Analysis of 4-Cyliner Diesel Engine Crankshaft by Using Aluminum Alloy

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Abstract- Crankshaft is a key element of in an engine assembly. Crankshaft consists of two web sections and one crankpin. Crankshaft converts the reciprocating displacement of the piston to a rotary motion with a four link mechanism. This paper is related to design and finite element analysis of crankshaft of 4 cylinder diesel engine of heavy vehicle like truck. Engine capacity is 3785.1cc. The finite element analysis in ANSYS software by using three materials based on their composition viz. FG260, FG300 and Aluminum based composite material. The parameter like von misses stress, deformation; maximum and minimum principal stress & strain were obtained from analysis software. The results of Finite element shows that the Aluminum based composite material is best material among all.

Keywords— **Crankshaft, Finite Element Analysis, Creo 2.0, ANSYS 14.5**

I. Introduction

In various mechanical engineering applications the most widely used machine elements is Shaft. The crankshaft, impeller shaft, propeller shafts, camshafts etc. use shaft. Crankshaft is one of the most important moving parts consisting of two web sections and one crankpin that convert the piston reciprocating displacement to a rotary motion with a four link mechanism. Crankshaft experiences large forces from gas combustion. This force is applied to the top of the piston and since the connecting rod connects the piston to the crank shaft, the force will be transmitted to the crankshaft. The magnitude of the forces depends on many factors which consist of crank radius, connecting rod dimensions, and weight of the connecting rod, piston, piston rings, and pin. Since the crankshaft experiences a large number of load cycles during its service life, fatigue performance and durability of this component has to be considered in the design process. So the life of internal combustion engine and reliability depend on the strength of the crankshaft largely and as the engine runs, the power impulses hit the crankshaft in one place and then another. Failures of shafts not only result in replacement cost, but also in process downtime. The one of the most common causes of shaft failure is fatigue fracture. Fatigue failures are important considerations in mechanical designs. To determine the stress, geometry and dimensions Analysis, evaluations and engineering principles are employed in the design. To ensure the life of engine, strength calculation of crankshaft becomes a key factor. Traditionally beam and space frame model were used to calculate the stress of crankshaft but in these models the number of node is limited. With the development of computer, more and more design of crankshaft has been utilized finite element method (FME) to calculate the stress of crankshaft.

In this paper we create a three dimensional model of crankshaft in Creo 2.0 modeling software. Static structural analysis done in Ansys 14.5 (FEA) software by using different material based on their composition. Crankshafts are manufactured using cast iron and forged steel. So the crankshaft material is replaced by material such as FG260, FG300, and Aluminum based composite material. We find the deformation, von misses stress and maximum and minimum principal stresses and strain in structural analysis. The knowledge gained from this project is to be able to understand the steps needed in structural analysis for crankshaft by using FEA method. Crankshaft could be studied on the various areas such as material improvement on the crankshaft, crankpin.

II. Literature Review

Wei Li, Qing Yan, and Jianhua Xue [1] this paper analyzed through the chemical composition, mechanical properties, macroscopic feature, microscopic structure and theoretical calculation methods. The analysis results show that the crankshaft which has obvious fatigue crack belongs to fatigue fracture. The crankshaft fatigue fracture was only attributed to the initiation and propagation of the fatigue cracks on the lubrication hole under cyclic bending and torsion.

M. Fonte, P. Duarte, V. Anes, M. Freitas, L. Reis [2] The fatigue strength and its correct assessment play an important role in design and maintenance of marine crankshafts to obtain operational safety and reliability. Crankshafts are under alternating bending on crankpins and rotating bending combined with torsion on main journals, which mostly are responsible for fatigue failure. The commercial management success substantially depends on the main engine in service and of its design crankshaft, in particular. The crankshaft design strictly follows the rules of classification societies. The present study provides an overview on the assessment of fatigue life of marine engine crankshafts and its maintenance taking into account the design improving in the last decades, considering that accurate estimation of fatigue life is very important to ensure safety of components and its reliability. An example of a semi-built crankshaft failure is also presented and the probable root case of damage, and at the end some final remarks are presented.

M. Fonte, V. Anes, P. Duarte, L. Reis, and M. Freitas [3] this paper reports a failure mode analysis of a boxer diesel engine crankshaft. Crankshafts are components which experiment severe and complex dynamic loadings due to rotating bending combined with torsion on main journals and alternating bending on crankpins. High level stresses appear on critical areas like web fillets, as well as the effect of centrifugal forces and vibrations. Since the fatigue fracture near the crankpin-web fillet regions is one of the primary failure mechanisms of automotive crankshafts, designers and researchers have done the best for improving its fatigue strength. The present failure has occurred at approximately 2000 manufactured engines, and after about 95,000 km in service. The aim of this work is to investigate the damage root cause and understand the mechanism which led to the catastrophic failure. Recommendations for improving the engine design are also presented.

M. Fonte, P. Duarte, L. Reis, M. Freitas, V. Infante [4] in this paper investigation is carried on two damaged crankshafts of single cylinder diesel engines used in agricultural services for several purposes. Recurrent damages of these crankshafts type have happened after approximately 100 h in service. The root cause never was imputed to the manufacturer. The fatigue design and an accurate prediction of fatigue life are of primordial importance to insure the safety of these components and its reliability. This study firstly presents a short review on fatigue power shafts for supporting the failure mode analysis, which can lead to determine the root cause of failure. The material of these damaged crankshafts has the same chemical composition to others found where the same type of fracture occurred at least ten years ago. A finite element analysis was also carried out in order to find the critical zones where high stress concentrations are present. Results showed a clear failure by fatigue under low stress and high cyclic fatigue on crankpins.

B. Kareem [5] in this study, mechanical crankshaft failures for automobiles are evaluated based on experts' opinion. This was done using data obtained using techniques based on oral interviews and questionnaire administration on mechanical failure of crankshafts from the experts working in the areas of automobile maintenance and crankshafts reconditioning. The data collected were analyzed using statistical methods based on probability. With this technique, probability of failure for each category of automobiles namely private, commercial cars and buses were evaluated. The results obtained show that private cars had lowest failure rate at the initial stage while commercial buses had the highest failure rate. At later periods all categories of automobile crankshafts considered had their failure rates converged steadily with stable reliability. Application of 6-sigma continuous improvement tool to the process indicated a further reliability improvement through improved oil lubrication system, especially in the thrust bearing. This showed that increased enlightenment campaign among the various stakeholders in automobile industries will improve on the choice of reliable mechanical crankshafts.

Xiaoping Chen, Xiaoli Yu, Rufu Hu, Jianfen [6] Crankshaft fatigue problem has long been a headache and frequent phenomenon in combustion engine which attracts various efforts especially including fundamental fatigue experimental data. In this paper, the rational experimental method is employed to study the crankshaft fatigue phenomenon based on a customized experiment platform, mimicking the real-world crankshaft working condition physically. Then, based on the experiment data, the statistical regression analysis of eight commonly used hypothesis distributions is conducted. The degrees of fitting effects of the chosen statistical model are evaluated individually. Results show that the three-parameter Weibull distribution model fits the data best which may be used as the fundamental model in future analysis. This study provides a solid foundation for better understanding the mechanism of crankshaft fatigue phenomenon.

A. Ktari, N. Haddar, H.F. Ayedi [7] a failure investigation has been conducted on three cases of failed diesel engine crankshafts used in train and made up of forged carbon steel. The chemical composition and the mechanical properties of the crankshafts material including tensile properties, micro-hardness and toughness were evaluated. The crankshafts examination shows that all failures occurred after a fatigue process. The failure zones comprise the fractured surfaces observation, show the presence of beach marks with semi-elliptical shape surrounding the fracture origins indicate its progressive growth character. The cracks initiation can occur as a result of mechanical and thermal fatigue loads, due to the high stress concentration on fillet radius and the unusual friction between journals and bearings, respectively. Nevertheless, the cracks propagation was only attributed to the mechanical fatigue produced under cyclic bending and torsion loadings.

J.A. Becerra, F.J. Jimenez, M. Torres, D.T. Sanchez, E. Carvajal [8] Analysis of the compressor revealed that the torsional dynamic controls the stress in the crankshaft and that the influence of the gas forces on the crankshaft stress is only minor. The appearance of the fracture was consistent with a torque overload. The maximum stress in the crankshaft, as obtained from the FEM and lumped model was located in the keyway, and this location belongs to the fracture surface in most of the broken crankshafts. The influence of the stress concentration factor imposed by this geometry is therefore very high. The compressor speed range was found to continuously cross the three lower resonance frequencies. The exhaust valve of the compressor should be redesigned in order to reduce gas forces, power consumption and pressure drop.

Gustavo Rocha da Silva Santos, Guilherme Vinícuis França dos Santos [9] this work presents the design of a crankshaft for a lightweight mono-cylinder spark-ignition four-stroke internal combustion engine using topology optimization. The topology optimization method implies the use of FE analysis combined with an optimization algorithm to find the optimum mass distribution of the crankshaft to minimize the component weight while satisfying manufacturing and maximum stress (yield strength) constraints. In addition, the application of this method allows control over the crankshaft natural frequencies by avoiding a spectrum around a specified Eigen frequency where no resonance occurs. This leads to a reduction of its torsional vibration, which is the leading cause of crankshaft failure. This methodology modifies the traditional mechanical design by placing structural analysis before the CAD design

M. Fonte, M. de Freitas [10] case study of a catastrophic failure of a web marine crankshaft and a failure analysis under bending and torsion applied to crankshafts are presented. A microscopy (eye seen) observation showed that the crack initiation started on the fillet of the crankpin by rotary bending and the propagation was a combination of cyclic bending and steady torsion. The crack front profile approximately adopts a semi-elliptical shape with some distortion due to torsion and this study is supported by a previous research work already published by the authors. The number of cycles from crack initiation to final failure of this crankshaft was achieved by recording of the main engine operation on board, taking into account the beach marks left on the fatigue crack surface.

III. PROBLEM STATEMENT

- Reduce the weight of the crankshaft without compromising on the properties of the material. As the crankshafts are made from ferrous material, it becomes bulky and heavy weight.

IV. OBJECTIVES

- Design of the crankshaft for the new proposed material and create 3D model using Creo 2.0.
- To study von misses stresses, deformations and maximum principal stresses.

V. METHODOLOGY

- Literature Survey of various research papers.
- Theoretical calculation of four cylinder diesel engine crankshaft.
- Solid model of four cylinder diesel engine crankshaft.
- Meshing of 3-D entity of crankshaft.
- Finite element analysis in ANSYS14.5
- Compare theoretical and FEA.
- Conclusion.

VI. DESIGN PROCEDURE

a) Design of Crankshaft

For the theoretical calculation of crankshaft we will consider the configuration of heavy vehicle like truck 4-cylinder diesel engine to calculate the theoretical static result:

Specification of 4 Cylinder Diesel Engine

Sr. No.	Type	4 Cylinder Diesel Engine (Value)
1	Capacity of engine	3785.1cc
2	Number of cylinder	4
3	Bore × Stroke	97mm × 128 mm
4	Compression Ratio	18:1
5	Maximum power	100hp @ 2300 rpm.
6	Maximum Torque	475 Nm @ 2300 rpm.
7	Max. gas pressure	25 bar or 2.5 N/mm^2.

When the crank has turned through 35$_o$ from the top dead center, the pressure on the piston is 1 N/m m$_2$ and the torque on the crank is maximum. The ratio of the connecting road length to the crank radius is 4. Assuming suitable data wherever is required. We will design the crankshaft for two position of the crank.

i) Design of crankshaft when the crank is at top dead center of piston where maximum bending moment occurs.

At this position of the crank, the maximum gas pressure on the piston will transmit maximum force on the crankpin in the plane of the crank causing only bending of the shaft. The crankpin as well as ends of the crankshaft will be only subjected to bending moment. Thus, when the crank is at the top dead center, the 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 (F_p).

We know that piston gas load,

$$Fp = \frac{\pi}{4} \times D^2 \times p = \frac{\pi}{4} \times (97)^2 \times 14$$
$$= 103.457kN$$

Assume that the distance (b) between the bearing 1 and 2 is equal to twice the piston diameter (D)

$$b_1 = b_2 = \frac{194}{2} = 97mm$$

We Know that due to the piston gas load, there will be two horizontal reactions H1 and H2 at bearing 1 and 2 respectively, such that

$$H_1 = \frac{F_p \times b_1}{b} = \frac{103.457 \times 97}{194} = 51.7286kN$$

$$H_2 = \frac{F_p \times b_2}{b} = \frac{103.457 \times 97}{194} = 51.7286kN$$

And assume that the length of the main bearing to be equal, i.e. $c_1 = c_2 = c/2$.

We know that due to the weight of the flywheel acting downwards, there will be two vertical reactions V_2 and V_3 at bearings 2 and 3 respectively, such that

$$V_2 = \frac{W \times c_1}{c} = \frac{W \times c/2}{c} = \frac{W}{2} = \frac{0.6}{2} = 0.3kN$$

$$V_3 = \frac{W \times c_2}{c} = \frac{W \times c/2}{c} = \frac{W}{2} = \frac{0.6}{2} = 0.3kN$$

And due to the resultant belt tension $(T_1 + T_2)$ acting horizontally, there will be two horizontal reaction $H_2^{'}$ and $H_3^{'}$ respectively, such that

$$H_2^{'} = \frac{(T_1 + T_2)c_1}{c} = \frac{(T_1 + T_2)c/2}{c} = \frac{(T_1 + T_2)}{2}$$
$$= 1/2 = 0.5kN$$

$$H_3^{'} = \frac{(T_1 + T_2)c_2}{c} = \frac{(T_1 + T_2)c/2}{c} = \frac{(T_1 + T_2)}{2}$$
$$= 1/2 = 0.5kN$$

Now the various parts of the crankshaft are designed such as:

(a) Design of crankpin:-

Let, $d_c =$ Diameter of the crankpin in mm;

$l_c = \frac{F_p}{d_c \times P_b}$ =Length of the crankpin in mm; and

$\sigma_b =$ Allowable bending stress for the crankpin, it may assumed that as 75 MPa or N/mm^2

We know that the bending moment at the center of the crankpin (M_c):-

$M_c = H_1 \times b_2 = 51.7286 \times 97 = 5017.67 kN - mm$

We also know that, $M_c = \frac{\pi}{32}(d_c)^3 \sigma_b$

Therefore, for solving the above equation we get,

$d_c = 90mm$

Length of the crankpin is (l_c) = 0.8d$_c$ = 0.8 x 90= 72mm

(b) 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 compressive stress & other one is bending stress.

We know that thickness of thickness of the crank web (t):-

$t = 0.7d_c$

$t = 63mm$

And width of the crank web (w):- $w = 1.14d_c$

$w = 103mm$

We know that maximum bending moment on the crank web,

$$M = H_1\left(b_2 - \frac{l_c}{2} - \frac{t_c}{2}\right)$$

$$M = 51.7286\left(97 - \frac{72}{2} - \frac{63}{2}\right)$$

$$M = 1525.99 kN - mm$$

Section modulus, $Z = \frac{1}{6} \times w \times t^2 = 68134.5 mm.^2$

Bending stress, $\sigma_b = \frac{M}{Z} = 22.39 N / mm^2$

We know that direct compressive stress on the crank web,

$$\sigma_c = \frac{H_1}{w \times t} = 51.72 \times 10^3 / (103 \times 63)$$

$$\sigma_c = 7.97 N / mm^2$$

$\therefore$ Total stress on the crank web (σ)

$$\sigma = \sigma_b + \sigma_c = 22.39 + 7.97 = 30.36 N / mm^2$$

Since, the total stress on the crank web is less than the allowable bending stress of the 75 MPa, therefore the design of the left hand crank web is safe.

(c) Design of right hand crank web:-

From the balancing point of view, the dimensions of the right hand crank web are made equal to the dimensions of the left hand crank web is safe.

ii) Design of the crankshaft when the Crank is at an angle of maximum twisting moment

Force on the Piston (F_P) = Area of the bore x Maximum combustion pressure.

$$Fp = \frac{\pi}{4} \times D^2 \times p' = \frac{\pi}{4}(97)^2 \times 1 = 7389.81N$$

In order to find the thrust in the connecting rod (F_Q), we should first find out the angle of inclination of the connecting rod with the line of stroke (ϕ).

We know that, $\sin\phi = \frac{\sin\theta}{l/r} = \frac{\sin 35^0}{5} = 0.1147$

$$\therefore \phi = \sin^{-1}(0.1147) = 6.58°$$

We know that thrust in connecting rod,

$$F_Q = \frac{F_p}{\cos\phi} = \frac{7389.81}{\cos 6.58^0} = 7438.81N$$

Thrust on the crankshaft can be split into tangential component and the radial component.

Tangential force acting on the crankshaft (F_T):-

$$F_T = F_Q \sin(\theta + \phi)$$

$$F_T = 4936.87N$$

And Radial force,

$$F_R = F_Q \cos(\theta + \phi)$$

$$F_R = 5564.45N$$

Due to tangential force (F_T), there will be two reactions at the bearings 1 and 2, such that

$$H_{(T1)} = \frac{F_T \times b_1}{b} = 4936.87 / 2$$

$$H_{(T1)} = 2468.43N$$

$$H_{(T2)} = \frac{F_T \times b_2}{b} = 4936.87 / 2$$

$$H_{(T2)} = 2468.43N$$

Due to radial Force is given by, Reaction at bearing (a & b)

$$H_{(R1)} = \frac{F_R \times b_1}{b} = 5564.45 / 2$$

$$H_{(R1)} = 2782.225N$$

$$H_{(R2)} = \frac{F_R \times b_2}{b} = 5564.45 / 2$$

$$H_{(R2)} = 2782.225N$$

VII. Analysis By Ansys Software

Building an accurate and reliable calculating model is one of the key steps of analysis with finite element analysis. The model was created using Creo 2.0 software. The following are the steps involved in modelling of the crankshaft.

In the 2D drawing of original of crankshaft crankpin diameter is 90 mm, journal diameter is 60 mm, crank length 72 mm & web thickness (Left and Right Hand) 63 mm and width (Left and Right Hand) 103 mm is respectively shown in Figure.

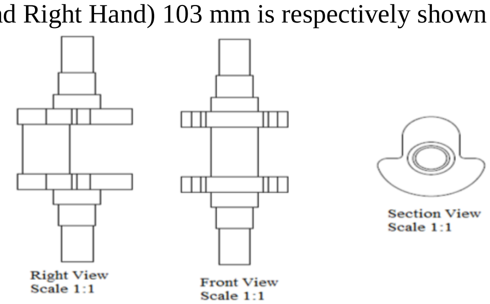

Figure 7.1 Sectional view of crankshaft

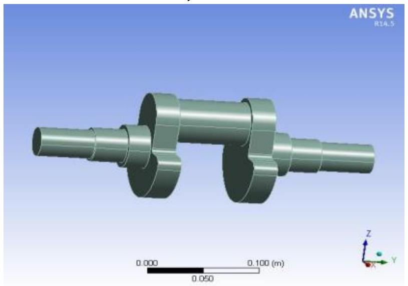

Figure 7.2 3D modeling original crankshaft in Creo 2.0

a) Apply Load and boundary condition

Boundary conditions play an important role in finite element analysis. Crankshaft is constricts with a ball bearing from one side and with a journal on other side. The ball bearing is press fit to the crankshaft and does not allow the crankshaft to have any motion other than rotation about its main axis. Since 180 degrees of the surface facing the load 104kN in downward z direction constricts the motion of the crankshaft.

b) Finite Element Analysis of Material

Analysis of crankshaft- FG260 Material

Material: - FG 260
Poisson ratio: - 0.26
Density: - 7197 kg/m3
Young's Modules: - 198 GPa.
Chemical Composition of FG 260Material

Sample Identification	Chemical Composition %								
	C	Si	Mn	S	P	Cr	Ni	Mo	Fe
FG 260	3.39	1.8	0.63	0.08	0.088	--	--	--	Bal

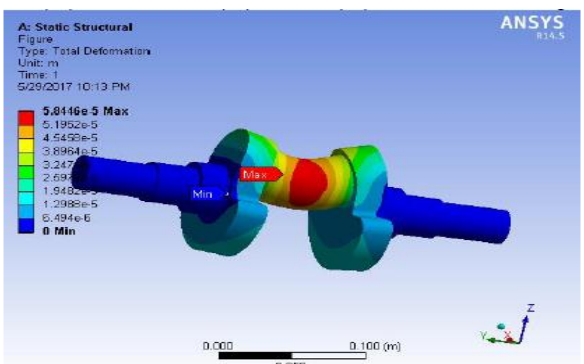

Figure 6.3 Deformation of (FG 260 material) crankshaft

From the above result for material FG 260 maximum deformation occurs at the center. Maximum deformation is noted as 0.05844mm.

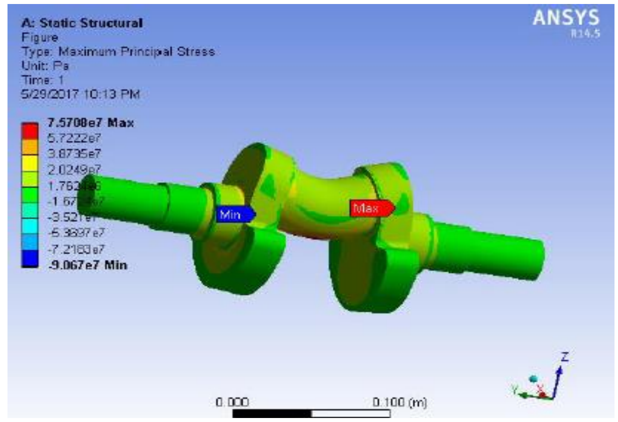

Figure 6.4 Maximum Principal Stress (FG 260 Material) of crankshaft

Maximum Principal Stress for the given condition is observed from -9.067e^7 to 7.57e^7 for FG260 material.

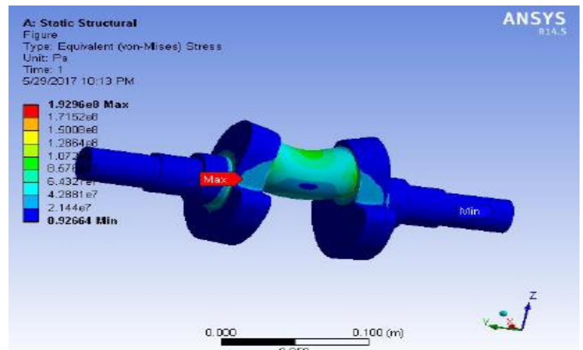

Figure 6.5 Von Misses Stress distribution (FG 260 Material) of Crankshaft.

For the given load conditions the range of Equivalent Von-Mises stress is observed from 0.92664 Pa to 1.92e8 Pa.

Analysis of crankshaft- FG300 Material

Material: - FG300
Poisson ratio: - 0.26
Density: - 7200 kg/m^3
Young's Modules: - 205 Gpa
Chemical Composition of FG 300Material

Sample Identification	Chemical Composition %								
	C	Si	Mn	S	P	Cr	Ni	Mo	Fe
FG 260	3.21	1.82	0.81	0.09	0.07	--	--	--	Bal

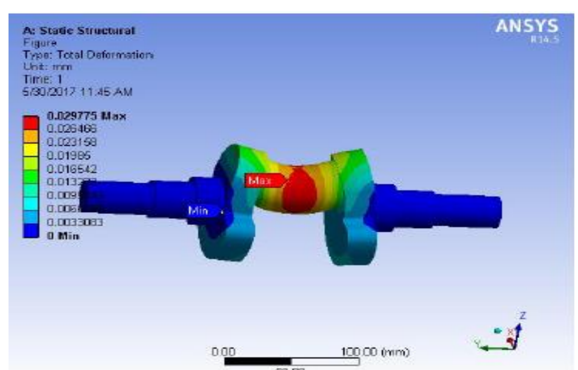

Figure 6.6 Deformation of (FG 300 material) crankshaft

From the above result for material FG 300 maximum deformation occurs at the center. Maximum deformation is noted as 0.029775mm.

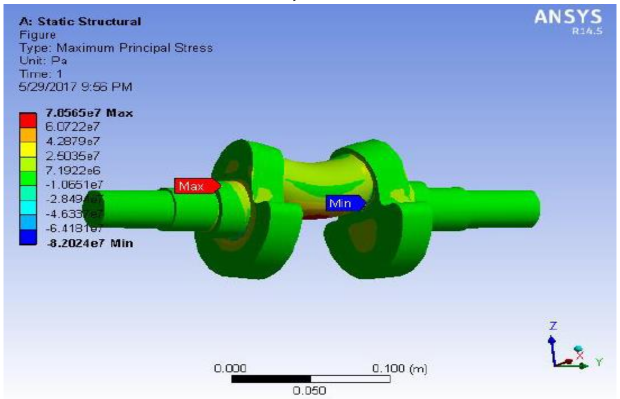

Figure 6.7 Maximum Principal Stress (FG 300 Material) of Crankshaft

Maximum Principal Stress for the given condition is observed from -8.202e^7 to 7.85e^7 for FG300 material.

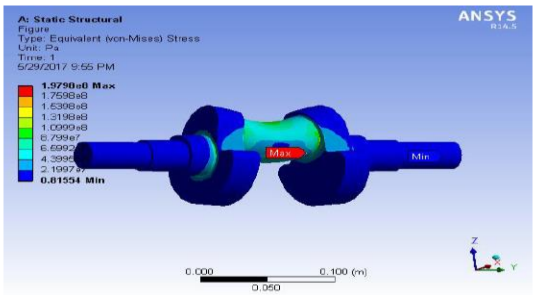

Figure 6.8 Von Misses Stress distribution (FG 300 Material) of Crankshaft.

For the given load conditions the range of Equivalent Von-Mises stress is observed from 0.81554 Pa to 1.9796e8 Pa.

Analysis of crankshaft- Aluminum Composite Material

Material: - Aluminum composite material

Yield Strength (Mpa):- 410 Mpa

Ultimate tensile strength: - 510 Mpa

Poisson ratio: - 0.32

Density: - 3100 kg/m3

Young's Modulus: - 72Gpa

Chemical Composition of Aluminum Composite Material

Sample Identification	Chemical Composition %								
	Mg	Si	Fe	Cu	Cr	Zn	Ti	Mn	Al
Al Composite	2.5	0.34	0.42	1.85	0.21	5.87	0.153	0.236	Bal

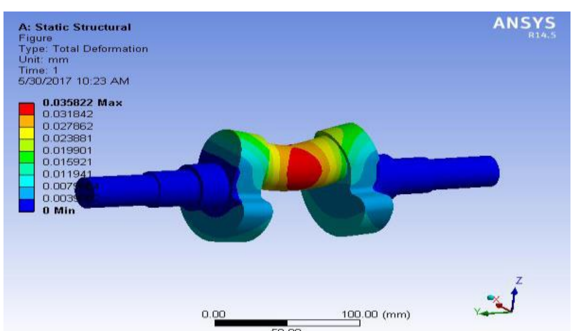

Figure 6.9 Deformation of (Aluminum Composite Material) Crankshaft

From the above result for material Aluminum composite material maximum deformation occurs at the center. Maximum deformation is noted as 0.035822mm.

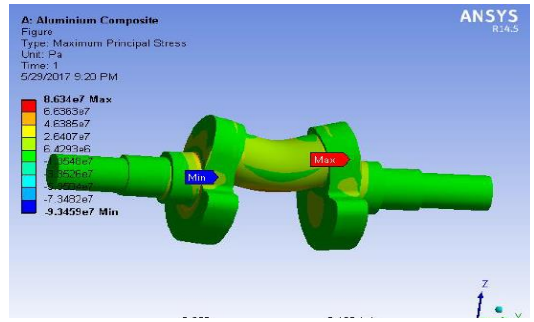

Figure 6.10 Maximum Principal Stress (Aluminum Composite Alloy) of Crankshaft

Maximum Principal Stress for the given condition is observed from -9.345e^7 to 8.63e^7 for Aluminum Composite Alloy material.

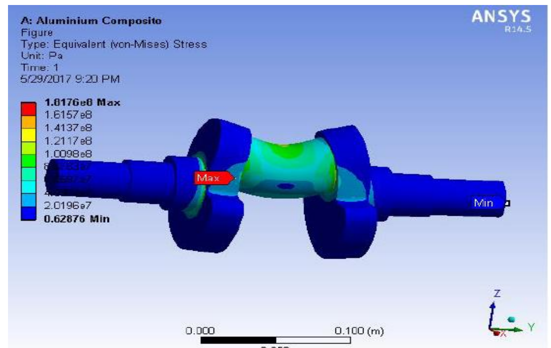

Figure 6.11 Von Misses Stress distribution (Aluminum Composite Material) of Crankshaft.

For the given load conditions the range of Equivalent Von-Mises stress is observed from 0.62876 Pa to 1.8176e8 Pa.

VIII. CONCLUSION

From the Analysis Software we can conclude that the deformation is maximum at the center of crankshaft. FEA results i.e. von mises stress, max principal stress and deformation shows comparative readings for FG260, FG300 and aluminum based alloy within 5-7% difference. Hence we can suggest aluminum based alloy material can be for crank shaft.

REFERENCES

[1] Wei Li, Qing Yan, Jianhua Xue, "Analysis of a crankshaft fatigue failure", Engineering Failure Analysis 55 (2015) 139–147.

[2] M. Fonte, P. Duarte, V. Anes, M. Freitas, L. Reis, "On the assessment of fatigue life of marine diesel engine Crankshafts", Engineering Failure Analysis (2015).

[3] M. Fonte, V. Anes, P. Duarte, L. Reis, M. Freitas, "Crankshaft failure analysis of a boxer diesel motor", Engineering Failure Analysis (2015).

[4] M. Fonte, P. Duarte, L. Reis, M. Freitas, V. Infante, "Failure mode analysis of two crankshafts of a single cylinder diesel engine", Engineering Failure Analysis (2015).

[5] B. Kareem, "Evaluation of failures in mechanical crankshafts of automobile based on expert opinion", Case Studies in Engineering Failure Analysis 3 (2015) 25–33.

[6] Xiaoping Chen, Xiaoli Yu, Rufu Hu, Jianfeng Li, "Statistical distribution of crankshaft fatigue: Experiment and modeling", Engineering Failure Analysis 42 (2014) 210–220.
[7] A. Ktari, N. Haddar, H.F. Ayedi, "Fatigue fracture expertise of train engine crankshafts", Engineering Failure Analysis 18 (2011) 1085–1093.
[8] J.A. Becerra, F.J. Jimenez, M. Torres, D.T. Sanchez, E. Carvajal, "Failure analysis of reciprocating compressor crankshafts", Engineering Failure Analysis 18 (2011) 735–746.
[9] Gustavo Rocha da Silva Santos, Guilherme Vinícuis França dos Santos, "Design of a Crankshaft for Internal Combustion Engine by Using Topology Optimization", 20th International Congress of Mechanical Engineering (2009).
[10] M. Fonte, M. de Freitas, "Marine main engine crankshaft failure analysis: A case study", Engineering Failure Analysis 16 (2009) 1940–1947.