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A New Approach to Explore the Weight Reduction Opportunities of Forged Steel Connecting Rod

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Abstract: **The connecting rod forms an integral part of an internal combustion engine and it is acted upon by different types of loads while undergoing its operation. The main objective of this study is to explore the weight reduction opportunities, without compromising the factor of safety and durability of the forged steel connecting rod. This has entailed performing a detail load analysis therefore this study has dealt with two subjects. In the first part of this study, the dynamic load, various stress analysis of the connecting rod and gudgen pin. In the Second part of this study is Optimization of weight and factor of safety. In the conclusion of this study the connecting rod can be designed and optimized under the effect of load range comprising the peak compressive gas load and the peak dynamic tensile load at 2400 rev/min. (at 360° crank angle) such that the maximum, minimum, and the equivalent stress amplitude are within the limits of the allowable stresses. Furthermore, the buckling load factor under the peak gas load has to be permissible. 3D modeling by Pro-ENGINEER 4.0, dynamic analysis by ADAMS and simulation work by ANSYS 14.5 has been done.**

1. INTRODUCTION

The connecting rods are of high volume production, and usually manufactured by drop forging process. The material mostly used for connecting rods varies from mild carbon steels (0.35 to 0.45 % carbons) to alloy steels (Chrome Nickel to Chrome molybdenum steels). The functions of connecting rod includes, providing a connecting link between Piston and Crankshaft to convert the reciprocating motion to rotary motion and conveying lubricating oil from Crankshaft (big end) to Piston pin (small end) through its central oil hole. Connecting rods are generally subjected to two types of inertia forces, one due to masses and friction induced by the reciprocating parts and other, due to the gas load generated from combustion process. The small end of the connecting rod is provided with a bush of phosphor bronze and connected to gudgeon pin. The big end of the connecting rod is usually made split into two halves. The split cap is fastened to the big end with two cap bolts. The bearing shells are made up of steel, brass or bronze with a thin lining about 0.75 mm of white metal. When the power intensity ratio (rated power/cm^2 of piston area of cross section) of a diesel engine goes above 0.18, an extra cooling should be provided to the piston other than the normal water-cooling system in order to reduce the emission levels.

In this subject, I have chosen a Turbo-Charged air cooled diesel engine, which has the power intensity ratio of 0.24, thus piston cooling is necessary .This is achieved by piston-cooling nozzles (cooling of piston through a separate jet from oil gallery to dissipate combustion heat and to control the piston ring sticking).

2. FORCES ACTING ON CONNECTING ROD

The forces acting on the connecting rod are:
1. Force due to the gas pressure and the piston inertia
2. Force due to friction of piston rings and piston
3. Inertia of the connecting rod and
4. Frictional force in the big and small end bearings.

Maximum force due to gas pressure, $F_a = (^{\pi}/_4) * d^2 * p_e$

The gas pressure is a function of the crank angle and can be obtained from the indicator diagram. Force due to piston inertia, given by

$$F_i = M\,\omega^2 r\,(\cos\theta + r / \ell \cos 2\theta)$$

Where
$M = 4.15 \times 10^{6} d^3$ kg for cast iron pistons.
$=1.78 \times 10^{6} d^3$ kg for aluminium pistons.
The force in the connecting rod will be maximum when the crank angle $\theta = 90^0$.

The force of friction due to piston rings and piston is:
$F_f = h\,\pi\,d\,i\,P_r\,\mu$

The maximum bending stresses is given by the formula,

$$\sigma_{max} = \frac{M_{max}}{Z}$$

Due to the gas force the connecting rod is subjected to buckling. The buckling load may be calculated by Rankine-Gordon formula:

$W_B = ((\sigma_c * A) / \{1+a [\ell/K_{xx}{}^2]\})$

3. CALCULATION OF FACTOR OF SAFETY

AIM: To calculate existing factor of safety of the connecting rod using Rankine's formula in an Excel Spread sheet.

STEP 1:
Yield Strength of connecting Rod $(\sigma_Y) = 570$ N/mm^2
Tensile Strength of connecting Rod $(\sigma_u) = 770$ N/mm^2

STEP 2:
Load due to Gas pressure $(F_G) = (\pi/_4 * d_1{}^2 * P_{max})$
$= (\pi/_4 * 107.277^2 * 12.236)$
$= 110600.3997$ N

STEP 3:
For an I-section $A = 11t^2$, $K_{xx} = 1.78t$.
Therefore, $A = 283.87$ mm^2 & $K_{xx} = 9.0424$ mm

Shank Design
Flange Thickness = t
Depth of section = 5t
Width of section = 4t

Area of section $(A) = 11t^2$
$I_{xx} = {}^1/_{12} (BH^3 - bh^3)$
$= 1/_{12} [(1t) \text{ x } (5t^2) - (3t) \text{ x } (3t^3)]$
$= 419t^2 / 12$
$I_{yy} = {}^1/_{12} (2t*B^3 + ht^3)$
$= {}^1/_{12} [(2t * 4t^3) + (3t * t^3)]$
$= {}^{131}/_{12} t^4$
$K^2{}_{xx} = I_{xx} / A = 419t^2 / (12*11t^2)$
$K^2{}_{xx} = 3.18 t^2$
Similarly,
$K^2{}_{yy} = I_{yy} / A = 131t^2 / (12*11t^2)$
$K^2{}_{yy} = 0.995 t^2$

STEP 4:
Rankine's formula:
The buckling load (W_B) of the component can be calculated as
$W_B = ((\sigma_c * A) / \{1+a [\ell/K_{xx}{}^2]\})$
Where, σ_c (Direct compressive stress + Bending Stress) = 770 N/mm^2

Soft wear.
W_B = Maximum gas force * Factor of safety
= P_{max} * Piston area * n
= 110600 n
F.O.S without hole $(n_1) = 5.7371$
F.O.S with hole $(n_2) = 5.533$

The above F.O.S values fits only to the I-Section of the Connecting rod.

4.1 Parametric Modeling of Connecting Rod

A solid model of the connecting rod was generated using Pro/Engineer Wildfire2.0. Due to the symmetry of the geometry, the component was first half modeled, and then the entire geometry was created by reflecting (mirror imaging) the half geometry. The model was designed without forging flash, bolts, and crank or pin hole bearings, as these details are not expected to have any significant influence on the obtained results at the critical regions (i.e. failure locations, which were at or near the transition of the crank end with the shank), and their removal allowed simplification of the model. Different modeling techniques have been adopted on modeling of the existing connecting rod. Thus, a final optimized geometry of connecting rod has been created using assembly cut procedure from the Pro/Engineer.

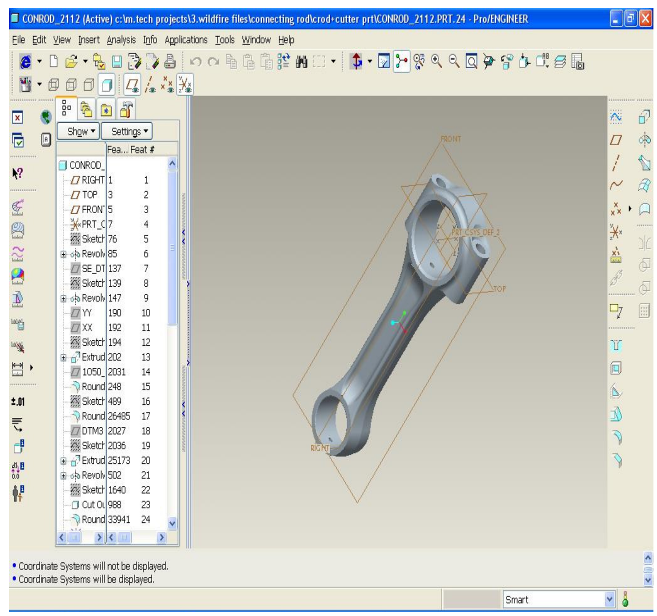

Fig 4.1 Parametric model of existing connecting rod

4.2 Modeling Process

The step-by-step modeling and analysis procedure of a connecting rod is described below:

1. ***Creating a model, setting units and gravity***

- The assembled 3D model of connecting rod, piston, piston pin part files are being imported to ADAMS physical model environment through Step / iges file format.

2. ***Creating parts and joints***

- The connecting rod is connected to the piston pin with a revolute joint; similarly, the either ends of pins are connected to movement in the piston using a lock joint to prevent the all directions.

- The Big end side of connecting rod is connected to a bar link with a spline defined revolute joint, i.e., the movement of connecting rod depends upon the Pressure Vs Crank angle diagram.
- This spline curve is defined by a set of xy axis values.
- Other end of the bar link is provided a revolute joint.

3. Running and animating a simulation

- The maximum gas load, taken as a point load on the piston surface at C.G. axis. Now, the force vectors are made visible (Both the axial force at top and bottom).
- From the geometry of 3D model & material properties, the corresponding mass properties are being calculated.
- The material properties include:
- Piston : Aluminium alloy
- Piston pin : 15 Cr Ni6
- Connecting rod: Steel to DIN 17200 41 Cr 4
- The masses of reciprocating parts are taken as tensile inertia force.
- Thus, the simulation is made to run for few steps.

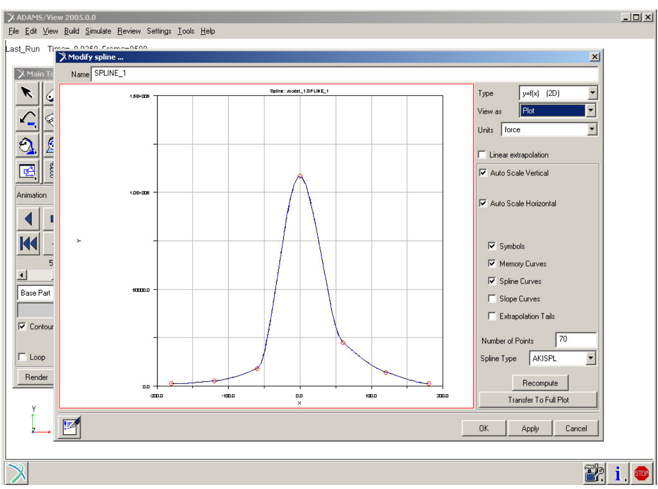

Figure 4.2.1 Pressure Crank angle diagram for the existing engine showing the peak firing pressure as 120 bars

4. PLOTTING RESULTS

After the completion of simulation, the corresponding graphs are obtained from the ADAMS/View

- Maximum Bending force perpendicular to connecting rod axis.
- Maximum compressive force obtained from peak firing pressure

4.2.1 Condition – I

The maximum gas load acts at the small end region, caused by peak combustion pressures. The corresponding axial load developed in this region is obtained through this analysis.

- Peak firing pressure = 120 bars
- Cylinder bore / Piston diameter = 107.25 mm

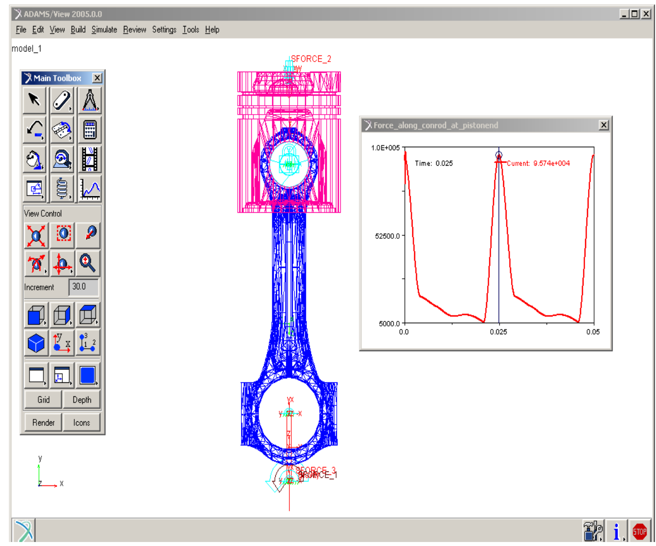

Figure 4.2 Maximum compressive load condition

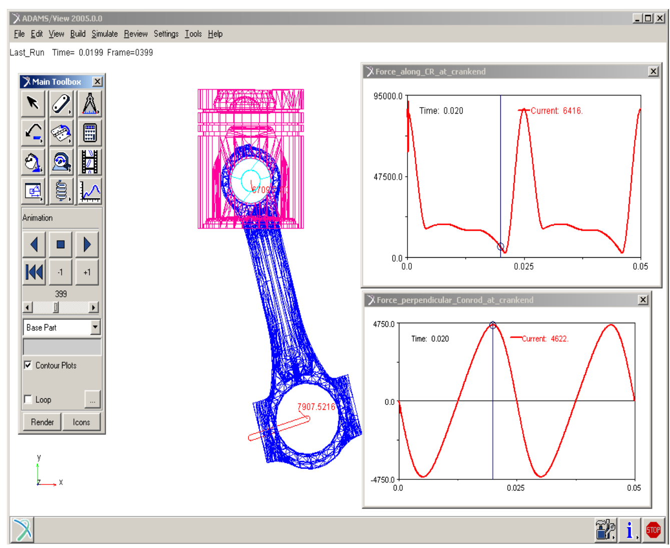

Fig 4.3 Angular loads caused due to mass of reciprocating parts

Results

The Axial load developed upon the connecting rod with respect to maximum gas load is 95740 N.

4.2.2 Condition – II

The tensile inertia load is caused by mass of reciprocating parts. Now, the ADAMS software calculates the corresponding bending force. The maximum force perpendicular to connecting rod axis in both bending and axial moments is shown below.

Results

When the force is 8127 N, then the maximum bending force obtained perpendicular to the connecting rod axis is 4622 N and the corresponding axial force is 6416 N.

5. CURRENT DESIGN- CONNECTING ROD WITH HOLE

Boundary Condition I

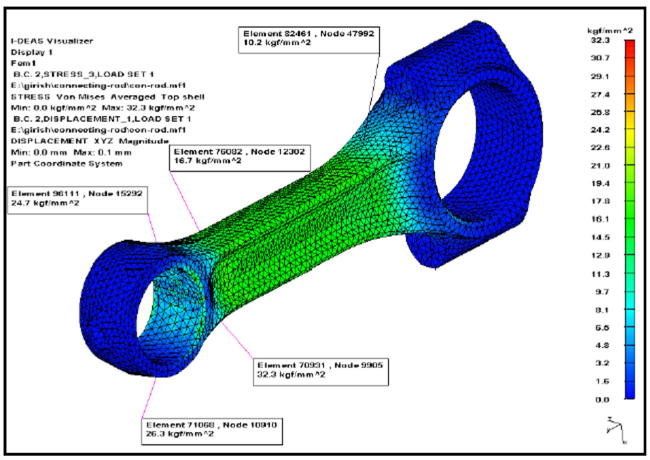

Figure 5.1.1 Von Mises stress distribution plot with compressive load of 95.74 kN at Piston pin end while the crank end was restrained

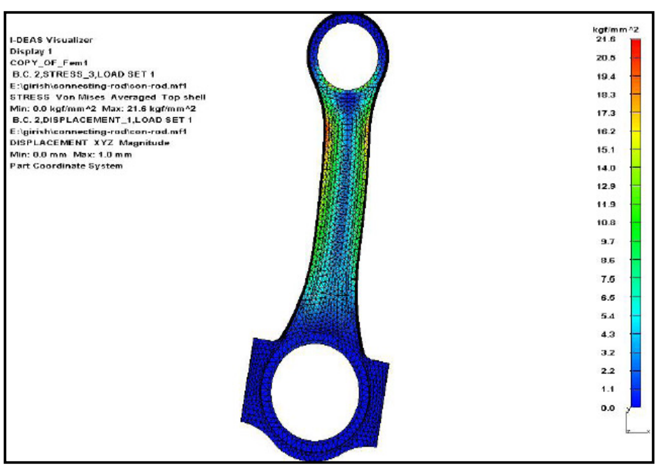

Figure 5.1.2 Displacement plot with compressive load of 95.74 kN at piston pin end while the crank end was restrained.

Observations

- Big end fully constrained and a vertical load of 95.74 kN
- It can be observed that big end bottom and small end top almost of zero stress due to rigid
- Maximum displacement of 0.1 mm observed for the given constraint
- Maximum stress of 36.5 kgf /mm^2 observed near the small end, may be due to rigid connection given to apply load at small end, that need to be verified by defining a contact.
- Near the web, stress is around 17.8 kgf / mm^2

Boundary Condition II

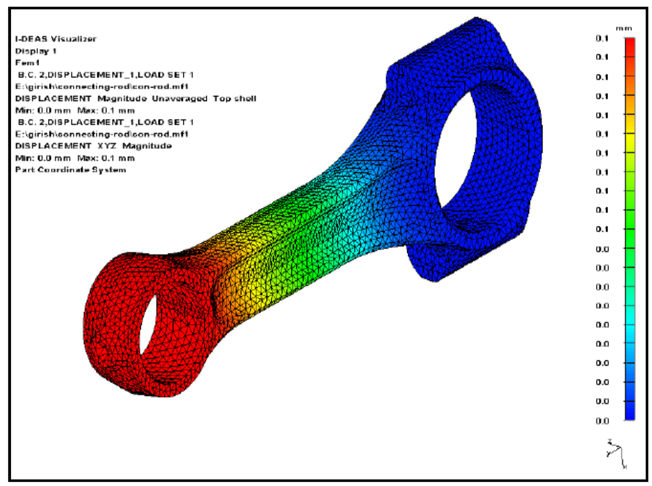

Figure 5.1.3 Von Mises stress distribution plot with static tensile load of 8.127kN at an angle 34.66^0 to vertical (Fy = 6.685kN, Fx = 4.622kN) at the crank end while the pin end was restrained.

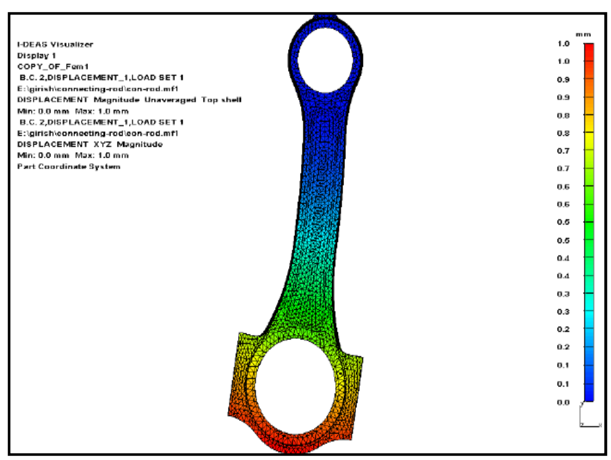

Figure 5.1.4 Displacement plot with static tensile load of 8.127kN at an angle 34.66^0 to vertical (Fy = 6.685kN, Fx = 4.622kN) at the crank end while the pin end was restrained.

Observations

- Small end constrained fully and a resultant load of 8.127kN at an angle of 34.66^0 to vertical (Fy = 6.685kN, Fx = 4.622kN)
- It can be observed that big end bottom and small end top almost of zero stress due to rigidity.
- Maximum displacement of 1.1 mm observed for the given constraint
- Maximum stress of 21.9 kgf/mm^2 observed near the web and 9 kgf/mm^2 observed near the hole at small end

5.2 Current Design - Connecting Rod Without Oil Hole

Boundary Condition I

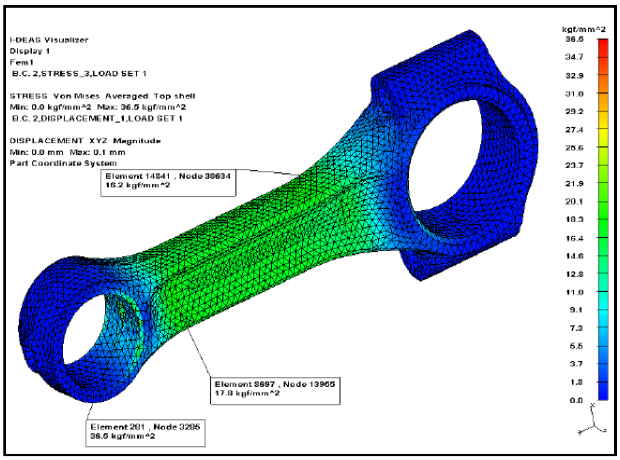

Figure 5.2.1 Von Mises stress distribution plot with compressive load of 95.74 kN at piston pin end while the crank end was restrained.

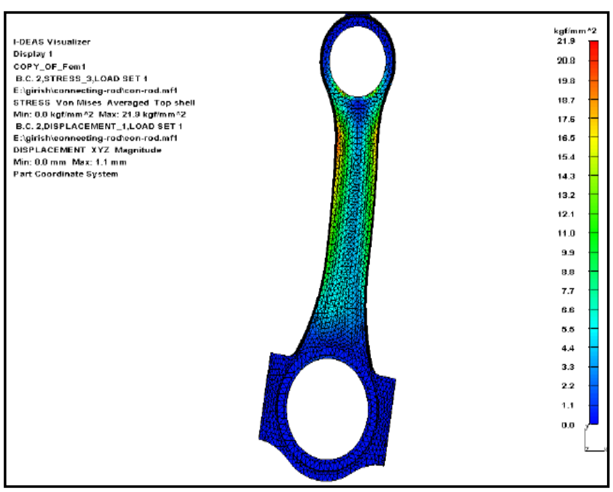

Figure 5.2.2 Displacement plots with compressive load of 95.74 kN at piston pin end while the crank end was restrained.

Observations

- Big end fully constrained and a vertical load of 95.74 kN
- It can be observed that big end bottom and small end top almost of zero stress due to rigid
- Maximum displacement of 0.1 mm observed for the given constraint
- Maximum stress of 32.3 kgf /mm^2 observed near the small end, may be due to rigid connection given to apply load at small end, that need to be verified by defining a contact.
- Near the oil hole removed region small end stress of 24.5 kgf/ mm^2
- Near the web, stress is around 16.7 kgf/mm^2

Boundary Condition II

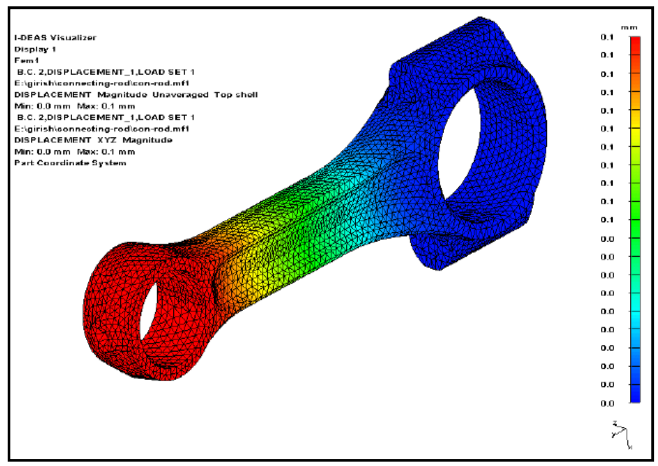

Figure 5.2.3: von Mises stress distribution plot with static tensile load of 8.127kN at an angle 34.660 to vertical (Fy = 6.685kN, Fx = 4.622kN) at the crank end while the pin end was restrained

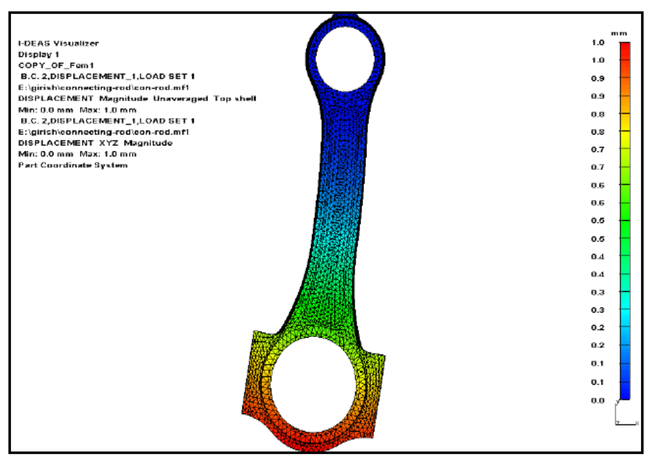

Figure 5.2.3: Displacement plot with static tensile load of 8.127kN at an angle 34.66^0 to vertical (Fy = 6.685kN, Fx = 4.622kN) at the crank end while the pin end was restrained.

Observations

- Small end constrained fully and a resultant load of 8.127kN at an angle of 34.66^0 to vertical (Fy = 6.685kN, Fx = 4.622kN)
- It can be observed that big end bottom and small end top almost of zero stress due to rigidity.
- Maximum displacement of 0.1 mm observed for the given constraint
- Near the web, stress is around 21.6 kgf/mm^2

Table 5.1 Shows the optimized result of the connecting rod assembly

Parameters	Existing Connecting rod	Optimized Connecting rod	Existing Gudgeon pin 1.6”	Optimized Gudgeon pin 1.3”
Mass	2.157 kg	1.8537 kg	0.677 kg	0.431 kg
Area of I–Section	65977.93 mm^2	61514.71 mm^2	18678.45 mm^2	15075.99 mm^2
Density	2.016e-04 kg / mm^3	2.016e-04 kg/ mm^3	7.20e-06 kg/mm^3	7.20e-06 kg/mm^3
F.O.S under downward load	1.63	1.58	1.42	1.32
F.O.S under vertical load	2.72	1.84	21.15	20.12

5.3 Result and Discussion

Below table describes the comparative chart of change in the mass, area of I-section, density and F.O.S values for the existing and optimized connecting rod.

Results

- Weight reduction contribution by Connecting Rod = 0.303Kg
- Weight reduction contribution by Gudgeon Pin = 0.246

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