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Design and Analysis of Crankshaft of Four Cylinder Diesel Engine for Heavy Vehicle

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ABSTRACT

Crankshaft is an important part of in an engine assembly. Crankshaft consists of two web sections and one crankpin. Crankshaft converts the reciprocating motion of the piston to a rotary motion with a four bar 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 has capacity of 3785.1cc. The finite element analysis in ANSYS software by using five materials based on their composition viz. FG260, FG300, EN8, EN24 and Aluminum Alloy material. The parameter like von misses stress, deformation; maximum principal stress were obtained from analysis software. The results of Finite element indicate that the Aluminum alloy material can be best suitable material among all.

Keywords: Crankshaft, Aluminium Alloy, Creo 2.0, ANSYS 14.5, FEA

I. INTRODUCTION

Crankshaft is an important part of in an engine assembly. Crankshaft consists of two web sections and one crankpin. Crankshaft converts the reciprocating motion of the piston to a rotary motion with a four bar 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 has capacity of 3785.1cc. The finite element analysis in ANSYS software by using five materials based on their composition viz. FG260, FG300, EN8, EN24 and Aluminum Alloy material. The parameter like von misses stress, deformation; maximum principal stress were obtained from analysis software. The results of Finite element indicate that the Aluminum alloy material can be best suitable material among all.

II. LITELATURE 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.

III. 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:

Table No.01 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 35o from the top dead center, the pressure on the piston is 1 N/m m2 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.

B) Design of Crankshaft When the Crank is at TDC 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/mm2

The thrust in the connecting rod will be equal to the gas load on the piston (F_p).

We know that piston gas load,

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

$$= 103.457 kN$$

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):-

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

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= 72*mm*

(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,

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

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

$$M = 1525.99kN - mm$$

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

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

We know that direct compressive stress on the crank web,

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

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

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

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

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.

C) 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)

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

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

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

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

IV. ANALYSIS USING ANSYS 14.5

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.

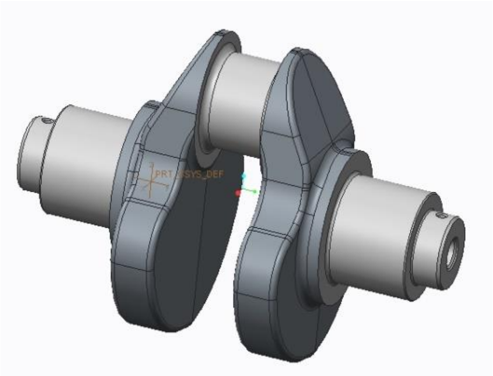

Figure No:- 1 Three Dimensional Modeling Original Crakshaft in Creo 2.0

A) Finite Element Analysis of Material

1) Analysis of crankshaft- FG 260 Material :

We select FG 260 material for analysis following are the properties of material for structural analysis,

Material: - FG 260

Poisson ratio: - 0.26

Density: - 7197 kg/m3

Young's Modules: - 198 GPa.

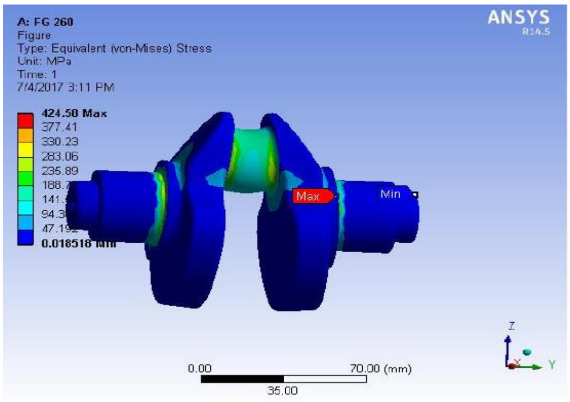

Figure No : 2 Von Misses Stress Distribution (FG 260) of Crankshaft

For the given load conditions the range of Equivalent Von- Mises stress is observed from 0.018518 Pa to 424.58MPa.

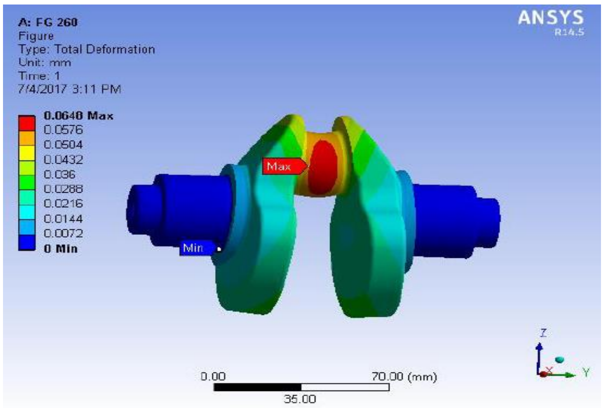

Figure No: 3 Deformation of (FG 260 material) crankshaft

From the above result for material FG 260 maximum deformation occurs at the center. Maximum deformation is noted as 0.0648mm

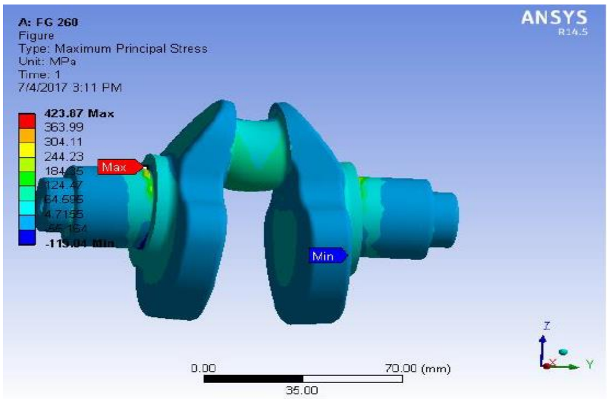

Figure No: 4 Maximum Principal Stress (FG 260) of Crankshaft

Maximum Principal Stress for the given condition is observed from -119.45 MPa to 423.87 MPa for FG260 material.

2) Analysis of Crankshaft- FG 300 Material :

We select second material FG 300 for analysis following are the properties of material for structural analysis,

Material: - FG300

Poisson ratio: - 0.26

Density: - 7200 kg/m3

Young's Modules: - 205 Gpa

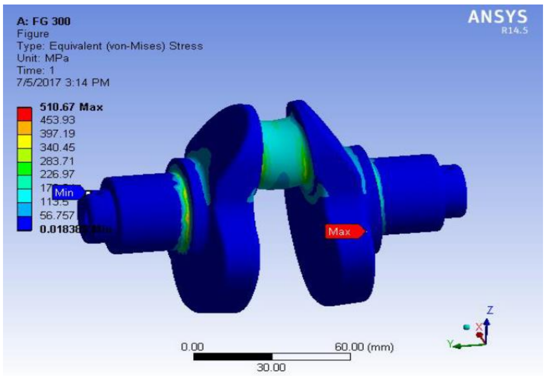

Figure No : 5 Von Misses Stress Distribution (FG 300) of Crankshaft.

For the given load conditions the range of Equivalent Von- Mises stress is observed from 0.018518 Pa to 510.67MPa.

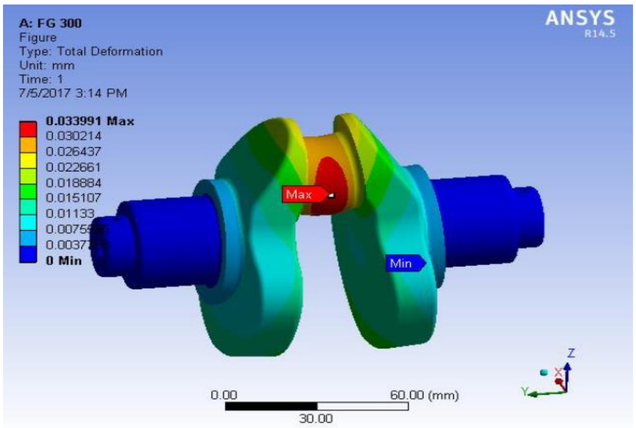

Figure No :-6 Deformation of (FG 300) Crankshaft

From the above result for material FG 300 maximum deformation occurs at the center. Maximum deformation is noted as 0.03391mm

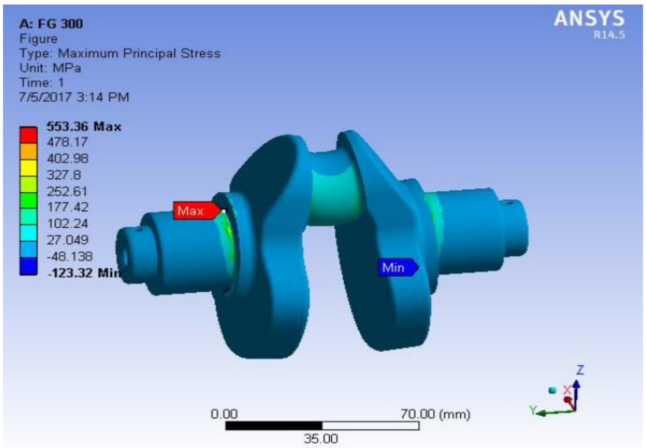

Figure No : 7 Maximum Principal Stress (FG 300) of Crankshaft

Maximum Principal Stress for the given condition is observed from -123.32 MPa to 553.36 MPa for FG300 material.

3) Analysis of Crankshaft- EN 8 Material :

We select third material EN 8 for analysis following are the properties of material for structural analysis,

Material: - EN 8

Poisson ratio: - 0.300

Density: - 7850 kg/mm3

Young's Modules: - 210 GPa

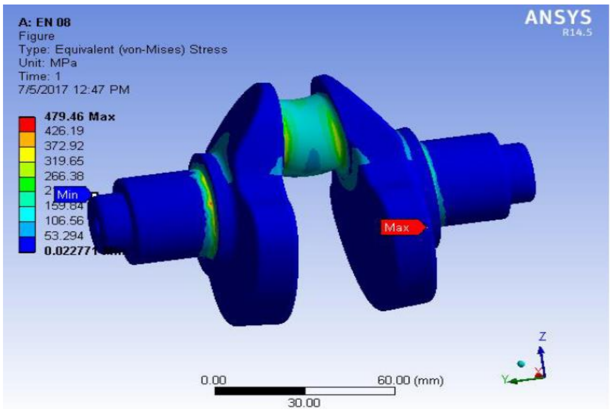

Figure No : 8 Von Misses Stress Distribution (EN 8) of Crankshaft.

For the given load conditions the range of Equivalent Von- Mises stress is observed from 0.022771 Pa to 479.46MPa.

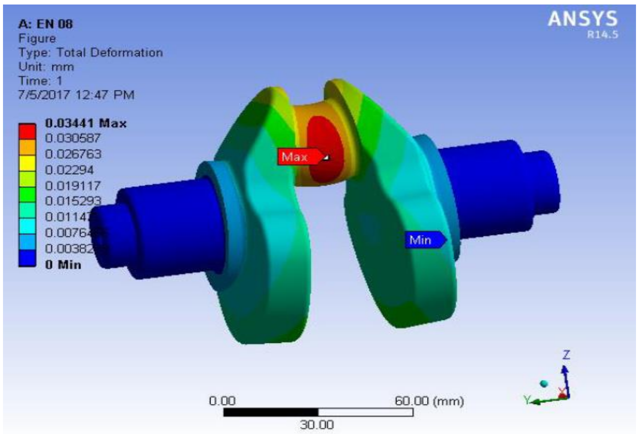

Figure No : 9 Deformation of (EN 8 material) Crankshaft

From the above result for material FG 300 maximum deformation occurs at the centre. Maximum deformation is noted as 0.03441mm.

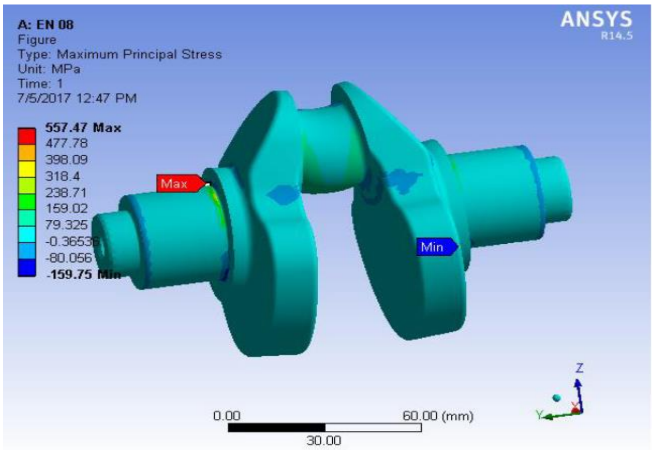

Figure No : 10 Maximum Principal Stress (EN 8 material) of Crankshaft

Maximum Principal Stress for the given condition is observed from -159.75 MPa to 557.47 MPa for EN 8 material.

4) Analysis of Crankshaft- E 24 Material:

We select fourth material EN 24 for analysis following are the properties of material for structural analysis,

Material: - EN 24

Poisson ratio: - 0.300

Density: - 7800 kg/mm3

Shear Modules: - 205 GPa

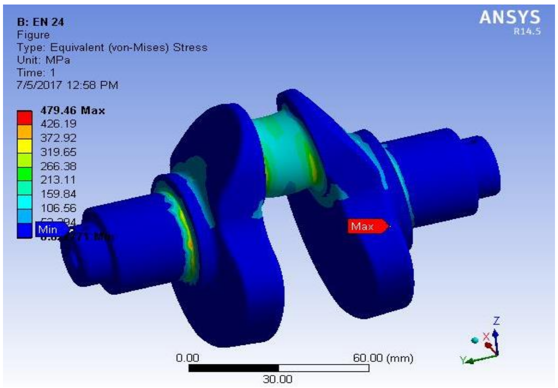

Figure No : 11 Von Misses Stress (EN 24) of Crankshaft.

For the given load conditions the range of Equivalent Von- Mises stress is observed from 0.02771 Pa to 479.46MPa.

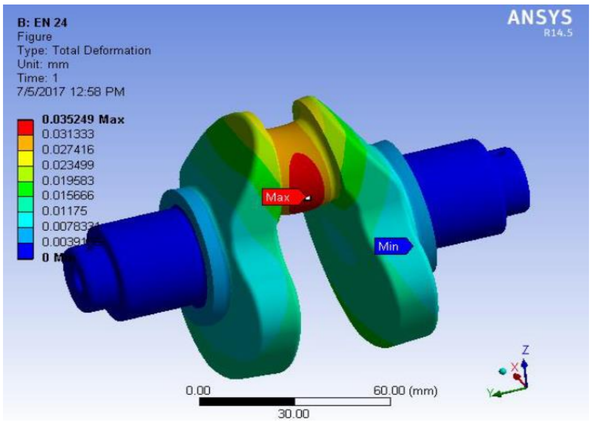

Figure No : 12 Deformation of (EN 24) Crankshaft

From the above result for material EN 24 maximum deformation occurs at the centre. Maximum deformation is noted as 0.035249mm.

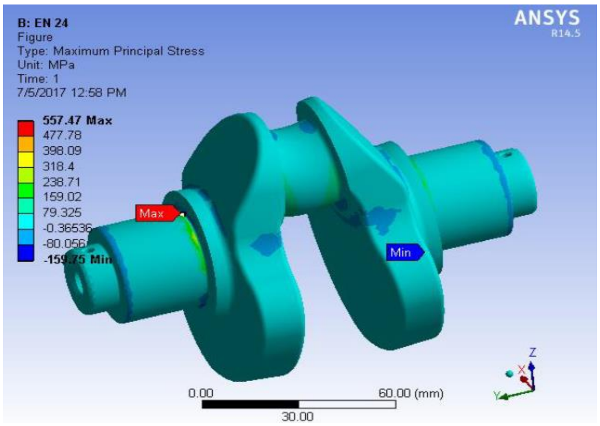

Figure No : 13 Maximum Principal Stress (EN 24) of Crankshaft

Maximum Principal Stress for the given condition is observed from -159.75 MPa to 557.47 MPa for EN 24 material.

5) Analysis of Crankshaft- Aluminium Alloy Material:

We select fifth material Aluminium Alloy for analysis following are the properties of material for structural analysis,

Material: - Aluminium Alloy

Poisson ratio: - 0.32

Density: - 3100 kg/m3

Young's Modulus: - 72Gpa

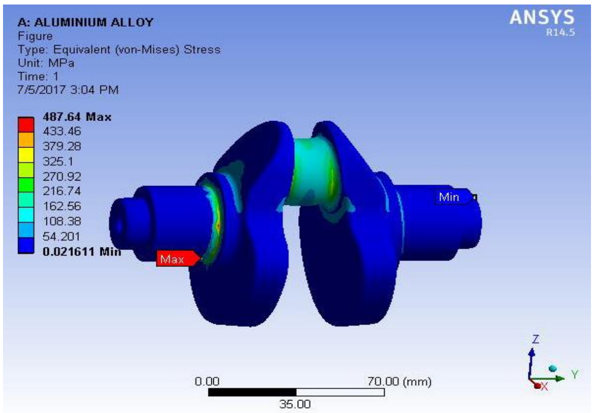

Figure No : 14 Von Misses Stress (Aluminiumm Alloy) of Crankshaft.

For the given load conditions the range of Equivalent Von- Mises stress is observed from 0.021611 Pa to 487.64MPa.

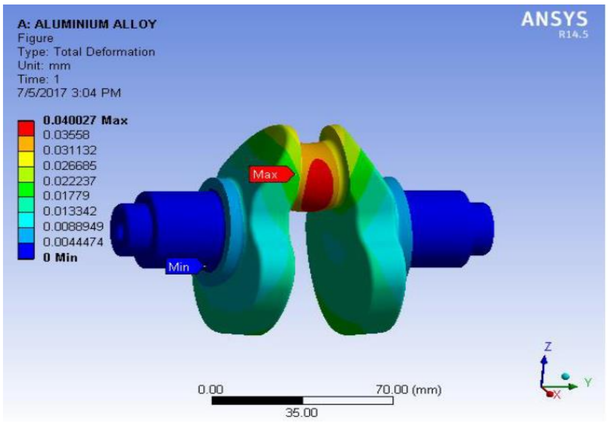

Figure No : 15 Deformation of (Aluminium Alloy) Crankshaft

From the above result for material Aluminium Alloy maximum deformation occurs at the centre. Maximum deformation is noted as 0.040027mm.

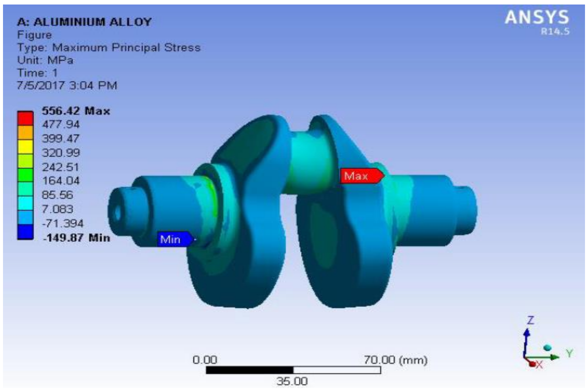

Figure No : 16 Maximum Principal Stress (Aluminum Alloy) of Crankshaft

Maximum Principal Stress for the given condition is observed from -149.87 MPa to 556.42 MPa for Aluminium Alloy material.

V. CONCLUSION

From the Analysis Software we can say that the maximum deformation at the centre of crankpin of crankshaft. FEA results i.e. von mises stress, max principal stress and deformation shows comparative readings for FG260, FG300, EN8, EN24 and Aluminium Alloy within 5-7% difference. Hence we can conclude aluminium based alloy material can be suitable for crank shaft.

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