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Buckling Analysis of Connecting Rod

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Abstract - The connecting rod is a major link inside a combustion engine. It connects the piston to the crankshaft and is responsible for transferring power from the piston to the crankshaft and sending it to the transmission. There are different types of materials and production methods used in the creation of connecting rods. The most common types of Connecting rods are steel and aluminum. The most common types of manufacturing processes are casting, forging and powdered metallurgy. The result predicts the maximum buckling load and critical region on the connecting rod using ANSYS. It is important to locate the critical area of concentrated stress for appropriate modifications. To find the stresses developed in connecting rod under static loading with different loading conditions of compression and tension at crank end and pin end of connecting rod.

Index terms - Connecting Rod, PRO-E, ANSYS12.

I. Introduction

The connecting rod is a major link inside of a combustion engine. Lighter connecting rods help to decrease load caused by forces of inertia in engine as it does not require big balancing weight on crankshaft it connects the piston to the crankshaft and is responsible for transferring power from the piston to the crankshaft and sending it to the transmission. There are different types of materials and production methods used in the creation of connecting rods. The most common types of materials used for connecting rods are steel and aluminum.

The most common types of manufacturing processes are casting, forging and powdered metallurgy. Connecting rods are widely used in variety of engines such as, in-line engines, V-engine, opposed cylinder engines, radial engines and opposed-piston engines. A connecting rod consists of a pin-end, a shank section, and a crank-end. Pin-end and crank-end pinholes at the upper and lower ends are machined to permit accurate fitting of bearings. These holes must be parallel. The upper end of the connecting rod is connected to the piston by the piston pin. If the piston pin is locked in the piston pin bosses or if it floats in the piston and the connecting rod, the upper hole of the connecting rod will have a solid bearing (bushing) of Bronze or a similar material.

A. Material for Connecting Rod

The most common types of materials used for connecting rods are steel and aluminum.

II. Analytical Calculations

For analytical calculations 'I' section shown in figure 1 is considered and by using Rankines formula all dimensions are calculated and then from these calculations FE Analysis is performed following are the calculations:

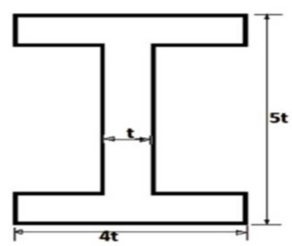

Figure 1. Dimensions of I section of connecting rod

Let us consider I section of connecting rod as shown in figure with following proportions.
Flange and web thickness of the section = t
Width of the section, B = 4t
Depth or height of the section, H = 5t

First of all let us find whether the section chosen is satisfactory or not. The connecting rod is considered like both ends hinged for buckling about X-axis and both ends fixed for buckling about Y-axis.

So connecting rod should be equally strong in buckling about both axes. in order to have a connecting rod equally strong at both the axes.
Ixx = 4Iyy
Ixx = Moment of inertia of the section about X-axis
Iyy = Moment of inertia of section about Y-axis
(Note: Ixx is kept slightly less than 4Iyy)
Area of the cross section = 2[4t x t] + 3t x t

$=11t^2$

Moment of inertia about x-axis

$I\ xx = 1/12(BD^3\text{-}bd^3)$

$=1/12\ [4t\ \{5t\}^3 - 3t\ \{3t\}^3]$

$= 419[t4]/12$

And moment of inertia about y-axis

$I\ yy = 2\times 1/12\times t\times\{4t\}^3+1/12\{3t\}t^3$

$=134/12[t^4]$

$I\ xx/I\ yy = [419/12]x[12/134]=3.12$

Since the value of I xx/I yy lies between 3 and 3.5 m therefore I-section chosen is quite satisfactory.

Now, Let us find the dimension of I- section. Since the connecting rod is designed by taking the force on the connecting rod (Fc) equal to the maximum force on the Piston (FL) due to gas pressure.

$F_C= F_L=\pi/4\times D^2\times P$

$F_C= F_L=\pi/4\times D^2\times 3.15$

$=\pi/4\times(100)^2\times 3.15$

$= 24740N$

We know that the connecting rod is designed for buckling about X – axis in plane of motion of connecting rod assume that both ends are hinged. Since the factor of safety is 6 therefore the buckling load.

$W_B=F_C\times FOS$

$=24740\times 6$

$=148440N$

We know that radius of gyration of section about X axis:

$K_{XX}=\sqrt{I_{XX}/A}$

$=\sqrt{419/12t^4\times 1/11t^2}$

$=1.78t$

Length of crank

r =stroke of piston/2

=190/2

=95mm

Length of connecting rod

Equivalent length of the connecting rod

For both ends hinged

L=l=252.5mm

Now according to Rankine's formula we know that Buckling load (W_B)

$W_B=[\sigma_c\times A]/1+\alpha(L/Kxx)^2$

Where $\sigma_{c=}320N/mm^2$(for mild steel)

$\alpha =1/7500$ (for mild steel)

Now,

$148440=320\times 11t^2/1+ (1/7500(252.5/1.78t)^2)$

$464=11t^2/ (1+2.67/t^2)$

$11t^4\text{-}464t^2\text{-}1238.88=0$

Put $t^2=x$

$11x^2\text{-}464x\text{-}1238.88=0$

$\therefore x=44.70 \qquad x=-2.519$

t =6.68mm

∴t=7mm say

Width of the section, B = 4×t =28mm

Depth or height of the section, H = 5 × t =35mm

Inner diameter of small end,

$d_{1=}F_L/P_{b1}\times l_1$

$d_{1=}24740/12.5\times 1.5d_1$

$d_{1=}36.32mm$

Outer diameter of small end

$_{=}d_1+2t_b+2t_m$

=37+2×3+2×6

=55mm

Where,

Thickness of bush (t_b) =2to5mm

Marginal thickness(t_m) =5to15mm

Inner diameter of big end$d_2=F_L/Pb_2\times l_2$

Pb_2 range =10.8to12.6N/mm^2

L_2 range=1to1.25d^2

$\therefore d_2=24740/10.8\times 1\times d_2$

=48mm

Outer diameter of big end

$=55+2t_b+2d_b+2t_m$

$_{=}55+2\times 3+2\times 5+2\times 6$

=85mm say 90mm

From above calculations an image is drawn which is shown in figure 2.

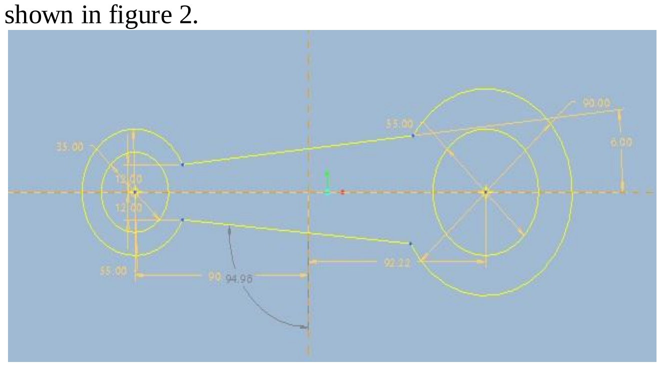

Figure 2.2D Image of connecting rod

A. Calculations for stress

These dimensions are at the middle of the connecting rod. The width (*B*) is kept constant throughout the length of the rod, but the depth (*H*) varies. The depth near the big end or crank end is kept as 1.1*H* to 1.25*H* and the depth near the small end or piston end is kept as 0.75*H* to 0.9*H*. Let us take Depth near the big end,

$H_1 = 1.2H = 1.2 \times 35 = 42$ mm

and depth near the small end,

$H_2 = 0.85H = 0.85 \times 35 = 29.75$ say 30 mm

Therefore Dimensions of the section near the big end

= 42 mm × 28 mm

and dimensions of the section near the small end

= 30 mm × 28 mm

Since the connecting rod is manufactured by forging, therefore the sharp corners of *I*-section are rounded off, for easy removal of the section from the dies.

Dimensions of the crankpin or the big end bearing and piston pin or small end bearing.

Let dc = Diameter of the crankpin or big end bearing,
lc = length of the crankpin or big end bearing = 1.0 dc
p_{bc} = Bearing pressure = 10.8 N/mm^2
We know that load on the crankpin or big end bearing
= Projected area × Bearing pressure
= $dc \cdot lc \cdot pbc = dc \times 1.0\, dc \times 10.8 = 10.8\,(dc)^2$

Since the crankpin or the big end bearing is designed for the maximum gas force (FL), therefore, equating the load on the crankpin or big end bearing to the maximum gas force, *i.e.*
$10.8\,(dc)^2 = F$L = 24 740 N
Therefore $(dc)^2$ = 24 740 / 10.8 = 2290.74 or dc = 47.86 say 55 mm
and lc = 1.0 dc = 1.0 × 55 = 55

The big end has removable precision bearing shells of brass or bronze or steel with a thin lining (1mm or less) of bearing metal such as babbit.
Again,
let dp = Diameter of the piston pin or small end bearing,
lp = Length of the piston pin or small end bearing = 1.5dp
p_{bp} = Bearing pressure = 12.5N/mm^2
We know that the load on the piston pin or small end bearing
= Project area × Bearing pressure
= $dp \cdot lp \cdot pbp = dp \times 1.5\, dp \times 12.5 = 18.75\,(dp)^2$

Since the piston pin or the small end bearing is designed for the maximum gas force (FL), therefore, equating the load on the piston pin or the small end bearing to the maximum gas force, *i.e.*
$18.75\,(dp)^2$ = 24 740 N
$(dp)^2$ = 24 740 / 18.75 = 1319.46 or dp = 36.32 mm
and lp = 1.5 dp = 1.5 × 36.32 = 54.48 mm.
The small end bearing is usually a phosphor bronze bush of about 3 mm thickness
Outer diameter of small end
$=d_1+2t_b+2t_m$
=37+2×3+2×6
=55mm
Where,
Thickness of bush (t_b) =2to5mm
Marginal thickness(t_m) =5to15mm
Outer diameter of big end
$=55+2t_b+2d_b+2t_m$
=55+2×3+2×5+2×6
=85mm say 90mm

Size of bolts for securing the big end cap

Let dcb = Core diameter of the bolts,
σ_t = Allowable tensile stress for the material of the bolts
= 60 N/mm^2
and n_b = Number of bolts. Generally two bolts are used.
We know that force on the bolts

$$= \frac{\pi}{4} dc^2 \sigma_t n_b$$

$$= \frac{\pi}{4} dc^2 \times 60 \times 2$$

The bolts and the big end cap are subjected to tensile force which corresponds to the inertia force of the reciprocating parts at the top dead centre on the exhaust stroke. We know that inertia force of the reciprocating parts,

$$FI = m\, r\, w^2 \times (cos\theta + \frac{cos2\theta}{\frac{l}{r}})$$

We also know that at top dead centre on the exhaust stroke,

$$FI = 2.25 \times 0.095 \times (188.48)^2 \left(cos\theta + \frac{cos\,2\theta}{\frac{l}{r}}\right) \quad (1)$$

Fi = 9490N
N = 1800 rpm
Theta = 0 degree
Equating the inertia force to the force on the bolts, we have
9490 = 94.26 $(dcb)^2$ or $(dcb)^2$ = 9490 / 94.26 = 100.7
dcb = 10.03 mm
and nominal diameter of the bolt,
db = $\frac{dcb}{0.84}$ =11.94
Thickness of the big end cap
Let tc = Thickness of the big end cap,
bc = Width of the big end cap. It is taken equal to the length of the
crankpin or big end bearing (lc) = 55 mm (calculated above)
σb = Allowable bending stress for the material of the cap = 80 N/mm^2 (Assume from data book)

The big end cap is designed as a beam freely supported at the cap bolt centres and loaded by the inertia force at the top dead centre on the exhaust stroke (*i.e.* FI when theta = 0). Since the load is assumed to act in between the uniformly distributed load and the centrally concentrated load, therefore, maximum bending moment is taken as

$$M_c = \frac{Fi \times x}{6} \quad (2)$$

where x = Distance between the bolt centres
x = Dia. of crank pin or big end bearing + 2 × Thickness of bearing liner + Nominal dia. of bolt + Clearance
= (dc + 2 × 3 + db + 3) mm = 55 + 6 + 12 + 3 = 76 mm
Maximum bending moment acting on the cap,

$$M_c = \frac{9490 \times 76}{6}$$

$$M_c = 120206.66\ N-mm$$

Let us now check the design for the induced bending stress due to inertia bending forces on the connecting rod (*i.e.* whipping stress).
We know that mass of the connecting rod per metre length,
m_1 = Volume × density = Area × length × density
= $A \times l \times \rho = 11t^2 \times l \times \rho$ ...($A = 11t^2$)

$= 11(0.007)^2 (0.254) *7850 = 1.0742$ kg
Maximum bending moment,

$$Mmax = m \times w^2 \times r \times \frac{l}{9\sqrt{3}} \qquad m = m_1 \times l$$

$$Mmax = 1.0742 \times w^2 \times r \times \frac{l}{9\sqrt{3}}$$

$$Mmax = 150.79\ N - m$$

Section Modulus

$$Zxx = \frac{Ixx}{\frac{5t}{2}}$$

$$Zxx = \frac{419t^4}{\frac{5t}{2}}$$

$$Zxx = 4792mm3$$

Maximum bending stress (induced) due to inertia bending forces or whipping stress,

$$\sigma_b = \frac{Mmax}{Zxx}$$

$$\sigma_b = \frac{150.9 \times 10^3}{4792}$$

$$\sigma_b = 31.48\ N/mm2$$

TABLE I
MATERIAL PROPERTIES OF CONNECTING ROD

Parameters	Unit	Structural Steel
Modulus of Elasticity	MPa	200×10^3
Poisson's Ratio	--	0.3
Tensile Yield Strength	MPa	250
Tensile Ultimate Strength	MPa	460
Density	Kg/m^3	7850

III. DESIGNING OF CONNECTING ROD

The Connecting Rod is designed by giving the dimensions into the modeling software PRO-E. The geometry of the Connecting Rod is designed in PRO-E is imported to the analysis software in the IGES format. The figure of the designed Connecting Rod is below Fig.

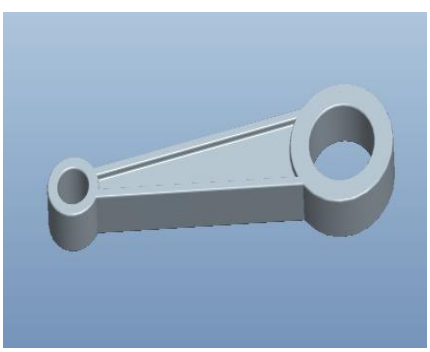

Figure3. Design of Connecting Rod

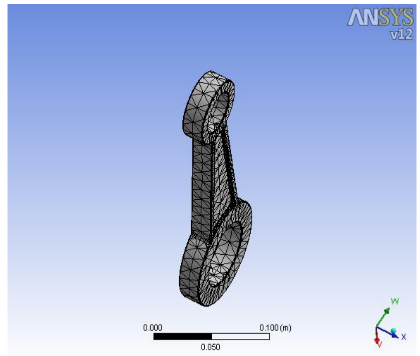

Figure 4. Meshed Model of Connecting Rod

IV. ANALYSIS OF CONNECTING ROD

FEM analysis of a connecting rod is done in ansys workbench 12.0 software first connecting rod model is imported to ansys by converting the PRO E file into .igs extension file format after successful import of model material property is defined. After applying $1.4844*10^5$N force the total deformation and bending stress is calculated and compare these results with analytical calculations. Figure 5 shows the stress and figure 6 shows the total deformation.

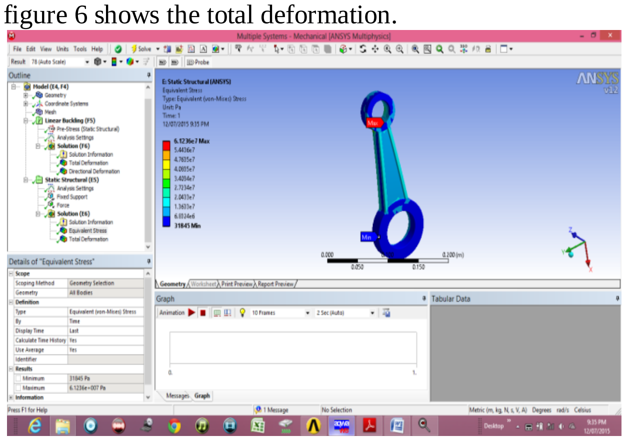

Figure5. Stress Analysis

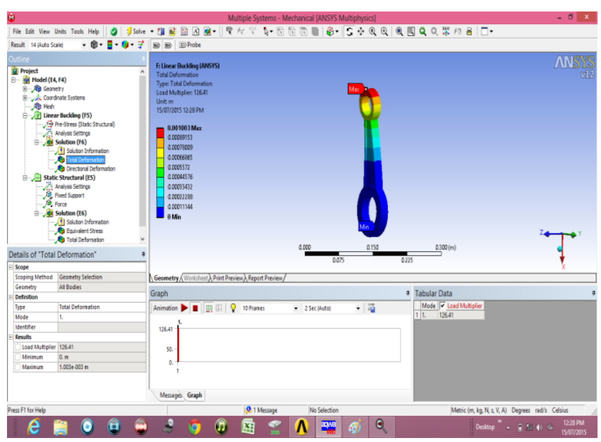

Figure6. Total Deformation

V. Comparison Of Analytical And Fe Analysis Result

TABLE II
RESULTS FOR DEFORMATION

Analysis	Analytically Calculated deformation (mm)	Deformation by Fem Analysis (mm)	Error
Circular bar	0.0006	0.0003	0.5
Rectangular bar	0.0005	0.0001	0.8
Tapered circular bar	0.015	0.015	0.0
Tapered rectangular bar	0.3	0.28	0.06

TABLE III
RESULTS OF STRESS ANALYSIS

Analysis	Analytically calculated stress(Mpa)	Stress by FEM Analysis (Mpa)	Error
Stress Analysis	31.84	31.48	0.01

VI. Conclusion

CAD model of the connecting rod is generated in PRO E and this model is imported to ANSYS for processing work. Following are the conclusions from the results obtained:

1) In present work analytical result compare with numerical result among all load conditions the minimum stress among all loading conditions was found at crank end cap as well as at piston end.
2) Buckling analysis of tapered circular and rectangular rod is to be performed and results obtained from analytical and finite element method are similar so it is concluded that the approach is correct for analyzing the buckling analysis of connecting rod and results obtained are also similar.
3) In this analysis there is possibility of further reduction in mass of connecting rod. For further work the thermal stresses are developed on different parts of connecting rod during dynamic conditions.
4) From the above result of comparison we conclude that analytical and FE analysis results for all types of bar and connecting rod are approximately similar. The percentage difference between analytical results & analysis results of connecting rod are very small.

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