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Design And Analysis of Connecting Rod Using Forged steel

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Abstract – The connecting rod is the intermediate member between the piston and the Crankshaft. Its primary function is to transmit the push and pull from the piston pin to the crank pin, thus converting the reciprocating motion of the piston into rotary motion of the crank. This thesis describes designing and Analysis of connecting rod. Currently existing connecting rod is manufactured by using Carbon steel. In this drawing is drafted from the calculations. A parametric model of Connecting rod is modeled using CATIA V5 R19 software and to that model, analysis is carried out by using ANSYS 13.0 Software. Finite element analysis of connecting rod is done by considering the materials, viz... Forged steel. The best combination of parameters like Von misses Stress and strain, Deformation, Factor of safety and weight reduction for two wheeler piston were done in ANSYS software. Forged steel has more factor of safety, reduce the weight, increase the stiffness and reduce the stress and stiffer than other material like carbon steel. With Fatigue analysis we can determine the lifetime of the connecting rod.

Keywords: connecting Rod, Analysis of connecting rod, four stroke engine connecting rod, forged steel connecting rod, design and analysis of connecting rod.

Nomenclature

A = cross sectional area of the connecting rod.
L = length of the connecting rod.
C = compressive yield stress.
W_{cr} *= crippling or buckling load.*
I_{xx} *= moment of inertia of the section about x-axis*
I_{yy} *= moment of inertia of the section about y-axis respectively.*
K_{xx} *= radius of gyration of the section about x-axis*
K_{yy} *= radius of gyration of the section about y- axis respectively.*
D = Diameter of piston
r = Radius of crank

1. INTRODUCTION

In a reciprocating piston engine, the connecting rod connects the piston to the crank or crankshaft. In modern automotive internal combustion engines, the connecting rods are most usually made of steel for production engines, but can be made of aluminum (for lightness and the ability to absorb high impact at the expense of durability) or titanium (for a combination of strength and lightness at the expense of affordability) for high performance engines, or of cast iron for applications such as motor scooters. The small end attaches to the piston pin, gudgeon pin (the usual British term) or wrist pin, which is currently most often press fit into the con rod but can swivel in the piston, a "floating wrist pin" design. The connecting rod is under tremendous stress from the reciprocating load represented by the piston, actually stretching and being compressed with every rotation, and the load increases to the third power with increasing engine speed. Failure of a connecting rod, usually called "throwing a rod" is one of the most common causes of catastrophic engine failure in cars, frequently putting the broken rod through the side of the crankcase and thereby rendering the engine irreparable; it can result from fatigue near a physical defect in the rod, lubrication failure in a bearing due to faulty maintenance or from failure of the rod bolts from a defect, improper tightening, or re-use of already used (stressed) bolts where not recommended. Despite their frequent occurrence on televised competitive automobile events, such failures are quite rare on production cars during normal daily driving. This is because production auto parts have a much larger factor of safety, and often more systematic quality control. When building a high performance engine, great attention is paid to the connecting rods, eliminating stress risers by such techniques as grinding the edges of the rod to a smooth radius, shot peening to induce compressive surface stresses (to prevent crack initiation), balancing all connecting rod/piston assemblies to the same weight and Magnafluxings to reveal otherwise invisible small cracks which would cause the rod to fail under stress. In addition, great care is taken to torque the con rod bolts to the exact value specified; often these bolts must be replaced rather than reused. The big end of the rod is fabricated as a unit and cut or cracked in two to establish precision fit around the big end bearing shell. Recent engines such as the Ford 4.6 liter engine and the Chrysler 2.0 liter engine have connecting rods made using powder metallurgy, which allows more precise control of size and weight with less machining and less excess mass to be machined off for balancing. The cap is then separated from the rod by a fracturing process, which results in an uneven mating surface due to the grain of the powdered metal. This ensures that upon reassembly, the cap will be perfectly positioned with respect to the rod, compared to the minor misalignments which can occur if the mating surfaces are both flat. A major source of engine wear is the sideways force exerted on the piston through the con rod by the crankshaft, which typically wears the cylinder into an oval cross-section rather than circular, making it impossible for piston rings to correctly seal against the cylinder walls. Geometrically, it can be seen that longer connectin rods will reduce the amount of this sideways force, and therefore lead to longer engine life. However, for a given engine block, the sum of the length of the con rod plus the piston stroke is a fixed number, determined by the fixed distance between the crankshaft axis and the top of the cylinder block where the cylinder head fastens; thus, for a given cylinder block longer stroke, giving greater engine displacement and power, requires a shorter connecting rod (or a piston with smaller compression height), resulting in accelerated cylinder wear.

2. SPECIFICATION OF THE PROBLEM

The objective of the present work is to design and analyses of connecting rod made of Forged steel. Steel materials are used to design the connecting rod. In this project the material (carbon steel) of connecting rod replaced with Forged steel .Connecting rod was created in CATIAV5 R19. Model is imported in ANSYS 13.0 for analysis. After analysis a comparison is made between existing steel connecting rod viz., Forged steel in terms of weight, factor of safety, stiffens, deformation and stress.

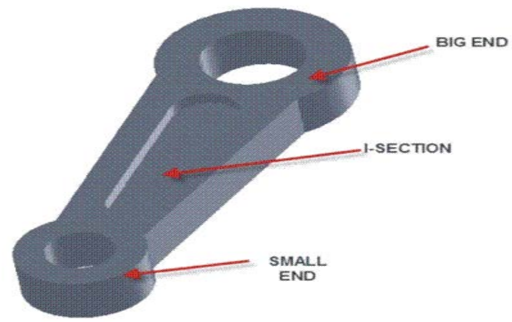

Fig 2.1 Schematic Diagram of Connecting Rod

3. DESIGN OF CONNECTING ROD

A connecting rod is a machine member which is subjected to alternating direct compressive and tensile forces. Since the compressive forces are much higher than the tensile force, therefore the cross-section of the connecting rod is designed as a strut and the rankine formula is used. A connecting rod subjected to an axial load W may buckle with x-axis as neutral axis in the plane of motion of the connecting rod,{or} y-axis is a neutral axis. The connecting rod is considered like both ends hinged for buckling about x-axis and both ends fixed for buckling about y-axis. A connecting rod should be equally strong in buckling about either axis.

According to rankine formulae

W_{cr} about x-axis

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

$$[\therefore for\ both\ ends\ hinged\ L = l]$$

W_{cr} about y-axis

$$= \frac{[\sigma_c \times A]}{1+a[\frac{L}{Kyy}]^2} = \frac{[\sigma_c \times A]}{1+a[\frac{l}{2Kyy}]^2} \qquad [\therefore for\ both\ ends\ fixed\ L = l/2]$$

In order to have a connecting rod equally strong in buckling about both the axis, the buckling loads must be equal. i.e.

$$= \frac{[\sigma_c \times A]}{1+a[\frac{l}{Kxx}]^2} = \frac{[\sigma_c \times A]}{1+a[\frac{l}{2Kyy}]^2} \text{ [or]}$$

$[\frac{l}{Kxx}]^2 = [\frac{l}{2Kyy}]^2$

$K^2_{xx} = 4K^2_{yy}$ [or] $I_{xx} = 4I_{yy}$ [$\therefore I = A \times K^2$]

This shows that the connecting rod is four times strong in buckling about y-axis than about-axis. If $I_{xx} > 4I_{yy}$, Then buckling will occur about y-axis and if $I_{xx} < 4I_{yy}$, then buckling will occur about x-axis .In Actual practice I_{xx} is kept slightly less than $4I_{yy}$. It is usually taken between 3 and 3.5 and the Connecting rod is designed for buckling about x-axis. The design will always be satisfactory for buckling about y-axis. The most suitable section for the connecting rod is I-section with the proportions shown mfg.

Area of the cross section = 2[4t x t] + 3t x t=11t²

Moment of inertia about x-axis = 2[4txt]+3txt=11t²

Moment of inertia about x-axis

$I_{xx} = \frac{1}{12}[4t\,\{5t\}^3 - 3t\,\{3t\}^3] = \frac{419}{12}[t^4]$

And moment of inertia about y-axis

$I_{yy} = \frac{2\times1}{12} \times t \times \{4t\}^3 + \frac{1}{12}\{3t\}t^3 = \frac{131}{12}[t^4]$

I_{xx}/I_{yy} = [419/12]x[12/131]=3.2

Since the value of I_{xx}/I_{yy} lies between 3 and 3.5 m therefore I-section chosen is quite satisfactory.

3.1 Pressure Calculation for 150cc Engine

Suzuki 150 cc Specifications

Engine type air cooled 4-stroke

Bore x Stroke (mm) = 57×58.6

Displacement = 149.5 CC

Maximum Power = 13.8 bhp @ 8500 rpm

Maximum Torque = 13.4 Nm @ 6000 rpm

Compression Ratio = 9.35/1

Density of Petrol C_8H_{18} = 737.22 kg/m³

= 737.22E-9 kg/mm³

Temperature = 60 °F

= 288.855 °K

Mass = Density × Volume

= 737.22E-9 x149.5E3

= 0.11kg

Molecular Weight of Petrol 114.228 g/mole

From Gas Equation,

PV = Mrt R

$= \frac{R_x}{M_w}$

= 8.3143/114228

= 72.76

$P = \frac{(0.11\times72.786\times288.85)}{149.5E^3}$

P = 15.5 Mpa.

3.2 Design Calculations for Existing Connecting Rod

Thickness of flange & web of the section = t

Width of section B= **4t**

The standard dimension of I - SECTION.

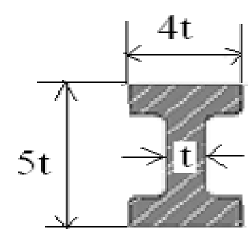

Fig 3.1 Standard Dimension of I – Section

Height of section H **= 5t**

Area of section A= 2(4t×t) +3t×t

A **= 11t²**

M.O.I of section about x axis:

$I_{xx} = \frac{1}{12}[4t\,\{5t\}^3 - 3t\,\{3t\}^3]$

$= \frac{419}{12}[t^4]$

MI of section about y axis:

$I_{yy} = \frac{2\times1}{12} \times t \times \{4t\}^3 + \frac{1}{12}\{3t\}t^3$

$= \frac{131}{12}[t^4]$

$\frac{Ixx}{Iyy}$ **= 3.2**

Length of connecting rod (L) = 2 times the stroke

L **= 117.2 mm**

Buckling load W_B = maximum gas force × F.O.S

$W_B = \frac{(\sigma_c \times A)}{(1+a\ (L/Kxx)2}$

= 37663N

σ_c= compressive yield stress = **415MPa**

$K_{xx} = \frac{I\,xx}{A}$

K_{xx} = 1.78t

$a = \frac{\sigma_c}{\pi^2 E}$

a = **0.0002**

By substituting σ_c, A, a, L, K_{xx} on W_B then

$= 4565t^4 - 37663t^2 - 81639.46 = 0$

$t^2 = 10.03$

t = 3.167mm

t = 3.2mm

Width of section B = 4t

= 4×3.2

= 12.8mm

Height of section H = 5t

= 5×3.2

= 16mm

Area A = $11t^2$

=11×3.2×3.2

= 112.64mm²

Height at the big end (crank end) = H_2

= 1.1H to 1.25H

= 1.1×16

H_2 **=17.6mm**

Height at the small end (piston end) = 0.9H to 0.75H

= 0.9×16

H_1 **=12mm**

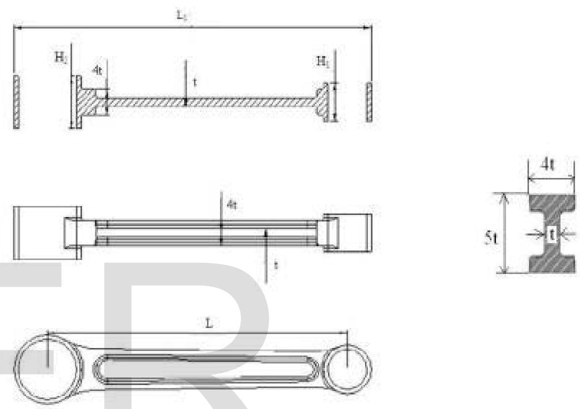

Fig 3.2 2D Drawing for Connecting Rod

Stroke length (l) =**117.2mm**

Diameter of piston (D) =**57mm**

P=**15.5N/mm²**

Radius of crank(r) =stroke length/2

=58.6/2

=29.3

Maximum force on the piston due to pressure

$F_l = \frac{\pi}{4xD^2xp}$

$=\pi/4 \times (57)^2 \times 15.469$

=39473.16N

Maximum angular speed $W_{max} = \frac{[2\pi Nmax]}{60}$

$= \frac{[2\pi\times8500]}{60} A = \pi r^2$

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=768 rad/sec

Ratio of the length of connecting rod to the radius of crank

N= $\frac{l}{r}$ =112/ (29.3) = **3.8**

Maximum Inertia force of reciprocating parts

$F_{im} = Mr\,(W_{max})^2\, r\,(\cos\theta + \frac{\cos 2\theta}{n})$ (Or)

$F_{im} = Mr\,(Wmax)^2\, r\,(1+\frac{1}{n})$

= 0.11x (768)^2 x (0.0293) x (1+ (1/3.8))

F_{im} = **2376.26N**

Inner diameter of the small end $d_1 = \frac{F_g}{Pb_1 \times l_1}$

$= \frac{6277.167}{12.5 \times 1.5 d_1}$

= 17.94mm

Where,

Design bearing pressure for small end p_{b1}=12.5 to 15.4N/mm^2

Length of the piston pin l_1= (1.5to 2) d_1

Outer diameter of the small end = $d_1+2t_b+2t_m$

= 17.94 + [2×2] + [2×5]

= **31.94mm**

Where,

Thickness of the bush (t_b) = 2 to 5 mm

Marginal thickness (t_m) = 5 to 15 mm

Inner diameter of the big end $d_2 = \frac{F_g}{Pb_2 \times l_2}$

$= \frac{6277.167}{10.8 \times 1.0 d_1}$

=23.88mm

Where,

Design bearing pressure for big end p_{b2} = 10.8 to 12.6N/mm^2

Length of the crank pin l_2 = (1.0 to 1.25) d_2

Root diameter of the bolt = $(\frac{(2F_{im})}{(\pi x St)})^{1/2}$

$= (\frac{2 \times 6277.167}{\pi \times 56.667})^{1/2}$

= **4mm**

Outer diameter of the big end = $d_2 + 2t_b + 2d_b + 2t_m$

= 23.88+2×2+2×4+2×5

= 47.72mm

Where,

Thickness of the bush [t_b] = 2 to 5 mm

Marginal thickness [t_m] = 5 to 15 mm

Nominal diameter of bolt [d_b] = 1.2 x root diameter of the bolt

= 1.2× 4 = **4.8mm**

3.3 Specifications of connecting rod

Table 3.3.1

Sno	Parameters (mm)
1	Thickness of the connecting rod (t) = 3.2
2	Width of the section (B = 4t) = 12.8
3	Height of the section(H = 5t) = 16
4	Height at the big end = (1.1 to 1.125)H = 17.6
5	Height at the small end = 0.9H to 0.75H= 14.4
6	Inner diameter of the small end = 17.94
7	Outer diameter of the small end = 31.94
8	Inner diameter of the big end = 23.88
9	Outer diameter of the big end = 47.72

4. MODELING OF CONNECTING ROD

Making of Stem

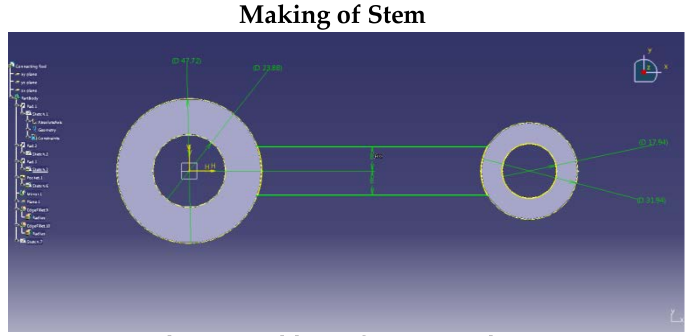

Fig 4.1 Making of Stem Pad

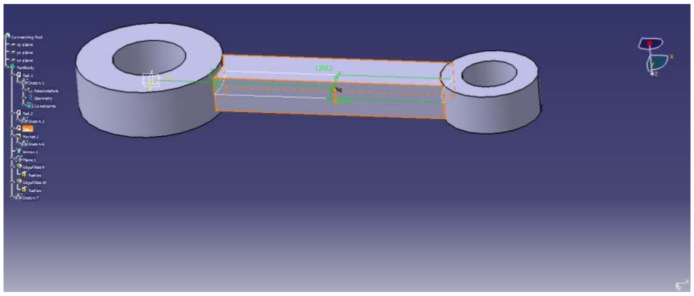

Fig 4.2 Stem Pad Sketch

Weight Reduction in Stem Pocket

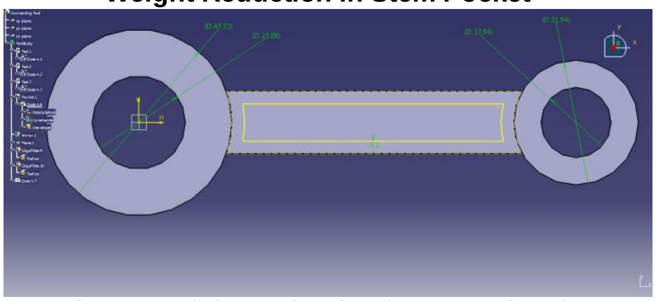

Fig 4.3 Weight Reduction in Stem Sketch

Pocket

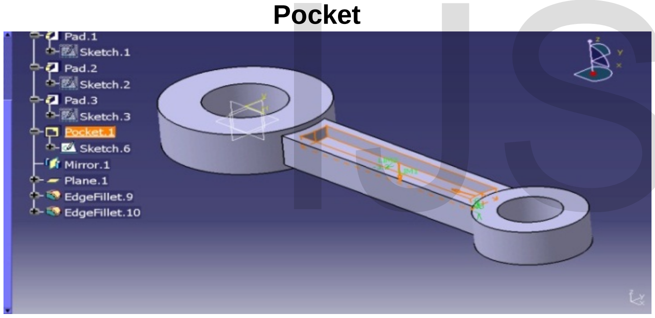

Fig 4.4 Pocket Sketch

Mirror Pocket

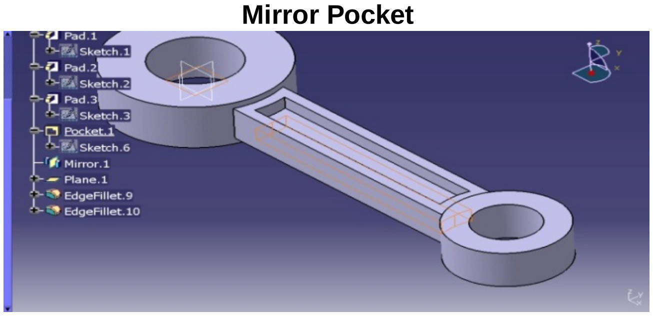

Fig 4.5 Mirror Pocket Sketch

Making of Edge Fillet (Radius =4.8mm)

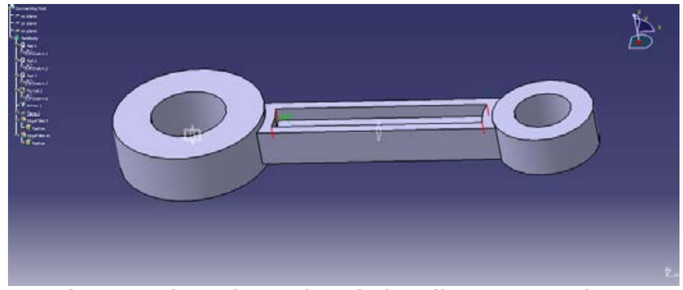

Fig 4.6 Edge Fillet Sketch (Radius =4.8mm)

Making of Edge Fillet (radius =8mm)

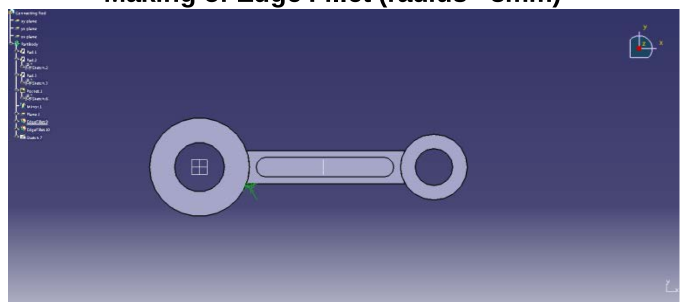

Fig 4.7 Edge Fillet Sketch (radius =8mm)

Connecting rod

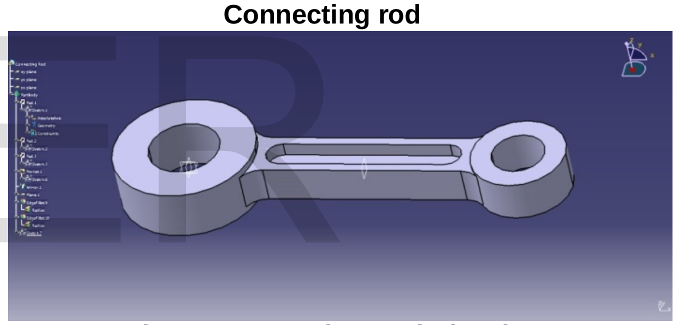

Fig 4.8 Connecting Rod Sketch

5. ANALYSIS OF THE CONNECTING ROD

Modified Connecting Rod (Forged Steel)

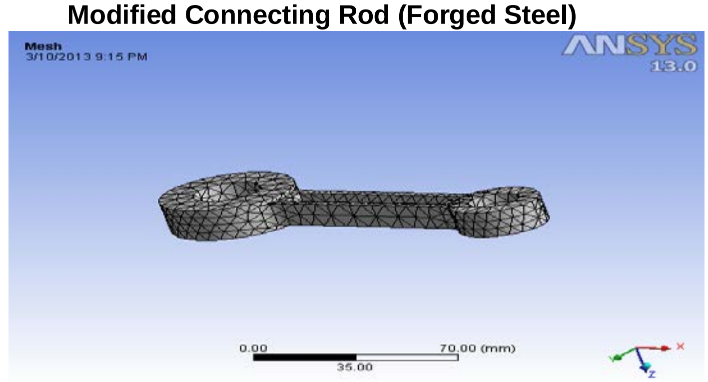

Fig 5.1 Meshing of Connecting Rod in Tetrahedral

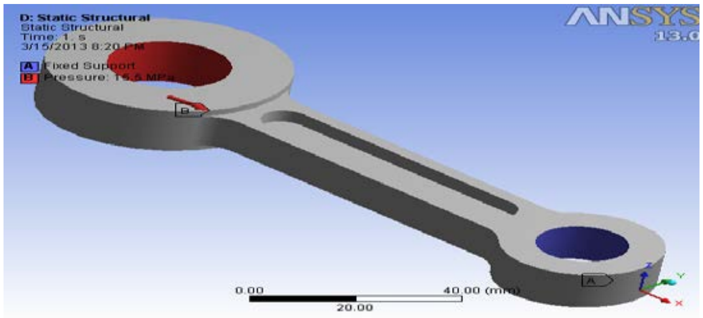

Fig 5.2 Loads at Boundary Conditions

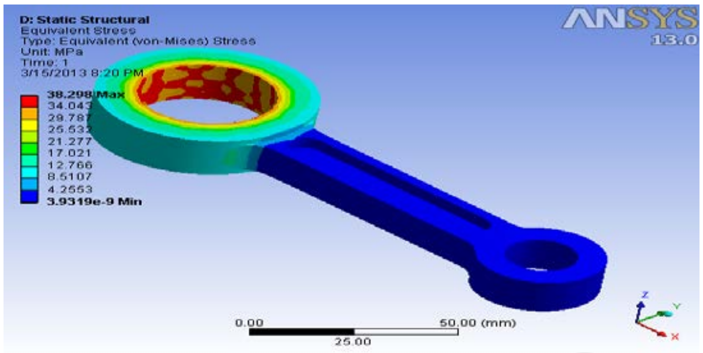

Fig 5.3 Equivalent Stress

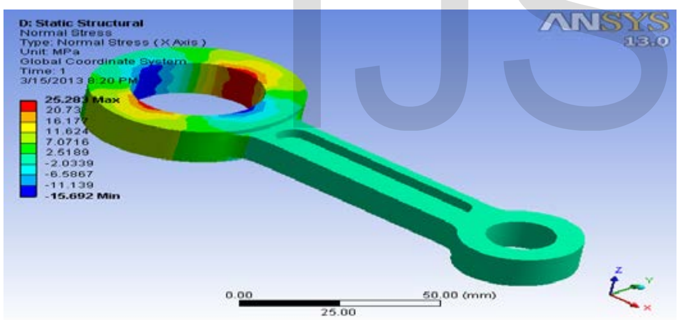

Fig 5.4 Normal Stress (X-Axis)

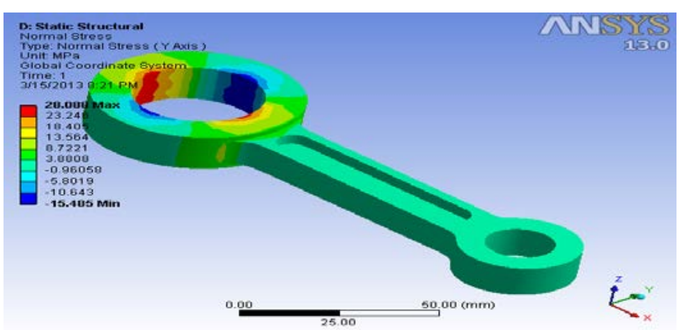

Fig 5.5 Normal Stress (Y-Axis)

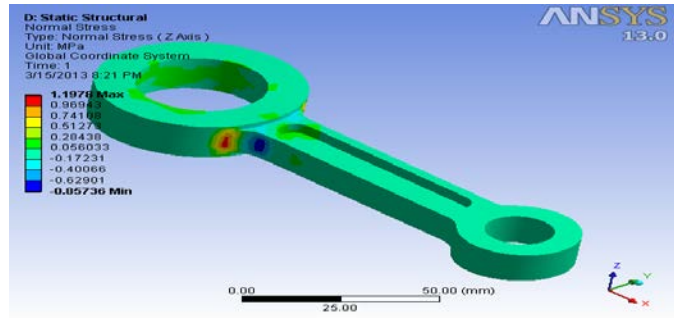

Fig 5.6 Normal Stress (Z-Axis)

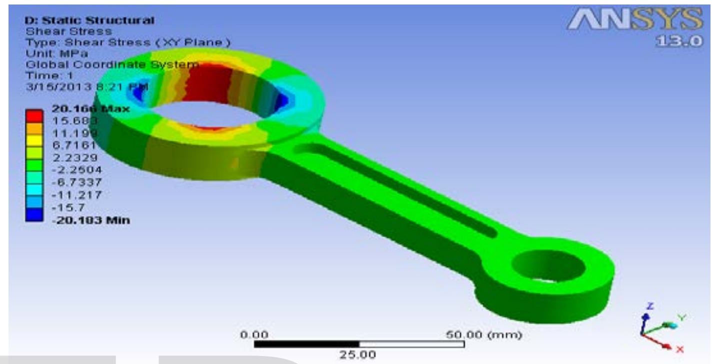

Fig 5.7 Shear Stress (XY Plane)

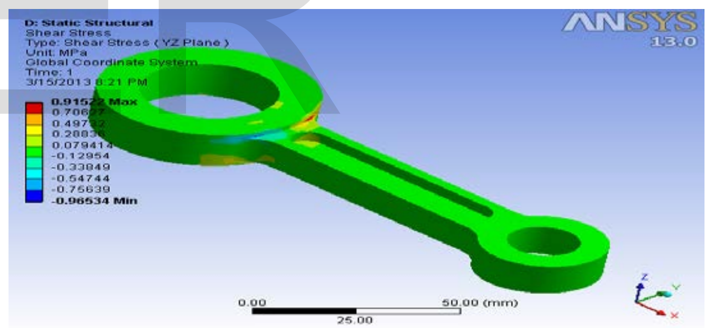

Fig 5.8 Shear Stress (YZ Plane)

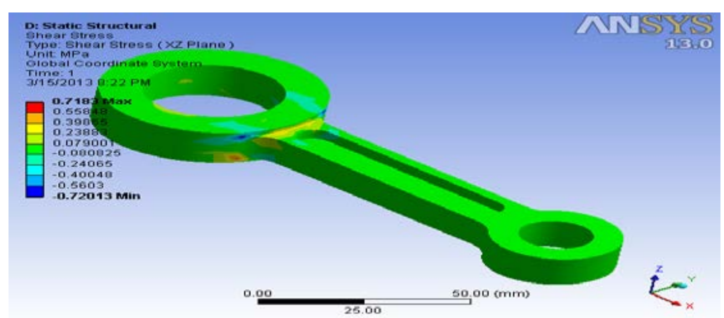

Fig 5.9 Shear Stress (ZX Plane)

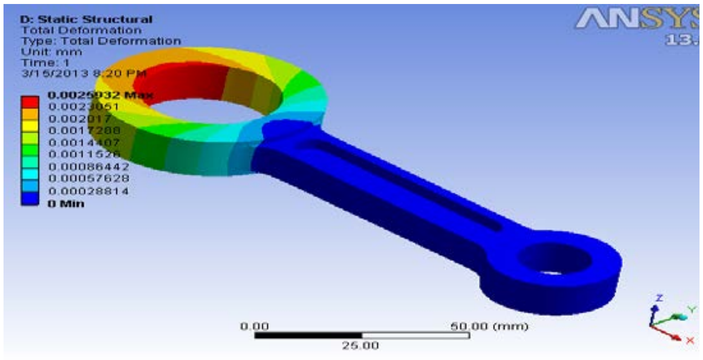

Fig 5.10 Total Deformations

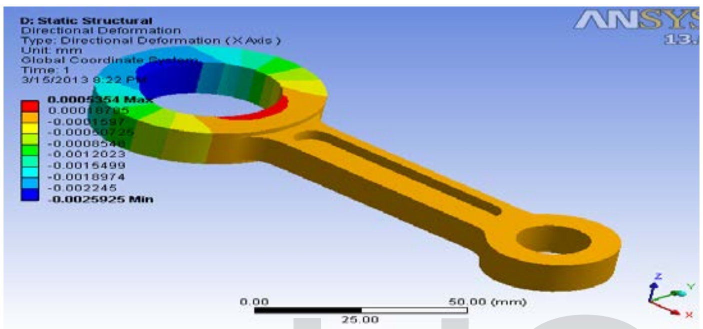

Fig 5.11 Directional Deformations (X Axis)

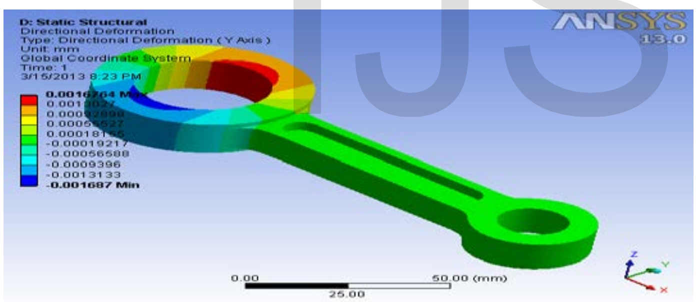

Fig 5.12 Directional Deformations (Y Axis)

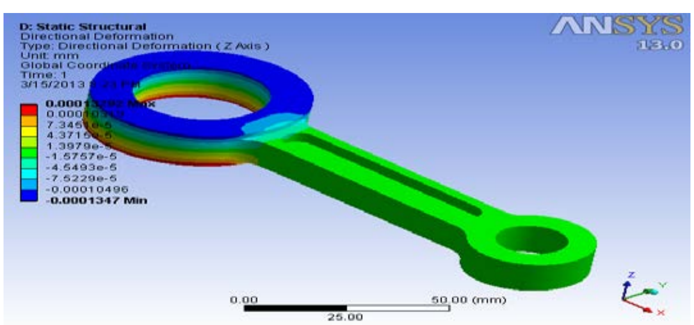

Fig 5.13Directional Deformations (Z Axis)

TABLE 5.1

Stresses and Deformation of Forged Steel

Sno	Types	Max (Mpa)	Min (Mpa)
1.	Equivalent stress	38.298	$4.0317e^{-9}$
2.	Normal stress(x-axis)	25.283	-15.692
3.	Normal stress(y-axis)	28.088	-15.485
4.	Normal stress(z-axis)	1.1978	-0.85736
5.	Shear stress(xy plane)	20.166	-20.183
6.	Shear stress(yz plane)	0.91522	-0.96534
7.	Shear stress(zx plane)	0.7183	-0.72013
8.	Total deformation	0.0025932	0
9.	Directional deformation (x-axis)	0.0005354	-0.0025925
10.	Directional deformation (y-axis)	0.0016764	-0.007687
11.	Directional deformation (z-axis)	0.00013292	-0.0001347

TABLE 5.2

Mechanical properties for forged steel

Sno.	Mechanical Properties	Forged Steel
1.	Density(g/cc)	7.7
2.	Average hardness(HRB)	101
3.	Modulus of elasticity,(Gpa)	221
4.	Yield strength, YS,(Mpa)	625
5.	Ultimate strength ,S_u,(Mpa)	625
6.	Percent reduction in area,%, RA	58
7.	Poison ratio	0.29

6. CHEMICAL COMPOSITION OF FORGED STEEL

Forged Steel 0.61-0.68%C, 0.2-0.4%S, 0.5-1.2%Mn, 0.04%S, 0.04%P, 0.9-1.2%Cr

7. CALCULATION

7.1 Calculation for factor of safety of connecting rod

f.s = factor of safety

σ_m = mean stress

σ_y = yield stress

σ_v = variable stress

σ_e = endurance stress

$$\frac{1}{f.s} = \frac{\sigma m}{\sigma y} + \frac{\sigma v}{\sigma e}$$

For Forged Steel

σ_{max} = 38.298 σ_{min} = 4.0317×10^{-9}

$\sigma_m = \frac{\sigma max + \sigma min}{2} = 19.149$

σ_y = 625Mpa

$\sigma_v = \frac{\sigma max - \sigma min}{2} = 19.149$

$\sigma_e = 0.6 \times 625 = 375$

$\frac{1}{f.s} = 0.081 = 12.23$

Factor of safety [F.S] = 12.23

7.2 Calculation for Weight and Stiffness

For Forged Steel:

Density of forged steel = 7.7×10^{-6} kg/mm^3

Volume = 41050 mm^3

Deformation = 0.0025932 mm

Weight of forged steel = volume × density

= $41050 \times 7.7 \times 10^{-6}$

= 0.31kg

= 0.31×9.81 = **3.10 N**

Stiffness = $\frac{\text{weight}}{\text{deformation}}$

$= \frac{3.10}{0.0025932}$

=1195.74 N/mm

7.3 Fatigue calculation

Result for fatigue of connecting rod:

$$N = 1000\left(\frac{s_f}{0.9\sigma_u}\right)^{\frac{3}{\log}} \left(\frac{\sigma_e'}{0.9 \times \sigma_u}\right)$$

Where,

N = No. of cycles

σ_e = Endurance Limit

σ_u = Ultimate Tensile Stress

σ_e' = Endurance limit for variable axial stress

k_a = Load correction factor for reversed axial load = 0.8

k_{sr} = Surface finish factor = 1.2

k_{sz} = Size factor = 1

$\sigma_e' = \sigma_e \times k_a \times k_{sr} \times k_{sz}$

$s_f = \frac{f.s\sigma_v}{1 - \frac{f.s\sigma_m}{\sigma_u}}$

For Forged Steel

σ_u=827 Mpa

$\sigma_e = \sigma_u \times 0.5$

= 827×0.5

= 413.5 Mpa

$\sigma_e' = \sigma_e \times k_a \times k_{sr} \times k_{sz}$

$= 413.5 \times 0.8 \times 1.2 \times 1$

= 396.96 Mpa

$s_f = \frac{f.s\sigma_v}{1 - \frac{f.s\sigma_m}{\sigma_u}}$

$= \frac{12.23 \times 19.149}{1 - \frac{12.23 \times 19.149}{827}}$

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$$=\frac{234.193}{0.7168}$$

$$= 326.713\text{Mpa}$$

$$N = 1000\left(\frac{s_f}{0.9\sigma_u}\right)^{\frac{3}{\log}}\left(\frac{\sigma_e'}{0.9\times\sigma_u}\right)$$

$$= 1000\left(\frac{326.713}{0.9\times827}\right)^{\frac{3}{\log}}\left(\frac{396.96}{0.9\times827}\right)$$

$$= 8500\times 10^3 \text{ cycles}$$

CONCLUSION

By checking and comparing the results of materials in finalizing the results are shown in below.

Considering the parameters,

1. ANSYS Equivalent stress for the both the materials are same.
2. For the forged steel material factor of safety (from Soderberg's) and stiffness is increased compared to existing carbon steel.
3. The weight of the forged steel material is less than the existing carbon steel.
4. From the fatigue analysis life time of the connecting rod can be determined.
5. And also no. of cycles for forged steel (8500×10^3)is more than the existing connecting rod (6255×10^3).
6. When compared to both of the materials, forged steel is cheaper than the existing connecting rod material.

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REFERENCES

[1] Afzal, A. and A. Fatemi, 2004. "A comparative study of fatigue behavior and life predictions of forged steel and PM connecting rods". SAE Technical Paper

[2] Chen, N., L. Han, W. Zhang and X. Hao, 2006. "Enhancing Mechanical Properties and Avoiding Cracks by Simulation of Quenching Connecting Rod". Material Letters, 61: 3021-3024.

[3] El – Sayed, M.E.M. and E.H. Lund, 1990. "Structural optimization with fatigue life constraints," Engineering Fracture Mechanics, 37(6): 1149-1156.

[4] Jahed Motlagh, H.M. Nouban and M.H. Ashraghi, 2003. "Finite Element ANSYS". University of Tehran Publication, PP: 990.

[5] Khanali, M., 2006. "Stress analysis of frontal axle of JD 955 combines". M.Sc. Thesis. Thran University, 124.

[6] Repgen, B., 1998. "Optimized Connecting Rods to Enable Higher Engine Performance and Cost Reduction," SAE Technical Paper Series, Paper No. 980882.

Books

1. Machine design by R.S. KHURMI, J.K GUPTA.
2. Design data by PSG.
3. A text book of Machine Design by S.Md. Jalaludeen.

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