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Dynamic Load Analysis of Carbon Fiber Connecting Rod

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Abstract- The main objective of this study was to explore weight reduction opportunities for a production of carbon fiber connecting rod. This has entailed performing a detailed dynamic load stress analysis of the connecting rod. In the first part of the study, the static load analysis, selection of material of the connecting rod are considered. Then we go for design of connecting rod in "inventor2014" Then component is imported to the Ansys 15.0 and analysis is done.
Keywords- Dynamic Load Analysis of Connecting Rod Using Vibration Analysis.

1. INTRODUCTION

A. Material Selection

Different types of materials are used in manufacturing of the connecting rods. The material for a connecting rod is selected based on the purpose of the connecting rod and depending upon the requirement of the I.C engines.

Some of the materials used in the manufacturing of connecting rod are

- Cast iron
- Aluminum alloys
- Carbon steel
- Stainless steel
- Magnesium
- Titanium

Generally forged materials are used for the manufacturing of connecting rods into account or neglected during the optimization. Nevertheless, a proper picture of the stress variation during a loading cycle is essential from fatigue point of view and this will require FEA over the entire engine cycle.

The objective of this chapter is to determine these loads that act on the connecting rod in an engine so that they may be used in FEA. The details of the analytical vector approach to determine the inertia loads and the reactions were discuss

This approach is explained by Wilson and Sadler (1993). The equations are further simplified so that they can be used in a spreadsheet format. The results of the analytical vector approach have been enumerated in this chapter.

B .Investigation Plan

The connecting rod undergoes a complex motion, which is characterized by inertia loads that induce bending stresses. In view of the objective of this study, this is optimization of the connecting rod. it is essential to determine the magnitude of the loads acting on the connecting rod. In addition, significance of bending stresses caused by inertia loads needs to be determined, so that we know whether it should be taken.

2. STATIC FORCES ON CONNECTING ROD

The stresses in the connecting rod are set up to the following forces acting on it

- Direct load on piston due to gas pressure.
- Inertia of connecting rod.
- Friction of the piston rings and of the piston.
- The friction of the piston pin bearing and the crank pin bearing.

Gas pressure force F_P

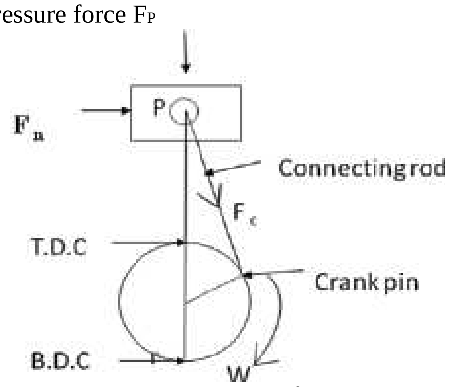


Fig1: General diagram of IC Engine.

The load due to piston inertia is

= weight of the reciprocating masses X accelerations

$$Fi = \frac{1000\ wrv^2}{gv} \times \cos\theta \pm \frac{\cos 2\theta}{n'}$$

Buckling load on connecting rod:

$$Wb = \frac{fc \times A}{1+a[\frac{L}{Kxx}]^2}$$

Where a = Rankin constant
Total force on the connecting rod :
$F = F_p - F_i$

A. Theoretical Calculation

3. Maruti Suzuki SX4 Specifications

Engine type water cooled 4-stroke
Bore x Stroke (mm) = 78×83
Displacement = 1586 CC
Maximum Power = 104.7 PS @ 5600 rpm

Force Acting on piston

$$Fp = \frac{\pi}{4}\, d^2 \ \text{X Gas Pressure}$$

Gas pressure
Density of Petrol C8H18 = 737.22 kg/m^3
= 737.22E-9 kg/mm^3
Flash point for petrol (Gasoline)
Flash point = -43°c (-45°F)
Auto ignition temp. = 280°c (536°F) = 288°k

Mass = Density x volume
= 737.22E-9 x 396.5E3
= 0.29kg

Molecular weight of petrol = 114.228g/mole
= 0.11423 kg/mole

From gas equation,
PV=m * Rspecific * T

Where, P = Pressure, MPa
V = Volume
m = Mass, kg
Rspecific = Specific gas constant
T = Temperature, °k
Rspecific = R/M
Rspecific = 8.3143/0.29
Rspecific = 28.67 Nm/kg K

P = m.Rspecific.T/V

$$P = \frac{(0.29 \times 28.67 \times 288.85)}{396E^3}$$

P = 6.06 MPa

P = 6.10 MPa

Force acting on Piston

$$Fp = \frac{\pi}{4}\, d^2 \ \text{X Gas Pressure}$$

$$Fp = \frac{\pi}{4}\, 78^2 \ \text{X } 6.1$$

Fp = 29148.01 N

Total Force acting F = Fp - Fi

Where Fp = force acting on the piston
Fi = force of inertia

$$Fi = \frac{1000\ wrv^2}{gv} \times \cos\theta \pm \frac{\cos 2\theta}{n'}$$

wr = weight of the reciprocating parts
wr = 0.673 x 9.81 = 6.24 N
r = crank radius, r = 41.5
Also, θ = Crank angle from dead center = 0 considering connecting rod is at TDC position
n' = length of connecting rod / crank radius
Angular velocity,w= $\frac{2\pi N}{60}$ = $\frac{2\pi\ 5600}{60}$ = 586.43
Crank velocity V= rw = 41.5E^{-3} x 586.43
= 24.33m/sec

$$Fi = \frac{1000 \times 6.24 \times 24.33^2}{9.81 \times 41.5} \times \cos 0 \pm \frac{\cos 2(0)}{4}$$

Fi = 9078.39 N

Therefore, total force acting F = Fp – Fi

F = 29148.01- 9078.39
F = 20069.61 N

According to Rankin's Formulae F,

$$F = \frac{fc \times A}{1 + a[\frac{L}{Kxx}]^2}$$

A = c/s area of connecting rod
L = Length of connecting rod
Fc = Compressive yield strength
F = Buckling load

$K_{xx} = \sqrt{\frac{Ixx}{A}} = 1.7t$

$a = \frac{\delta c}{\pi E^2} = 0.0004$

$$F = \frac{fc \times A}{1 + a[\frac{L}{Kxx}]}$$

$$20069.61 = \frac{196 \times 11t^2}{1 + 0.0004[\frac{83}{1.78t}]^2}$$

t = 3.18 mm
t = 3.5 mm

In general,

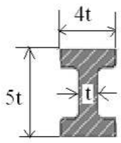


Fig 2: I Section Standard Dimensions of connecting rod

Therefore
Width B = 4t = 14 mm
Height H = 5t = 17.5 mm
Area A = $11t^2$ = 134.75 mm^2

Height at the piston end, H_1 =0.75H– 0.9H
H_1= 0.82X17.5 = 14.35mm
Height at the crank end,H_2=1.1H – 1.25H
H_2= 1.18 X 17.5 = 20.65 mm
Length of the connecting rod (L) = 166mm

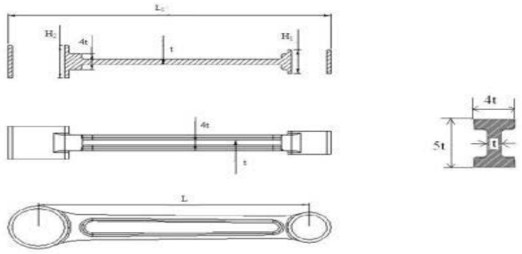


Fig 3: Cross sectional View of connecting rod

Design of small end:
Load on the piston pin or the small end bearing (Fp)
= Projected area x Bearing pressure

*Fp= dp X lp * P_{bp}*

Fp= 29148.01 N load on the piston pin,
dp = Inner dia. of the small end
P_{bp} = Bearing pressure
= 10.0 for oil engines.
= 12.7 for automotive engines.
We assume it is a 150cc engine, thus
P_{bp} = 12 MPa
lp = length of the piston pin
lp = 1.75 *dp*
Substituting,

29148.01 = 1.75 *dp X dp*.x 12

dp = 37.25 mm
dp = 38.00mm
lp = 1.75X38 = 67.0 mm
Outer diameter of small end =1.3dp
= 1.3 X 38 = 49.4
Od= 50mm

Design of Big end:
Load on crankpin or the big end bearing (Fp) =Projected Area * Bearing pressure Fp
= *dp X lp X* P_{bp}

Fp = 29148.01Nforce or load on piston pin
dp = Inner dia. of big end
lp = length of crankpin = 1.3 dp
P_{bp} = 9 MPa
Putting these,

29148.01 = 1.3dp X dp X 9

Dp = 49.91 =50 mm
Lp = 1.3 X 50 = 65 mm

Design of Big end Bolts:

Force on bolts $= \frac{\pi}{4} d^2 \times \delta t \times Nb$

d_{cb} = Core dia. of bolts
δt = Allowable tensile stress for material of bolts (SAE 3130 = 156.667 MPa)
n_b = Number of bolts(2 bolts are used)

Force on bolts $= \frac{\pi}{4} d^2 \times \delta t \times Nb$

$9078.39 = \frac{\pi}{4} d^2 \times 156.667t \times 2$

D= 6.20mm

Nominal Dia of Bolt Db = $dcb/0.84$
Diameter of bolt = 7.38/ 0.84
Diameter of bolt = 8mm
Use M8 bolt.

Design of Big end Cap:

Maximum bending moment is taken as

$$B_{max} = \frac{Fi \times Lo}{6}$$

Lo= distance between bolt centre
= dia of crank pin + Nominal dia of bolt+ (2x thickness of bearing liner)+ Clearance

= 50+14+(2 X(0.05*50+1))+3
Lo = 74 mm

$$B_{max} = \frac{9078.39 \times 74}{6}$$

B_{max} = 111966.81 N.mm

Section Modulus for the cap

$$Z = \frac{b \times h^2}{6}$$

$$Z = \frac{65 \times h^2}{6}$$

$Z = 10.83\ h^2$

We know that bending stress

$$\delta b = \frac{Bmax}{z} \qquad \delta b = 120\text{MPa}$$

$$h^2 = \frac{111966.81}{120 \times 10.83}$$

h^2 =86.15
H = 9.28 mm
h = 10.00 mm

Sr.No	Parameters (mm)
01	Thickness of the connecting rod (t) = 4.5mm
02	Width of the section (B = 4t) = 18 mm
03	Height of the section(H = 5t) = 22.5 mm
04	Height at the big end =(1.1 to 1.125H) = 26.55 mm
05	Height at the small end =(0.9Hto0.75H)= 18.45mm
06	Inner diameter of the small end = 43mm
07	Outer diameter of the small end = 56mm
08	Inner diameter of the big end = 58mm
09	Outer diameter of the big end = 88mm
10	Centre distance of bolt = 76mm
11	Length of connecting rod =166mm.

Table 1: Dimensional Specification of connecting rod

4. MODELING OF THE CONNECTING ROD USING Inversion

Inventor software is used to create a complete 3D digital model of manufactured goods. The models consist of 2D and 3D solid model data which can also be used downstream in finite element analysis, rapid prototyping, tooling design, and CNC manufacturing.
Connecting rod of a Light Vehicle Engine easily available in the market is selected and its dimensions are calculated based on the design and working parameters. According the dimensions obtained the model of the connecting rod is developed in the Inventor.

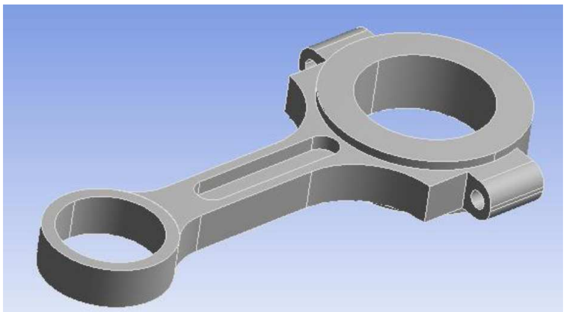


Fig 4: Model of connecting rod in Inventor

5. MATERIAL PROPERTIES

	Carbon Fiber
Young modulus	133.9 GPa
Poisson Ratio	0.10
Density	0.155 g/cc
Shear modulus	30 GPa
Tensile Strength, Yeild	1050MPa
Shear Strength	600 MPa

Table 2: Mechanical Properties used for Analysis.

Here we are using Ansys 14.5 to find the stresses, strain developed, deformation and safety factor of connecting rod at two variable speed.

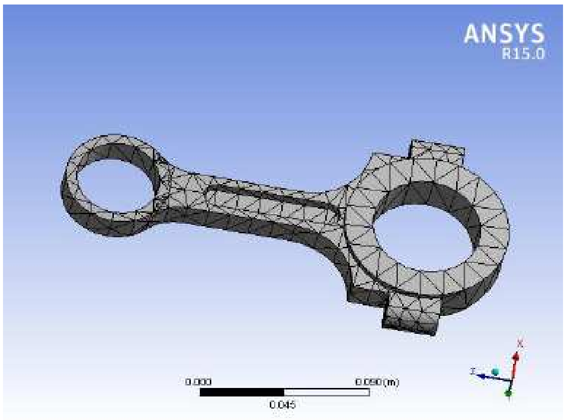


Fig 5: Mesh generation in Ansys 15.0

6. DYNAMIC LOAD ANALYSIS OF ROD

The objective of this chapter is to determine these loads that act on the connecting rod in an engine so that they may be used in FEA. The details of the analytical vector approach to determine the inertia loads and the reactions.

A. Analytical Vector Approach

The analytical vector approach has been discussed With reference to Figure for the case of zero offset (e = 0), for any given crank angle θ.

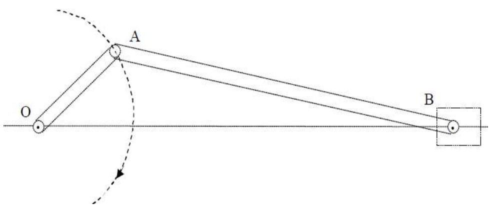


Fig 6: Slider Crank mechanism .

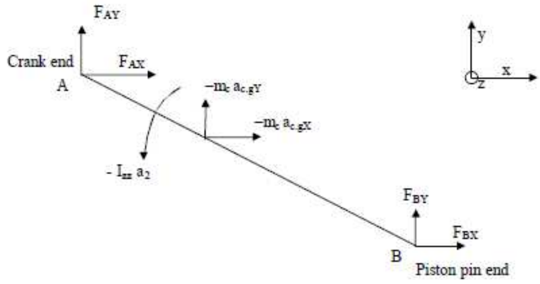


Fig 7: Free body diagram of connecting rod

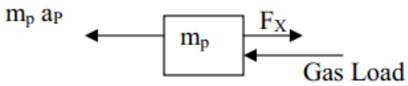


Fig 8: Free body diagram of connecting rod and piston.

The orientation of the connecting rod is given by:
$\beta = \sin\text{-}1\{-r1 \sin\theta / r2\}$.
Angular velocity of the connecting rod is given by the expression:
$\omega2 = \omega2\ k$
$\omega2 = -\ \omega1\cos\theta / [(r2/r1)2 - \sin2\theta]^{0.5}$

The angular acceleration of the connecting rod is given by the following relations
The angular acceleration of the connecting rod is given as
$\alpha2 = \alpha2\ k$
$\alpha2 = (1/\cos\beta)[\ \omega12\ (r1/r2) \sin\theta - \omega22 \sin\beta\ ]$

Forces at the piston pin and crank ends in X and Y directions are
given by:
$FBX = -(m_p\ a_p + \text{Gas Load})$
$FAX = m_c\ a_{c.gx} - FBX$
$FBY = [m_c\ a_{c.gY}\ u\cos\beta - m_c a_{c.gx}\ u \sin\beta + I_{zz}\ \alpha^2 + FBX\ r2 \sin\beta] / (r2\cos\beta)$
$FAY = m_c a_{c.gy} - FBY$.

Where
$a = (-r1\ \omega1^2 \cos\theta - \omega^2 2\ u \cos\beta - \alpha^2\ u \sin\beta)\ i + (-r1\ \omega1^2 \sin\theta - \omega^2\ u \sin\beta + \alpha^2\ u \cos\beta)\ j$

Acceleration of the piston is given by
$ap = (-\omega1^2\ r1 \cos\theta - \omega^2\ r^2 \cos\beta - \alpha^2\ r^2 \sin\beta)\ i + (-\omega1^2\ r1 \sin\theta - \omega^2 r^2 \sin\beta + \alpha^2\ r^2 \cos\beta)\ j$

Configuration of the engine connecting rod
Crank shaft radius = 41.5 mm
Connecting rod length = 166 mm

Piston diameter = 78 mm
Mass of the piston assembly =0. 250 kg
Mass of the connecting rod = 0.253 kg
Izz about the center of gravity = 0.00144 kg mm2
Distance of C.G. from crank end center = 36.44 mm
Maximum gas pressure = 6.1 Bar

B. Dynamic Load Analysis of Connecting rod @5600 rpm of crank speed

Consider piston movement inside the cylinder as slider crank mechanism.

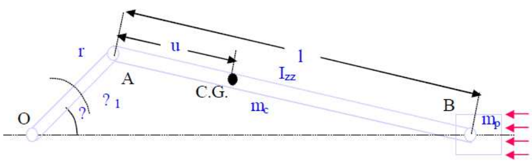


Fig 9: Piston Movement.

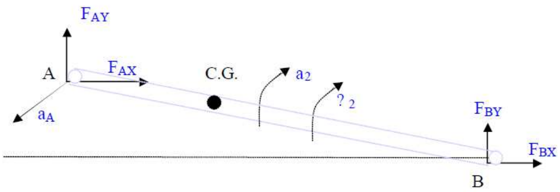
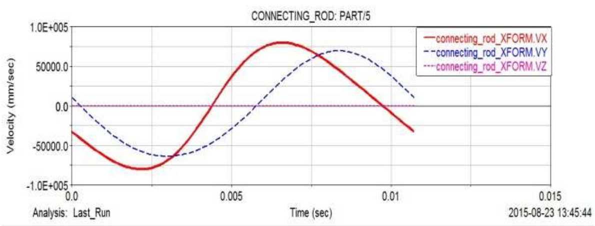
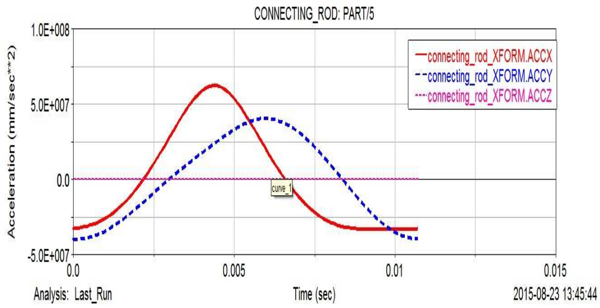


Fig 10: Input requires to perform load analysis on connecting rod.

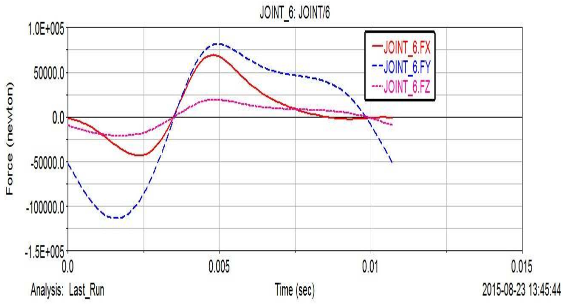
Typical input required for performing load analysis on the connecting rod and the expected output graphs are shown using Adams-View 2012. Here we have to calculate the angular acceleration of the connecting rod about centre of gravity



Graph 1: Angular velocity of link AB at 5600 rev/min crank speed (ccw).

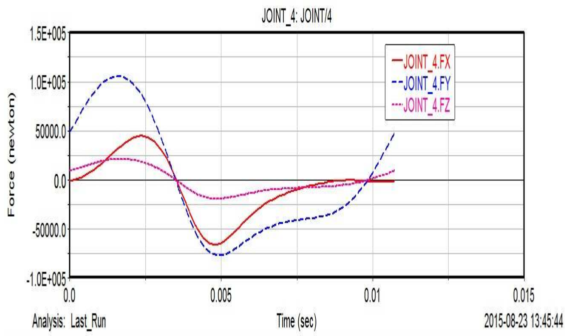


Graph 2: Angular acceleration of link AB at 5600 rev/min cranks speed (ccw).



Graph 3: Forces at piston Pin End Vs Time

Forces at the joint 6 (Piston Pin End) at 5600 rev/min crank speed. Fx Corresponds to F_{BX} and Fy corresponds to F_{BY}.



Graph 4: Forces at Crank Pin End Vs Time

Forces at the joint 4 (Crank Pin End) at 5600 rev/min crank speed. Fx Corresponds to F_{AX} and Fy corresponds to F_{AY}.
The data obtained by Adams View 2012 is used for the Dymanic Analysis of connecting rod at maximum engine speed of 5600 rpm of the crank shaft.

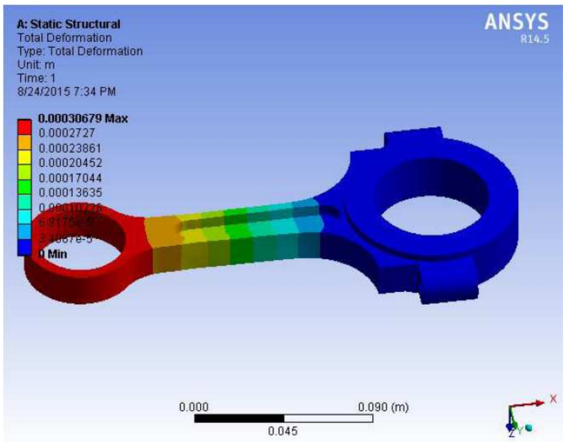


Fig 11: Total deformation of CF Connecting Rod @1.05E5 N at crank speed of 5600 rpm

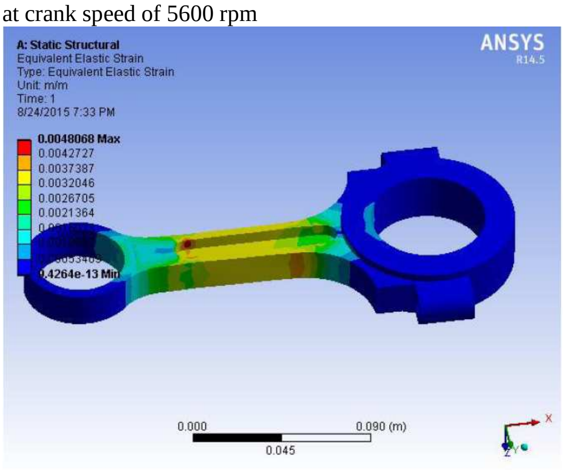


Fig 12: Elstic Strain of CF Connecting Rod @ 1.05E5 N at crank speed of 5600rpm

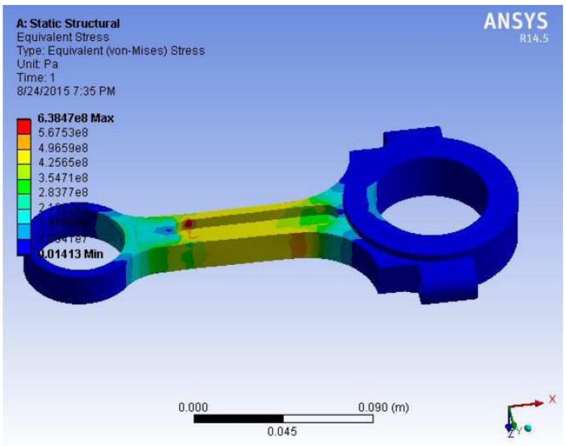


Fig 13: Von-Mises Stress of CF Connecting Rod @ 1.05E5 N at crank speed of 5600rpm.

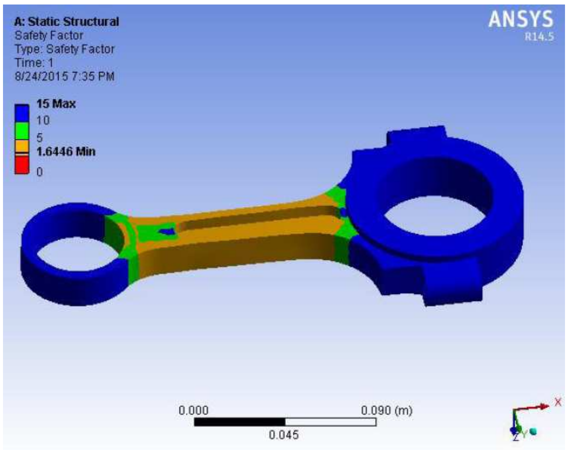
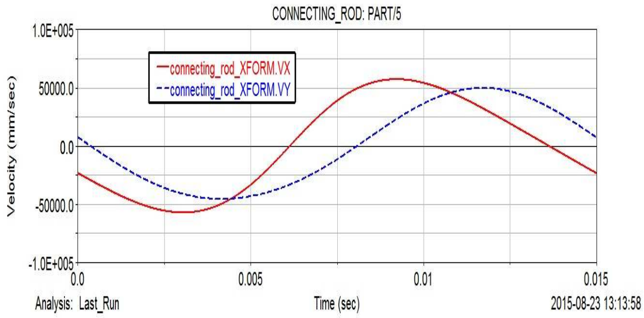


Fig 14: Safety Factor of CF Connecting Rod @ 1.05E5 N at a speed of 5600rpm.

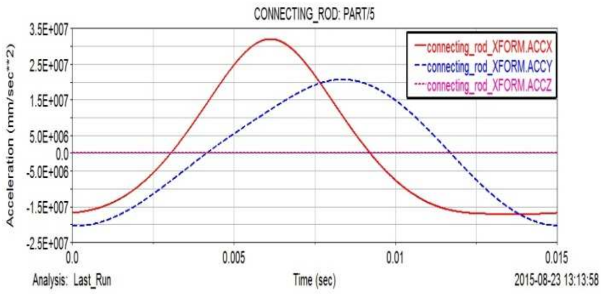
Crank speed (rpm)	Total Deform -ation (m)	Elastic Strain (m/m)	Von mises stress (MPa)	Safety Factor	Mass (Kg)
@5600	0.0003	0.0048	6.384e8	1.6447 to 15	0.253

Table 2: Results For Dynamic load Analysis of Connecting Rod @105 KN at crank speed of 5600rpm.

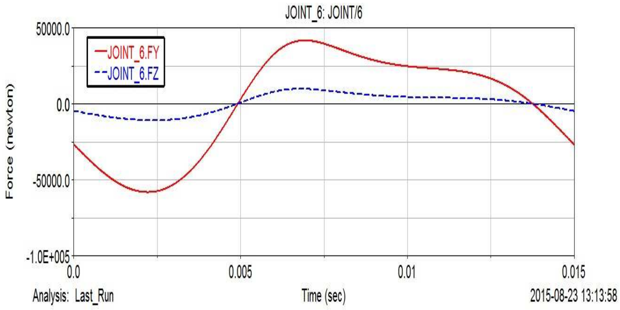
C. Dynamic Load Analysis of Connecting rod @5600 rpm of crank speed



Graph 5: Angular velocity of link AB at 4000 rev/min crank speed (ccw).

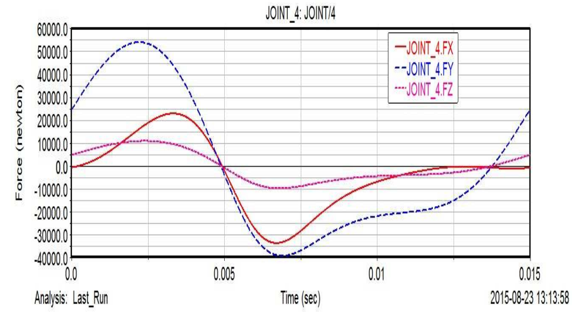


Graph 6: Angular acceleration of link AB at 5600 rev/min cranks speed (ccw).



Graph 7: Forces at piston Pin End Vs Time

Forces at the joint 6 (Piston Pin End) at 4000 rev/min crank speed. Fx Corresponds to F_{BX} and Fy corresponds to F_{BY}.



Graph 8: Forces at Crank Pin End Vs Time

Forces at the joint 4 (Crank Pin End) at 4000 rev/min crank speed. Fx Corresponds to F_{AX} and Fy corresponds to F_{AY}.

Results of Dymanic Load Analysis of connecting rod

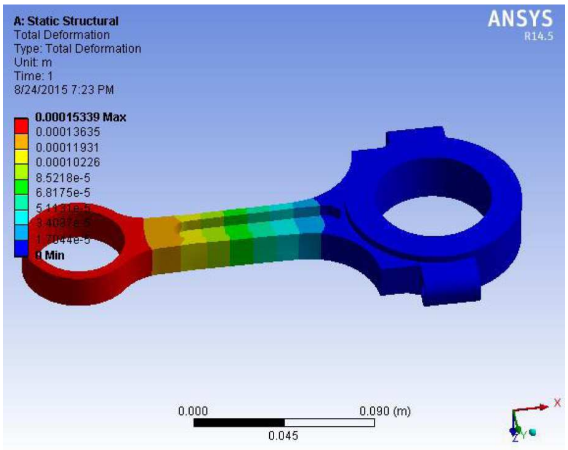


Fig 15: Total deformation of CF Connecting Rod @50KN at crank speed of 4000 rpm

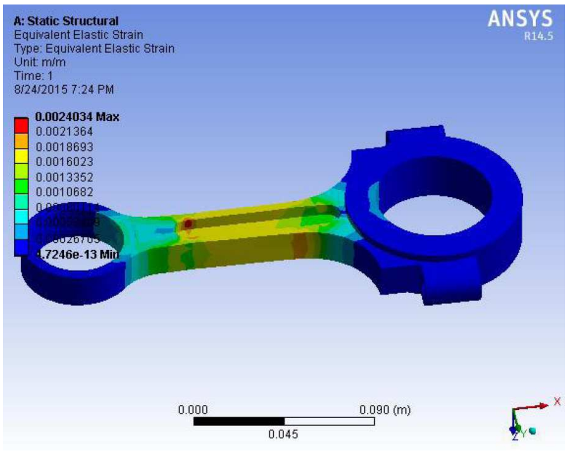


Fig 16: Total Elastic Strain of CF Connecting Rod @50KN at crank speed of 4000 rpm

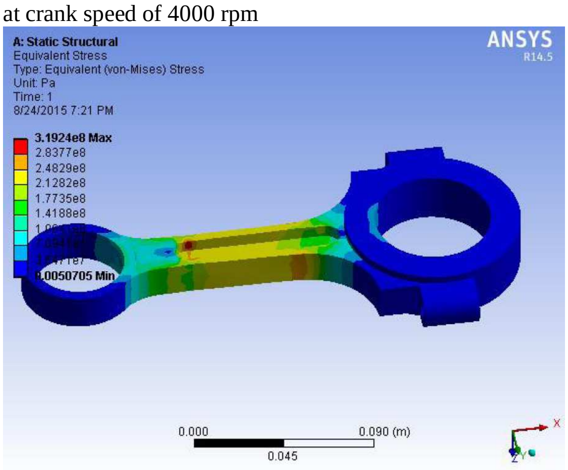


Fig 17: Von-Mises Stress of CF Connecting Rod @50KN at crank speed of 4000 rpm

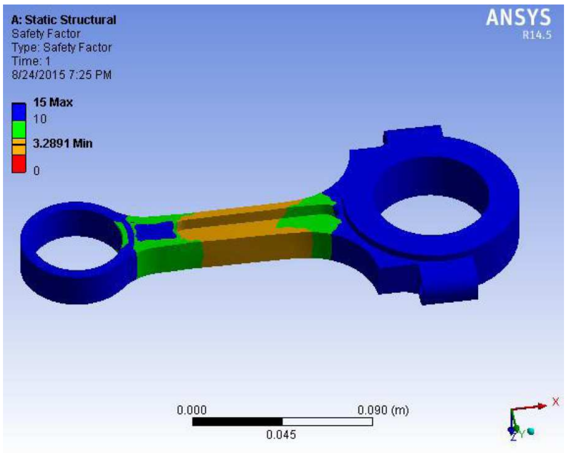


Fig 18: Safety Factor of CF Connecting Rod @ 50KN at a speed of 4000 rpm.

Crank speed (rpm)	Total Deform -ation (m)	Elastic Strain (m/m)	Von mises stress (MPa)	Safety Factor	Mass (Kg)
@4000	0.00015	0.0024	3.192e8	3.289 to 15	0.253

Table 3: Results For Dynamic load Analysis of Connecting Rod @50 KN at crank speed of 4000rpm.

7. CONCLUSION

The following conclusions can be drawn From this study:

1. There is considerable difference in the structural behavior of the connecting rod between axial loading and dynamic loading.
2. Dynamic load should be incorporated directly during design as the design loads, rather than using static loads.
3. Bending stresses and Tensile stresses of the connecting rod will be reduced. Bending stresses were also negligible at the piston pin end.
4. Due to less weight of carbon fiber inertia forces are neglected.
5. Due to light weight of connecting rod efficiency of engine will be increased.
6. In the cost orientation carbon fiber is more costlier as compared to Aluminum alloys, Titanium and Stainless Steel.

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