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CALCULATION METHODOLOGY FOR DESIGN OF CONNECTING ROD CONSIDERING FATIGUE STRESSES IN IC ENGINE

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***Abstract*— In this Literature, design and structural calculations of connecting rod of IC engine has been performed. Design point of view, connecting rod is most critical part as it the only part which converts reciprocating forces into rotating forces. It is subjected to complex loading of compression and tensile under cyclic process. Therefore, the design calculations are done considering the fatigue life of connecting rod. This work computes, the required strength and size in the critical areas of connecting rod. The calculation methodology is also verified with existing connecting rods of three different engine categories, like small petrol engine, SUV diesel engine and heavy commercial Engine.**

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

I. INTRODUCTION

The connecting rod calculations is the critical topic of design and research due to its motion and complex loading. Before detailed FEA analysis it is necessary to find out the dimensions by calculations to build the primary 3D model of connecting rod. So it becomes necessary to calculate the forces & there loading pattern acting on connecting rod. Normally it is made of H section. H Section is having high inertia and less weight compared to circle or rectangular sections. H section is also better in sustaining high buckling and bending stresses. [1][3] The connecting rod is subjected to complex cyclic forces. Which includes tensile stress due to inertia forces & compressive force due to combustion.[2]. Here infinity life of connecting rod is considered while laying down this methodology. Commonly material used are aluminum of steel alloys, such as 25CrMo4, 42CrMo4, C70, AISI 1070, etc.
This calculation can be used to optimize/weight reduction of existing or for design of new connecting rod.

II. PROPOSED ALGORITHM

This approach suggests improved method of calculations of connecting rod of internal combustion engine. Stress pattern of cyclic loads on connecting rod is already identified as fluctuating [2]. Load variation w.r.t crank angle [4][7], Max Compressive and maximum tensile loads considering p-theta diagram [4][7] , endurance limits, buckling, flexural rigidity etc are considered in proper maner to get more accurate results. Following algorithm is used to laid down for the calculation / design methodology:

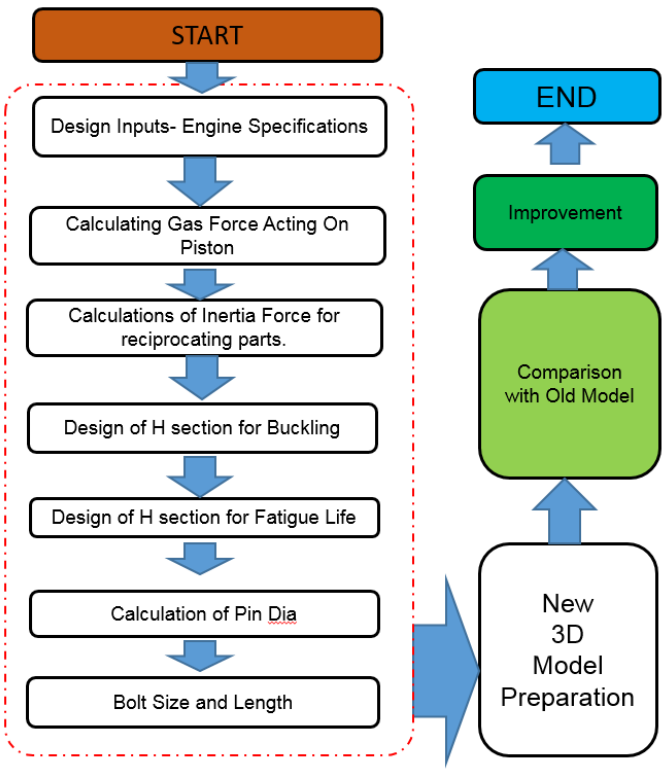

Fig. 1. Flow chart for Design of Connecting Rod.

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2.1 Engine Specifications:

Table 1. Required Design Inputs - Engine specifications.

S.N	Parameter	Sym	Unit
1	Engine Capacity	Vs	cc
2	Cyl. Bore Diameter	d	m
3	Piston Stroke	s	m
4	Crank Radius	r	m
5	Con rod length	l	m
6	Max. Engine Speed	V	rpm
7	Obliquity Ratio	n	-
8	Total Mass of connecting rod.	Mc	Kg
9	Mass of Piston Assy & Pin	Mp	Kg
10	Max. Combustion Pressure	Pg	N/m2

Design Inputs - Material Specifications:

Table 2. Material Properties of connecting rod - 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:

Table 3. Actual Material Properties of connecting rod -

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

2.2 Forces on Connecting rod:

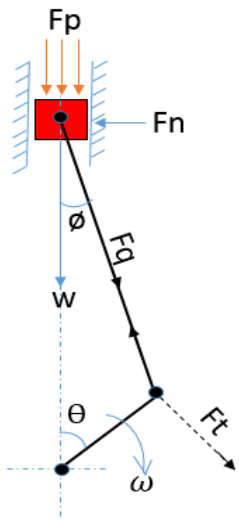

Figure 2. Slider Crank Mechanism for Connecting rod. ……[2]

Gas Force of Piston Top (Fg):

$$Fg = Pg.A = Pg.\frac{\pi d^2}{4}$$

2.2 Inertia force due to reciprocating mass (Fi):

Two mass equivalent system for connecting rod [2].

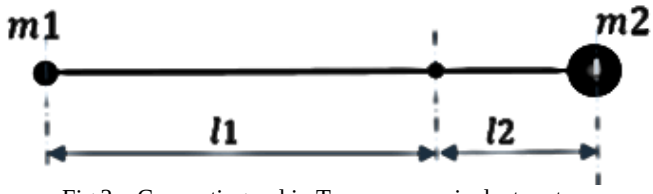

Fig.3 – Connecting rod in Two mass equivalent system.

$l1.m1 = l2.m2$, where $m1 + m2 = m$ and $l1 + l2 = l$ [2]

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

m1 = Reciprocating mass of connecting rod = **m1** kg.
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.

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

Inertia/acceleration force due to reciprocating mass,

Fi = Mr.a ……..…[2]

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

w = Mr.g

Friction force (Ff) between Piston and cylinder wall can be neglected [2], because it is reducing the applied force on connecting rod.

2.4 Net max. Force on piston pin, (Fp:)

Fp = Fg - Fi ± Ff + W ……..….[2]

Max. Compressive Force along connecting rod (**Fq**) can be found out using trigonometric relation between Θ and ø as shown in figure.2.[2]. And Inertia Force (Fi) will act as Maximum Tensile Force along connecting rod. Fq and Fi are the forces which are considered for further calculations.

H section Design:

2.5 H section design for buckling of con rod.

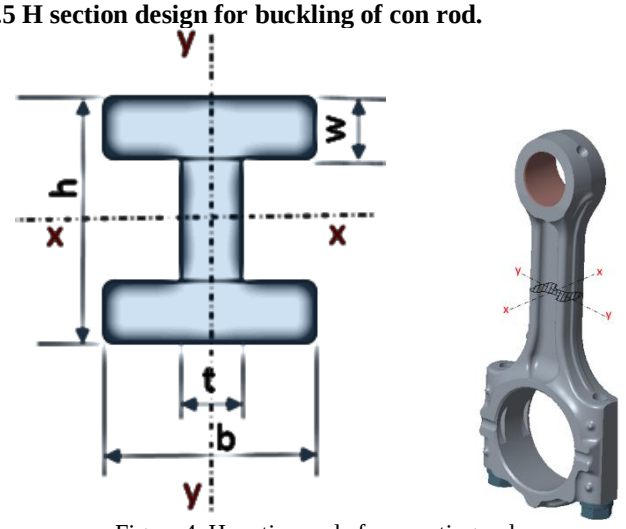

Figure 4. H section and of connecting rod.

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 = $\frac{b \times h^3}{12}$ mm^4…… [1]

Moment of Inertia about yy, Iyy = $\frac{h \times b^3}{12}$ 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 for given material, a = $\frac{\sigma c}{\pi^2 \,.\, E}$ = 0.000329 ...[1]

Factor of Safety = 2

Buckling load about xx axis, Fc:

$$Fc = \frac{\sigma c \; x \; A'}{1+a\left(\frac{Le}{Kxx}\right)^2} \; X \; \frac{1}{FOS} \quad \text{……..…. [1]}$$

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

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

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

---

Buckling load about yy axis, Fc :

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

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

If assumed H section Area A > A', design is safe in buckling about xx

---

2.6 Calculation of H section under axial loading on:

Considering Fq and Fi through 360 degree, we get resulting forces in fluctuating manner, so let's find out σ_{max} and σ_{min} and then σ_a and σ_m.

$$\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2} \text{ N/mm}^2 \qquad [3]$$

$$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2} \text{ N/mm}^2 \qquad [3]$$

Considering modified Goodman Theory, and selecting equation for Goodman line. [3]

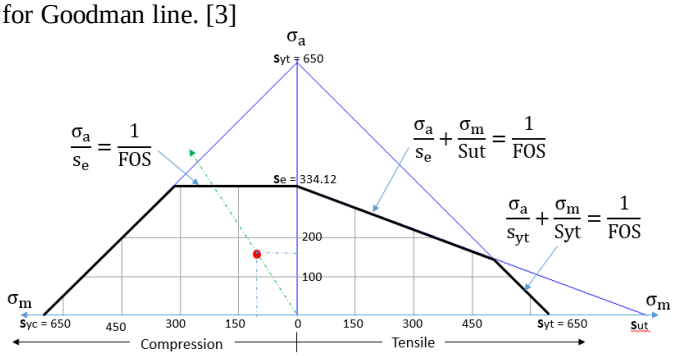

Figure 5. Modified Goodman diagram for 25CrMo4, 45CrMoo4, C70 and equivalent materials.

Equation of suitable line: $\frac{\sigma_a}{s_e} = \frac{1}{FOS}$ …….……..…..… [3]

Rearranging the Goodman Line equation to obtain area:

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

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

If assumed I section Area A > A', It is safe for axial loading.

2.7 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.[8]

Piston Pin Subjected to double shear stress:

$$Ses = \frac{F_P\ .FOS}{2\ .A}$$ *(Ses=210 Mpa - max shear theory)....* [3]

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

d = **x mm**

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

Piston Pin is subjected to Crushing stress, σc: –

$$\sigma c = Se = \frac{F_P\ .\ FOS}{l\ .\ (d-2\ .t)}$$ (*l* = small end width)……[3]

d = **x' mm**

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

Piston Pin subjected to Bending Moment:

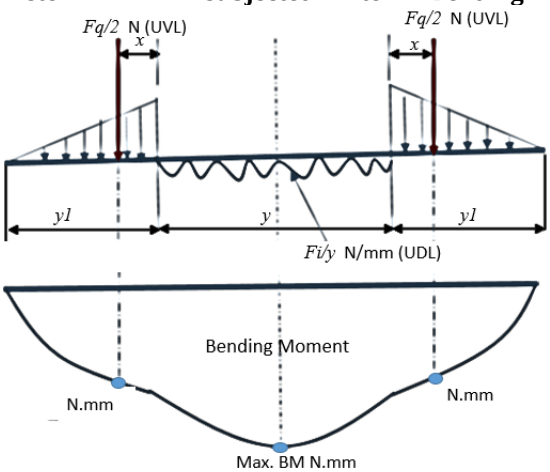

Figure 5. Force representation and bending moment of piston Pin. [8]

$$\sigma_{eb} = \frac{M\ .\ y}{Imin}.FOS$$ ……[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 x, x' and x''

2.8 Small End thickness design:

As the loads on Pin are completely reversed type, so considering S-N curve for infinity of life.

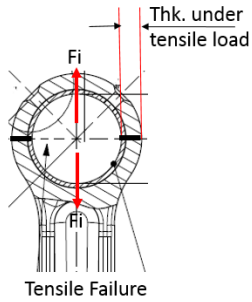

Figure 6. Tensile loading on connecting rod Small end.

Considering S-N curve for infinity of life

Thickness in Tensile loading

$$Se = \frac{F_i\ .FOS}{A}$$ ……[3]

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

t = y **mm**

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

Thickness in Shear loading (Van mosses):

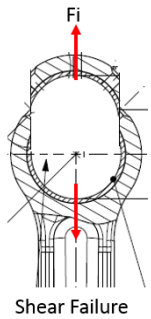

Figure7. Shearing of connecting rod Small end

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

$$= \frac{F_i .FOS}{2 \times (l \times t)} \quad (l = \text{to be assumed.})$$

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

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

Select max value out of t from y and y' for small end thickness.

2.9 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

Table 5. Material Properties of bolt - 16MnCr5

Bolt CS area in tensile stress,

Considering S-N curve for infinity life of Bolt

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

$$A = A \ mm^2$$

$$d = \sqrt{\frac{A \times 4}{\pi}}$$

$$d = x \ mm$$

Minor diameter of bolt = d = x mm.

Bolt Design as per IS:1367:

Impact Tensile load on each bolt = $\frac{2 \times Fi \times FOS}{No \ of \ bolts}$ **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 50
M10×1	64,5	11 600	14 500	20 000	18 100	24 500	28 400	37 400	41 900	53 50
M10×1,25	61,2	11 000	13 800	19 000	17 100	23 300	26 900	35 500	39 800	50 80

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.

III. EXPERIMENT AND RESULT

H or I section dimensions, gudgeon pin diameter, outer diameter of small end and big end, bolt size can be finalized using above calculations.

Newly built 3D model is analyzed through FEA simulations and proved the all critical areas are with in limit of allowable stress. Kumar PraKash Shankar el al. (2015) [10], Mohammed M. N el al. (2012) [11] and Rehman ek al.(2008) addressed the FEA for connectind rod is their work.

Additionally, This Calculation Methodology is applied on connecting rods of different categories of state of the art productionised engines (Passengers, commercial and two wheeler engines.) And it is found that this methodology works properly with in FOS 2.

Structure behavior of connecting rod is checked through FEA for fatigue analysis using Tool - PTC Creo-Simulation.
Meshing of connecting rod:

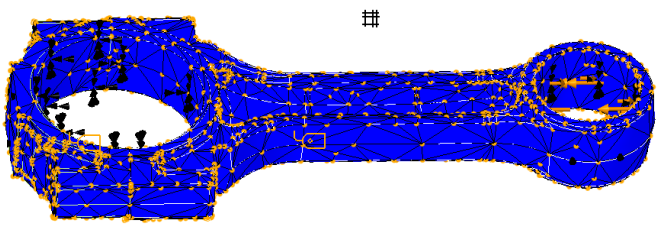

Figure8. Meshing of Connecting rod.

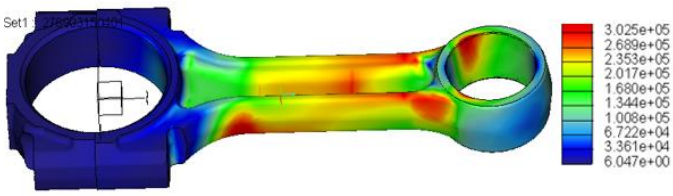

Figure 9. Stresse in KPa.

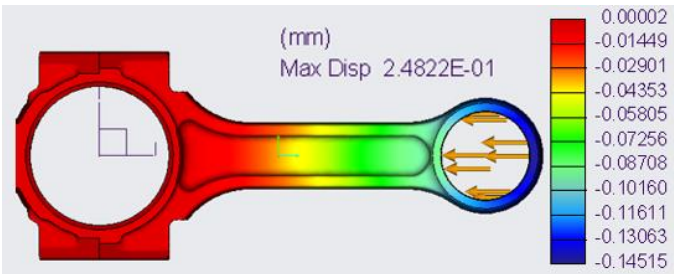

Figure 10. Deformation in compressive force.

Figure 11. FOS in Fatigue Analysis.

IV. CONCLUSION

It clearly shows that using above methodology and calculations, con rod can be designed or optimised within FOS as low as 2. It is verified using FEA analysis for fatigue failure as well as it is cross checked with connecting rod of actual working engines of different class from light to heavy four and two wheeler Engines. It resulted in weight reduction up to 10-15%. Even Log Life (infinity life) remains above log of 6.

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