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Design And Finite Element Analysis Of Aluminium-6351 Connecting Rod

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Abstract

The connecting rod is the most relevant pats of an automotive engine. The connecting rod is subjected to an extremely complex state of loading. High compressive and tensile loads are due to the combustion and connecting rod's mass of inertia respectively. The objective of this research is to investigate the failure analysis of the connecting rod of the automotive engine. Apart from conventional material of connecting rod I choose the connecting rod of FM-70 Diesel engine which is made of Aluminium 6351. static analysis is done to determine the von Misses stress, elastic strain, total deformation in the present design connecting rod for the given loading conditions using the FEM Software Ansys 12.1 .In the starting of the work, the static loads acting on the connecting rod, After that the work is carried out for safe design and life in fatigue. Fatigue Analysis is compared with the Experimental results.

1. Introduction

Connecting rod is the main component of the combustion engines which main purpose are transfer the energy from the pistons to crankshafts and convert the linear, reciprocating motion of a piston into the rotary motion of a crankshaft, from the viewpoint of functionality; also it connects reciprocating piston to rotating crankshaft and some other design connected direct to the crosshead and then transmitting the thrust of the piston to the crankshaft of the Automobile Engine. Where in the automobile engine con rod moves are in both a rotating motion in to a big end and reciprocating motion in to a small end.

Connecting rods must have the highest possible rigidity at the lowest weight. In Automobile internal combustion engine connecting rod is a high volume production component subjected to complex loading.

Bending stresses appear due to eccentricities, crankshaft, case wall deformation, and rotational mass force. Therefore, a connecting rod must be capable of transmitting axial tension, axial compression, and lower cost due to their simple bending stresses caused by the thrust and pull on the piston and by centrifugal force.

During the operation of the engine, the connecting rod undergoes is prone to tensile, compression, and buckling loading. In many cases, the major reason behind or causing catastrophic engine failure is the occurrence of the connecting-rod failure and sometimes, such a failure can be attributed to the broken connecting rod's shank especially when there is a probability of being pushed through the side of the crank-case, thereby making the engine irreparable [1].

However, specifically describing such a failure, it is important to point out at the different reasons for this failure such as fatigue near a physical defect in the rod, the overheating of the engine, cracking, lubrication failure in a bearing which is usually caused by inaccurate or faulty maintenance, failure of the rod bolts which is due to defect, improper tightening ,and re-use of already used (stressed) bolts where not recommended. Figure 1 shows the failure on connecting rod [2].

2. Specification of Existing Connecting rod

Table 1 shows the specifications of the connecting rod for Aluminium 6351 (FM 70 Diesel Engine.). Where tested chemical composition of the material is 0.87%Si, 0.49% Mn, 0.006%Ch, 0.005%Co, 0.013% Ti, 0.14% Fe, 0.02%Pb, 0.005% Sn, 0.87% Mg, and 97.53% Al.

Table: 1 Specification of Al 6351 Connecting rod

	ENGINE DIMENSION	
1	LENGTH (mm.)	355
2	WIDTH (mm.)	552
3	HEIGHT (mm.)	465
4	ENGINE WEIGHT	18
5	SPARK PLUG	MICO M 45 Z48
6	FUEL CAPACITY	4.5 (Lit)
7	OPERATING HOUR	2.3
8	P.T.O. SHAFT	ANTI CLOCKWISE

	ROTATION	
9	METHOD OF STARTING	ROPE / RECOIL
10	FUEL	PETROL (START) DIESEL (RUN)
11	IGNITION SYSTEM	ELECTRONIC

3. Theoretical Calculation of Connecting Rod

A connecting rod is a Reciprocating part which is deformed in the compressive and tensile forces. Since the Tensile forces are much lower than the Compressive forces, therefore, design of the cross-section of the connecting rod we consider the Rankin's formula.

Here,
σ_c = Compressive yield stress
A = C/S area of Connecting rod
L = Length of Connecting rod
P_{cr} = Critical buckling load
Ixx and Iyy =moment of inertia of the section about x-axis and y-axis respectively.
Kxx and Kyy =radius of gyration of the section about x-axis and y- axis respectively. Rankin formula = (Ixx=4Iyy).

3.1 Pressure Calculation for 256 CC Diesel Engine

Let,
Con rod length to crank ratio, $\frac{l}{r} = 3.80$
Length of the connecting rod= 3.80*33.35 = 127 mm.
Indicated power, $I_p = \frac{BP}{\eta} = \frac{2.238}{0.80} = 2.7975$ Kw.
Indicated power, $I_p = p_m$ L A n, (n=3000 rpm)
Mean Pressure, $p_m = 0.4295$ N/mm^2
C/S area of piston = $\left(\frac{\pi D^2}{4}\right)$ = A =3848.45 mm^2
Force on the piston due to gas pressure, $P_c = \left(\frac{\pi D^2}{4}\right) P_{max} = P_c = 14876.83$ N
Critical buckling load is given by, $P_{cr} = P_c\ (f_s)$ = 37190 N

3.2 Design Calculation for the Al-6351 Connecting rod

The most suitable section for the connecting rod is I-section with the proportions as shown in Fig. 1

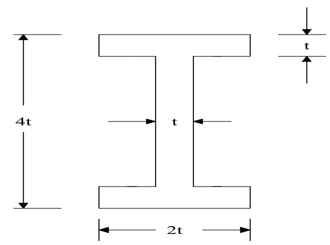

Figure 1: I-Section of the connecting rod

Thickness of the flange and web of the section = t
Width of the section = B =2t
Height of the section = H = 4t
Area of the section = A = [2 (4t * t) + 2t*t] = 6t^2
Moment of inertia about x- axis = $I_{xx} = \left(\frac{120}{12}\right) t^4$
Moment of inertia about y-axis = $I_{yy} = \left(\frac{18}{12}\right) t^4$
So now, I_{xx} / I_{yy} = 6.6667

We know that radius of gyration of the section about X-axis,
$K_{xx} = \sqrt{I_{xx}} / \sqrt{A} = 1.291\ t$

After deciding the proportions for I-section of the connecting rod, its dimensions are determined by considering the buckling of the rod about X-axis (assuming both ends hinged) and applying the Rankin's formula.
We know that buckling load, $P_{cr} = \frac{\sigma_c}{1+a\left(\frac{L}{K_{xx}}\right)}$

By putting all the defined values we can find the,
$t^4 - 21.37t^2 - 88.199 = 0$
So, t = 5mm. So by the calculation, B = 10 mm And H = 20 mm.
$\therefore$ Depth at crank end H_1 = 1.1H = 22.34 mm
$\therefore$ Depth at piston end H_2 = 0.8H = 17.2 mm

3.3 Dimension at small end or at piston end

Max gas load = Bearing load, $P_c = d_p\, l_p (p_b)_p$
Here $\left(\frac{l_p}{d_p}\right) = 1.7$
By making calculation we find out;
Diameter of the piston pin, d_p = 15.60 mm. And Length of piston pin, l_p = 26.2 mm.

3.4 Dimension at big end or crank end

$P_c = d_c l_c (p_b)_c,\ \frac{l_c}{d_c} = 0.89$, Bearing pressure of crank pin, $(p_b)_c = 18.926$ N/mm^2 , so we can find out,

d_c = Diameter of the crank pin = 30 mm.
l_c = Length of crank pin = 26.2 mm.

3.5 Big End Cap and Bolts

The maximum force acting on the cap and two bolts consists only of inertia force at the top dead centre on the exhaust stroke. The inertia force acting on the bolts or cap is given by,

$$\therefore \quad P_i = m_r \omega^2 r \left[\cos\theta + \frac{\cos 2\theta}{n_1}\right]$$

Where, the angular velocity of the crank is, $\omega = \left(\frac{2\pi N}{60}\right)$, and the mass of reciprocating parts is given by,m_r = [mass of piston assembly + (1/3) mass of connecting rod] = 2.51 kg. = 24.615 N

The inertia force on the connecting rod will be maximum at the top dead centre position where (θ = 0). So when $\theta = 0$ then $\cos\theta = 1$ and $\cos 2\theta = 1$ and substituting the above values and get;

$(P_i)_{Max} = m_r\, \omega^2 r\left[1 + \frac{1}{n_1}\right]$ so by solving the equation get the value of; $(P_i)_{Max} = 10431.1867$ N.

So $(P_i)_{Max} = 2\left(\frac{\pi d_{co}^2}{4}\right)\frac{\sigma_t}{Fs}$,Core dia of bolt dco=6.4mm

Nominal diameter of the bolt is calculated by, $d = \left(\frac{dco}{0.7}\right)$ = 10 mm

The distance between the centers of bolts is, $L_h = d_c + d + C_L$ = 40.2 mm.

It is treated as a beam freely supported at the bolt centers and loaded in a manner intermediate and centrally concentrated load in which case the bending moment is WL/6 so,

$$M_b = \frac{(P_i)max.}{6} * L_h = \frac{10431.1867}{6} * 40.2$$

M_b = 69888.95 N.mm

For thickness of the cap (t_c) is obtained by,

$$\sigma_{pb} = \frac{(M_b)Y}{I}, \; Y = \frac{t_c}{2}, \; I = \frac{b_c}{12} t_c^{\,3}$$

Solving the above equations we get t_c = 7.5 mm

It may be noted that the inertia force of reciprocating parts opposes the force on the piston when it moves during its downward stroke (i.e. when the piston moves from the top dead centre to bottom dead centre). On the other hand, the inertia force of the reciprocating parts helps the force on the piston when it moves from the bottom dead centre to top dead centre. Figure 2 shows the Inertia Force acting on the connecting rod at different crank angles.

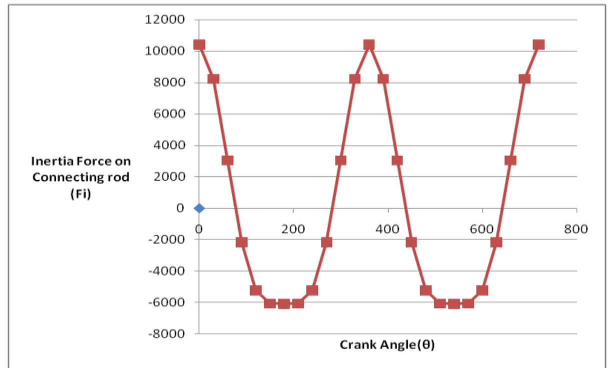

Figure 2: Relation between Inertia Force to Crank Angle.

Now force acting in the connecting rod at any instant due to effect of inertia force,

$P_r = \frac{P}{\cos\emptyset}$. Where $\sin\emptyset = \frac{\sin\theta}{n_1}$

Let, P_r = Force on Connecting rod at different crank angle.

$\emptyset$ = Inclination angle of line of stroke, θ =crank angle

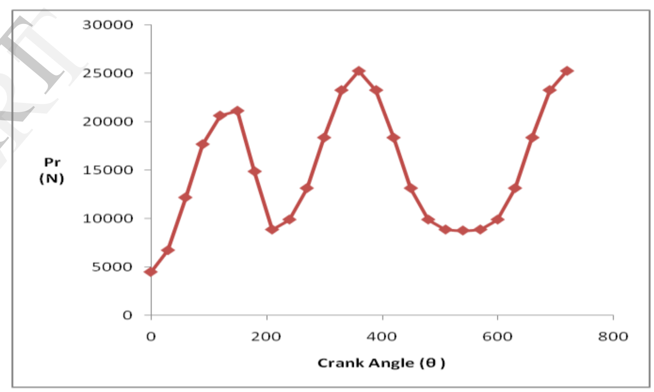

Figure 3: Crank angles to Force acting on the connecting rod.

4. Modeling and FEM Analysis of Connecting rod

Aluminium 6351 Connecting rod was modeled by taking the designed parameter of rod and then by using the Pro-e Wild Fire 5.0 software solid modeling has done which is shown in Fig.4. And saved within this program in *.IGES format. The model is imported in Ansys and then the mechanical characteristics of the connecting rod are established: density - 2800 kg/m3, Young’s modulus – 68.9 GPa, Poisson’s ratio - 0.3, etc.

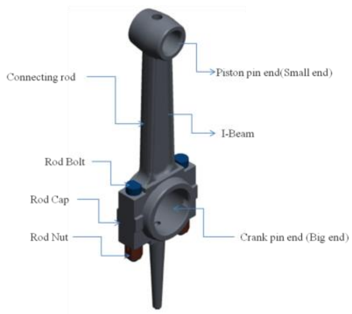

Figure 4: Schematic of a typical Connecting rod [Pro-e modal]

4.1 Meshing

Here Meshing element chooses is 10 nodes Tetrahedron named Solid187 as shown in fig 5. First convergence was checked by finding deformation against different element size then plotting graph of deformation versus no of elements. Here element size is found out to be 2mm for working in convergence zone. Total No of element was 13403 and Nodes were 23962. Fig. 6 shown below is meshed model of connecting rod.

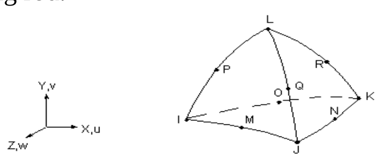

Figure 5: Schematic of 10 tetrahedron element shape used for FEA meshing.

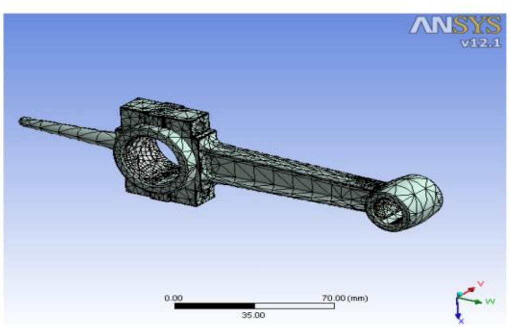

Figure 6: Meshed connecting rod in ANSYS Workbench 12.1 software.

4.2 Loading and Boundary Conditions

In this study four finite element models were analysed. FEA for both tensile and compressive loads were conducted. Two cases were analysed for each case, one with load applied at the crank end and restrained at the piston pin end, and the other with load applied at the piston pin end and restrained at the crank end. In the analysis carried out, the axial load was 14.88 KN in both tension and compression. The pressure constants for 14.88 KN are as follows. [8]

Compressive Loading Stress:

Crank End $P_{co}=\left(\frac{Pc}{rp\ H1\ \sqrt{3}}\right)=\frac{14876.83}{15*22.34*\sqrt{3}}=25.637$ MPa

Piston End $P_{co}=\left(\frac{Pc}{rc\ H2\ \sqrt{3}}\right)=\frac{14876.83}{7.8*17.2*\sqrt{3}}=64.269$ MPa

Tensile Loading Stress:

Crank End $P_{to}=\left(\frac{Pc}{rp\ H1\left(\frac{\pi}{2}\right)}\right)=\frac{14876.83}{15*22.34*\left(\frac{\pi}{2}\right)}=28.263$ MPa

Piston End $P_{to}=\left(\frac{Pc}{rc\ H2\left(\frac{\pi}{2}\right)}\right)=\frac{14876.83}{7.8*17.2*\left(\frac{\pi}{2}\right)}=70.594$ MPa

Since the analysis is linear elastic, for static analysis the stress, displacement and strain are proportional to the magnitude of the load. Therefore, the obtained results from FEA readily compare with the above values.

So, Net force acting on connecting rod = Gas Forces-Inertia Forces Acting on piston

$\therefore$ P = 14876.508 – 10431.1867, P = 4469.6 N

Boundary Condition:

Table 2: Loading Conditions

Type of Loading	Applied Net force at	Restrained end at
Tensile Loading	Crank end (180^0)	Pin end (180^0)
	Pin end (180^0)	Crank end (180^0)
Compressive Loading	Crank end (120^0)	Pin end (120^0)
	Pin end (120^0)	Crank end (120^0)

FEA Result:

The load analysis was carried out to obtain the loads acting on the connecting rod at any given time in the loading cycle and to perform FEA. Most investigators have used static axial loads for the

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design and analysis of connecting rods. In the static analysis von Mises stress, critical locations observed under tension and compression loadings at the piston and crank ends. The most highly stressed areas are in the transition regions between the shank and the crank end, as well as the shank and the pin end. Stresses are all symmetric over the entire rod, since geometry and loading were symmetrical.

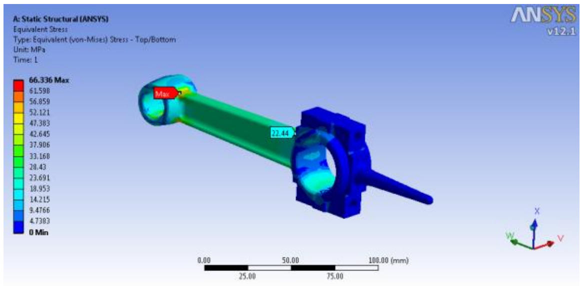

Figure 7 Equivalent (von-Mises) Stress (Tensile load at piston end)

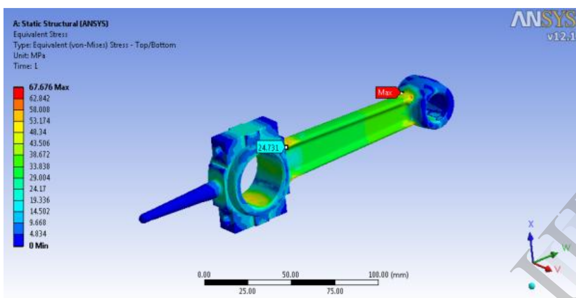

Figure 8 Equivalent (von-Mises) Stress (Tensile load at crank end)

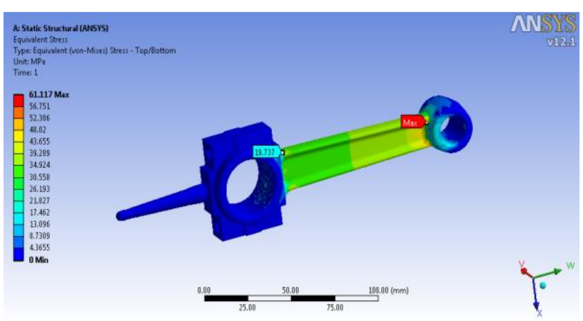

Figure 9 Equivalent (von-Mises) Stress (Compressive load at piston end)

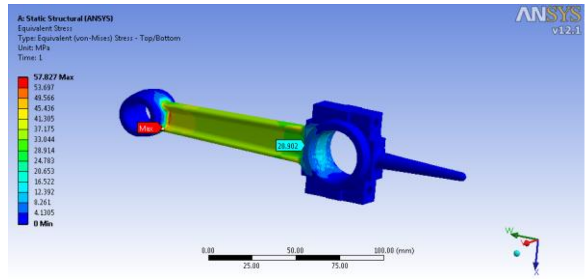

Figure 10 Equivalent (von-Mises) Stress (Compressive load at crank end)

TABLE 3: Comparative result of Equivalent Stress

Type of Loading	Applied Net force at	Restrained end at	Equivalent (von-Mises) Stress (MPa)			
			Analytical		FEM Analysis	
			At Crank end	At pin end	At Crank end	At pin end
Tensile Loading	Crank end (180^0)	Pin end (180^0)	28.263	70.59	24.73	67.68
	Pin end (180^0)	Crank end (180^0)	28.263	70.59	22.44	66.34
Compressive Loading	Crank end (120^0)	Pin end (120^0)	25.637	64.27	20.90	57.83
	Pin end (120^0)	Crank end (120^0)	25.637	64.27	19.74	61.12

5. Fatigue Behavior and Life prediction

Fatigue failure of mechanical components is a process of cyclic stress/strain evolution and redistribution in the critical stressed volume. It may be imagined that due to stress concentration (notches, material defect or surface roughness) the local material yields firstly to redistribute the loading to the surrounding material, and then follows with cyclic plastic deformation and finally crack initiates and the resistance is lost. Therefore, the simulation for cyclic stress/strain evolutions and improving the accuracy of fatigue life prediction of mechanical components.

The details view of the fatigue tool is used to define the various aspects of a fatigue analysis such as loading type, handling of mean stress effects and more. Several results for evaluating fatigue are available to the user. Outputs include fatigue life, damage, factor of safety, stress biaxiality, fatigue sensitivity.

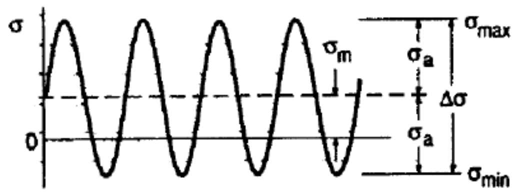

Figure 11 Reversible cyclic loading

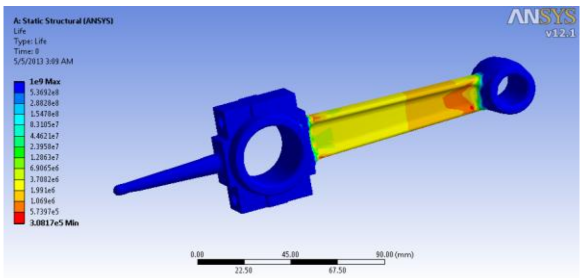

Figure 12 Fatigue life

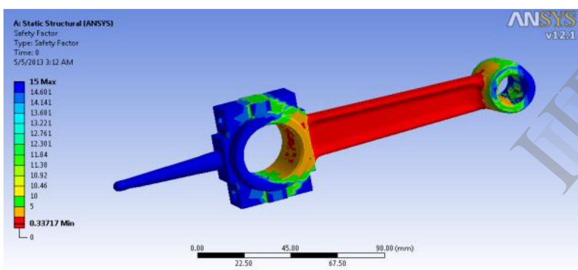

Figure 13 Safety Factor

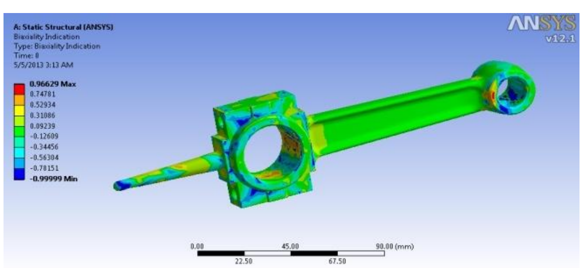

Figure 14 Biaxiality Indication

Fatigue Experimental Analysis

The experimental fatigue analysis has been done for finding the life of and probably to find failure area of the Aluminium 6351 connecting rod.

Figure 15 shows the modal of fatigue Experiment. Where Fig 16-17 shows the experimental set-up of the analysis.

Figure 15 Aluminium-6351 Connecting rod

Figure 16 High and Low Cycle Fatigue machine

Figure 17 Arrangement of jig and Fixture for Experiment

Result of Analysis

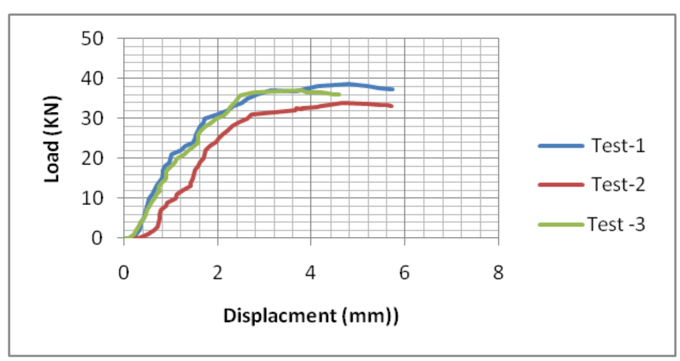

Figure 18 Displ. Vs. Load in experimental Fatigue Testing

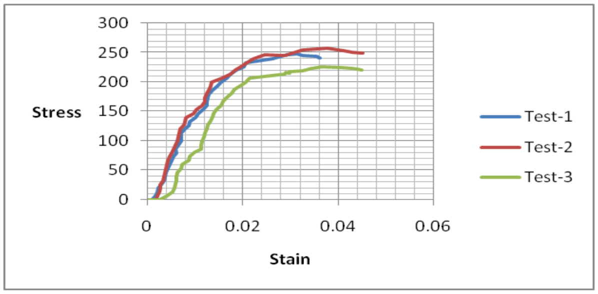

Figure 19 Stresses vs. Strain in Experimental Fatigue Testing

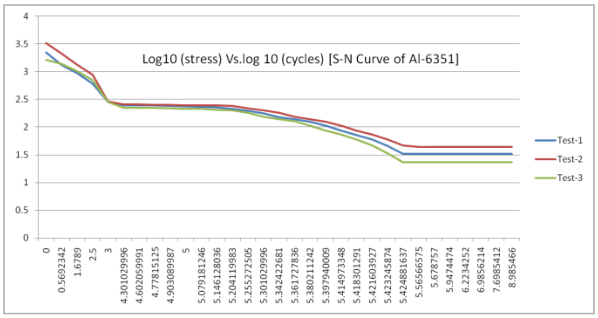

Figure 20 S-N curve of Aluminium 6351

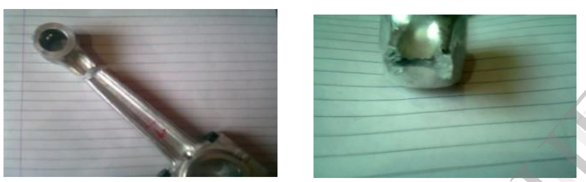

Figure 21 Fatigue test of Spacimen-1

Figure 22 Fatigue test of Spacimen-2

Figure 23 Fatigue test of Spacimen-3

Figure 24 Failure result of 3- specimen

Factor of Safety

Material Properties	Aluminium 6351
Yield Strength (Mpa)	285
Theotitical Factor of Safety	4
Allow Stress (Mpa)	71.25
Ansys Result (Mpa)	67.68
Working Factor of Safety	4.21

6. Conclusions

Figure 7 to 10 shows the FEM Analysis of the connecting rod of Aluminium 6351 applying load 4469.6 at the crank and piston pin end while loading condition are in tensile and compressive.

As from calculation of Factor of Safety here it is (4.21) appropriate with considering designer experience. The working factor of safety is nearer to theoretical factor of safety in aluminum 6351 connecting rod.

As showing the results the peak stresses mostly occurred in the transition area between pin end, crank end and shank region. The value of stress at the middle of shank region is well below allowable limit.

Based on the results of the experiments obtained by analyzing the finite element in the present study, it can be concluded that the occurrence of the connecting rod failure was due to the fatigue crack growth mechanism which came as a result of higher stress being combined with the porosity (manufacturing defect) in initiation and growth of a fatigue crack followed by catastrophic failure. Finally, lubrication engine system should be regularly checked, and all these are highly recommended to ensure long life connecting rod.

Figure 18-24 shows the Ansys and experiments results of the fatigue of three spacimen.Experiment result for spacimen-1 life was 3.125×10^5,Spacimen-2 life is 2.984×10^5 ,and spacimen-3 life is 3.096×10^5.where Ansys software result was 3.0817×10^5.So here Experimental results are nearly equal to the Analysis software results. So we can now successfully implement this part in to a FM-70 Diesel Engine.

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