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Research Paper

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Methodology for Calculations and Design of Connecting Rod for Fatigue Loads in IC Engines

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Abstract—In this work, design and structural calculations of connecting rod of IC engine has been performed. From design point of view, connecting rod is most critical part as it the only part which converts reciprocating forces into rotating forces and thus creates unbalance in engine. From the functionality point of view, connecting rod must have the higher inertia at the lowest weight. The forces acting on connecting rod are: - Peak combustion pressure force, inertia force of reciprocating masses, Weight of Reciprocating parts and frictional forces due to cylinder wall thrust. It experiences complex loading of compression and tensile loadings under cyclic process, which repeats after each 360 phase. Therefore, the design calculations are analyzed for the axial compressive as well as axial tensile loads and considering the fatigue life of connecting rod. This work computes, the required strength and size in the critical areas of connecting rod. Connecting rod calculations, Fatigue, Endurance, Con rod, I C Engine.

Keywords— Connecting rod calculations, Fatigue, Endurance, Con rod, I C Engine.

I. Introduction

Calculations of connecting rod is the critical topic of design due to its critical motion and complex loading pattern. It is the only part in IC engines which simultaneously rotates as well as reciprocates while transmitting the forces. To initiate 3D modelling of connecting rod and before performing FEA, it is necessary to calculate the forces acting on connecting rod and required dimensions of critical sections to sustain it. A connecting rod is of H or I section. As a column or strut, H section has to sustain high buckling as well as flexural bending stress. The connecting rod also experiences multiple and complex forces in cyclic manner. Which includes high compressive force due to combustion and high tensile loads due to inertia of reciprocating masses. Therefore, infinity life of connecting rod needs to be considered while designing. In most of cases material used are aluminum of steel alloys, such as 25CrMo4, 42CrMo4, C70, AISI 1070, etc.

II. Related Work

Major amount of work and research have been conducted on optimization of connecting rod in IC engine areas. While out of those, most of work is involved geometrical or material related changes. Dynamic analysis of Loads and stress of connecting rod is discussed in [5]. Design and structural analysis of connecting rod can be read from [6]. Buckling of beams, flexural rigidity is discussed in book [1]. Inertia Forces and two mass equivalent system is referred from book [2]. Fatigue /endurance calculations, fatigue failure criterion is discussed in book [3].

III. Methodology

In this literature, Improved force computation for connecting rod has been performed, which can be used for design calculation for new connecting rod or for optimization and weight savings of existing connecting rods.

For improving accuracy of calculation, Load fluctuation w.r.t crank rotation, Max Compressive load, max tensile loads, buckling and bending stress, flexural rigidity and fatigue strength etc are taken into account while performing calculations.

Fatigue analysis is done using PTC Creo-Simulate 4.0 to find out high Factor of safety areas for weight reduction.

Finally, calculation Approach and FOS is cross verified by applying it on existing three production engines of different categories from light to heavy vehicles.

3.1 Design Inputs - Engine Specifications

Table 1. Engine specifications for which connecting rod to be designed.

S.N	Parameter	Sym	Value	Unit
1	Engine capacity	Vs	2200	cc
2	Bore Diameter	d	0.085	m
3	Stroke	s	0.095	m
4	Crank Radius	r	0.05	m
5	con rod length	l	0.150	m
6	Max. Engine Speed	V	4000	rpm
7	Obliquity Ratio	n	3.10	-
8	Mass of connecting rod.	Mc	0.700	Kg
9	Mass of Piston Assy & Pin	Mp	0.94	Kg
10	Max. Combustion Pressure	Pg	13000000	N/m2

3.1.1 Design Inputs - Material Specifications

Table 2. Material Properties of connecting rod Material 42CrMo4.

S.N	Parameter	Value	Unit
1	Young's modulus (E)	200	Gpa
2	Ultimate Tensile Strength (Sut)	900	Mpa
3	Yield Strength (Syt)	650	Mpa
4	Compressive/crushing stress (σc)	650	Mpa
5	Endurance Strength (Se')	420	Mpa
6	Endurance Bending Strength (σ_{eb})	530	Mpa

Se = endurance limit of actual connecting rod.

Se = Ka . Kb . Kc . Kd . Ke . Kg . Se' ………………..…[3]

= 0.78 x 1 x 1 x 1.02 x 1 x 1 x 420

= 334.152 MPa

Actual connecting rod Specification for endurance

S.N	Parameter	Value	Unit
1	Endurance Strength (Se)	334.15	Mpa
2	Endurance Shear Str. (Max shear) (Ses)	167	Mpa
3	Endurance shear Str (Von mises) (Ses)	193	Mpa
4	Endurance Bending Strength (σ_{eb})	422	Mpa

3.2 Forces on Connecting rod:

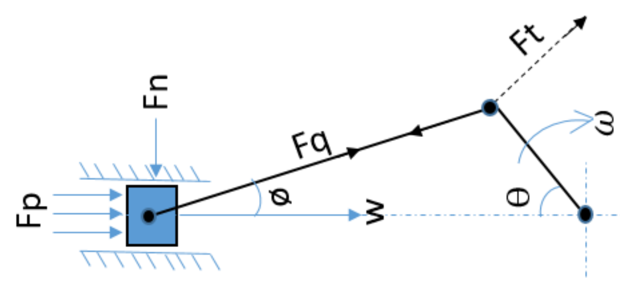


Figure 1. Forces on connecting rod. Represented in Slider Crank Mechanism.

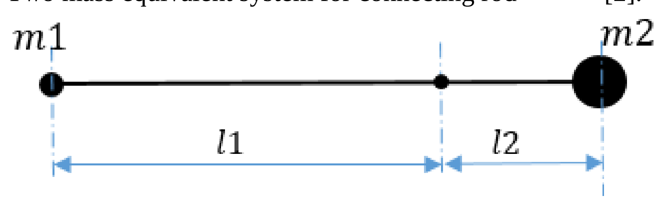
3.2.1 Gas Force of Piston Top (Fg):

$$Fg = Pg.A$$

----**3.2.2 Inertia force due to reciprocating mass (Fi):**

Fi = mass X acceleration

Two mass equivalent system for connecting rod [2].



(Fig.2 - Two mass equivalent system for connecting rod.)

$$m1 = \frac{(l - l1).(Mc - m1)}{l1} \ \& \ m2 = \frac{(l - l2).(Mc - m2)}{l2}$$

Mr=Total Reciprocating mass=*m1*+*Mp* = **Mr** Kg

Piston acceleration = a = $\omega^2.r.(\cos\Theta + \frac{\cos 2\Theta}{n})$……………...[2]

Piston acceleration will be maximum when piston will be at TDC i.e Θ is zero.

$$Fi = Mr.\omega^2.r.\left(\cos(0) + \frac{\cos(2 \times 0)}{n}\right) \ m/s^2$$

3.2.3 Force due to weight of reciprocating mass (w):

$$w = Mr.g$$

---Friction force can be neglected, because it is reducing the applied force on connecting rod.

3.2.4 Net max. Vertical Force on Connecting Rod, (Fp:)

Fp = Fg - Fi ± Ff + W ……………..….[2]

Max. Force **Fq** can be found out using trigonometry and relation of Θ as shown in fig.1.

3.3 H section design:

3.3.1 H section design for buckling of con rod.

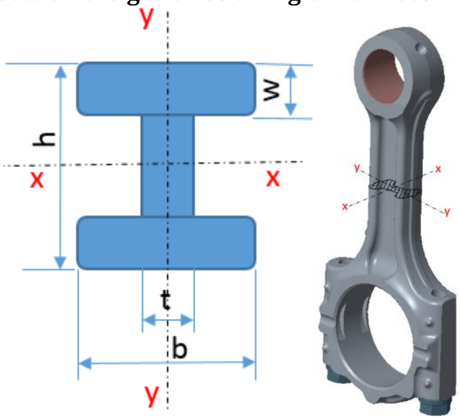


Figure 3. H section and of connecting rod.

Assumed or primary dimensions of H section-
b = Width h = Height w = Thickness t = Thickness.
Area of Assumed I section, A = (b x h)-[(b-t) x (h-2w)]
= **A** mm^2
Effective length of Connecting rod about xx, Le = ***l*** mm.
Effective length of connecting rod about yy, Le = ***l*/2** mm.

Moment of Inertia about xx, Ixx = $\frac{b \, x \, h^3}{12}$ mm^4
Moment of Inertia about yy, Iyy = $\frac{h \, x \, b^3}{12}$ mm^4
Radius of gyration about xx, $Kxx = \sqrt{\frac{Ixx}{A}}$ mm
Radius of gyration about yy, $Kyy = \sqrt{\frac{Iyy}{A}}$ mm
Rankine constant, a = $\frac{\sigma c}{\pi^2 \, . \, E}$ = 0.000329
FOS = 2

Safe load for Buckling about xx axis, Fc:

$$Fc = \frac{\sigma c \; x \; A\prime}{1+a\left(\frac{Le}{Kxx}\right)^2} \; X \; \frac{1}{FOS} \; \ldots\ldots\ldots\ldots\ldots [1]$$

Min. required area of I section to avoid buckling about x-x axis.

$$A' = \frac{Fc.Fos.\left[1+a\left(\frac{Le}{Kxx}\right)^2\right]}{\sigma c} \; mm^2$$

Assumed H section Area A > A', so design is safe in buckling about xx

Safe load for Buckling about yy axis, Fc :

$$Fc = \frac{\sigma c \; x \; A}{1+a\left(\frac{Le}{Kxx}\right)^2} \; x \; \frac{1}{FOS}$$

Min. required area for I section to avoid buckling about y-y axis.

$$A' = \frac{Fc.Fos.\left[1+a\left(\frac{Le}{Kyy}\right)^2\right]}{\sigma c} \; mm^2$$

Assumed H section Area A > A', so design is safe in buckling about xx

3.3.2 Calculation of H section under axial loading on :

Forces are fluctuating so let's find out σ_{max} and σ_{min} and then σ_a and σ_m.

$$\sigma_{min} = \frac{Fq}{A} \; \mathbf{N} \text{ (- ve for Compressive)}$$

$$\sigma_{max} = \frac{Fi}{A} \; \mathbf{N}$$

$$\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2} \; \mathbf{N/mm^2}$$

$$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2} \; \mathbf{N/mm^2}$$

Considering modified Goodman Theory, and selecting equation for Goodman line.

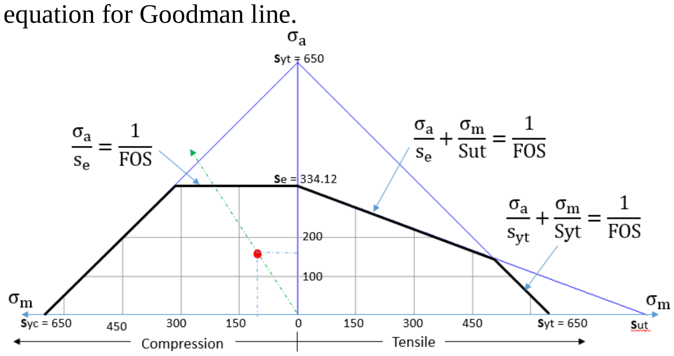


Figure 4. Modified Goodman diagram for connecting rod of 25CrMo4, 45CrMoo4, C70 and equivalent materials.

Equation of suitable line: $\frac{\sigma_a}{S_e} = \frac{1}{FOS}$ …………….…..… [3]
Rearranging the Goodman Line equation to obtain area:

$$\frac{\frac{\sigma_{max} - \sigma_{min}}{2}}{S_e} = \frac{1}{FOS}$$

$$A' = \frac{(Fi-(-\;Fq)}{Se\;.\;2} \, .FOS \; \mathbf{mm^2}$$

Assumed I section Area A > A', so design is safe for axial loading.

3.4 Small End / Piston Pin Dia Calculation:

Table 4. Material Properties of Piston Pin - 16MnCr5.

S. N	Parameter	Value	Unit
1	Endurance Strength (Se)	420	Mpa

2	Endurance Shear Str (Max shear stress) (Ses)	210	Mpa
3	Endurance shear Strength (Von mises stress) (Ses)	240	Mpa
4	Endurance Bending Strength σ_{eb})	350	Mpa

In case of Piston pin, lads are completely reversed so Considering S-N curve for infinity life of Pin.

3.4.1 Piston Pin Subjected to double shear stress:

$$Ses = \frac{F_P . FOS}{2 . A} \; (Ses=210 \text{ Mpa - max shear theory})$$

$$A = \frac{\pi[d^2-(d-2.t)^2]}{4} \quad (t = \text{thickness of hollow Pin})$$

$$d = \mathbf{x\ mm}$$

Min. required small end ID of connecting rod subjected to shear, d = **x mm**.

3.4.2 Piston Pin is subjected to Crushing stress, σc: –

$$\sigma c = Se = \frac{F_P . FOS}{l . (d - 2 .t)} \quad (l = \text{small end width})$$

$$d = \mathbf{x' \ mm}$$

Min. required small end ID of connecting rod subjected to crushing, d = **x' mm.**

3.4.3 Piston Pin subjected to Bending Moment:

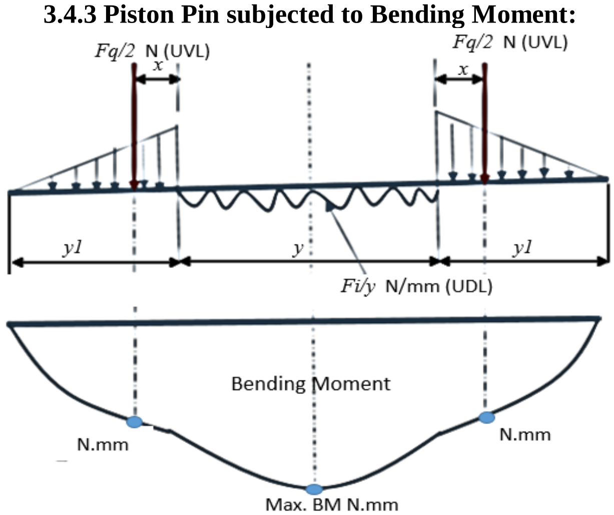


Figure 5. Force representation and bending moment diagram of piston Pin.

$$\sigma_{eb} = \frac{M . y}{Imin} . FOS \text{ ....[1]}$$

$$\sigma_{eb} = \frac{M \, x \, d/2}{\frac{\pi d^4}{64}} \; x \; 2$$

$$d = \mathbf{x'' \ mm.}$$

Min. required small end ID of connecting rod subjected to crushing, d = **x" mm.**

Select max diameter out of 3.4.1, 3.4.2 and 3.4.3.

3.5 Small End thickness design:

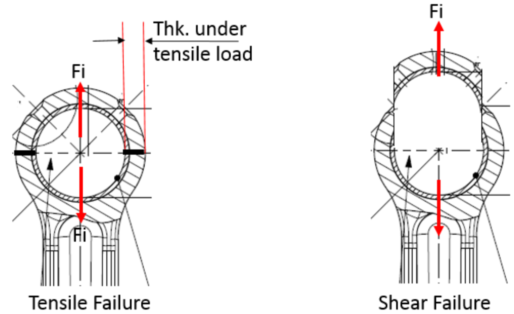


Figure 6. Tensile and shear loading on connecting rod Small end.

Considering S-N curve for infinity of life

3.5.1 Thickness in Tensile loading

$$Se = \frac{F_i . FOS}{A} \text{..............................[3]}$$

$$= \frac{F_i . FOS}{2 \, x \, (l \, x \, t)} \quad \textit{(l=to be assumed.)}$$

$$t = y \ \mathbf{mm}$$

Min required thickness of small end when subjected to tensile force, t = y mm.

3.5.2 Thickness in Shear loading (Van mosses):

$$Ses = \frac{F_i . FOS}{A}$$

$$= \frac{F_i . FOS}{2 \, x \, (l \, x \, t)} \quad \textit{(l=to be assumed.)}$$

$$t = \mathbf{y' \ mm}$$

Min required thickness of small end when subjected to Shear force, t = y' mm.

Select max value out of I and ii for small end thickness.

3.6 Design of Bolt

Table 5. Material Properties of bolt - 16MnCr5

S.N	Parameter	Value	Unit
1	Ultimate tensile Strength (SUT)	570	Mpa
2	Yield tensile Strength (Syt)	295	Mpa
3	Young's Modulus	200 000	Mpa
4	Endurance Strength-Material (Von Mises theory) (Se')	285	Mpa
5	Endurance Strength-Bolt (Se) (Ka.Kb.Kc,Ke,Kf-1 & Kd=1.05)	290	Mpa

Considering S-N curve for infinity life of Bolt,

3.6.1 Bolt CS area in tensile stress,

$$Se = \frac{F_i \, .FOS}{A \text{ x No of Bolts}}$$

$$A = A \text{ mm}^2$$

$$d = \sqrt{\frac{A \, x \, 4}{\pi}}$$

$$d = x \text{ mm}$$

Minor diameter of bolt = d = x mm.

3.6.2 Bolt Design as per IS:1367:

Impact Tensile load on each bolt = $\frac{2 \text{ x Fi x FOS}}{No \ of \ bolts}$ **N**

Table 9 — Proof loads – ISO metric fine pitch thread

Thread $(d \times P$ [a]$)$	Nominal stress area $A_{s, nom}$ [b] mm²	Property class								
		3.6	4.6	4.8	5.6	5.8	6.8	8.8	9.8	10.9
		Proof load $(A_{s, nom} \times S_p)$, N								
M8×1	39,2	7 060	8 820	12 200	11 000	14 900	17 200	22 700	25 500	32 500
M10×1	64,5	11 600	14 500	20 000	18 100	24 500	28 400	37 400	41 900	53 500
M10×1,25	61,2	11 000	13 800	19 000	17 100	23 300	26 900	35 500	39 800	50 800

Table 6. Bolt selection as per IS:1367

Suitable Bolt Size can be selected from IS: 1367 based on the calculated tensile load.

Selecting max bolt size out of 3.6.1 and 3.6.2.

IV. RESULTS AND DISCUSSION.

4.1.1 FEA modelling and results:

In order to compare the structure behavior of existing and modified connecting rod, FEA for fatigue analysis using PTC Creo-Simulation is performed.

FOS in Fatigue Analysis.

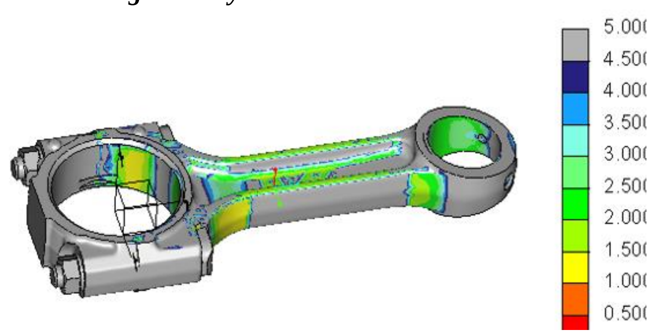


Figure 7. FOS Fatigue Analysis of Existing and Improved Connecting rod.

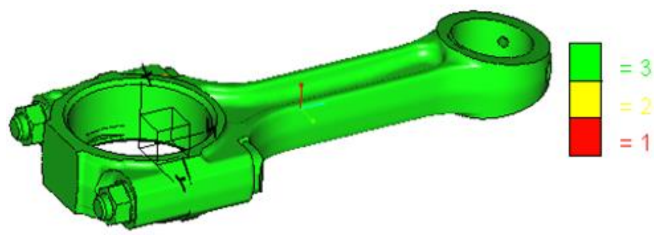


Figure 8. Confidence of Life – for Improved Connecting Rod.

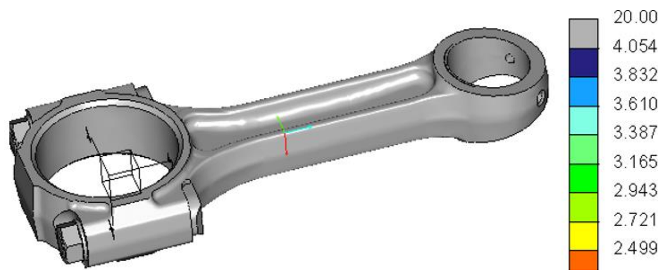


Figure 9. Log Life - for Improved Connecting Rod.

Initially, Cross section dimensions, OD of small end, bolt size has been decided using above mentioned calculation methodology.

Afresh created CAD model is evaluated using FEA-Simulations and it has been verified that the all critical regions of connecting rod are with in limit of FOS and stress.

Moreover, same has been applied on various categories of engines and it is found that this calculations work properly there also.

Structure behavior of one of such connecting rod is shown here, which are done for fatigue analysis using Tool - PTC Creo-Simulation.

V. CONCLUSION AND FUTURE SCOPE

Using above design considerations and methodology, connecting rod can be optimised within Factor of Safety up to two. Methodology is cross verified by applying it on existing three State of the art production engines of different categories from light to heavy four wheeler and two wheeler Engines. It resulted in weight reduction up to 10-15%. Comparison of Fatigue stress analysis of existing and optimized connecting rod shows that stresses are within acceptable limit. Even Log Life (infinity life) remains almost same.

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