iJRASET
International Journal For Research in
Applied Science and Engineering Technology

INTERNATIONAL JOURNAL
FOR RESEARCH
IN APPLIED SCIENCE & ENGINEERING TECHNOLOGY

Volume: 6 Issue: VII Month of publication: July 2018

DOI: http://doi.org/10.22214/ijraset.2018.7024

International Journal for Research in Applied Science & Engineering Technology (IJRASET)
ISSN: 2321-9653; IC Value: 45.98; SJ Impact Factor: 6.887
Volume 6 Issue VII, July 2018- Available at www.ijraset.com

Design, Stress and Modal Analysis of Connecting Rod for Linear Opposed Engine

Mahesh Kumar Sah
Siddhartha Institute of Science And Technology

I. INTRODUCTION

A. Background

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. 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. Bringing the part to final dimensions under tight tolerance results in high expenditure for machining, as the blank usually contains more excess material. Rods are 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 A comparison of the European and North American connecting rod markets indicates that according to an unpublished market analysis for the year 2000. 78% of the connecting rods in Europe (total annual production: 80 million approximately) are steel forged as opposed to 43%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 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. The first aspect of this research program has been dealt with in a master's project entitled "Fatigue Behavior and Life predictions of Forged Steel and PM Connecting Rods. This current project deals with the analysis of part.

Due to large volume production, it is only logical that analysis 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. LITERATURE SURVEY

Benson. R.S and Whitehouse (2001) explain the heat transfer analysis for IC engines in Advanced Engineering Thermodynamics. To analyze the shape of connecting rod subjected to a load cycle, consisting of the inertia load deducted from gas load as one extreme and peak inertia load exerted by the piston assembly mass as the other extreme, with fatigue life constraint. Fatigue life defined as the sum of the crack initiation and crack growth lives, was obtained using fracture mechanics principles. The approach used finite element routine to first calculate the displacements and stresses in the rod. The stresses and the life were used in routine to evaluate the objective function and constraints. The new search direction was determined using finite difference approximation with design sensitivity analysis. The author was able to reduce the weight by 28%, when compared with the original component. For this analysis, to be developed approximate mathematical formulae to define connecting rod weight and cost as objective functions and also the constraints. The analysis 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. The FEA analysis is identified design loads in terms of maximum engine speed, and loads at the crank and piston pin ends. They performed static tests in which the crank end and the piston pin end failed at different loads. Clearly, the two ends were designed to withstand different loads. Computational strategy used in Mercedes Benz using examples of engine components. In their opinion, 2d FE models can be used to obtain rapid trend statements, and 3d FE models for more accurate investigation. The various individual loads acting on the connecting rod were used for performing simulation and actual stress distribution was obtained by superposition. The stress variation at the column center and column bottom of the connecting rod, as well as the bending stress at the column center. The plots, to be shown indicate that at the higher engine speeds, the peak tensile stress does not occur at 360^0 crank angle or top dead center. It was also observed that the r ratio varies with location, and at a given location it also varies with the engine speed.

John. B. Heywood (1994) has formulated the literature for the study of fundamentals of engine system. This book gives the overall view about engine operation, operation members and requirements of that member. Variational equations of elasticity used material derivative idea of continuum mechanics and an adjoint variable technique to calculate shape design sensitivities of stress. The results were used in an iterative analysis algorithm, steepest descent algorithm, to numerically solve an optimal design problem. The focus was on shape design sensitivity analysis with application to the example of a connecting rod. The stress constraints were imposed on principal stresses of inertia and firing loads. But fatigue strength was not addressed. The other constraint was the one on thickness to bind it away from zero. They could obtain 20% weight reduction in the neck region of the connecting rod. Design methodology in use at piaggio for connecting rod design, which incorporates an analysis session. However, neither the details of analysis nor the load under which analysis was performed were discussed. Two parametric procedures using 2d plane stress and 3d approach developed by the author were compared with experimental results and shown to have good agreements. The analysis procedure they developed was based on the 2d approach. A method to consider fatigue life as a constraint in optimal design of structures. They also demonstrated the concept on a SAE key hole specimen. In this approach a routine calculates the life and in addition to the stress limit, limits are imposed on the life of the component as calculated using FEA results.

Kolchin. A and V.Demidov (1989) has written a book named Design of Automotive Engines, which includes the overall design of all automotive engine systems with balancing. This is the book translated from the Russian by P.Zabolotnyi. The connecting rod is subjected to a complex state of loading. it undergoes high cyclic loads of the order of 10^8 to 10^9 cycles, which range from high compressive loads due to combustion, to high tensile loads due to inertia. therefore, durability of this component is of critical importance. Due to these factors, the connecting rod has been the topic of research for different aspects such as production technology, materials, performance simulation, fatigue, etc. for the current study, it was necessary to investigate finite element modeling techniques, analysis techniques, developments in production technology, new materials, fatigue modeling, and manufacturing cost analysis. This project performed three dimensional finite element analysis of a high-speed diesel engine connecting rod. For this analysis they used the maximum compressive load which was measured experimentally, and the maximum tensile load which is essentially the inertia load of the piston assembly mass. The load distributions on the piston pin end and crank end were determined experimentally. They modeled the connecting rod cap separately, and also modeled the bolt pretension using beam elements and multi point constraint equations. 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 that of the powder metal component. it also 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 choice for steel forged connecting rods. it also mentions two other steels that are being tested, modified micro-alloyed steel and modified carbon steel. Repgen.K (1992) have discussed about 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. The micro structural behavior investigated 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 is 18% higher than the later.

The analysis of the connecting rod end used a fatigue load cycle consisting of compressive gas load corresponding to maximum torque and tensile load corresponding to maximum inertia load. Evidently, they used the maximum loads in the whole operating range of the engine. To design for fatigue, modified Goodman equation with alternating octahedral shear stress and mean octahedral shear stress was used. For analysis, they generated an approximate design surface, and performed analysis of this design surface. The objective and constraint functions were updated to obtain precise values. This process was repeated till convergence was achieved. They also included constraints to avoid fretting fatigue. The mean and the alternating components of the stress were calculated using maximum and minimum values of octahedral shear stress. Their exercise reduced the connecting rod weight by nearly 27%. The initial and final connecting rod end designs are shown.

International Journal for Research in Applied Science & Engineering Technology (IJRASET)
ISSN: 2321-9653; IC Value: 45.98; SJ Impact Factor: 6.887
Volume 6 Issue VII, July 2018- Available at www.ijraset.com

A. Multi Cylinder Opposed Engine

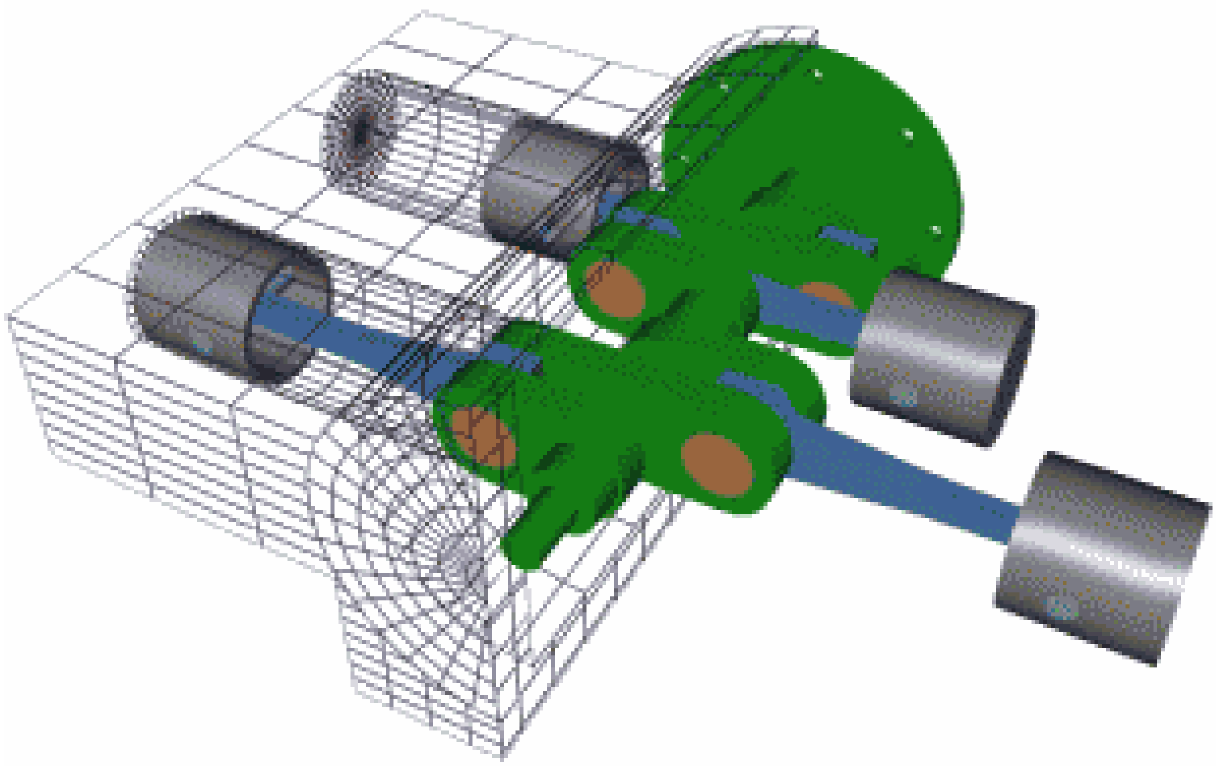


Fig 2.1 Multi Cylinder Opposed Engine

Engine design has been quite popular among researchers from the past and advancements are being done so that the design becomes more sophisticated. The design consideration depends on the purpose of calculations and the type of design. For the study of design even though there is no much literatures available, there are some useful literatures for this study.

One such literature is the "Design of Automotive Engines" by A.Kolchin and V.Demidov. This is the book translated from the Russian by P.Zabolotnyi. This is the book, which includes the overall design of all automotive engine systems with balancing.

The literature for the study of fundamentals of engine system is from "IC Engine Fundamentals" by John.B.Heywood. This book gives the overall view about engine operation, operation members and requirements of that member.

Another work done by P.C. Sharma is "Design of Machine Elements" which explains about the design of operating members of engine. P.K. Nag in "Engineering Thermodynamics" explains the heat transfer analysis in IC engines.

R.S. Benson and Whitehouse explain the heat transfer analysis for IC engines in "Advanced Engineering Thermodynamics".

III. DESIGN OF CONNECTING ROD

The connecting rod is the operating member of engine, which connects piston with crankshaft. It converts the reciprocating motion of piston into rotary motion of crankshaft. The connecting rod should have enough fatigue strength to withstand the gas forces transmitted through piston.

The material selection for connecting rod depends on the capacity of engine and operating conditions. The automobile and tractor engine employs a variety of connecting rod depending mostly on the type of engine and arrangement of the cylinders. The design elements of the connecting rod assembly are big end and small ends, connecting rod shank.

During the engine operation, the connecting rod is subject to the effect of alternating gas and inertial forces and sometimes these forces produce impact loads. Therefore, connecting rods are fabricated of alloyed or carbon steel which is highly resistant to fatigue. Connecting rods of carburetor engines are made of steel, grades 40, 45 and 45T2.

A. Calculations

Material

C 70 Steel

Ultimate Strength $\sigma_u = 800\ N/mm^2$

Yield Strength $\sigma_y = 600\ N/mm^2$

Specification of Engine

Engine Capacity 147.5 CC

Bore diameter 57 mm

Stroke length

Piston displacement = $\pi D^2 l\ / 4$

$147.5 = (3.14 \times 5.7^2 \times l) / 4$

$l = 57.8mm$

Gas Pressure

$P_1 = 1$ bar $T_1 = 300k$ $\gamma = 1.32$

Compression ratio = $\varepsilon = 9.4$

$P_2/P_1 = \varepsilon^\gamma$

$P_2 = (9.4)^{1.32}$

$P_2 = 19.25$ bar

$P_3 = 4 \times P_2$

$P_3 = 19.25 \times 4$

$= 77$ bar

Pressure will be lost due to constant volume, exhaust, blow down etc.,

(i.e.,) 40% of P_3

$P_3 = 77 \times 60\% = 46$ bar

Gas pressure = Cylinder Pressure - atmospheric Pressure

= 46-1

= 45 bar

= 4.5 Mpa

1) Force On The Connecting Rod

F_g = Gas pressure x Area of Piston

$= (4.5 \times 3.14 \times 0.0572 \times 106) / 4$

$= 11477N$

2) Determination Of Web Thickness

$I_{xx} = 1/12\ [BH3 - bh^3]$

$= 1/12\ [3t\ (4t)^3 - 2t\ (2t)^3]$

$= 1/12\ [192\ t^4 - 16t^4\ ]$

$= 14.76t^4$

$I_{yy} = 1/12\ [\ 2B^3t + ht^3]$

$= 1/12\ [2x\ (3t)^3\ xt + 2txt^3]$

$= 56t^3 / 12$

$= 4.66\, t^4$

$K^2_{xx} = I_{xx} / A$

$= 14.76t^4 / 8t^2$

$= 1.85t^2$

3) *Factor Of Safety* = 5

4) *Rankine's Formula*

$$fg = \frac{\sigma_y}{n}\left\{1 - \frac{\sigma_y}{4\pi^2 E}\left[\frac{Lo}{K_{xx}^2}\right]^2\right\}$$

L_0 → Length of connecting Rod (Centre to centre)

σ_y → 640 N/mm^2

E_1 → $2.1 \times 10^5\ N/mm^2$

N → Factor of Safety

F_g → Force on the connecting Rod

K_{xx} → radius of Gyration about xx axis

A → Area of Cross Section of I

$$\frac{11477}{8t^2} = \frac{640}{5}\left\{1 - \frac{640}{4 \times \pi^2 \times E} * \frac{[107^2]}{1.85\, t^2}\right\}$$

$$\frac{11477}{8t^2} = 128 \left\{\frac{[t^2 - 0.48]}{t^2}\right\}$$

t= 3.25mm

Width of connecting Rod = 3t = 3x3.25

= 9.75mm = 10mm

Height = 4t = 4x3.25 = 13mm

Depth at crank end = [1.1 to 1.25] depth at mid point

= 1.175 x 15

= 17.625

= 18mm

Depth at Piston end = (0.75 to 0.9) x depth at mid point

= 0.825 x 5

= 12.375

= 13mm

5) *Allowable Bending Stress*

$$[\sigma_b] = \frac{\sigma_y}{n} = \frac{640}{5} = 128\ N/mm^2$$

6) Bending Stress Due To Inertia Force

$$\sigma_{bmax} = \frac{\Upsilon a l^2 w^2 r}{9\sqrt{3}\, g\, z_{xx}}$$

Υ- Specific weight $= 7.8 \times 10^{-5} N/mm^3$

$a = 8t^2 = 8 \times 3.25^2 = 84.5mm^2$

$$z_{xx} = \frac{I_{xx}}{H/2} = \frac{14.66t^3}{2}$$

$= 7.33t^3$

$z_{xx} = 251.63mm^3$

$$r = \frac{Stroke}{2} = \frac{57.8}{2} = 28.9mm$$

$l = 107mm$

$n = 7500$ rpm

$$\omega = \frac{2\pi \times 7500}{60} = 785.4 \text{ rad /Sec}$$

$$\sigma_{bmax} = \frac{7.8 \times 10^{-5} \times 84.5 \times 107^2 \times 785.4^2 \times 28.9}{9\sqrt{3} \times 9.81 \times 1000 \times 251.63}$$

$\sigma_{bmax} = 35 N/mm^2$

$\sigma_{bmax} < \sigma[b]$ so satisfactory. (Therefore design is safe)

B. Design of Small End

Small end is designed by the following

1) To provide enough fatigue strength at section I-I when loaded by inertial force attaining their maximum with the engine operating at maximum speed under no load.
2) To stand to stresses occurring at the small end because of bushing pressed in.
3) To provide enough fatigue strength at section A-A to withstand the gas and inertial forces.

Calculation

$d_p \rightarrow$ Piston pin or gudgeon Pin outer dia

$d_p \rightarrow 0.22$ to $0.28 D$

$d_p \rightarrow 0.25 \times 57 = 14.25 \sim 14mm$

For carburetor Engine of small end

a) Inner diameter of small end D_S

W/o bushing $d_p \sim d_s$

With bushing $d_s = (1.1 – 1.25)\, d_p$

International Journal for Research in Applied Science & Engineering Technology (IJRASET)
ISSN: 2321-9653; IC Value: 45.98; SJ Impact Factor: 6.887
Volume 6 Issue VII, July 2018- Available at www.ijraset.com

The connecting rod is of type w/o bushing

$d_s = d_p = 14mm$

b) Outer diameter of end d_{se} = $(1.25 – 1.65)\ d_p$

= 1.45 x 14

= 20.3 ~ 20mm

c) Length of small end l_{se}

Retained pin (0.28- 0.45) B

Floating Pin (0.33- 0.45) B

L_{se} = 0.28 x 57 = 15.96 = 16mm

d) Maximum Radial Thickness of End Wall

t_{se} = $(0.16 \text{ to } 0.27)\ d_p$

= 0.21 x 14

= 2.94

= 3mm

C. Design Of Section I-I

1) Pulsating Cycle Max Stress

$$\sigma_{mcx} = \frac{(m_p + m_{se})\ w^2_{\ idmax}\ R^{(1+x)}]}{2\ t_{se} l_{se}}$$

Mass of piston (m_p) = 125gm = 0.125 kg

Mass of small end ($M_{s.e}$) = 6 to 9% of connecting Rod

= 7% of connecting Rod

= 7 % of 150

= 10gm – 0.01Kg

Maximum idling rpm (n_{idmax}) = 1.05 to 1.20 of engine rpm

= 1.05 x 7500

= 7875

$w = 2\pi N_{id}/60 = 824.25$

r- Crankradius = 28.9mm

$$\lambda = \frac{r}{L_{\ con.rod}} = \frac{28.9}{107} = 0.27$$

l_{se} - 16mm

t_{se} - 3mm

$$\sigma_{mox} = \frac{(.125 + 0.01)\ 824.25^2 \times 28.9\ (1 + 0.27)}{2 \times 16 \times 3} = 35MPa$$

The small end force attains its maximum with the Piston at TDC at the beginning of induction. This force is determined by

$P_{j.p}$ = $-m_p\ Rw^2\ (1+R)$

= -0.125 x 28.9 x 824.25 (1+0.27)

= - 3117N

D. Design Of Big End

1) Crankpin diameter dcp

= 0.56 – 0.75 B

→ 0.65 x 57→ 37 ~ 36mm

2) *Shell Wall Thickenss*

Thin walled (0.03 – 0.05) d_{co}
Thick walled (0.1 – 0.2) dep
Thick walled so – 5mm

3) *Big end length*

l_{be} → (0.45 – 095) d_p
→ .6 x30 - 18mm

E. Design of Connecting Rod Shank

h_{sh} min = (0.50 – 0.55) ^{de}=0.5 x 20 = 10mm
h_{sh} = (1.2 – 1.4) h&$_{nmin}$ = 1.3 x 10
= 13mm

b_{sh} = (0.50 – 0.60) = 7.8 = 10mm

IV. BOUNDARY CONDITIONS

A. Loading

The crank and piston pin ends are assumed to have a sinusoidal distributed loading over the contact surfaces area, under tensile loading.

The normal pressure on the contact surface is given by:

$p = p_o \cos$ □

The load is distributed over an angle of 180°. The total resultant load is given by:

$$P_t = \int_{-\pi/2}^{\pi/2} p_{o} (\cos^2 \Theta)\, r\, t\, d\Theta = p_o\, r\, t\, \pi/2$$

The normal pressure constant p_o for tensile load is, therefore, given by:

$p_o = P_t / (r\, t\, \pi / 2)$

The tensile load acting on the connecting rod can be obtained using the expression from the force analysis of the slider crank mechanism.

For compressive loading of the connecting rod, the crank and the piston pin ends are assumed to have a uniformly distributed loading through 120° contact surface, as
shown. The normal pressure is given by:

$p = p_o$

The total resultant load is given by:

$$P_t = \int_{-\pi/3}^{\pi/3} p_{o} (\cos \Theta)\, r\, t\, d\Theta = p_o\, r\, t\, \pi/3$$

The normal pressure constant for compressive load is then given by:

$P_O = P_C / (r\, t\, \sqrt{3})$

P_c can be obtained from the indicator diagram, such as the one shown in Figure of an engine.

In this study four finite element models were analyzed. FEA for both tensile and compressive loads were conducted. Two cases were analyzed for each case, one with load applied at the crank end and restrained at the piston pin end, and the other with load applied at the piston pin end and restrained at the crank end. In the analysis carried out, the axial load was 11.477 KN in both tension and compression. The pressure constants for 11.477 KN are as follows:

Compressive Loading:

Crank End: p_o = 11477/ (24 x 17.056 x $\sqrt{3}$) = 37.66 MPa
Piston pin End: p_o = 11477/ (11.97 x 18.402 x $\sqrt{3}$) = 69.98 MPa

Tensile Loading:

Crank End: p_o = 11477/ [24 x 17.056 x $(\pi/2)$] = 41.5 MPa
Piston pin End: p_o = 11477/ [11.97 x 18.402 x $(\pi/2)$] = 77.17 MPa

International Journal for Research in Applied Science & Engineering Technology (IJRASET)
ISSN: 2321-9653; IC Value: 45.98; SJ Impact Factor: 6.887
Volume 6 Issue VII, July 2018- Available at www.ijraset.com

Since the analysis is linear elastic, for static analysis the stress, displacement and strain are proportional to the magnitude of the load. Therefore, the obtained results from FEA readily apply to other elastic load cases by using proportional scaling factor.

B. Static FEA

As already mentioned FEA models to be solved. Tensile load is applied at the crank end as well as piston pin in FEA model. Similarly, when the connecting rod is under axial compressive load, 120^0 of Contact surface area is totally restrained. Project shows FEA model in which Compressive load is applied at the crank end and piston pin end is completely restrained.

C. Performance Evaluation

The opposed cylinder engine has the following advantages.

1) Well balanced engine
2) Vibration is less
3) Simpler connecting rod design
4) Simpler crankshaft design
5) Less space occupied
6) Simpler Casing design

V. MODELLING

A. Connecting Rod – Sectional View

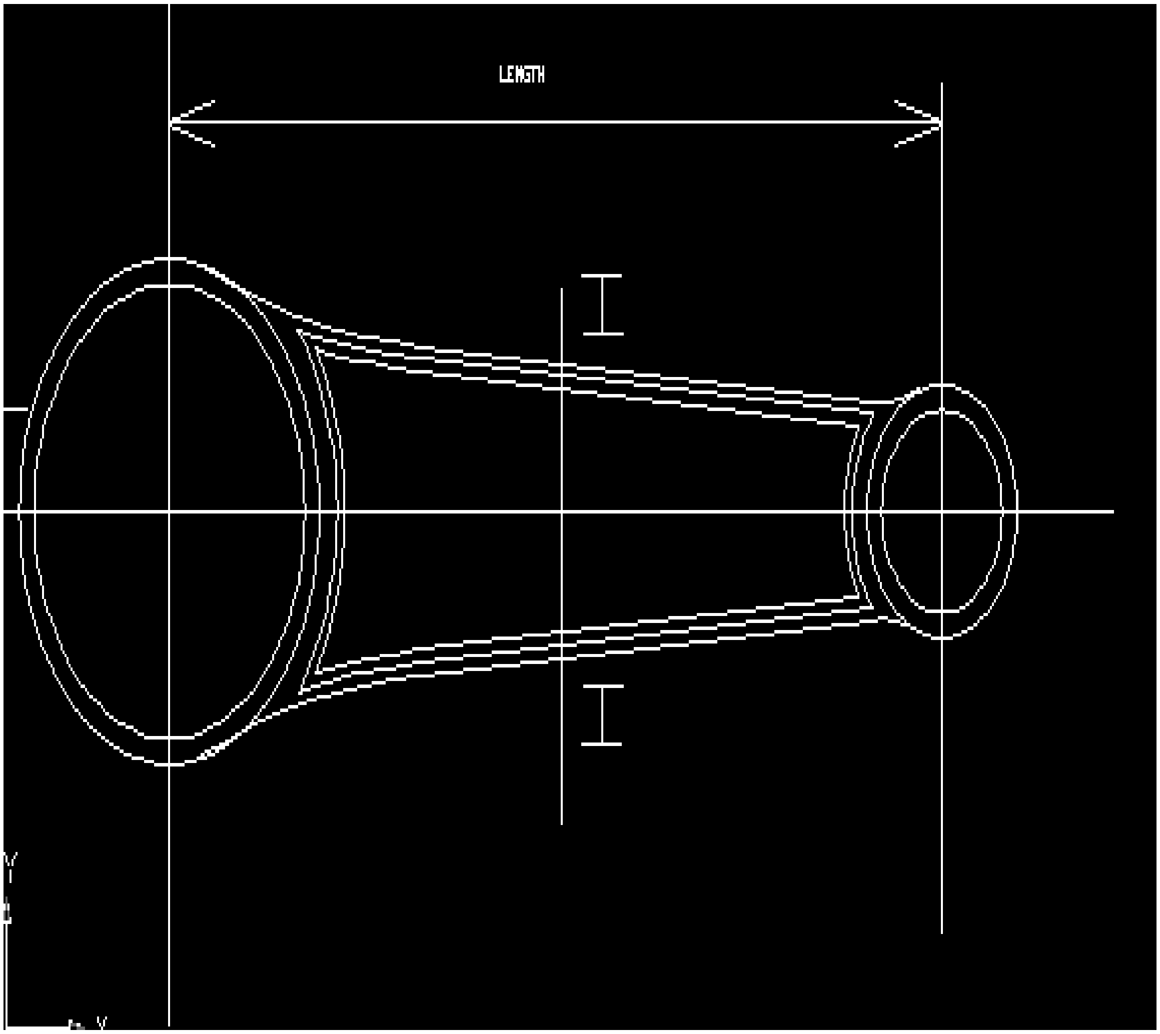


Fig 5.1 Connecting Rod – Sectional View

International Journal for Research in Applied Science & Engineering Technology (IJRASET)
ISSN: 2321-9653; IC Value: 45.98; SJ Impact Factor: 6.887
Volume 6 Issue VII, July 2018- Available at www.ijraset.com

B. Design Of I – Section

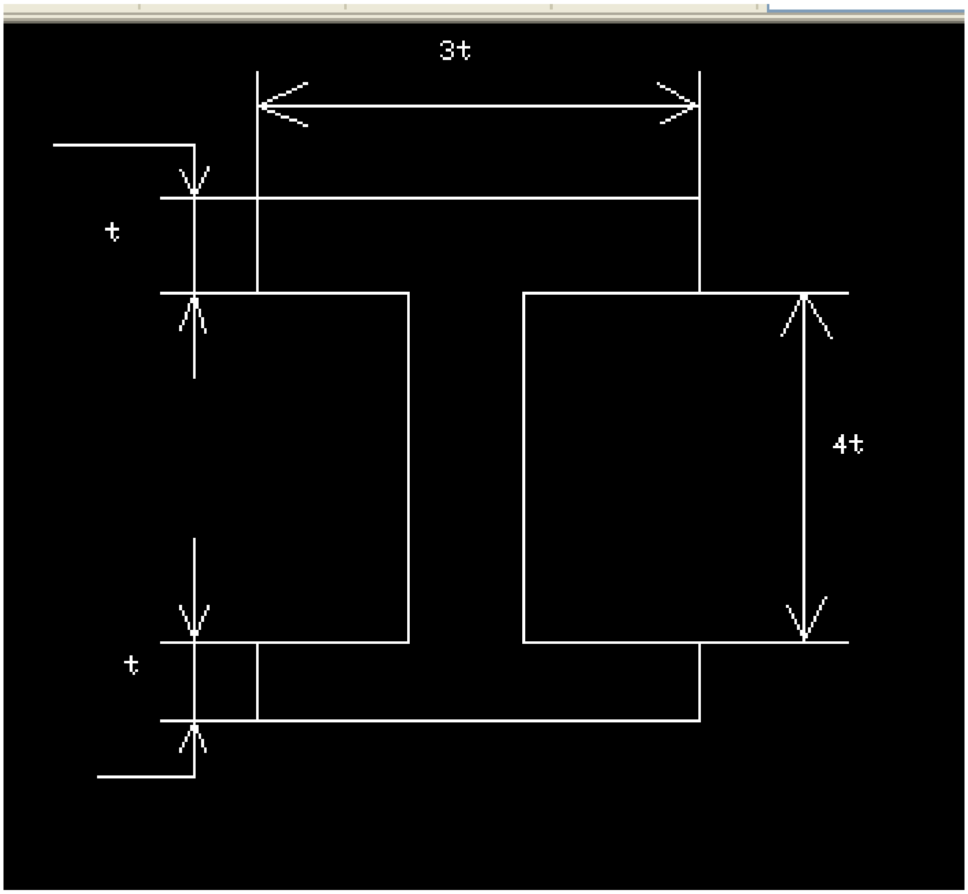


Fig 5.2 Design of I – Section

C. Connecting Rod – In Tension

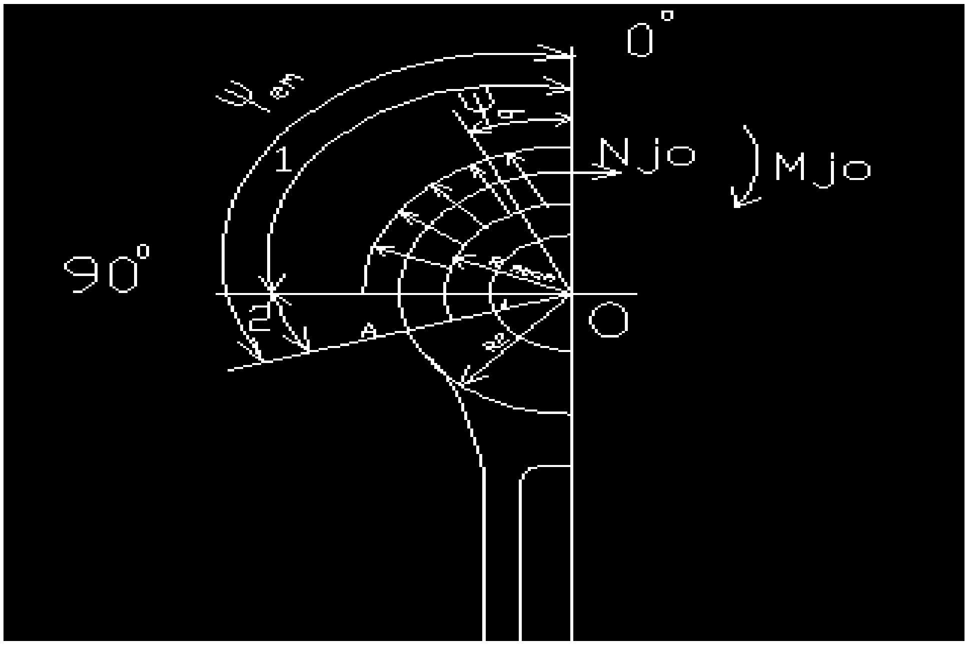


Fig 5.3 Connecting Rod – In Tension

International Journal for Research in Applied Science & Engineering Technology (IJRASET)
ISSN: 2321-9653; IC Value: 45.98; SJ Impact Factor: 6.887
Volume 6 Issue VII, July 2018- Available at www.ijraset.com

D. Connecting Rod In Compression

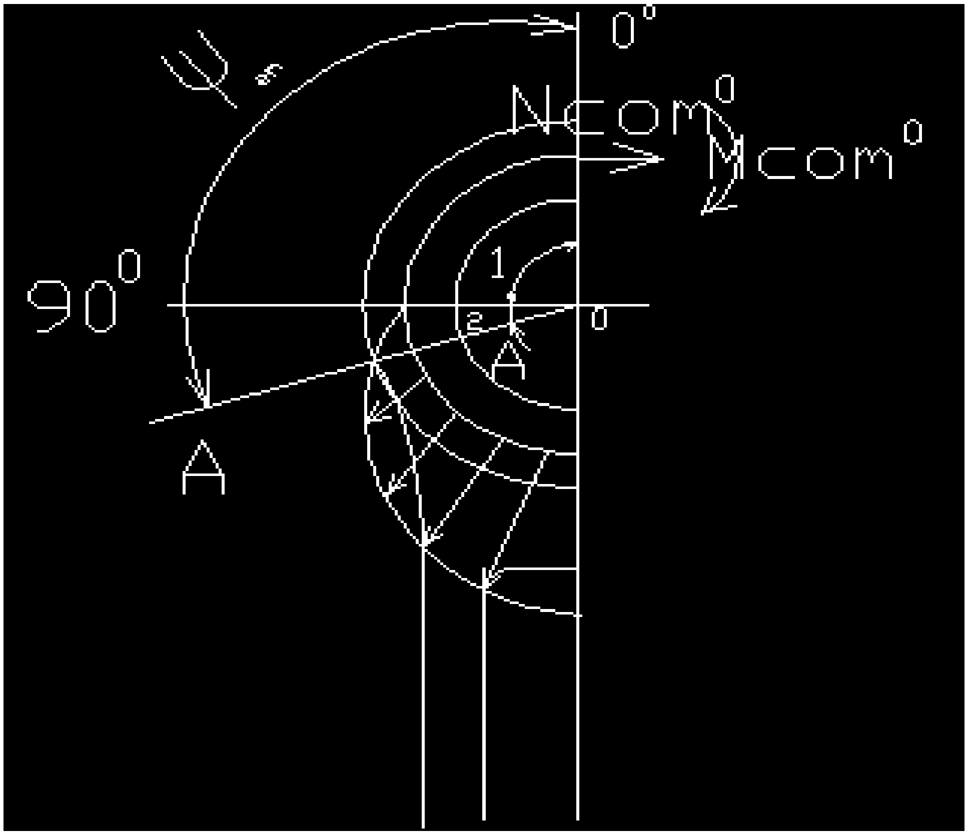


Fig 5.4 Connecting Rod In Compression

E. Connecting Rod – Existing

Fig 5.5 Connecting Rod – Existing

International Journal for Research in Applied Science & Engineering Technology (IJRASET)
ISSN: 2321-9653; IC Value: 45.98; SJ Impact Factor: 6.887
Volume 6 Issue VII, July 2018- Available at www.ijraset.com

F. Connecting Rod – Proposed

Fig 5.6 Connecting Rod – Proposed

G. Connecting Rod – Assembled

Fig 5.7 Connecting Rod – Assembled

H. Connecting Rod – Exploded View

Fig 5.8 Connecting Rod – Exploded View

I. Crank Shaft

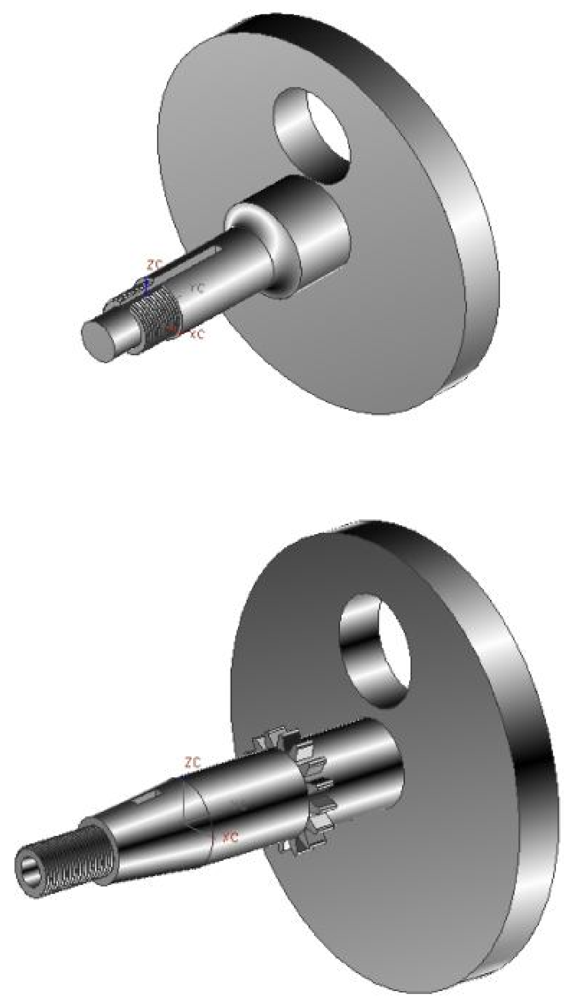


Fig 5.9 Crank Shaft

J. Piston

Fig 5.10 Piston

K. Gudgeon Pin

Fig 5.11 Gudgeon Pin

VI. WORK DONE SO FAR AND PROPOSED

A. Work done so far

1) Design of connecting rod

a) Determination of force acting on the connecting rod

b) Determination of web thickness

c) Width and height of the connecting rod

d) Depth at crank and piston end

e) Allowable bending stress

f) Bending stress due to inertia force

g) Design of smaller end of the connecting rod
h) Design of bigger end of connecting rod
i) Design of connecting rod shank

2) Pressure acting on the connecting rod under boundary conditions
a) Pressure at crank and piston end due to compressive load
b) Pressure at crank and piston end due to tensile load

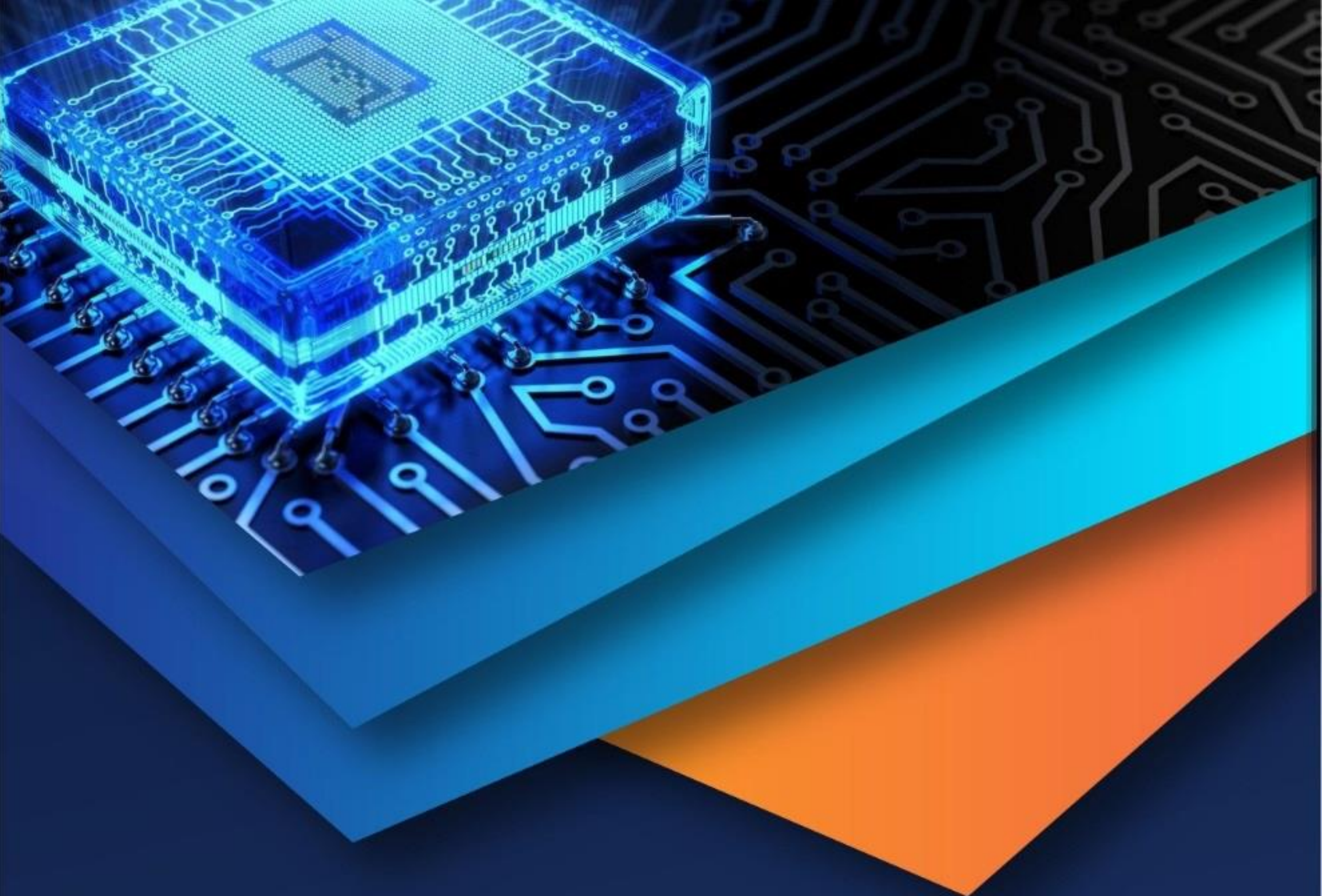
3) Modeling
a) Connecting rod assembly
b) Crank shaft
c) Piston and gudgeon pin

B. Work To Be Proposed

1) Stress analysis of connecting rod
a) Stresses in x-direction
b) Stresses in y-direction
c) Stresses in xy-direction
d) Von Mises stresses
e) Bending stresses
2) Modal analysis of connecting rod

REFERENCES

[1] Achten PAJ, van den Oever JPJ, Potma J, Vael GEM. Horsepower with brains: The design of the Chiron free piston engine, SAE Paper 2000-01-2545, 2000.
[2] Clark N, Nandakumar S, Atkinson C, Atkinson R, McDaniel T, Petreanu S et al. Operation of a small bore two-stroke linear engine, Proc. of the Fall Technical Conference of the ASME Internal Combustion Engine Division, 1998:31-1:33—40.
[3] Design Databook, department of mechanical engineering, PSG College of engineering, Coimbatore, 1998 edition
[4] Hibi A, Ito T. Fundamental test results of a free piston internal combustion engine, Proc. Inst. Mech. Eng., 2004:218:1149—1157.
[5] Tikkanen S, Herranen M, Savela R, Vilenius M. Simulation of a free piston engine: A dual piston case, Proc. Sixth Scandinavian international conference on fluid power, pp. 339—349, 1999.
[6] Uludogan A, Foster DE, Reitz RD. Modeling the effect of engine speed on the combustion process and emissions in a DI Diesel engine, SAE Paper 962056, 1996.



10.22214/IJRASET

INDEX COPERNICUS

45.98

IMPACT FACTOR: 7.129

TOGETHER WE REACH THE GOAL

IMPACT FACTOR: 7.429

INTERNATIONAL JOURNAL FOR RESEARCH

IN APPLIED SCIENCE & ENGINEERING TECHNOLOGY