

Comparission of two Different Materials for Connecting Rod using Ansys

T Arun Kumar Reddy
M. Tech Student (Computer Aided Design)
Department of Mechanical Engineering
SRM University, Chennai

M.Sachidhanandam
Assistant Professor (O.G)
Department of Mechanical Engineering
SRM University, Chennai

Abstract

The function of the connecting rod is to convert the reciprocating motion of the piston into the rotary motion of the crankshaft. To reduce the obliquity of the rod with the cylinder axis, its length should be kept as large as possible. Reduced obliquity decreases the oscillating angular motion of the rod about its small end, thereby decreasing the piston side-thrust and improving the reciprocating balance of the engine. The connecting rods are generally made by forged steel or duralumin or carbon steel. The connecting rods these days are also cast from malleable or spheroidal graphite cast iron. In the present work, the modeling of connecting rod is developed using CATIA V5/R20. Later, the analysis is carried out for different end conditions of connecting rod using ANSYS12 software package. From the analysis, the 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.

Keywords: Connecting Rod, Modeling, Stress Analysis of Connecting Rod, Fatigue Analysis, Optimization

I. INTRODUCTION

The automobile engine connecting rod is a high volume production, critical component. It connects reciprocating piston to rotating crankshaft, transmitting the thrust of the piston to the crankshaft. Every vehicle that uses an internal combustion engine requires at least one connecting rod depending upon the number of cylinders in the engine. Connecting rods for automotive applications are typically manufactured by forging from either wrought steel or powdered metal. They could also be cast. However, castings could have blow-holes which are detrimental from durability and fatigue points of view. The fact that forgings produce blow-hole-free and better rods gives them an advantage over cast rods (Gupta, 1993). Between the forging processes, powder forged or drop forged, each process has its own pros and cons. Powder metal manufactured blanks have the advantage of being near net shape, reducing material waste. However, the cost of the blank is high due to the high material cost and sophisticated manufacturing techniques (Reppen, 1998). With steel forging, the material is inexpensive and the rough part manufacturing process is cost effective.

Bringing the part to final dimensions under tight tolerance results in high expenditure for machining as the blank usually contains more excess material (Reppen, 1998). A sizeable portion of the US market for connecting rods is currently consumed by the powder metal forging industry. A comparison of the European and North American connecting rod markets indicates that according to an unpublished market analysis for the year 2000 (Ludenbach, 2002), 78% of the connecting rods in Europe (total annual production: 80 million approximately) are steel forged as opposed to 43% in North America (total annual production: 100 million approximately), as shown in Figure. In order to recapture the US market, the steel industry has focused on development of production technology and new steels. AISI (American Iron and Steel Institute) funded a research program that had two aspects to address. The first aspect was to investigate and compare fatigue strength of steel forged connecting rods with that of the powder forged connecting rods. The second aspect was to optimize the weight and manufacturing cost of the steel forged connecting rod. Due to its large volume production, it is only logical that optimization of the connecting rod for its weight or volume will result in large-scale savings. It can also achieve the objective of reducing the weight of the engine component, thus reducing inertia loads, reducing engine weight and improving engine performance and fuel economy.

II. OBJECTIVE

The main aim of the project is to determine the Von Misses stresses, Shear stresses, and Equivalent Alternating stress, Total Deformation, Fatigue Analysis and Optimization in the existing Connecting rod. If the existing design shows the failure, then suggest the minimum design changes in the existing Connecting rod. A lot has been done and still a lot has to be done in this field. In this Project, only the static FEA of the connecting rod has been performed by the use of the software. This work can be extended to study the effect of loads on the connecting rod under dynamic conditions. Experimental stress analysis (ESA) can also be used to calculate the stresses which will provide more reasons to compare the different values obtained. Now a day a lot is being said about vibration study of mechanical component important role in its failure. So the study can be extended to the vibration analysis of the connecting rod. The study identified fatigue strength as the most significant design factor in the

optimization process. Then the combination of finite element technique with the aspects of weight reduction is to be made to obtain the required design of connecting rod.

III. MATERIALS USED IN THIS PROJECT

The connecting rods are usually forged out of the open hearth steel or sometimes even nickel steel or vanadium steel. For low to medium capacity high speed engines, these are often made of duralumin or other aluminum alloys. However, with the progress of technology, the connecting rods these days are also cast from malleable or spheroidal graphite cast iron. The different connecting rod steels are (40C8, 37Mn6, 35Mn6 MO3, 35Mn6 Mo4, 40Cr4, 40Cr4 Mo3, 40NiCr4MO2) etc. In general, forged connecting rods are compact and light weight which is an advantage from inertia view point, whereas cast connecting rods are comparatively cheaper, but on account of lesser strength their use limited to small and medium size petrol engines.

IV. 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.

A. Gas Pressure Force F_p

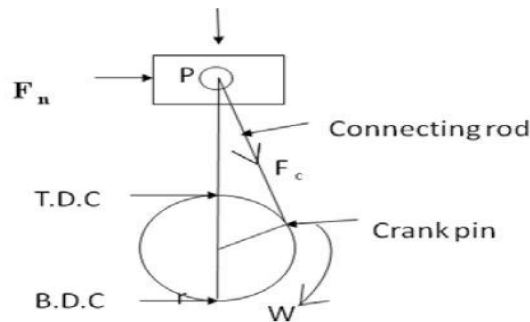


Fig. 1: Forces on connecting rod and crankshaft

The load due to piston inertia is = weight of the reciprocating masses X accelerations

$$F_i = F/g \omega^2 r (\cos \theta + r (\cos 2\theta) / l)$$

Centrifugal force acting on the a unit length at the crank pin, then

$$C = \rho A \omega^2 r \sin (\theta + \Phi)$$

Where

θ = angle between rod and vertical line.

Φ = angle between crank and vertical line.

Buckling load on connecting rod:

$$W_b = \sigma_c A / \{1 + a (L / k)^2\}$$

Where a = Rankin constant

Total force on the connecting rod:

$$F = F_p - F_i$$

B. Theoretical Calculations:

Engine type - Air cooled, 4-stroke, Single cylinder

Bore x Stroke (mm) = 62.5x57.8

Displacement = 177.4 m³ Maximum Power = 17.03 bhp @ 8500 rpm

Maximum Torque = 15.5Nm @ 6500 rpm

Compression Ratio = 9.5: 1

Temperature = 32⁰ F to 140⁰ F

= 273.15 k to 333.15 k

Length of the connecting rod = 231 mm

Factor of safety = 6

Density of the rod material = 737.22 kg/m³

Yield in compression = 415 Mpa

$$\begin{aligned} \text{Mass} &= \text{Density} \times \text{Volume} \\ &= 737.22 \times 10^{-9} \times 177.4 \times 10^3 \\ &= 0.13 \text{ kg} \end{aligned}$$

Molecular Weight of Petrol 114.228 g/mole

From Gas Equation,

$$PV = nRT$$

The chemical amount (n) (in moles) is equal to the mass (m) (in grams) divided by the molar mass (M) (in grams per mole)

By replacing n with m / M,

$$PV = mRT/M$$

$$R/M = 8.3143/0.114228$$

$$= 72.786$$

$$P = (0.13 \times 10^3 \times 72.786 \times 295) / 177.4 \times 10^3$$

$$P = 15.7 \text{ Mpa}$$

1) Dimensions of Cross-Section of Connecting Rod:

Let us consider an I-section,

Flange and web thickness of the section = t

Width of section = 4t

Depth or height of section = 5t

$$\begin{aligned} \text{Gas load on the piston, } F_g &= (\pi D^2/4) \times p = \pi/4 \times (62.5)^2 \times 15.7 \\ &= 48166.99 \text{ N} \end{aligned}$$

Maximum inertia load on the piston, $F_i = m \omega^2 r (1+1/n)$

Here $\omega = 2\pi N/60 = 2\pi \times 8500/60 = 890 \text{ rad/s}$

$$r = L/2 = 57.8/2 = 28.9 \text{ mm}$$

$$n = l/r = 231/28.9 = 7.99$$

$$\begin{aligned} F_i &= 0.13 \times (890)^2 \times 0.0289 \times (1+1/7.99) \\ &= 3348.37 \text{ N} \end{aligned}$$

Net load on the piston, $F = F_g - F_i = 48166.99 - 3348.37 = 44818.62 \text{ N}$

Buckling load, $F_c = F_g \times F.S = 48166.99 \times 6 = 289001.98 \text{ N}$

Considering the buckling of the rod about X-X axis (i.e., in the plane of motion of the connecting rod), and assuming that both the ends are hinged,

Area of cross-section, $A = 2(4t \times t) + t \times 3t = 11t^2, \text{ mm}^2$

Area moment of inertia, $I_{xx} = 1/12[4t(5t)^3 - 3t(3t)^3]$
 $= 419/12 \times t^4, \text{ mm}^4$

$$K^2 = I_{xx}/A = (419/12) \times t^4/11t^2 = 3.17 t^2$$

Using Rankine's equation,

$$F_c = (\sigma_{yc} \times A) / (1+a(l/k)^2)$$

$$289001.98 = 330 \times 11t^2 / [1 + (0.002 \times (231)^2/3.17 t^2)]$$

$$t = 13 \text{ mm}$$

Width of the section = $4t = 4 \times 13 = 52 \text{ mm}$

Depth or height of the section = $5t = 5 \times 13 = 65 \text{ mm}$

These are the dimensions at the middle of the connecting rod. The width is kept constant throughout the length of the rod, but the depth or height varies.

The depth or height near the big end or crank pin is 1.1H to 1.25H.

The depth or height near the small end or piston end is 0.75H to 0.9H.

Depth or Height near the big end, $H_1 = 1.1H \text{ to } 1.25H = 1.2H = 1.2 \times 65$

$$H_1 = 78 \text{ mm}$$

Depth or Height near the small end, $H_2 = 0.75H \text{ to } 0.9H = 0.85H = 0.85 \times 65$

$$H_2 = 55.25 \text{ mm}$$

2) Dimensions of Big End Bearing:

Let, d_1 = diameter of the crank pin or big end bearing

l_1 = length of the crank pin or big end bearing

p_1 = permissible bearing pressure = 7.5 Mpa

Load on the bearing = bearing area \times bearing pressure

$$F_g = d_1 l_1 p_1$$

Assuming $l_1/d_1 = 1.3$,

$$48166.99 = d_1 \times 1.3 d_1 \times 7.5$$

$$d_1 = 70.21, \text{ say } 70 \text{ mm,}$$

$$l_1 = 1.3 d_1 = 1.3 \times 70 = 91 \text{ mm}$$

$$\text{Force on the bolts} = \pi/4(d_{cb})^2 \times \sigma_t \times n_b$$

$$n_b = \text{number of bolts} = 2$$

$$F_1 = \pi/4(d_{cb})^2 \times 60 \times 2$$

$$3348.37 = 94.24(d_{cb})^2$$

$$(d_{cb})^2 = 35.47$$

$$d_{cb} = 5.96$$

Nominal diameter, $d_b = d_{cb}/0.84 = 5.96/0.84$

$d_b = 7.47$ mm say 7mm

$$\text{Outer diameter at big end} = d_1 + 2t_b + 2d_b + 2t_m$$

$$= 70 + 2 \times 2 + 2 \times 7 + 2 \times 5$$

$$= 98 \text{ mm}$$

3) Dimensions of Small End Bearing:

Let d_2 = diameter of the piston pin or small end bearing

l_2 = length of the piston pin or small end bearing

p_2 = permissible bearing pressure = 10 Mpa

Load on the bearing = bearing area \times bearing pressure

$$F_g = d_2 l_2 p_2$$

Assuming $l_2/d_2 = 2$, $48166.99 = d_2 \times 2 d_2 \times 10$

$d_2 = 49.04$, say 49 mm,

$l_2 = 2 d_2 = 2 \times 49 = 98$ mm

Outer diameter at small end = $d_2 + 2t_b + 2t_m = 49 + 2 \times 2 + 2 \times 5 = 64$ mm

V. MODELING OF CONNECTING ROD

Modeling has been done in catiaV5/R20 software with calculated dimensions.

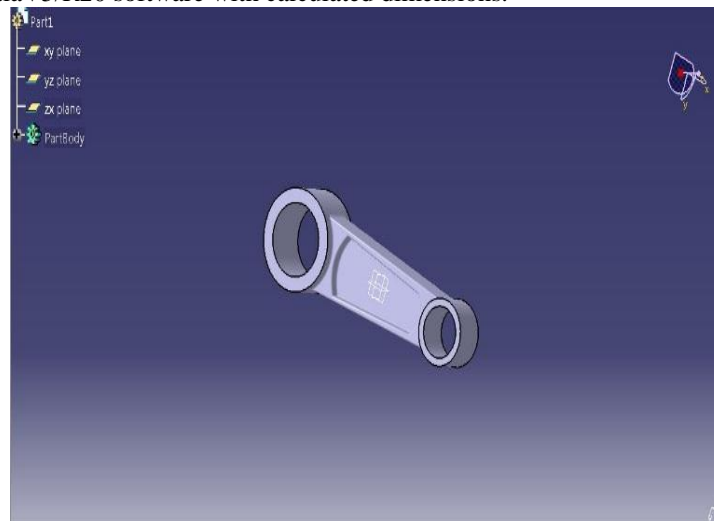


Fig. 2: Modeling of connecting rod

VI. MESHING OF CONNECTING ROD:

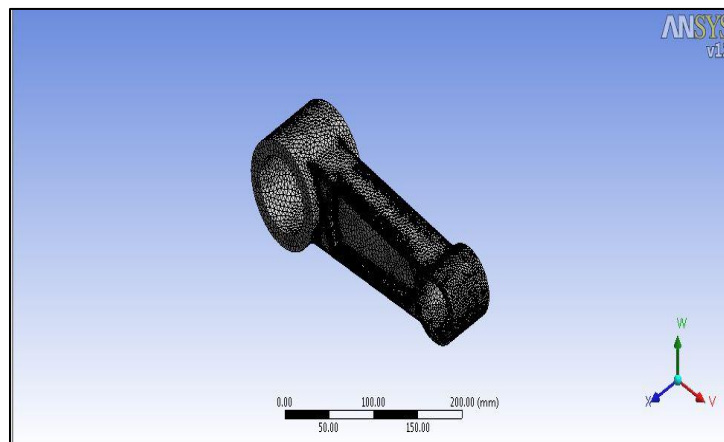


Fig. 3: Meshing of connecting rod

VII. STATIC ANALYSIS OF CONNECTING ROD

A 3d model of connecting rod is used for analysis in Ansys 12.0. The loading conditions are assumed to be static. Analysis done with Force applied at the piston end and restrained at crank end or other end Load applied at the crank end and restrained at piston end. The element chosen is solid 45, it was used with the tetrahedral option, making it a 10-node element with 3 degrees of freedom at each node. The finite element. Analysis is carried out on forged steel of connecting rod as well as the carbon steel. The material properties are used and necessary for analysis.

Table - 1
Material property of Connecting Rod

Mechanical properties	forged steel	carbon steel
Density (g/cc)	7.7	7.85]
Tensile yield strength (Mpa)	625	186-758
Ultimate tensile strength (Mpa)	625	276-1882
Poison's ratio	0.27-0.30	0.29
Young's modulus (Gpa)	221	190-210

A. For Carbon Steel:

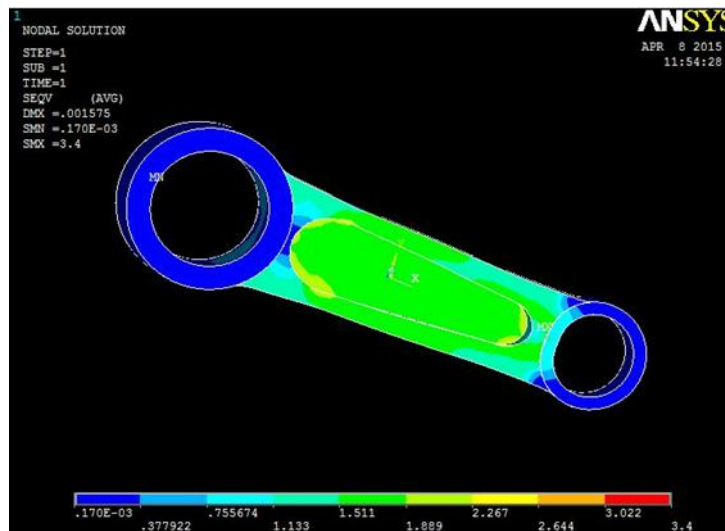


Fig. 4: Equivalent von-mises stress



Fig. 5: Equivalent von-mises strain

B. For Forged Steel:

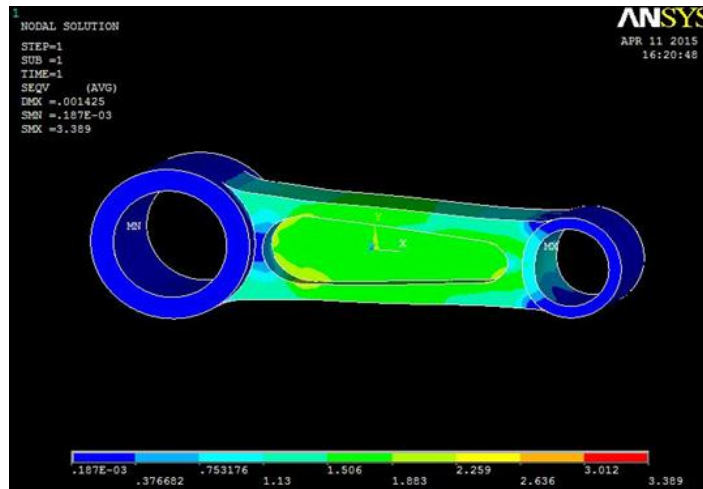


Fig. 6: Equivalent von-mises stress



Fig. 7: Equivalent von-mises strain

VIII. FATIGUE ANALYSIS

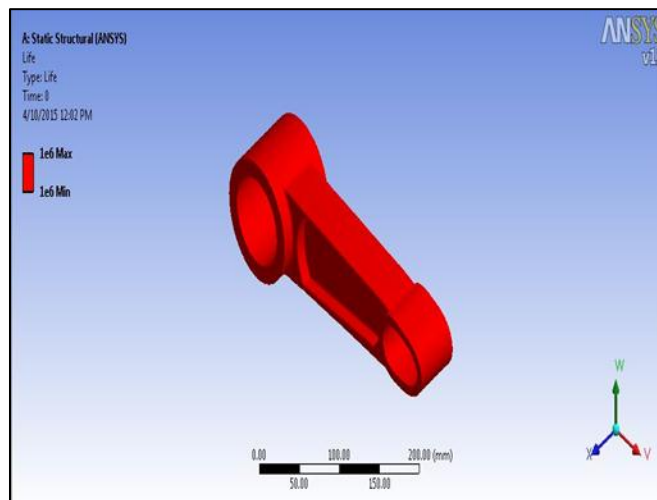


Fig. 8: Life

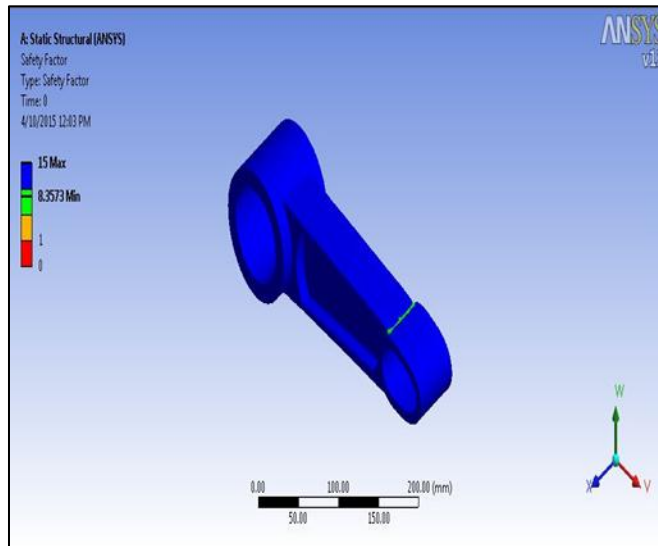


Fig. 9: Safety Factor

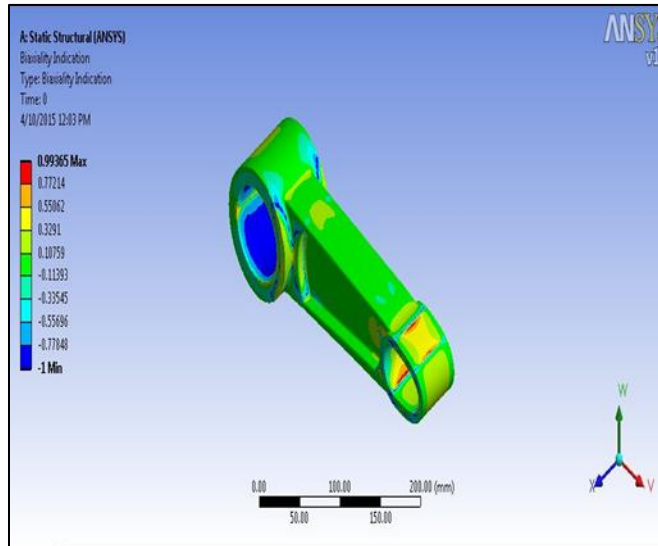


Fig. 10: Biaxiality Indication

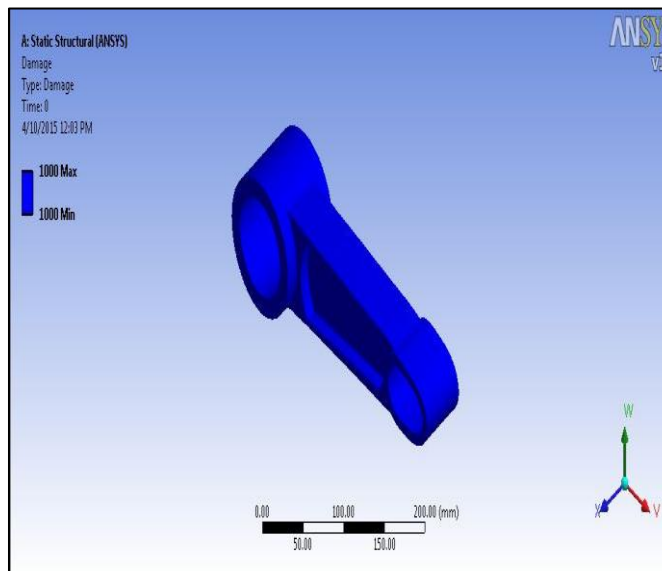


Fig. 11: Damage

IX. OPTIMIZATION OF CONNECTING ROD

The optimization task was to minimize the mass of the connecting rod under the effect of a load range for two extreme loads, the peak compressive gas load and the tensile load within the limits of the allowable stresses. The result of weight reduction for optimized connection rod is given below table2.

Table – 2
Values for Weight Reduction

Name	Scope	Target Reduction	Predicted Reduction
Shape Finder	Model	20.0%	9.24% to 9.51%

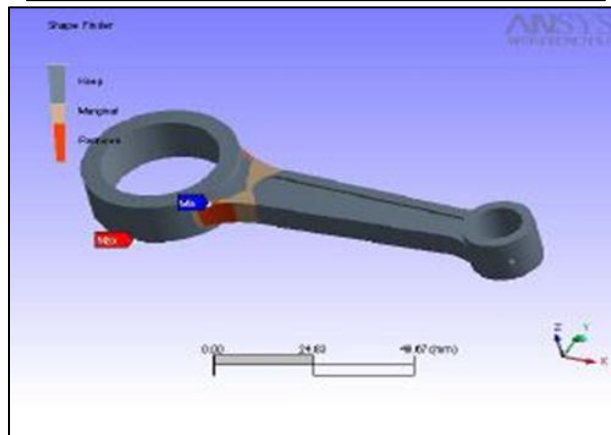


Fig. 12: Shape Finder

X. CONCLUSION

It was found that the design parameter of connecting rod with modification gives sufficient improvement in the existing results. The weight of connecting rod for forged steel is reduced as compared with carbon steel. Fatigue strength plays the significant role in the optimization of the connecting rod. Optimization was performed to reduce the weight of connecting rod. The optimization can also be achieved by the carbon steel connecting rod into forged steel.

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