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Analysis of Connecting Rod Using Analytical and Finite Element Method

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ABSTRACT: *The connecting rod is a major link inside of 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 materials used for connecting rods are steel and aluminum. Connecting rods are widely used in variety of engines such as, in-line engines, V-engine, opposed cylinder engines, radial engines and oppose-piston engines.*

For the project work we have selected connecting rod used in light commercial vehicle of tata motors had recently been launched in the market. We used PRO-E wildfire 4.0 software for modeling of connecting rod and ANSYS 11 software for analysis. ANSYS Workbench module had been used for analysis of connecting rod. We found out 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. We have also designed the connecting rod by machine design approach. Design of connecting rod which is designed by machine design approach is compared with actual production drawing of connecting rod. We found that there is possibility of further reduction in mass of connecting rod.

Keywords: *Analysis, Connecting rod, Machine Design*

I. INTRODUCTION

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. 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. As the lower end of the connecting rod revolves with the crankshaft, the upper End is forced to turn back and forth on the piston pin. Although this movement is slight hence the bushing is necessary because of the high pressure and temperatures.

The lower hole in the connecting rod is split to permit it to be clamped around the crankshaft. The bottom part, or cap, is made of the same material as the rod and is attached by two bolts. The surface that bears on the crankshaft is generally a bearing material in the form of a separate split shell. The two parts of the bearing are positioned in the rod and cap by dowel pins, projections, or short brass screws. Split bearings may be of the precision or semi precision type. From the viewpoint of functionality, connecting rods must have the highest possible rigidity at the lowest weight.

The function of connecting rod is to transmit the thrust of the piston to the crankshaft. Figure1.2 shows the role of connecting rod in the conversion of reciprocating motion into rotary motion. A four-stroke engine is the most common type. The four strokes are intake, compression, power, and exhaust. Each stroke requires approximately 180 degrees of crankshaft rotation, so the complete cycle would take 720 degrees. Each stroke plays a very important role in the combustion process. In the intake cycle, while the piston moves downward, one of the valves open. This creates a vacuum, and an air-fuel mixture is sucked into the chamber (Figure 1 (a)). During the second stroke compression occurs. In compression both valves are closed, and the piston moves upward and thus creates a pressure on the piston, see Figure 1 (b). The next stroke is power. During this process the compressed air-fuel mixture is ignited with a spark, causing a tremendous pressure as the fuel burns. The forces exerted by piston transmitted through the connecting rod moves the crankshaft, see Figure1(c). Finally, the exhaust stroke occurs. In this stroke, the exhaust valve opens, as the piston moves back upwards, it forces all the air out of the chamber and thus which completes the cycle of crankshaft rotation Figure 1(d).

Fig 1 (a) Intake stroke

Fig 1 (b) compression stroke

Fig 1 (c) Power stroke

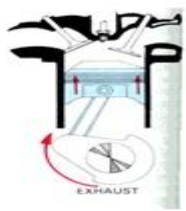


Fig 1.2(d) Exhaust stroke

II. ANALYTICAL DESIGN OF CONNECTING ROD

Piston Diameter =112m
Piston Weight =1.450 kg
=22.12 N
Weight of connecting rod = 2.731 kg
Weight of reciprocating parts = Piston Weight + 0.33*Weight of connecting rod
=1.450 + 0.33*2.731
=2.351 kg
= 2.351*9.81 N
=23.06 N
Piston Material = C-70
Stroke =112x127
Engine RPM = 1200-1700 rpm with maximum over speed of 2200 rpm
Compression Ratio of Engine = 17:1
Maximum explosion pressure = 2.63 MPa
Length of connecting rod = 214 mm

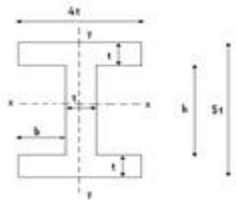
Section of rod : In I.C. engines, the most widely used section of rod is I due to its easiness as it will keep the inertia forces small and it can withstand the high gas pressure also.

In the plane of motion, the ends of the rod are direction free and so freely hinged at the piston pin and the crank pin. Hence for buckling about neutral axis, the strut is freely hinged (figure 5.1). in the plane perpendicular to plane of motion, for buckling about axis yy, the strut is is fixed ended due to the effect of bearings at piston and crankpins. Therefore for buckling about axis yy, the rod is four times as strong as for buckling about axis xx. But the rod should be equally strong in both the planes.

4Iyy = Ixx
kyy2 = (1/4)kxx2
let the flange and web thickness of section = t
depth of section = 5t
width of section = 4t
area of section =11t2
Ixx= (BH3 – bh3)/12
= [4t*(5t)3 – 3t*(3t)3]/12
= 410t4/12
Iyy = (2tB3 - ht3)/12
= [2t*(4t)3 + 3t*(t)3]/12
= 131t4/12
Kxx2 = Ixx/ A
= [419t4] / [12*11t2]
= 3.18t2
Kyy2 = Iyy/A
= [131 t4] / [12*11t2]
= 0.995t2
Kyy2/kxx2 = [0.995t2] / [3.18 t2] = 1/3.2
Kxx2=3.2*kyy2
By Rankine formula

$$\text{Buckling load} = \frac{Fcu*A}{1+ a*(l/k)2}$$

Buckling load = Maximum explosion load* factor of safety
= (3.14/4)*0.1122*2.63*106*6
(taking factor of safety = 6)
= 155465 N



Fcu = 870MPa
A = 11t2
a = 1/7500
l = 0.214 m
k = kx

$$155465 = \frac{870*106*11t2}{1+ (1/7500)*(0.214*(3.18\ t2))}$$

t = 4.6* 10-3 m
t = 5 mm
Depth = 5t = 25mm
Width = 4t =20mm
Section at crank end = 32*20
Section at pin end = 22*20.

5.2.2 Dimensions of big end
P = lc*dc*Pb
Where
P = maximum gas load
lc = length of crank pin (mm)
dc = diameter of crank pin(mm)
Pb = allowable bearing pressure= ranges from 8MPa 15 MPa
P = (3.14/4)*0.1122 *2.63*103
= 26 kN
= 26000 N
But lc/dc= 1.25 to 1.5
Taking lc/dc=1.25
26000 = 1.25*dc*dc*15
dc = 50mm
lc = 1.25*dc
=1.25*50
=62.5mm
Diameter of crank pin=50mm
Length of crank pin =62.5mm
Empirically dc should be = (0.5 to 0.65)*bore
and lc should be =(0.4 to 0.6)*bore

5.2.3 Design of small end
P = lp*dp*Pb
Where
P = maximum gas load
lc =length of piston pin (mm)
dc = diameter of piston pin(mm)
Pb = allowable bearing pressure
Pb = usually ranges from 15 to 31.5 MPa
But lp/dp=1.5 to 2
Taking lp/dp=1.5
26000 =1.5*dp*dp*15

dp= 34 mm
lp = 52.5 mm

5.2.4 Bolts for Big End Cap

Inertia force is

Fi = (F/g)*(ω2)*r[cosθ + (r/l)cos2θ]

Where

F = Weight of reciprocating parts
ω =angular velocity
r = radius of
l = length of

ω = (2*3.14*N)/60
= (2*3.14*2250)/60
= 236 rad/s

θ =00
F=23.6 N
l=0.214m
r=0.0635m

Fi = (F/g)*(ω2)*r[cosθ + (r/l)cos2θ]
=10780 N

If 2 bolts are used and core diameter is dc

2*(3.14/4)*dc2 *ft=10780N

(ft = 120 MPa = allowable stress ranges from 120 to 175MPa)

dc= 8 mm

Outer diameter =8/0.8
=10mm

Connecting rod big end Bearing cap

Maximum bending moment may be taken as
= (Fi*s)/6

Where

Fi = load on cap i.e. the inertia load
s = distance between bolt centers
= diameter of bearing + twice the thickness of bearing liner +diameter of
Bolt + some clearance

Thickness of bearing liner = thickness of shell + thickness of bearing metal

Thickness of bearing metal shell may be taken empirically as
=0.05*cylinder bore
=0.05*112
=5.6mm

Thickness of bearing metal may be taken as 1mm

s = 50+2*(5.6+1) + 10+ say 1.5mm
=75mm

Bending moment
= (Fi*s)/6
= 10780*75/6
= 134.750 Nm

Now

Mb= fb*Z

fb may be taken as 120 MPa

Z = $(b*c^2)/6$

Where

b= width of cap
= length of bearing (mm)
= 50mm
c = thickness of cap (mm)

134750=120*$(50 *c^2)/6$
c =11mm

5.2.6 Transverse Inertia Bending Stress

Centripetal force per unit of connecting rod, acting at crankpin,

C= ρ A ω2 r, N/m

Now

ρ for C-70 =7848kg/m^3

A= $11t^2$
= 11*(0.005)2
= 2.75*10-4 mm^2
= (2*3.14*N)/60
= (2*3.14*2250)/60
= 236rad/s

C=7848*2.75*10-4*2362*0.0635
= 7632.9N/m

Now

Maximum B.M. =0.128*Fn*l

Now

Fn = 0.5*C*l

Maximum B.M.
= (0.128*C*l2)/2
= 0.128 * 7633*0.2142/2
= 22.37 Nm

Now

fb=M/Z
Z=Ixx/(depth/2)
Ixx = (419*t4)/12
= (419*0.0054)/12
= 2.18*10-8m4
Z = 2.18*10-8/(0.05/2)
= 8.72*10-7
fb =22.33/(8.72*10-7)
=25.60 Mpa

III. FINITE ELEMENT ANALYSIS OF CONNECTING ROD

In Figure 2 shown that solid model of Connecting Rod.

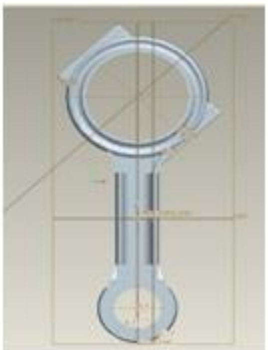


Fig 2 Solid Model of Connecting rod

Meshing of Connecting rod: - In Figure 3 shown that meshed model of connecting rod.

Fig 3.Meshed model of Connecting Rod

IV. LOAD DISTRIBUTION ON CONNECTING ROD

Tension loading

In tension, the connecting rod experiences cosine distribution over 180 of the contact area. The pressure is acting on the contact surface area of the connecting rod. The normal pressure (po) was calculated from the following equations:

$P = Po \cos \theta$

$Po = (2Pt) / (rt\Pi)$

Where,

θ = Crank angle, 0 degree for top dead center position

r = Radius of crank or pin end

t = Thickness of the connecting rod at the loading surface

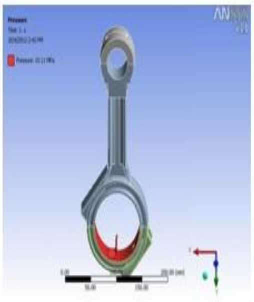
Pt = Force magnitude in tension

Table 1 loading data

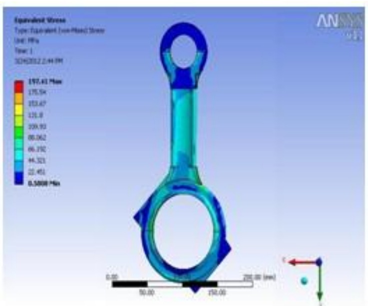
Parameter	Crank end loading		Pin end loading	
	Tension	Compression	Tension	Compression
Load magnitude	26000N	26000N	26000N	26000N
Load Distribution	Cosine Distribution1800	Uniform Distribution 1200	Cosine Distribution 1800	Uniform Distribution 1200
Pressure Function	Po=2Pt/Rct π	Po=Pc/Rct √3	Po=2Pt/Rptπ	PO=Pc/Rpt √3
Pressure On the Surface	10.11MPa	20.67MPa	22.24MPa	20.17MPa

A Tension loading at Big End (Crank end)

Fixed supports

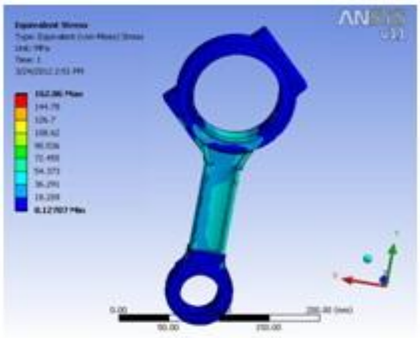


Application of Pressure



Equivalent stresses developed

Compression loading:-



Equivalent stresses developed

Conclusion and future scope

Solid modeling of connecting rod was made in Pro-E according to production drawing specification and analysis under the effect of tensile and compressive loads in terms of pressure is done in ANSYS Workbench. In present work analytical result compare with numerical result among all load conditions the maximum value of equivalent stress was found to be 197.41 MPa when crank end of connecting rod is in tension. This stress is less than yield strength of material. It gives a factor of safety of 3.2.So the existing design is oversafe but It is consider for only static load condition.

From analysis it is observed that the minimum stresses among all loading conditions, were found at crank end cap as well as at piston end. So the material can be reduced from those portions, thereby reducing material cost. For further optimization of material dynamic analysis of connecting rod is needed. After considering dynamic load conditions once again finite element analysis will have to be performed. It will give more accurate results than existing

REFERENCES

[1] FEM analysis of connecting rod by R.Vozenilek,C.Scholz (The Technical University of Liberec,Halkova)

[2] Dynamic Load Analysis and Fatigue Behavior of Forged Steel vs. Powder Metal Connecting Rods by Adila Afzal and Pravardhan Shenoy (The University of Toledo 2003)

[3] Moon Kyu Lee ,Hyungyil "Buckling sensitivity of connecting rod to the shank sectional area reduction original research article" Material and Design vol 31 Issue 6 Page 2796-2803

[4] Saharash khare,O.P.Singh "Spalling investigation of connecting rod original research article" Engineering Failure Analysis Vol 19,Jan 2012 page 77-86

[5] S.griza,F.Bertoni,G.zanon,A.Reguly "fatigue in engine connecting rod bolt due to forming laps original research article" Engineering failure Analysis Vol 16,issue 5 july 2009,page 1542-1548.

[6] R.J.yang, D.L Dewhirst "Shape optimization of connecting rod pin end using a generic model original research article" Finite element in analysis and design vol 11,issue 3 july 1992,page 257-264.