Weight, Stiffness, Life for of AL SIC (10% SIC)

i) Safety factor for AlSiC

$\sigma max = 144.02$ $\sigma min = 0.0001799$

$\sigma m = \sigma max + \sigma min/2 = 72.01$

$\sigma y = 430$Mpa

$\sigma v = \sigma max - \sigma min/2 = 71.99$

$\sigma e = 0.6\times 430 = 258$ Mpa

$1/f.s = .446$

Factor of safety [F.S] = 2.25

ii)Calculation for Weight and Stiffness For Al SiC

Density of AlSiC = 2.9 x 10^{-6} kg/mm^3

Volume = Area x length=378.6x 97.6=37829.8mm^3

Deformation = 0.032199 mm

Weight of forged steel = volume ×density

= 37829.8x 2.9x10^{-6}

= 0.109kg

= 0.29×9.81 = **1 N**

Stiffness = weight/deformation

= 0.109/0.032199 =3.385 kg/mm=33.85 N/mm

iii) Fatigue calculation for Life forAlSiC

Result for fatigue of connecting rod:

$N=1000(sf/0.9\sigma u)^{3/\log}(\sigma e'/0.9\times\sigma u)$

$\sigma u = \sigma e \times 2 = 516$ MPa

$\sigma e' = \sigma e\times ka\times ksr\times ksz$

= 258×0.8×1.2×1

= 247.68Mpa

$sf = f.s.\sigma v/(1-f.s.\sigma m/\sigma u)$

= 2.25x 71.99/(1-2.25x 72.01/516)

= 236.11MPa

$N=1000(sf/0.9\sigma u)^{3/\log}(\sigma e'/0.9\times\sigma u)$

$= 1000(236.11/0.9\times 516)^{3/\log}(247.68/0.9\times 516)$

= 1.83x 10^{6}cycles

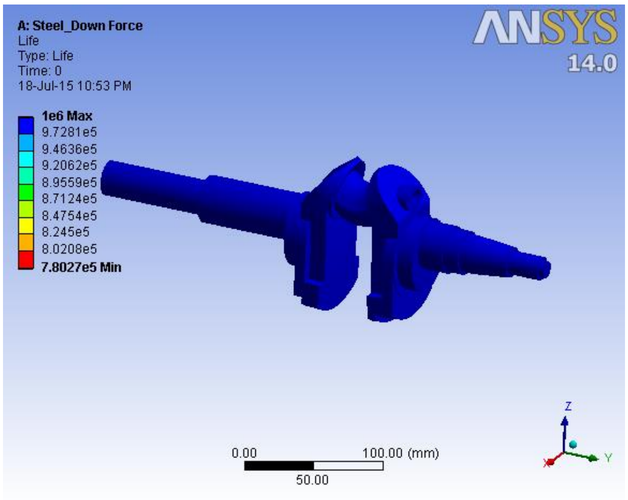

Fig.9 Life for steel

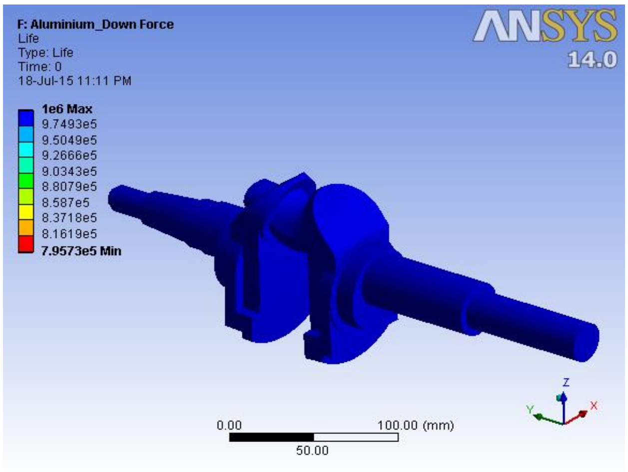

Fig.10 Life for Al sic

After calculating the alternate and mean stresses, we can plot the Soderberg diagram. With the alternate and mean stresses, and using the Modified Goodman diagram for the crankshaft material, it is possible to evaluate the fatigue factors.

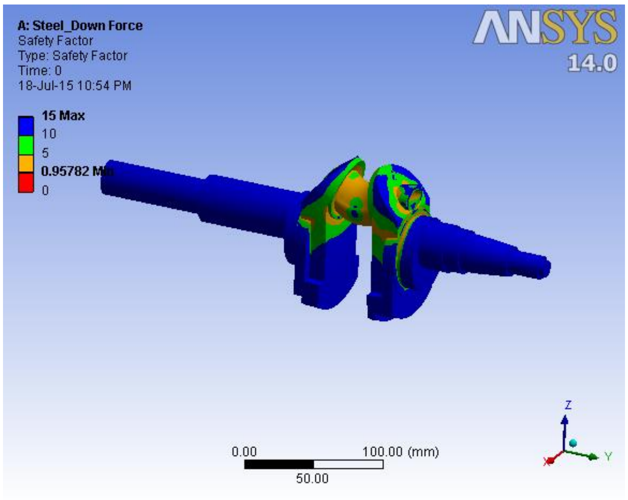

Fig.11 Safety Factor for Steel

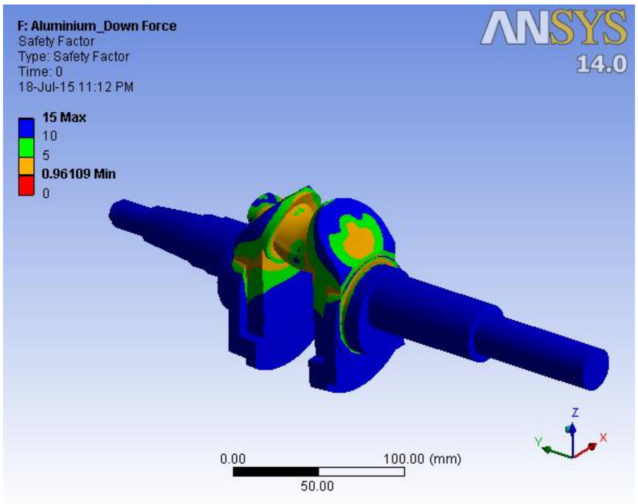

Fig12.safety factor for Al Sic

VI .RESULT

Parameter	**C45**	**Al SiC**
Von Mises stress(Mpa)Ansys	143.84	144.02
Total deformation (mm)(ansys)	.032985	.032199
Equivalent strain	.00079	.00076
Safety factor(Ansys)	1.0094 to15	0.9 to 15
Life (Cyckes)(ansys)	$1x10^6$	$1x10^8$
Bending stress (analytical)	103	36.88
Life (analytical)	$1x10^6$	$1.8\ x10^6$
Safety factor (analytical)	1.88	2.15

VII. CONCLUSION

The aluminum composite crankshaft has light weight about 1/3 of steel . Equivalent elastic strain , total deformation, and stresses are approximately equal in Al alloy composite crankshaft and structural steel crankshaft but it comes under the permissible tolerance limit. The maximum life value is more in an aluminum alloy crankshaft as compared to the crankshaft made of steel. Thus a steel crankshaft can be replaced with a developed Al alloy crankshaft.

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REFERENCES

[1] Yenetti Srinivasa Rao, Gouthaman Murali, Udupa S. Srinivasa," Cost and Weight Optimization of a Transport Refrigeration Compressor Crankshaft" International Compressor Engineering Conference 2008 Purdue University.
[2] Jonathan Williams and Ali Fatemi," Fatigue Performance of Forged Steel and Ductile Cast Iron Crankshafts", SAE International 2007-01-1001.
[3] C.M.Balamurugan, R.Krishnaraj, Dr.M.Sakthivel, K.Kanthavel, DeepanMarudachalam M.G, R.Palani ," Computer Aided Modeling and Optimization of Crankshaft "International Journal of Scientific & Engineering Research Volume 2, Issue 8, August-2011 1 ISSN 2229-5518 .
[4] Bhumesh J. Bagde, Laukik P. Raut ," Finite Element Analysis of Single Cylinder Engine Crankshaft "International Journal of Advances in Engineering & Technology, May 2013. ©IJAET ISSN: 2231-1963
[5] C. Azoury, A. Kallassy, B. Combes, I. Moukarzel, R. Boudet," Experimental and Analytical Modal Analysis of a Crankshaft "IOSR Journal of Engineering Apr. 2012, Vol. 2(4) pp: 674-684
[6] K. Thriveni , Dr. B. Jaya Chandraiah , "Modeling and Analysis of the Crankshaft Using Ansys Software "International Journal of Computational Engineering Research||Vol, 03||Issue, 5||
[7] Ram.R.Wayzode, P. G. Mehar, V. N. Mujbaile, "Modelling, Analysis And Comparison Of Forged Steel And Cast Iron Crankshaft "Indian Streams Research Journal Volume 2, Issue. 7, Aug 2012, ISSN:-2230-7850