OPTIMIZATION OF STEEL CONNECTING ROD BY ALUMINUM CONNECTING ROD USING FINITE ELEMENT ANALYSIS

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***Abstract*-The connecting rod connects reciprocating piston to rotating crankshaft, transmitting the thrust of the piston to the crankshaft. In this research paper the connecting rod for Alloy Steel AISI 4340 and Aluminum 7068 Alloy are analytically design for the given configuration of the engine and the Finite Element model has been generated in Pro-E Wildfire 5.0 software. Then they were imported in ANSYS workbench 12 for finite element analysis, which is one of the most popular CAE tools. Also the weight and Von miss stresses for aluminum connecting rod have been compared with steel connecting rod and factor of safety under the effect of applied loads have been calculated. It has been shown that steel connecting rod optimization by aluminum connecting rod causes reduction of weight and maximum stress considerably.**

***Keywords*- connecting rod, finite element analysi, CAE tools, Von miss stresses**

I. INTRODUCTION

The function of connecting rod is to transmit the thrust of the piston to the crank shaft, and as the result the reciprocating motion of the piston is translated into rotational motion of the crank shaft. It consist of a pin – end. A shank section, and crank an end. One end of the connecting rod is connected to the piston by the piston pin. Connecting rods are subjected to forces generated by mass and fuel combustion .Theses two forces results in axial load and bending stresses. A connecting rod must be capable of transmitting axial tension, axial compression, and bending stress caused by the thrust and full of the piston and by centrifugal force. [1]

Connecting rods are highly dynamically loaded components used for transmission in combustion engines. The optimization of connecting rod had already started as early year 1983 by Webster and his team. Optimization of connecting rod is to make the less time to produce the product that is stronger, lighter and less cost. The design and weight of the connecting rod influence on car performance. Hence, it is effect on the car manufacture credibility. [5]

The objective of this study was to optimize a steel connecting rod by Aluminum connecting rod for its weight and manufacturing cost.

II. DESIGN OF CONNECTING ROD

Design of connecting rod from the Configuration of engine for pressure 37.3 bar [2]

TABLE 1 ENGINE SPECIFICATIONS

Particulars	Dimensions
Diameter of piston (D)	86 mm
Maximum explosion pressure (pmax)	37.3 bar
Crank shaft radius(r)	48.5 mm
Mass of piston assembly, m_2	0.434 kg
Connecting rod length	141 mm
Distance of C.G from crank center	36.4 mm
IZZ about the centre of gravity	0.00144 kgm_2
Maximum engine speed (Nmax)	5700 RPM

The connecting rod converts the reciprocating motion of the piston to oscillatory motion of itself which is finally converts to rotary motion of the crank shaft. The main parts of connecting rod are:

1. The small end which connect the connecting rod to the piston through the piston pin.
2. The shank, usually of I-section and
3. The big end which is usually split to surround crank pin.

The length of connecting rod is usually kept 3 to 4.5 times the crank radius. The shorter length of connecting rod increases obliquity and there by the side thrust on the cylinder whereas the longer length increases the height of the engine. Maximum gas force

We know that, the maximum gas force,

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$$P_{gas} = \frac{?}{4} \times (86)^2 \times 3.73$$

$$P_{gas} = \frac{?}{4} \times D^2 \times p_{\max}$$

$$P_{gas} = 21666.84N$$

Crippling load

Since a connecting rod is subjected to severe load conditions including fatigue load, a high factor of safety (5-6) is used while treating it as static strut. Thus according to Rankine formula,

$$F_{cr} = (?_{cr}\ A)/\left(1 + a(I / K_{xx})^2\right)$$

Where,

F_{cr} = Crippling load.

For crippling load,

$F_{cr} = P_{gas}$ × factor of safety

F_{cr} = 21666.84 × 5

F_{cr} = 108334.21 N

Dimension of small end

Considering bearing failure of the pin:-

$P_{br} = P_{gas} / (l_s \times d_{ps})$

Assuming allowable bearing pressure, $P_{br} = 15\ N/mm^2$,

$l_s / d_{ps} = 2$

$d_{ps} = (P_{gas} / 2_{pbr})\ 0.5$

$d_{ps} = (21666.84 / 2\times15)\ 0.5$

$d_{ps} = 26.87 \cong 27$ mm

Inner diameter of the small end,

$d_{si} = (1.1\text{-}1.25)\ d_{ps}$

$d_{si} = 1.18\times dps = 1.18\times27$

$d_{si} = 31.86 \cong 32$ mm

Outer diameter of the small end,

$d_{so} = (1.25\text{-}1.65)\ d_{ps}$

$= 1.45 \times d_{ps} = 1.45 \times 27$

$d_{so} = 39.15 \cong 40$ mm

Length of small end,

$l_s = (0.3\text{-}0.45)\ D = 0.380\times86 = 32.68 \cong 33$ mm

Dimension for Big end

Considering bearing failure of crank and assuming empirical relations:

Diameter of crank pin, $d_{pc} = (0.55\text{-}0.75)\ D$

$d_{pc} = 0.65\ D = 0.65\times86 = 55.9 \cong 56$ mm

Inner diameter of the big end,

$d_{bi} = (1.1\text{-}1.25)\ d_{pc}$

$d_{bi} = 1.18 d_{pc} = 1.18\times56 = 66.08 \cong 67$mm

Outer diameter of the big end,

$d_{bo} = (1.25\text{-}1.65) d_{pc}$

$= 1.45 d_{pc} = 1.45\times56 = 81.25 \cong 82$mm

Length of the big end:-

$l_c = (0.45\text{-}1.0)\ d_{pc}$

$l_c = 0.73 d_{pc} = 0.73\times56 = 40.88 \cong 41$mm

Bearing pressure (p_{br}),

$p_{br} = P_{gas} / l_c\ d_{pc}$

$p_{br} = 21666.84/41\times56 = 9.14\ N/mm^2$

Length of crank pin,

$l_c = 1.0 d_{pc} = 1.0\times56 = 56$ mm

This is reasonable for the crank pin bearing.

Thickness of bush,

$t_{bush} = (0.03\text{-}0.1)\ d_{pc}$

$= 0.065 \times 56 = 3.64$ mm

Cap of the big end:-

It is designed as a beam supported at the bolt centre. The cap may be assumed to be loaded at the centre by a concentrated load.

Bending moment, $M = (P_{inertia} \times c)/6$

Where c = Distance between the centres of bolts.

c = diameter of the crank pin+ 2 × thickness of bush

liner + Diameter of the bolt + clearance

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

= 0.05D + 2 mm = 0.05 × 86 + 2 = 6.3 mm

c = 56 + (2 × 6.3) + 8 + 6.3 = 82.9 ≅ 83 mm

Bending moment, M=(2403.40×83)/6 = 33247.03 N-mm

Cap is made of $37Mn^2$ having allowable strength, 90 N/mm^2

Section modules, $Z = 1/6 \times l_c\ t_c^2$

$Z = 1/6 \times 41\ t_c^2$

$Z = 6.83 t_c^2$

Bending stress, $?_b = M / Z \leq ?$ allowable

$$90 = 33247.03 / 6.83t_c^2$$

$$t_c = 7.35 \text{ mm}$$

a) For AISI4340 Alloy Steel

Thickness of connecting rod shank:-

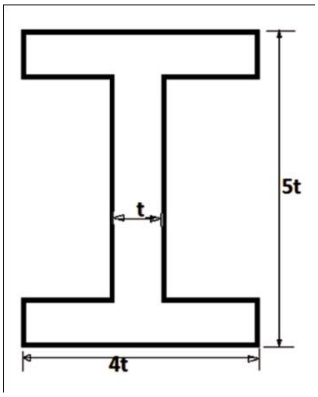


Fig. 1. I section of Connecting Rod Shank

The I-section is the most suitable for the Connecting rod, with the usual proportions as Shown in the adjoining figure.

Width of flange, B = 4t

Height of I-section, H = 5t

Where,

t = Web thickness

Radius of gyration (K_{xx}) is given by,

$K_{XX} = (I_{XX}/A)^{0.5}$

$A = 11t^2 \text{ mm}^2$

$K_{XX} = (419\, t^4/11\times12\times t^2)^{0.5}$

$K_{XX} = 1.78t$

Where,

I_{XX} = Mass moment of inertia

A = Area of I-section

Putting the value of K_{XX} in Rankine formula

$F_{cr} = (?_{cr} \times A) / (1+ a\, (l / K_{XX})^2)$

$108334.21=(470\times11t^2) / (1+ (1/6250) \times (141/1.78t))$

t = 4.68 . 4.7 mm

Height of I-section, H = 5 × 4.7 =23.5 mm

Width of flange, B = 4× 4.7=18.8 mm

Length near big end, H_1=1.2H=1.2×23.5 =28.2 mm

Length of pinion end, H_2=0.85H=0.85×23.5=19.975mm

Dimensions near big end =28.8×18.8

Dimensions near small end = 19.975× 18.8

b) For Aluminum 7068 Alloy

Thickness of connecting rod shank:-

Radius of gyration (K_{xx}) is given by,

$K_{XX} = (I_{XX}/A)$ 0.5

$A = 11t^2 \text{ mm}^2$

$K_{XX} = (419\, t^4/11\times12\times t^2)^{0.5}$

$K_{XX} = 1.78t$

Where,

I_{XX} = Mass moment of inertia

A = Area of I-section

Putting the value of K_{XX} in Rankine formula

$F_{cr} = (?_{cr} \times A) / (1+ a\, (l / K_{XX})^2)$

108334.21 = (540×11t2) /(1+ (1/6250) × (141/1.78t))

t = 4.38 . 4.4 mm

Height of I-section, H = 5 × 4.4 =22 mm

Width of flange, B = 4× 4.4=17.6 mm

Length near big end, H_1= 1.2H =1.2 ×22 =26.4 mm

Length of pinion end, H_2= 0.85H = 0.85 ×22 =18.7 mm

Dimensions near big end =26.4 × 17.6

Dimensions near small end = 18.7 × 17.6

For the analysis of I.C. Engine connecting rod the most critical area is considered and accordingly the two dimensional model of connecting rod is formed. The different

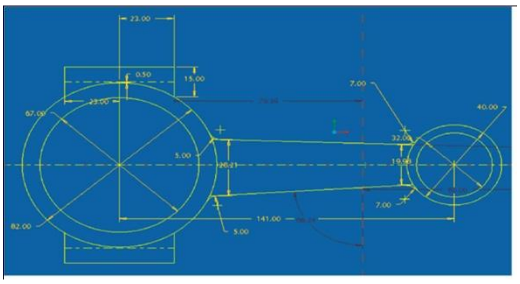


Figure 2. 2D drawing for Alloy Steel AISI 4340

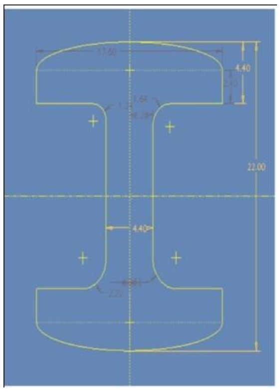


Figure 3. I-Section for Alloy Steel AISI 4340

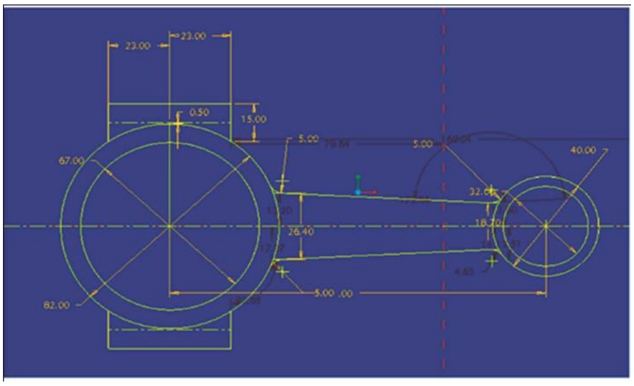


Figure 4. 2D drawing for Aluminum 7068 Alloy

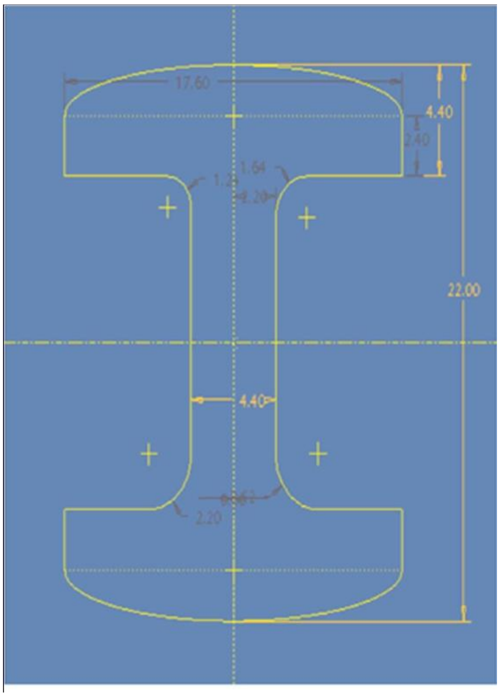


Figure 5. I-Section for Aluminum 7068 Alloy

Following tables show the properties of Steel and

Aluminum material used in this paper

TABLE 2 MATERIAL PROPERTIES OF AISI 4340

Density	7.85	g/cm^3
Modulus of elasticity E	200	GPa
Poisson's ratio	.28	
Tensile Strength	745	MPa
Yield strength	470	MPa

TABLE 2 MATERIAL PROPERTIES OF ALUMINUM 7068 ALLOY

Density	2.85	g/cm^3
Modulus of elasticity E	73.1	GPa
Poisson's ratio	.33	
Tensile Strength	641	MPa
Yield strength	540	MPa

III. FINITE ELEMENT ANALYSIS

A stress analysis is performed using finite element analysis (FEA). The complete procedure of analysis has been done using ANSYS Workbench 12.0. Pro-E modeling

The design connecting rod for AISI 4340 Alloy Steel and Aluminum 7068 Alloy are modeled in Pro-E Wildfire 5.0.

Figure 6.Pro-E model of Alloy Steel AISI 4340

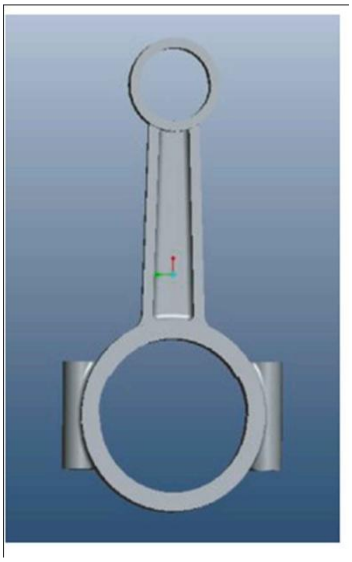


Figure 7..Pro-E model of Aluminum 7068 Alloy

Meshing

The connecting rod's model is made in Pro-E Wildfire 5.0 and the model is imported in ANSYS Workbench 12.0. The tetrahedral type meshing is used to mesh the models connecting rods with element size of 0.002mm.

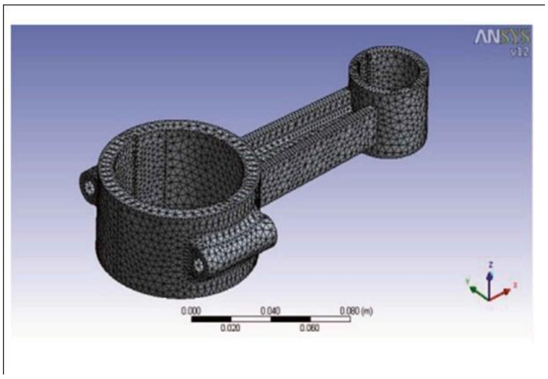


Figure 8.Meshing of AISI 4340

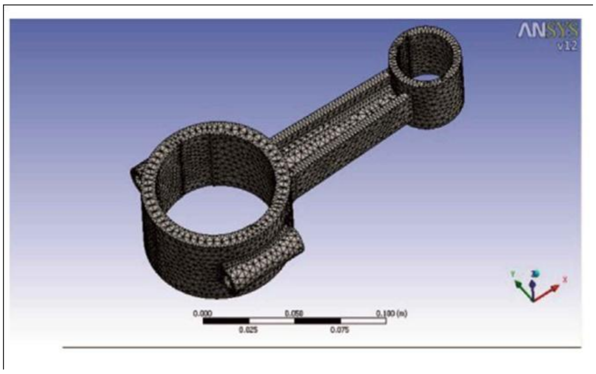


Figure 9.Meshing of Aluminum 7068 Alloy

IV. LOADING CONDITIONS AND CONSTRAINS

The loading conditions are assumed to be static. The applied load distributions were based on research by Webster et al. [2]

Two cases were analyzed for each case, one with load applied at the crank end and restrained at the piston pin end, and the other with load applied at the piston pin end and restrained at the crank end.

The tensile load was applied over 180o of crank contact surface with cosine distribution, whereas compressive load was applied as a uniformly distributed load over 120o of crank contact surface. [3]

For all practical purposes, the force in the connecting rod is taken equal to the maximum force on the piston due to pressure of gas (Pg), neglecting piston inertia effects.[4]

The maximum force due to pressure of gas is calculated as

$$F_{max} = 21666.84 \text{ N}$$

The pressure is acting on the contact surface area of the connecting rod. The normal pressure (po) was calculated from the following equations:

$$P = Po \cos \theta$$

$$po = Pt / (r\, t\, \pi / 2)$$

$$po = Pc / (r\, t \sqrt{3})$$

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

r = Radius of crank or pin end

t = Thickness of the connecting rod at the loading surface

Pt = Force magnitude in tension

Pc= Force magnitude in compression

Axial load for both tension and compression is Fmax

Compressive Loading:

Crank End: po = 21666.84/ (28 x 41 x √3) =10.89 MPa

Piston pin End: po = 21666.84/ (13.5 x 33 x √3) = 28.07MPa

Tensile Loading:

Crank End: po = 21666.84/ [28 x 41 x (π/2)] = 12.01MPa

Piston pin End: po = 21666.84/ [13.5 x 33 x (π/2)] =30.96 MPa

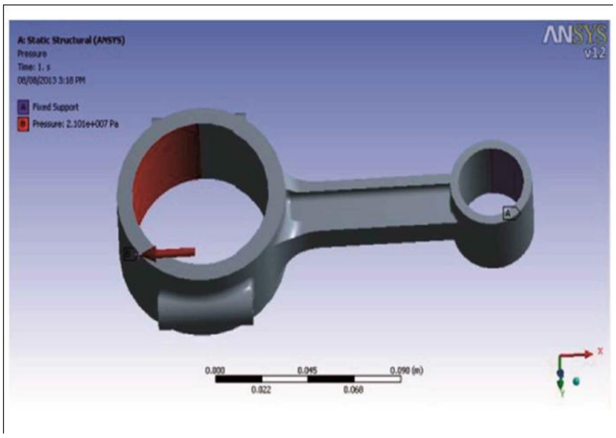


Figure 10. Tensile Load at Crank End for AISI 4340

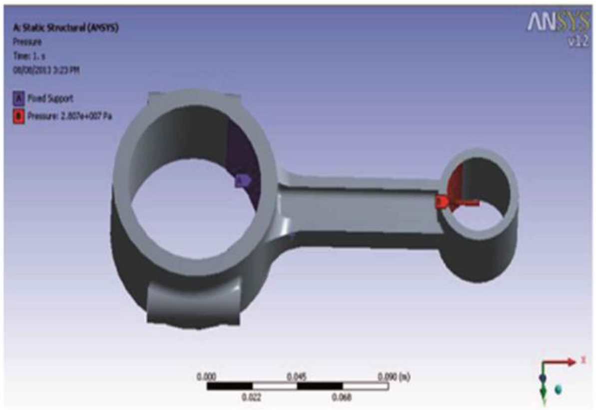


Figure 11. Compressive Load at Piston Pin End for AISI 4340

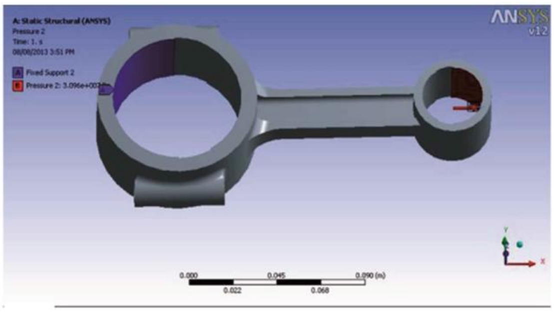


Figure 12.Tensile Load at Piston Pin End for Aluminum 7068 Alloy

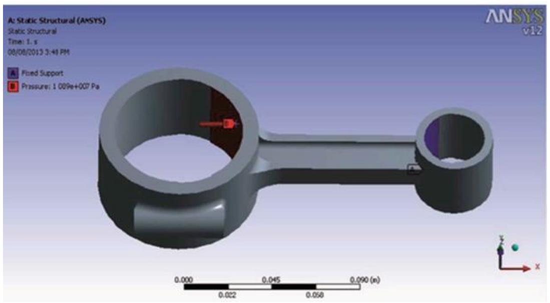


Figure 13.Compressive Load at Crank End for Aluminum 7068 Alloy

V. RESULT AND DISCUSSION

The Finite Element Analysis of both the connecting rod is done in ANSYS Workbench 12.0 considering all the loading conditions. Stress analysis has been done to calculate factor of safety.

Stress Results

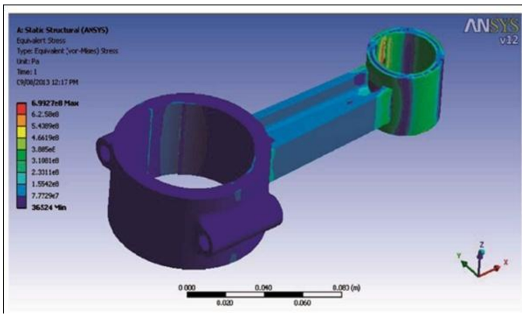


Figure 14. Stress analysis for Tensile load at Piston pin end for AISI4340 Steel

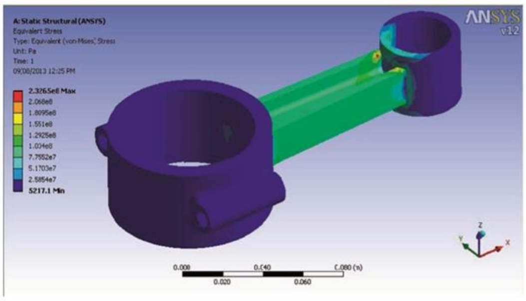


Figure 15. Stress analysis for Compressive load at Piston Pin End for AISI4340 Steel

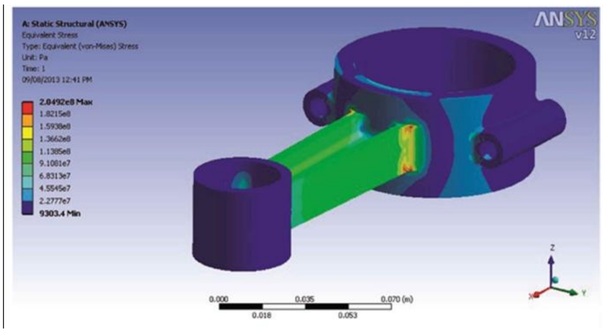


Figure 16.Stress analysis for Compressive load at Crank End for Aluminum 7068 Alloy

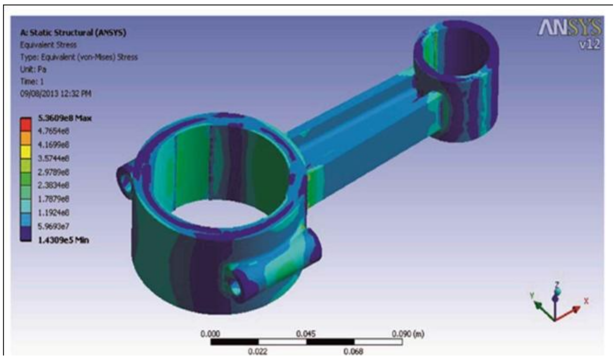


Figure 17. Stress analysis for Tensile load at Crank End for Aluminum 7068 Alloy

VI. COMPARISON OF WEIGHT, STRESS REDUCTION AND FACTOR OF SAFETY

Weight of AISI4340 Steel =Volume × Density

=1.185x10-4x7850

=.93025 Kg

Weight of Aluminum 7068 Alloy

=1.1766 x 10-4 x 2850

=.335331 Kg

% reduction in weight = 63.95%

For compressive loading at Crank End

Max equivalent stresses for AISI4340 Steel

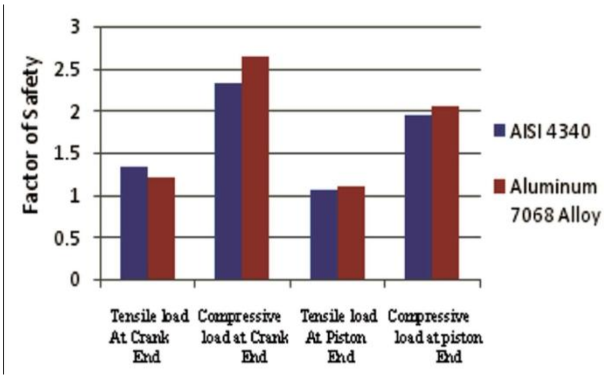
=556MPa

Max equivalent stress for Aluminum 7068 Alloy

=536MPa

% Reduction in stress = 3.59%

Following graph shows the comparison of factor of safety for AISI 4340alloy steel and aluminum 7068 Alloy for all loading conditions



VII. CONCLUSION

The stress analysis of connecting used in engine has been presented and discuss in this paper. The results obtain by FEA for both Aluminum 7068 alloy and AISI 4340 alloy steel are satisfactory for all possible loading conditions.

By using Aluminum 7068 alloy instead of AISI 4340 alloy steel can reduce weight up to 63.95%.

Also equivalent stresses for Aluminum 7068 alloy is less by 3.59%.

The factor of safety of connecting rod will reduce by 9.77% in case tensile load applied at crank end but it will increase in all other load conditions if Aluminum 7068 alloy is used instead of AISI 4340 alloy steel

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