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Size and Shape Optimization of a Two Wheeler Connecting Rod by Structural Analysis

Lingaraj. K. Ritti [1]
lingaraj.k.ritti@gmail.com

Pavan Kumar [2]
pavansmvitm@gmail.com

Ambarish M [3]
ambarish.mpm@gmail.com

Assistant Professors,
Department of Mechanical
Engineering, Shri Madhwa Vadiraja
Institute of Technology,
Bantakal-574115, Udupi, India

Abstract— **Connecting rod is the intermediate link between the piston and the crank and is responsible to transmit the push and pull from the piston pin to crank pin, thus converting the reciprocating motion of the piston to rotary motion of the crank. Connecting rod in automotives should be lighter and lighter, should consume less fuel. In the case of four stroke engines, during compression and power strokes the connecting rod is subjected to compressive loads and during the last part of the exhaust and the beginning of the suction strokes, to tensile loads. Since the compressive forces are much higher than the tensile force, therefore the cross-section of the connecting rod is analyzed and optimized based on the compressive load. In this work two wheeler connecting rod is selected, solid edge software is used for the modelling and ANSYS software is used for the analysis to find the stresses developed in connecting rod under static loading with compression at pin end and constraints at the crank end and these stress values are used to perform shape optimization. Size optimization is performed by reducing the web thickness of the connecting rod to optimize the size and the further analysis is carried out to find stress distribution, based on that shape optimization is done by considering shape finder as a design variable, maximum stress as a constraint and mass reduction as an objective of the connecting rod.**

Index terms— Connecting rod, Finite Element Method, Size and Shape Optimization, ANSYS.

I. INTRODUCTION

Connecting rod is the intermediate link between the piston and the crank. And is responsible to transmit the push and pull from the piston pin to crank pin, thus converting the reciprocating motion of the piston to rotary motion of the crank. Connecting rod, automotives should be lighter and lighter, should consume less fuel and at the same time they should provide comfort and safety to passengers, that unfortunately leads to increase in weight of the vehicle. This tendency in vehicle construction led the invention and implementation of quite new materials which are light and meet design requirements

In the case of four stroke engines, during compression and power strokes the connecting rod is subjected to compressive loads and during the last part of the exhaust and the beginning of the suction strokes, to tensile loads. Since the compressive forces are much higher than the tensile force, therefore the cross-section of the connecting rod is analyzed and optimized based on the compressive load.

Design optimization refers to the process of generating improved designs. That means a design that possesses some optimal characteristics, such as minimum weight, maximum first natural frequency. It is performed by an optimizer. An optimizer is nothing but a formal plan or algorithm that is used to search for a best design

Engineering applications include structural design, chemical process design, aerodynamic optimization, nonlinear control system design, mechanical component design, multi-discipline system design, and a variety of others. Because the statement of the numerical optimization problem is so close to the traditional statement of engineering design problems, the design tasks to which it can be applied are inexhaustible

Size optimization involves a modification of the cross-section or thickness of finite elements. The optimization is carried out by mathematical optimization algorithms with different objective functions e. g. maximum stiffness or minimum weight. Many programming approaches were tested and implemented in finite element programs or special optimization programs. Due to the easy calculation of sensitivities for size optimization purposes even realistic problems can be handled. Today these approaches can be considered as state of the art.

The basic concept of shape optimization design is to place material in areas that truly need it and thin out unnecessary material from areas that are not important for correct function in order to obtain the minimum shape that satisfies all the necessary functional requirements, such as mechanical strength and rigidity. Spurred by the lightening needs mentioned above, recently the demand for the ability to determine optimal shapes easily has been mounting. However, currently available software packages are not powerful enough to fulfill this demand.

II. DETERMINATION OF FORCES ACTING ON THE CONNECTING ROD.

A connecting rod is a machine member which is subjected to alternating direct compressive and tensile forces. In the case of four stroke engines, during compression and power strokes the connecting rod is subjected to compressive loads and during the last part of the exhaust and the beginning of the suction strokes, to tensile loads. Since the compressive forces are much higher than the tensile force, therefore the cross-section of the connecting rod is analyzed and optimized based on the compressive load.

Specifications of Connecting rod
Diameter at the crank end, di_c = 30.1mm
Diameter at the piston end, di_p = 14mm
Thickness at the crank end, t_1 = 4.4mm
Thickness at the crank end, t_2 = 2.5mm
Length of connecting rod, c = 92.5mm
Thickness of I – section = 3mm
Piston and cylinder
Diameter of piston, D = 50mm
Diameter Bore = 52mm

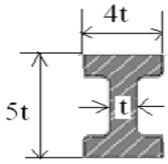

Fig 1: Standard I section of the connecting rod.

Properties of forged steel
Young's modulus, E = 221 GPa
Density, ρ = 7750 kg/m^3
Yield strength, y_c= 625 MPa
Ultimate strength, y_u= 827 MPa
Weight of connecting rod = 0.117 kg
Weight of piston = 0.068 kg
Weight of gudgeon pin = 0.02

Pressure Calculation for 100cc Engine
Density of Petrol C_8H_{18} = 737.22kg/m3

Mass of the petrol M = $737.22 \times 10^{-9} \times 97.5 \times 10^3$
= 0.072 kg
From gas equation, PV = MRT

$$R = \frac{8.3143}{0.114228} = 72.786$$

$$P = \frac{0.072 \times 72.786 \times 288.85}{97.5 \times 10^{\wedge}3} = 15.5 \text{ MPa}$$

Force Due to Gas Pressure:

$$F_g = \frac{\pi d^{\wedge}2}{4} \text{ x } P_f = \frac{\pi \text{ x } 50^{\wedge}2}{4} \text{ x } 15.45 = 30336.004 \text{ N}$$

Inertia Load at Θ = 0

$$F_I = \frac{0.01095 \text{ Wrn } 2}{g} x \left(1 + \frac{1}{n'}\right)$$

Where,
W = Weight of reciprocating parts = Piston weight + (0.33 x weight of connecting rod) = 0.088 + (0.33 x 0.117) = 0.1242 N
Crank radius, r = 24.83 mm
Max. Speed of engine, n = 8000 rpm

$$n' = \frac{l}{r} = \frac{92.5}{24.83} = 3.725$$

g = 9810 mm/sec^2

$$F_I = \frac{0.01095\ x\ 1.242 x\ 24.83\ x\ 8000^{\wedge}\mathbf{2}}{9810} x \left(1 + \frac{1}{3.725}\right)$$

F_I = 2794.47 N

Combined Load or Net Load acting on the connecting rod:
$F = F_g \pm F_I$
'-'ve → acceleration
'+'ve → retardation
Net load, F =F_g - F_I = 30336.004–2794.47= 27541.534 N
Hence, the compressive load acting along the connecting is equal to 27.5 kN

Inertia of the Connecting Rod:
Inertia/unit length
$F = \rho A\omega^2 r\sin\alpha$
When $\alpha = 90^o$, $F = F_{max}$
$F_{max} = \rho A\omega^2 r$
Where,
ρ = 7750 kg/m^3
A = 68.4×10^{-6} m^2

$$\omega = \frac{2\pi N}{60} = \frac{2\pi\ x\ 8000}{60} = 837.76 \text{ rad/sec}$$

$$r = \frac{Stroke\ lengt\ h}{2} = 24.83 \text{ mm} = 24.83 \times 10^{-3} \text{ m}$$

l = 92.5 mm = 92.5×10^{-3} m
F_{max} = $7750 \times 68.4 \times 10^{-6} \times (837.76)^2 \times 24.83 \times 10^{-3}$
F_{max} = 9237.91 N/m

$$F_{ic} = \frac{1}{2} F_{max} \text{ x l} = \frac{1}{2} \text{x } 9237.91 \text{ x } 92.5 \text{ x } 10^{-3} = 427.25 \text{ N}$$

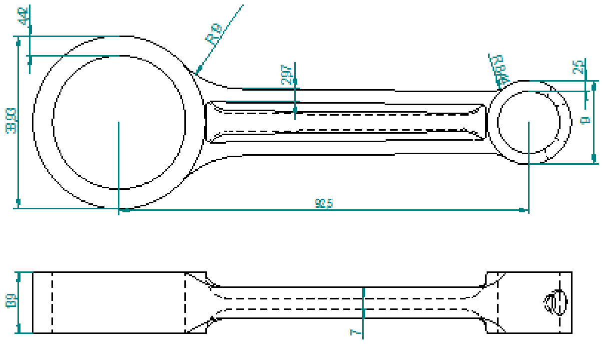

Fig .2: two wheeler connecting rod

III STATIC STRESS ANALYSIS OF CONNECTING ROD BY FINITE ELEMENT METHOD

The finite element analysis is carried out on forged steel connecting rod. A 3-D model of connecting is used for analysis in ANSYS. The loading conditions are assumed to be static. The pressure load is applied at the piston end .The crank end of the connecting rod is subjected to pure rotation therefore all the translations are fixed.

The element chosen is tetrahedral element with 3 degrees of freedom at each node. The meshed model of connecting rod is as shown in figure 3.

The following material properties are defined for the analysis of forged steel connecting rod Young's modulus, E= 221 GPa Poisson's ratio= 0.3, Density = 7750 kg/m^3.

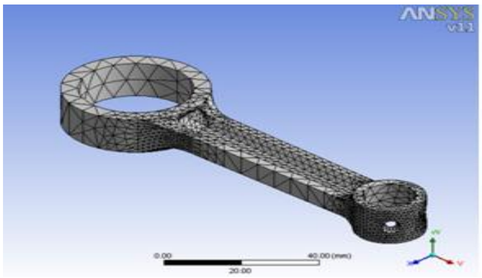

Fig 3: Meshed model of connecting rod

The boundary conditions are applied on the crank end of the connecting rod is shown in figure 4.The big end of the connecting rod is subjected to pure rotation therefore the big end of the connecting rod is fixed. (Translations along x, y and z directions are constrained).

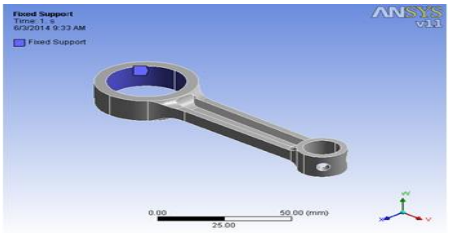

Fig 4 Fixed crank end of the connecting rod.

The figure 5 shows the pressure load applied on the connecting rod at the piston end. The Compressive Load 27.54kN is applied at the piston end of the connecting rod as pressure load.

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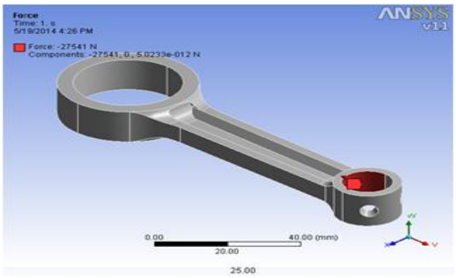


Fig 5: Loaded piston end of the connecting rod

IV. SIZE AND SHAPE OPTIMIZTION OF CONNECTING ROD

The objective of the optimization is to minimize the mass of the connecting rod under the effect of peak compressive gas load at 8000 rev/min (at 360° crank angle), such that the maximum, minimum, and the equivalent stress amplitude are within the limits of the allowable stresses. The production cost of the connecting rod is also minimized

A. Size optimization of a existing connecting rod

Size optimization is done based on the stress analysis of the existing connecting rod, and it is performed by reducing the web thickness of the connecting rod from 3mm to 2.5mm and once again the same stress analysis is carried out. The stress distribution along the shank of the connecting rod is increased as shown in figure 8. The stress distribution along critical section of the connecting rod is from 308 MPa to 387 MPa. The following figures show the displacements, strains and stress distribution along the connecting rod.

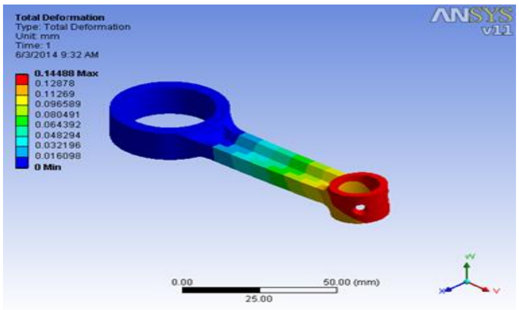


Fig 6: Deformation plot of the connecting rod having web thickness 2.5mm

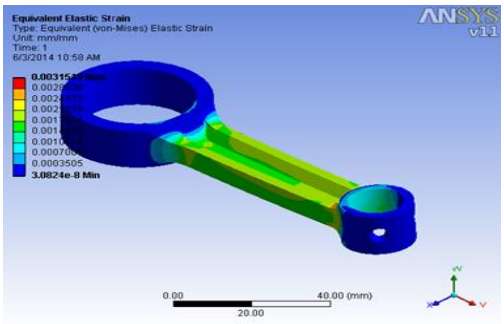


Fig 7: Strain plot of the connecting rod having web thickness 2.5mm

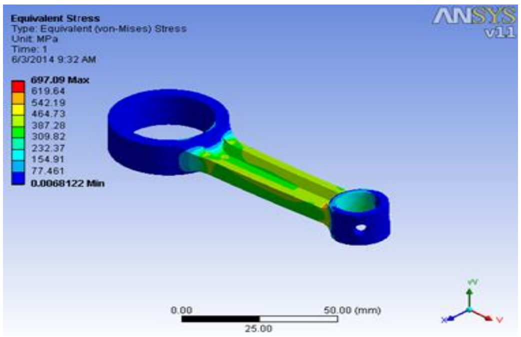


Fig 8: Von Misses stress distribution of the connecting rod having web thickness 2.5mm

B. Shape Optimization of a existing Connecting Rod

Objective of the shape optimization is to minimize the mass of the connecting rod (I-section). The weight of optimized connecting rod is certainly lower than the weight of original connecting rod. The following factors are considered during the optimization- load factor, stresses under the loads.

Objective: Minimize Mass and Cost
Subject to:
a) Compressive load = peak compressive gas load.
b) Maximum stress < Allowable stress
c) Constraints (Component dimensions)

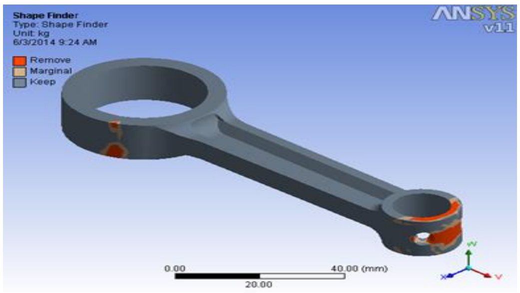


Fig 9: Shape optimization of existing connecting rod.

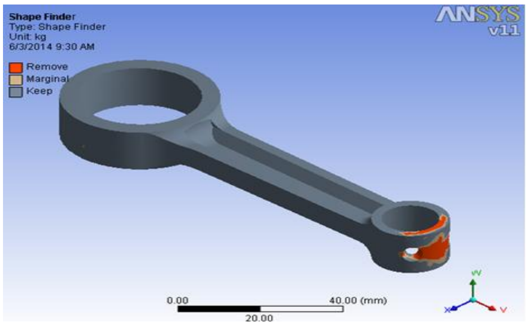


Fig 10 Shape optimization of connecting rod having web thickness 2.5mm

V. RESULTS AND DISCUSSIONS

The deformation strain and von misses stress distribution obtained by finite element method. The deformation is more at the piston end of the connecting rod and strain, stress value is maximum at the fillets of the crank end. The stress distribution along the shank of the connecting rod is 353.83 MPa and maximum stress is 778.4 MPa

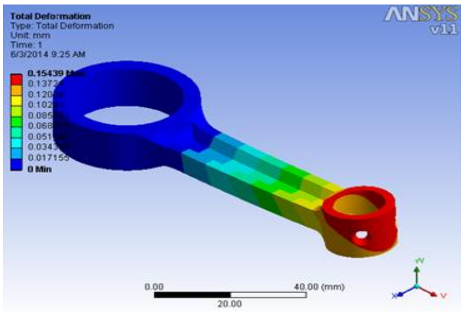


Fig 11: Deformation plot of the connecting rod

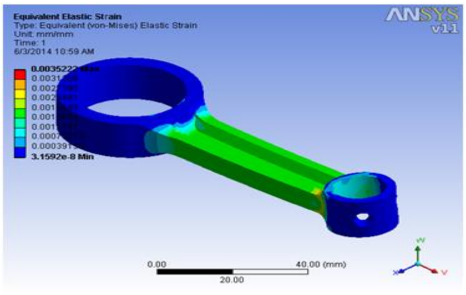
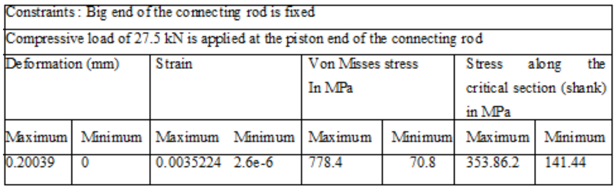


Fig 12: Strain plot of the connecting rod

Comparison of FEA results for the existing connecting rod against the allowable stresses indicate that the shank region of the connecting rod offers the greatest potential for weight reduction. Regions near fillets of the big end are already highly stressed.

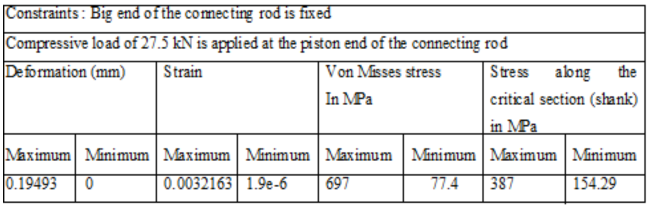
Table 1 Stress analysis of existing connecting rod (web thickness 3mm)



Constraints : Big end of the connecting rod is fixed							
Compressive load of 27.5 kN is applied at the piston end of the connecting rod							
Deformation (mm)		Strain		Von Misses stress In MPa		Stress along the critical section (shank) in MPa	
Maximum	Minimum	Maximum	Minimum	Maximum	Minimum	Maximum	Minimum
0.20039	0	0.0035224	2.6e-6	778.4	70.8	353.86.2	141.44

In the shank region, the rib and the web thicknesses are reduced, however, only to a certain limit to maintain forge ability. The section modulus of the optimized connecting rod should be high enough to prevent high bending stresses. The stress results are shown in table 2 the stress distribution along the critical section is 150.29 MPa to 395 Mpa that is less than the allowable stress

Table 2 Stress analysis of connecting rod with web thickness 2.5mm



Constraints : Big end of the connecting rod is fixed							
Compressive load of 27.5 kN is applied at the piston end of the connecting rod							
Deformation (mm)		Strain		Von Misses stress In MPa		Stress along the critical section (shank) in MPa	
Maximum	Minimum	Maximum	Minimum	Maximum	Minimum	Maximum	Minimum
0.19493	0	0.0032163	1.9e-6	697	77.4	387	154.29

After several iterations, which involved determining the loads and performing FEA for the resulting geometry of each iteration step, an optimized geometry was obtained, shown in Figure 9 and 10. Mass of the optimized connecting rod is 0.10041kg which is lower than the mass of the original connecting rod by 4%. This geometry was found to satisfy the design constraints

Table 3: Shape optimization results of existing connecting rod (web thickness 3mm)

Optimization type	Original weight in kg	Optimize d weight in kg	Weight reduction in %
Shape finder	0.10649	0.10224	4

Table .4: Shape optimization results of connecting rod having web thickness 2.5mm

Optimization type	Original weight in kg	Optimized weight in kg	Weight reduction in %
Shape finder	0.1057	0.10137	4

IV. CONCLUSIONS

In the case of four stroke engines, during compression and power strokes the connecting rod is subjected to compressive loads and during the last part of the exhaust and the beginning of the suction strokes, to tensile loads. Since the compressive forces are much higher than the tensile force, therefore the cross-section of the connecting rod is analyzed and optimized based on the compressive load. The following conclusions are made

1) Based on the analytical method the compressive load due to gas pressure is 27.5 kN.
2) Based on the numerical analysis the stress distribution along the critical section of the connecting rod is from 141.6MPa to 424.2MPa
3) Based on size optimization analysis the stress distribution along the critical section of the connecting rod is from 154.9MPa to 464.73MPa and these stress are less than the allowable stress, therefore the web thickness of connecting rod is reduced from 3mm to 2.5mm.
4) Based on shape optimization the mass of the connecting rod is reduced by 4%

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