EXPERIMENTAL ANALYSIS ON OPTIMIZATION OF AUTOMOBILE CONNECTING ROD

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Abstract

The main objective of this work is to explore weight reduction possibilities, in the design and production of a connecting rod. This can be implemented by detailed load analysis. Therefore, this study has been dealt with two steps. The load and stress analysis of the connecting rod is done and then optimization is achieved by reducing the weight and cost. First part of the study analysis includes the determination of loads acting on the connecting rod as a function of time. This is done for finding out the minimum stress area and to remove the material in those areas. The relationship between the load and acceleration of the connecting rod for a given constant speed of the crankshaft are also determined. Finite element analysis was done over several crank angles. The connecting rod can be designed and optimized under the loads ranging from tensile load, corresponding to 360 degree crank angle at the maximum engine speed as one extreme load and the compressive load proportionate to the peak gas pressure as the other extreme load. The fracture crack ability feature facilitates separation of cap from rod without additional machining of the mating surfaces. Yet, the same performance can be expected in terms of component durability. This aspect is considered in this work and the connecting rod is optimized by reducing the weight and cost.

Keywords— Connecting rod, Stress Analysis and Finite Element Analysis (FEA).

I. INTRODUCTION

The automotive vehicle has become a basic necessity of today. For any automotive vehicle, the human comfort is considered as a major criterion. Reducing vibration, noise and ensuring safety of the vehicle and pleasure and comfort of the passengers are the main consideration in the design of an automotive engine. The vibration and the noise are mainly due to the force transmitting parts of the automotive vehicle like piston, connecting rod, crank shaft, cam shaft and valve trains. The mass of the moving parts plays a major role in the generation of noise and vibration. So the weight of the moving parts must be reduced to reduce the undesirables while the automobile is in operation. The weights of the moving parts must be taken care of while designing these parts. These parts should be designed to withstand the force to make sure that the engine is reliable and durable.

The automobile engine connecting rod is a high volume production that is a critical component. This 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.

A. Connecting Rod

The connecting rod is the connection between the piston and the crankshaft. It joins the piston pin with the crankpin; small end of the connecting rod is connected to the piston and big end to the crank pin. The mechanism of the connecting rod is to convert linear motion of the piston into rotary motion of the crankshaft.

The lighter connecting rod and the piston greater than resulting power and less the vibration because of the reciprocating weight is less. The connecting rod carries the power thrust from piston to the crank pin and hence it must be very strong, rigid and also as light as possible. There are two types of small end and big end bearings. Connecting rods are subjected to fatigue due to alternating loads.

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. 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. With steel forging, the material is inexpensive and the rough part manufacturing process is cost effective.

B. *Overall Objectives*

1) Selection of materials
   Evaluate and compare fatigue performance of different material connecting rods.

2) Optimization of connecting rod
   Perform load analysis and optimization.

II. LITERATURE SURVEY

In a study reported by Repgen[12] (1998), based on fatigue tests carried out on identical components made of powder metal and C-70 steel (fracture splitting steel), he notes that the fatigue strength of the forged steel part is 21% higher than the powder metal component. Also he notes that using the fracture splitting technology results in a 25% cost reduction over the conventional steel forging process. These factors suggest that a fracture splitting material would be the material of alternative for steel forged connecting rods. He mentions two other steels that are being tested, a modified micro-alloyed steel and a modified carbon steel. Other issues mentioned by Repgen are the necessity to avoid jig spots along the parting line of the rod and the cap, need of consistency in the chemical composition and manufacturing process to reduce variance in microstructure and production of near net shape rough part.

For their optimization study, Serag et al.[15] (1989) developed approximate mathematical formulae to explain connecting rod weight and cost as objective functions and also the constraints. The optimization was achieved using a Geometric Programming technique. Constraints were imposed on the compression stress, the bearing pressure at the crank and the piston pin ends. Fatigue was not addressed. The cost function was expressed in some exponential form with the geometric parameters.

Hippoliti[7](1993) reported design methodology in use at Piaggio for connecting rod design, which incorporates an optimization session. However, neither the details of optimization nor the load under which optimization was performed were discussed. Two parametric FE procedures using 2D plane stress and 3D approach developed by the author were compared with experimental results and shown to have good agreements. The optimization procedure they developed was based on the 2D approach.

Park et al.[9] (2003) investigated micro structural behavior at various forging conditions and recommend fast cooling for finer grain size and lower network ferrite content. From their research they concluded that laser notching exhibited best fracture splitting results, when compared with broached and wire cut notches. They optimized the fracture splitting parameters such as, applied hydraulic pressure, jig set up and geometry of cracking cylinder based on delay time, difference in cracking forces and roundness. They compared fracture splitting high carbon micro-alloyed steel (0.7% C) with carbon steel (0.48% C) using rotary bending fatigue test and concluded that the former has the same or better fatigue strength than the later. From a comparison of the fracture splitting high carbon micro-alloyed steel and powder metal, based on tension-compression fatigue test they noticed that fatigue strength of the former was 18% higher than the later.

Athavale and Sajanpawar[2] (1991) modeled the inertia load in their finite element model. An interface software was developed to apply the acceleration load to elements on the connecting rod depending upon their location, since acceleration varies in magnitude and direction with location on the connecting rod. They fixed the ends of the connecting rod, to determine the deflection and stresses. This, however, may not be representative of the pin joints that exist in the connecting rod. The results of the detailed analysis were not discussed, rather, only the modeling technique was discussed. The connecting rod was separately analyzed for the tensile load due to the piston assembly mass (piston inertia), and for the compressive load due to the gas pressure. The effect of inertia load due to the connecting rod, mentioned above, was analyzed separately.

In a published SAE case study (1997), a replacement connecting rod with 14% weight savings was designed by removing material from areas that showed high factor of safety. Factor of safety with respect to fatigue strength was obtained by performing FEA with applied loads including bolt tightening load, piston pin interference load, compressive gas load and tensile inertia load. The study lays down certain guidelines regarding the use of the fatigue limit of the material and its reduction by a certain factor to account for the as-forged surface. The study also indicates that buckling and bending stiffness are important design factors that must be taken into account during the design process. On the basis of the stress and strain measurements performed on the connecting rod, close agreement was found with loads predicted by inertia theory. The study also concludes that stresses due to bending loads are substantial and should always be taken into account during any design exercise.

III. DESIGN METHODOLOGY

A. Analytical Evaluations

- Digitizing Connecting Rod Geometry
- Stress (FEA) Analysis
- Modal Analysis and Life Predictions
- Optimization Analysis.

B. Steps involved in methodology

Step 1: Modeling of connecting rod using 3D modeling software.

Step 2: Finite element modeling of the connecting rod.

Step 3: Analysis of connecting rod using Ansys software.

1) Element selection.
2) Discretization.
3) Mesh generation.

Step 4: Finite element stress analysis.

Step 5: Modal analysis.

Step 6: Optimization.

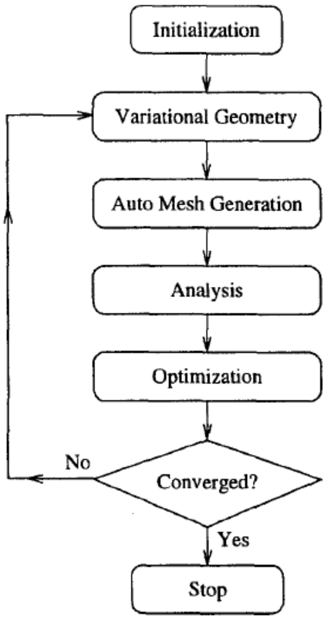

Fig 1. Optimization Flow Chart

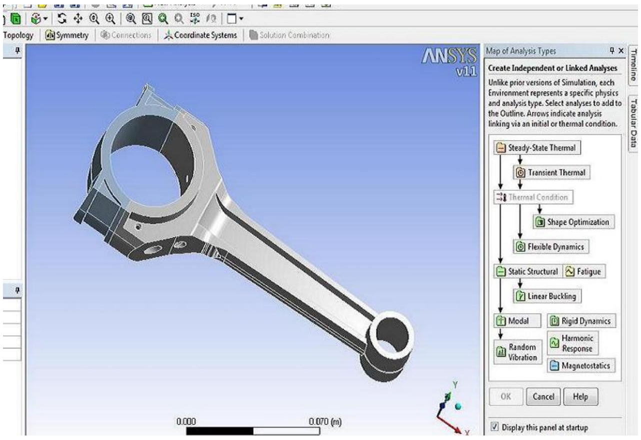

Fig 2. Map of analysis types

C. Optimization procedure

The main objective of optimization is to minimize the mass of the connecting rod such that the maximum, minimum and the equivalent stress amplitude are within the limits of allowable stress. The production cost also had to be minimized by reducing the machining costs respectively.

The key constraints used in the optimization procedure are,

a) Applied Load

The dynamic tensile load and compressive gas load. Connecting rods are subjected to

- Inertia forces due to mass.
- Forces generated from the combustion process.
- Forces due to wearing of forging flashes.

b) These forces produce

- Cyclic axial force and stress
- Cyclic bending moment and stress (perpendicular to the crankshaft axis)

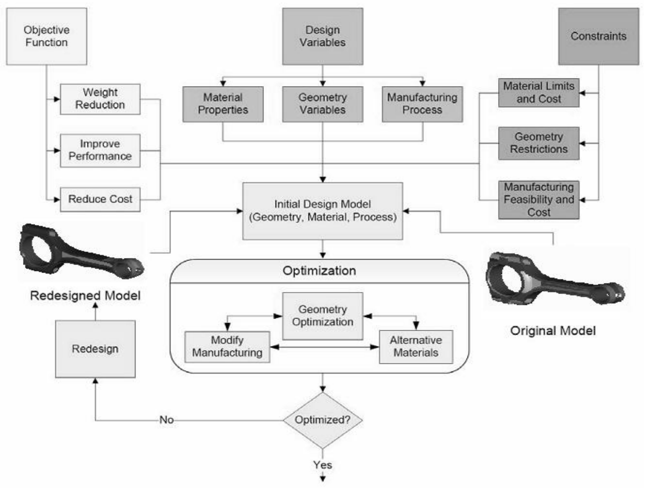

Fig 3. Optimization procedure

c) Allowable Stress

Allowable stress is the ratio of material strength to the factor of safety

d) Failure Index

Failure index (FI), which is the inverse of FOS is defined as ratio of the von misses stress to the strength of the material.

e) Geometry Constraints

Basically, diameter of crank pin, piston pin holes, crank pin center to piston pin centre remains unchanged.

IV. DESIGN OF CONNECTING ROD

For the initial design of the connecting rod, it is assumed that the peak cylinder pressure occurs at top dead centre position and the design is based on the axial loads. As there is no angularity of the connecting rod and the acceleration of the parts at this position is zero, the axial load is approximately equal to the gas load.

A. Design of the shank portion

The I-Section of the connecting rod is shown in fig. The safe load is given by Euler's formula,

$$\text{Axial load } Q = \frac{[\sigma_c]A_1}{[1+c(L/k)^2]}$$

$$(Q) = \pi D^2 P_{max}/4$$

B. Design of the eye end and big end bore

The big and small ends of the connecting rod serve as bearings for the crank pin and the gudgeon pin respectively. Since the dimensions of these parts are limited, fairly high bearing pressures are likely to be encountered at these bearings. The crank pin bearing is more severely loaded due to the larger relative rubbing speed in comparison to the small end bearing, which is subjected to oscillatory motion only. This necessitates a lower value of the recommended bearing pressure for the big end bearing.

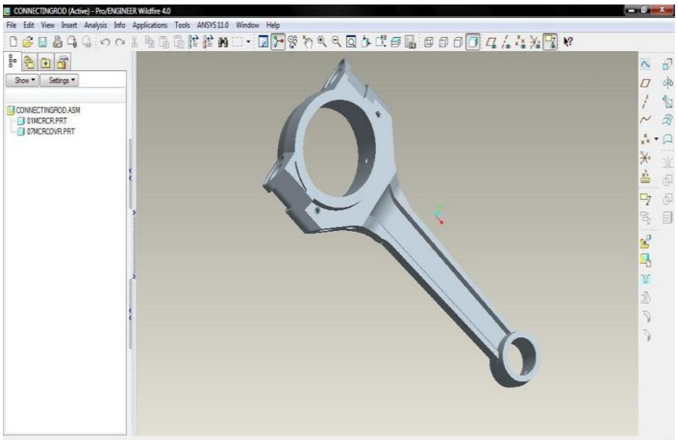

Fig 4. Model of the connecting rod

The shear failure occurs due to tensile load caused by the inertia of the reciprocating parts of the engine. The mass of the reciprocating parts is assumed to be equal to the sum of the mass of the piston assembly and one third of the mass of the connecting rod. The maximum value of this force is occurring at inner dead centre and is given by,

$$F_i = m_r^2 R\,(\cos\theta + \cos(2\theta/n))$$

$$m_r = m_c + m_p$$

$$n = L/R$$

Considering the shear failure,

$$F_i = (d_o - d_p)\times b\times\tau$$

Considering the tensile stress for the initial design of the connecting rod,

$$F_i = \sigma_t \times b \times t_c$$

As the big end cap of the connecting rod is inclined, it is subjected to tensile stress and bending stress.

V. STRESS ANALYSIS OF THE CONNECTING ROD

A. Calculation of loads

1) Axial Loads

- Gas Loads

The direct load on the piston due to gas pressure is calculated from the following equation.

$$\text{Gas load (P)} = \rho A N$$

$$A = \pi D^2 / 4 m^2$$

- Inertia Load

To calculate the inertia force, first two harmonies are taken into consideration. It is given by,

Inertia load **(F) =** maN

The axial load (Q) acting on the connecting rod is given by,

$$\mathbf{Q} = \frac{P}{R/L[(L/R)2 - \sin 2\theta]} N$$

2) Bending Load

This load is due to the inertia of the oscillating parts of the connecting rod. This force trends to bend the connecting rod outwards, away from the centre line. It is alternating one, and at high speed, it is considerable. Total inertia bending force is given by,

$$F_b = \frac{\rho A_i L^2 \sin(\theta + \emptyset)}{2} N$$

B. Calculation of stresses

1) Stress due to axial loads

These stresses are calculated as follows,

$$\sigma_a = Q/A_i \quad N/m^2$$

This load sets up a compressive stress except at the end of the exhaust stroke and beginning of the suction stroke. Tensile stress developed during these periods is very less when compared to the compressive stress, but produces severe stresses in the bolt and stud and big end cap. The direct inertia stresses considered alone also change sign every half revolution, but are about 90 out of phase with the inertia bending stress. The stress values set up by the axial loads are calculated. To calculate the axial stresses at the sections away from the smell end, acceleration along the connecting rod is found from that region. The acceleration along the connecting rod causes an axial force. The acceleration is found at the centre of gravity of the section which is under consideration, the inertia forces is given by,

$$F^i = m^i x a^i N$$

2) Stress due to inertia bending force

Inertia bending load sets up a stress which would be tensile on one side of the rod and compressive on another side and that these stresses change sign each half revolution. The bending moment at any section 'x' m from the small end is given by

$$M = \frac{X}{3}\left[1 - \frac{X^2}{L^2}\right] F_b \quad Nm$$

The stress is calculated by using the formula,

$$\sigma_b = M/Z$$

$$Z = I/(2.5 \times t)$$

$$I = 419 \times t^4$$

The axial and bending stresses calculated above produces a variation of stress over the I-section of the connecting rod. The stresses at the outer and inner fibre of the I-section are calculated as follows,

$$\sigma = \sigma_a + \sigma_b$$

A computer program is written to calculate the stress values at different crank angles measured from the centre.

VI. FINITE ELEMENT ANALYSIS OF THE CONNECTING ROD

A. Static analysis

This analysis evaluates the effects of steady loading conditions on the connecting rod. The Time-varying loads like inertia loads are approximated as static equivalent loads and their effects can also be evaluated using this analysis. It is used to determine the displacements, stresses, strains and forces on the structures. The kind of loading that can be applied in a static analysis include externally applied forces and pressures, steady state inertial forces such as gravity or rotational velocity, imposed displacements and temperatures.

B. Applying loads to the finite element model

The axial and bending loads as calculated above were applied to the finite element model of the connecting rod. The axial load is applied at the small end lower half of the connecting rod and bending load is applied in the transverse direction at the centre of gravity of the model. The above two types of loads calculated at different crank angles were applied to the model one after another and analyzed for each load sets separately.

C. Applying boundary conditions

The displacement perpendicular to the geometry symmetry plane is restrained for all the nodes on the geometry symmetry plane. The displacement in the load direction is restrained against axial loads. If the axial load• is compressive in nature, all the nodes on the bearing surface of the big end bore upper half were restrained and whereas in tension the lower half is restrained. For the transverse loads, displacement in the same direction is restrained for all the nodes of the big end and small end bearing surfaces. Loads that are to be applied the finite element model were calculated analytically.

D. Analysis of the connecting rod

After applying the loads and boundary conditions, ANSYS solution module solves the finite element problem. The forces found at different crank angles were applied to the model and solved.

The stress contours in the model due to combined axial compressive and inertia bending loads at different crank angle. And stress contours in the model due to combined tensile and inertia bending loads at different crank angle.

The maximum induced stress values developed in the model due to the application of loads. The values are obtained from the finite element analysis of the connecting rod.

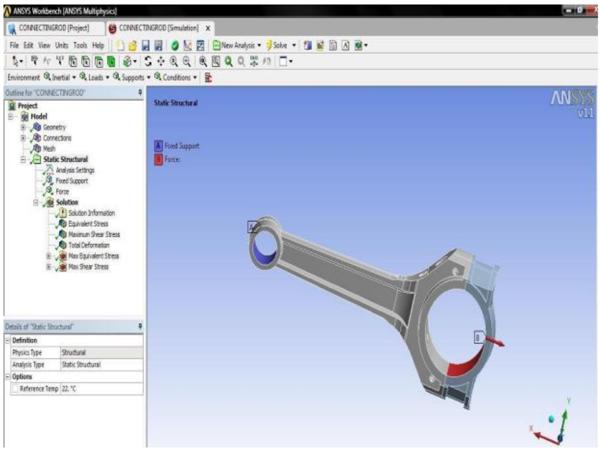

Fig 5. Tension on crank end and pin end restricted

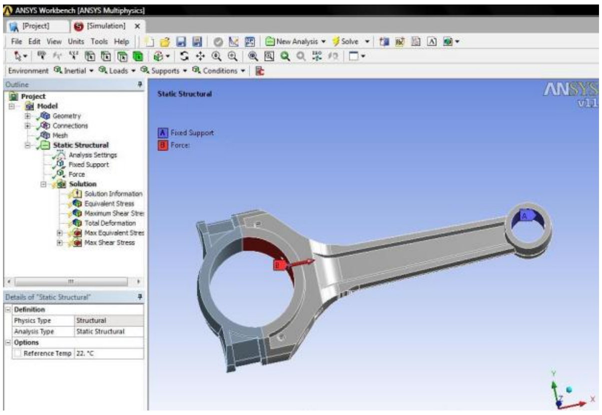

Fig 6. Compression on crank end and pin end restricted

VII. RESULTS AND DISCUSSIONS

The connecting rod of a high speed automotive compression ignition engine modeled using Modeling Technique and analyzed it using Finite Element Technique. Various alternatives are discussed for the assumed conditions in the above modeling and analysis of the connecting rod and the final dimensions obtained after finite element analysis, satisfy, the requirements.

The result of the finite element analysis is compared with analytical stress values. From the analysis it is found that the stress values are well below the yield strength of the connecting rod material.

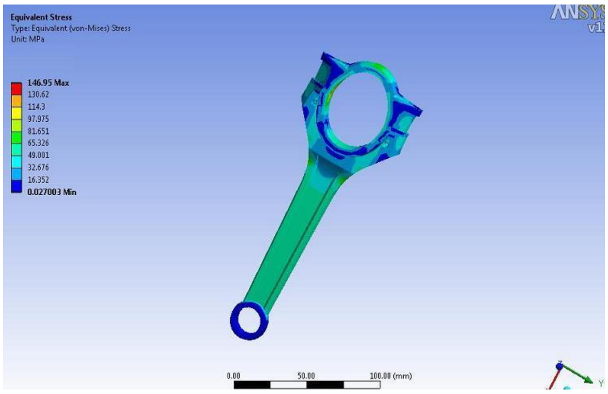

Fig 7. Structural analysis on existing connecting rod

Comparison of results indicates that the shank region of the connecting rod offers the highest potential for weight reduction. In the shank region, the rib and the web thickness were reduced only up to certain limits.

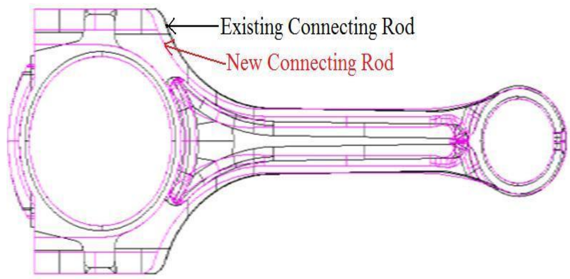

Fig 8. Shape optimization

After several iterations of calculating loads for different speeds, and performing analysis, a constrained model is obtained.

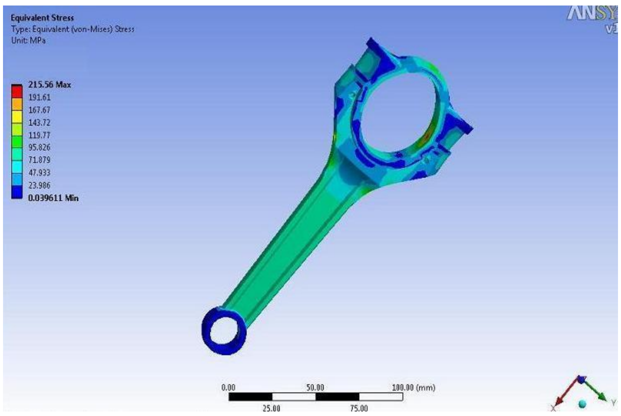

Fig 9. Structural analysis on new connecting rod

This study deals with the study of weight optimization performed under two cyclic loads comprising dynamic tensile and static compressive as the two extreme loads. Also consider the cyclic load conditions for life predictions analysis. In the optimization process, fatigue strength is very significant factor in this area of study. The study results in an optimized connecting rod that is about 10% lighter and 25% less expensive, as compared to the existing connecting rod.

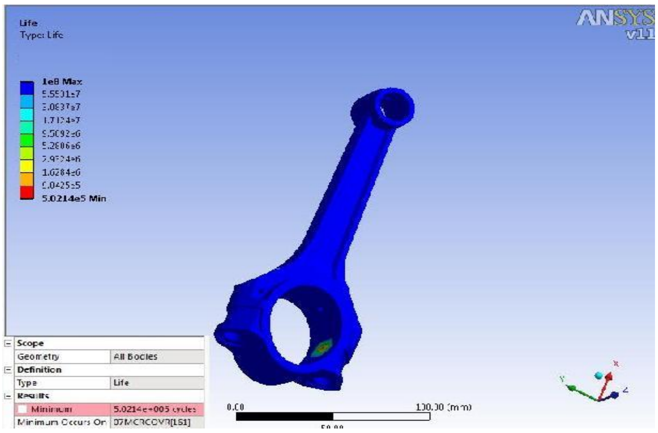

Fig 10. Life predictions analysis on connecting rod

Generally we use C45 steel for connecting rod manufacturing diesel engines. In this study I propose an alternative material C70 steel for connecting rod manufacturing with a purpose to reduce the weight of connecting rod. When we compare between C45 steel and C70 steel surprisingly it is noted that around 10% of weight reduction occurs.

TABLE I
MATERIAL PROPERTIES OF C45 AND C70 STEEL

S.No	**Material Properties**	**Values for C45 steel CR**	**Values for C70 steel CR**
1.	E (GPa)	207	212
2.	Yield strength(MPa)	700	574
3.	% Elongation	24	27
4.	% Reduction in area	42	25
5.	Tensile strength(MPa)	938	966
6.	Endurance limit(MPa)	415	399
7.	Density (kg/m^3)	0.000782	0.000796
8.	Weight(gms)	440	396
9.	I_{ZZ} (kg-m^2)	0.00144	0.00139
10.	Buckling load factor	7.8	9.6

VIII. CONCLUSION

It is an attempt in the development of a suitable less weight connecting rod for a multi-cylinder automotive compression-ignited engine using Modeling and Finite Element Techniques.

Optimization was performed to reduce weight and manufacturing cost of a steel connecting rod subjected to cyclic load comprising the compressive gas load and the dynamic tensile load at different speed, corresponding to various crank angles. Cost was reduced by changing the material of the existing C45 steel connecting rod to C70 steel.

- Optimization is performed to reduce weight and manufacturing cost of a steel connecting rod subjected to different load at different speed, corresponding to various crank angles.
- Material removal has been introduced in places where minimum stress is acting.

Cost is reduced by changing the material of the existing C 45 steel connecting rod to C 70 steel.

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