# CAE ANALYSIS ON A FORD ECOBOOST MUSTANG CONNECTING ROD FOR FORGED STEEL, ALUMINUM 7075

by

SHIVA PRASAD KERALAPURA BASAVARAJU

Presented to the Faculty of the Graduate School of

The University of Texas at Arlington in Partial Fulfillment

of the Requirements

for the Degree of

MASTER OF SCIENCE IN MECHANICAL ENGINEERING

THE UNIVERSITY OF TEXAS AT ARLINGTON

December 2015

Acknowledgements

I would like to thank Dr. Dereje Agonafer for complete support and guidance which was very valuable during completion of thesis.

I would like to thank my mentor Mark H Mouland at ROUSH for constant support each day by advising me how to successfully understand the topic and complete it on time.

I would like to thank Dr. Haji- Sheikh and Dr. Fahad Mirza for serving as thesis committee members.

I would like to thank Ms. Sally Thompson and Ms. Debi Barton for coordinating and scheduling my thesis defense dates and helping me in other educational matters.

I would like to thank my parents for their constant motivation and support. In my eyes my parents graduated and would like to dedicate my degree for them.

November 19, 2015

Abstract

CAE ANALYSIS ON A FORD ECOBOOST MUSTANG CONNECTING ROD

FOR FORGED STEEL, ALUMINUM 7075

Shiva Prasad Kerelapura Basavaraju, MS

The University of Texas at Arlington, 2015

Supervising Professor: Dereje Agonafer

Connecting rod is high volume production critical component in an automobile engine. Every Internal combustion engine uses a connecting rod. Connecting rod must have highest possible rigidity with lowest possible weight. The major stress developed in the connecting rod are axial stress, due to combustion chamber pressure, and bending stress, due to centrifugal effect. Connecting rod mainly fails due to fatigue; according to survey, about 90% of the time the failure is due to fatigue. CAE analysis on Ford Eco boost Mustang connecting rod is carried out to validate the life of the connecting rod for forged steel and Aluminium7075. CAE analysis is mainly aimed at the static structural case and fatigue life case using ANSYS workbench. Fatigue redesign is proposed wherever necessary. Results of CAE is compared between forged steel and Aluminium 7075 to conclude the best material to use for a Ford Eco Boost Mustang.

Table of contents

List of Illustrations

List of Tables

# Chapter 1

# INTRODUCTION

## 1.1 Piston-Connecting Rod Assembly

Connecting rod as the name says is a connecting member between piston and crank shaft. Material, such as structural steel (for high stress values), aluminum alloy (for lightness at the expenses of fatigue life and durability), titanium (for high stress and light weight at expense of affordability), and cast iron (for low stress application) are used [1]. The connecting rod package has to be custom tailored to the engine and the customer’s needs, says Kerry Novak of Crower [2]. The Small end of the connecting rod is connected to the piston end using a gudgeon pin/ wrist pin by press fit; big end is connected to the crank shaft using fasteners.

Stresses on the connecting rod are always high due to the combustion chamber pressure, inertia forces, which induces high value of stresses. According to Vegi [3] “failure of a connecting rod, usually called "throwing a rod" is one of the most common causes for catastrophic engine failure in cars, frequently putting the broken rod through the side of the crankcase and thereby rendering the engine irreparable; it can result from fatigue near a physical defect in the rod, lubrication failure in a bearing due to faulty maintenance or from failure of the rod bolts from a defect, improper tightening, or re-use of already used (stressed) bolts where not recommended”. However, failure of the connecting rod is not common since the big automobile companies try to keep very high factor of safety of 2 or 3 above.

Furthermore, most of the automobile companies provide 10 years or 100000 miles of warranty on engine and transmission; to provide this type of warranty, automobile companies should have the robust design and manufacture capability. By having all this factors in consideration, a lot of engines fail or cease due to failure of connecting rod assembly, which leaves the companies to consider that the connecting rod as a very high risk component. For example connecting rod failed for GM 2014 Chevy Malibu's, 2014 Buick Regal GS , 2014 Chevy Impala, 2014 Cadillac ATS and 2015 Porsche 911 GT3, which caused millions of dollars to be spent on recall to replace the whole engine and redesign the connecting rod [4]. While designing the connecting rod, Vegi [3] suggested that measures have to be taken to reduce the stresses in the connecting rod. Methods, like grinding the edges to give smooth surface and radius to prevent crack initiation shot peening method, are used which induces compressive surface stress to balance the weight of the connecting rod and piston assembly to reduce the bending stress due to centrifugal action. He suggest us to use high end equipment which zooms in the connecting rod to give minute invisible cracks, which lead to brittle fracture in the ductile material. Furthermore, he points out that torqueing the bolts, which connect the crank part of the connecting rod to design guide values, is very important and not to reuse the bolts, but instead to replace the bolts.

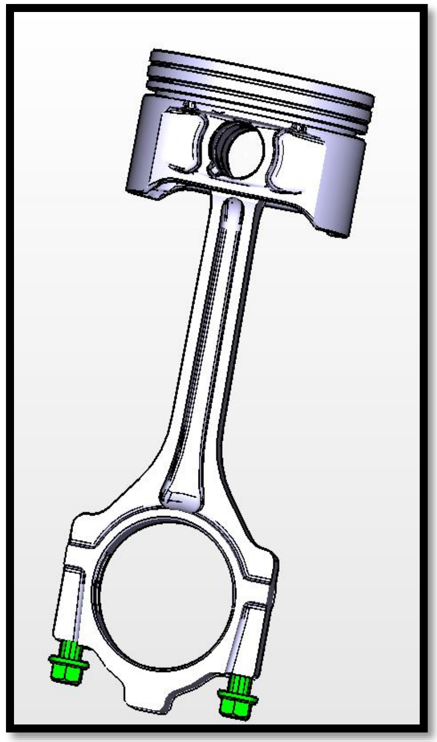


Figure 1.1 Connecting rod and bolts.

According to Yogesh N Dupare [5], axial stresses and bending stress are acted on the connecting rod inside the combustion chamber. He also says that axial stress is due to combustion chamber pressure and inertia forces and bending stress is due to centrifugal action of the connecting rod when connected to the crank shaft. Tony George Thomas [6] adds that fatigue failure is very high due to the fluctuation of these loads. Yogesh [5] says that 50-90% of the failure of the connecting rod are due to fatigue failure, thus it is very important to consider fatigue failure in the connecting rod design and great care must be taken by the Computer aided

Engineering (CAE) team in a company to perform analysis on fatigue and come up with the redesign proposal, if necessary [14] and [15].

2016 Ford Eco Boost Mustang uses forged steel as a connecting rod member. There is always been a thug of war in automobile industry to choose the type of connecting rod material. In this thesis forged steel and aluminum 7075 material is used as a connecting rod material. CAE analysis is carried out to pick the better material.

Computer aided Engineering (CAE) team in a company performs analysis on all the real world problems using many different software by applying real world constraints to get solutions. Every company is equipped with a CAE team, which performs a detailed analysis on the connecting rod in every automobile companies by applying combustion chamber constraints like pressure, inertia forces, suppress the linear motion of the connecting rod were ever necessary. This team comes up with a real time results after the analysis is carried out and suggestions are made to redesign, if necessary. Once the CAE team approves the design then the actual production of the part kicks off. The connecting rod selected in this analysis is under investigation to validate the stresses and fatigue life of the component. Furthermore, if the connecting rod fails the design requirement, a new design proposal is given where ever necessary.

## 1.2 Objective of the thesis

To validate,

1. Fatigue life and stress developed in the connecting rod for infinite life and propose design modification if necessary using forged steel and aluminum 7075 as a connecting rod material.
2. Compare analysis results of forged steel and aluminum 7075.

Chapter 2

# TERMINOLOGIES AND ASSUMPTIONS

## 2.1 Terminologies

a) Connecting Rod
b) CC – Combustion Chamber
c) FS – Forged Steel
d) AA – Aluminum 7075
e) FOS – Factor of Safety
f) BC – Boundary Condition
g) CAE- Computer Aided Engineering
h) rpm- Rotations per minute.
i) E – Young Modulus
j) V – Poisson Ratio

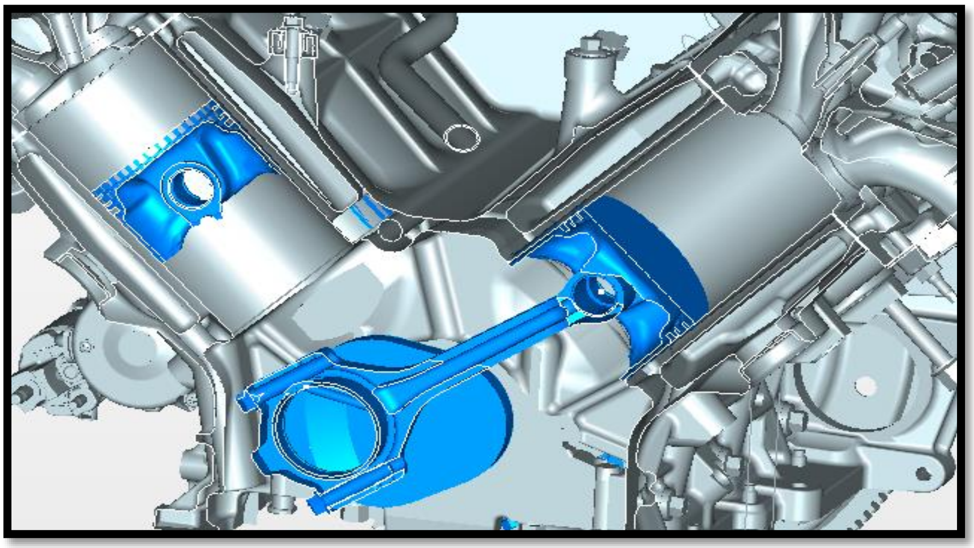


Figure 2.1 Sectional view of piston and connecting rod assembly.

## 2.2 Assumption

### *2.2.1 Connecting Rod Materials*

Forged steel is currently Ford Eco boost Mustang material. AA is used mostly in aerospace application; this material is used to handle high stress values.

Forged steel (FS) - A cosmetic trend has started by using Aluminum alloy as a CR member mainly to reduce the weight, however due to engine design evolving day by day, engineers have moved back to steel. Bryan Neelen [6] of late model Engines (LME) explains, “The weight below the wrist pin is not a big of a concern as the weight above it”. He also says that this is one of the biggest reason for moving back from Aluminum alloy.

Figure 2.2 Forged steel connecting rod assembly. [6]

One of the other big reason to use FS as CR material is due to minimum clearance, which is a drawback using Aluminum alloy. FS provides a good clearance between CR and camshaft. However, by using AA as a CR material, the extra material is always necessary to take high stress and run at higher rpm; this extra material interfere with the camshaft [6].

Aluminum 7075 (AA) - This material is used as CR to reduce the weight and it gives cushion effect between piston head and crank shaft at higher rpm [7].

Figure 2.3 Aluminum 7075 connecting rod assembly. [19]

AA CRs are generally manufactured by using CNC machines, which has high fatigue life and stronger. This CNC manufactured AA CR are very light weight and has high cushion effect. These types of CR works as a shock absorbers between the piston and crank shaft. However, the cost of manufacturing the rod is pretty high which also requires high tech tools and machinery.

AA is used in Aircraft fittings, gears and shafts, fuse parts, meter shafts and gears, missile parts, regulating valve parts, worm gears, keys, aircraft, aerospace and defense applications; bike frames, all-terrain vehicle (ATV) sprockets [8].

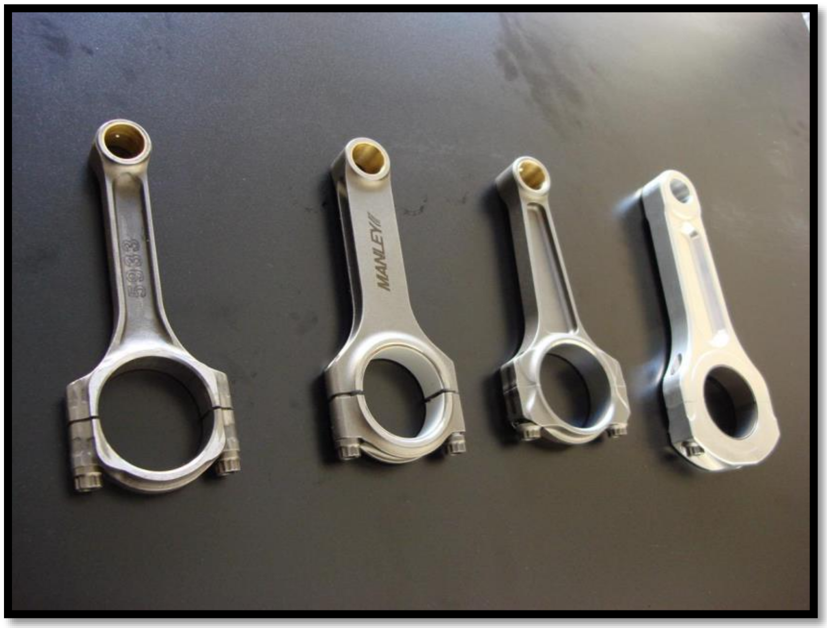



Figure 2.4 Forged steel and Aluminum 7075 as a connecting rod material [9].

Table 2.1 Material Properties [1] and [8].

| Properties/materials | Units | Aluminum 7075 | Forged Steel |
|---|---|---|---|
| Density, ρ | kg $m^{-3}$ | 2,810 | 8050 |
| Young modulus, E | MPa | 7.1E4 | 2.21E5 |
| Poisson ratio, ν | | 0.33 | 0.3 |

Table 2.2 - continued

| | | | |
|---|---|---|---|
| Yield strength | MPa | 503 | 625 |
| Ultimate strength | MPa | 572 | 827 |
| Infinite fatigue life | Min/cycles | N/A | 10E6 |
| Endurance limit for 10E8 fatigue cycles | MPa | 228.8 | 413.5 |
| Corrected Endurance limit | MPa | 192.5 | 309.3 |
| Percent of elongation | % | 11 | 54 |

### *2.2.2 Connecting Rod is used in 2016 Ford Eco boost Mustang*

The 3D CAD model is assumed to belong to 2016 Ford Eco Boost Mustang. CAE analysis is carried out on this model to validate for fatigue life. Furthermore, 2016 Ford eco boost mustang specification are used to find out the boundary conditions for both FS and AA materials.

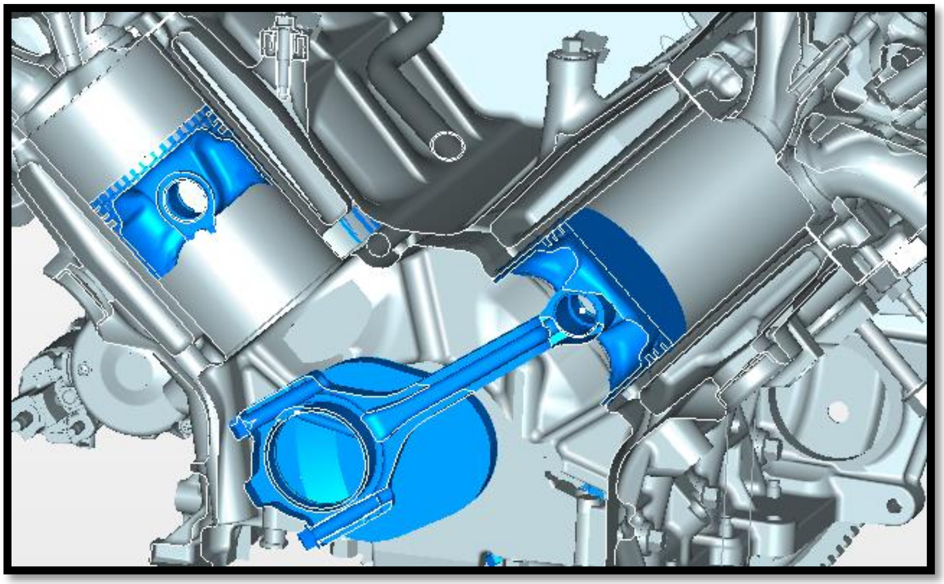


Figure 2.5 Engine block and combustion chamber

Table 2.2 2016 Ford Eco Mustang specs [10].

| Engines specification | Units | Mustang Eco Boost |
|---|---|---|
| Engine type | | 2.3L GTDI I-4 Engine |
| Displacement | Cu.in | 140 |
| Horsepower | SAE net @ rpm | 310@5500 (93-octane fuel) |
| Torque | Lb-ft @rpm | 320@3000 (93-octane fuel) |
| Compression Ratio | | 9.5.1 |
| Bore and stroke | mm | 87.5 X 94 |
| Main Bearing | | 5 |
| Valve filter | | Direct Acting Mechanical buckets |

Table 2.2 – continued

| | | |
|---|---|---|
| Fuel Delivery | | Direct injection |
| Recommended Fuel | | Unleaded Premium |
| Exhaust | | Dual bright slashed cut |
| MPG | | 22 city/ 31 high way |
| Transmission type | | 6- speed manual |
| Engine block material | | Cast Aluminum |
| Connecting rod material | | Forged steel |
| Cylinder head material | | Cast Aluminum |
| Piston material | | Cast Aluminum |

2D drawing of the Connecting rod-

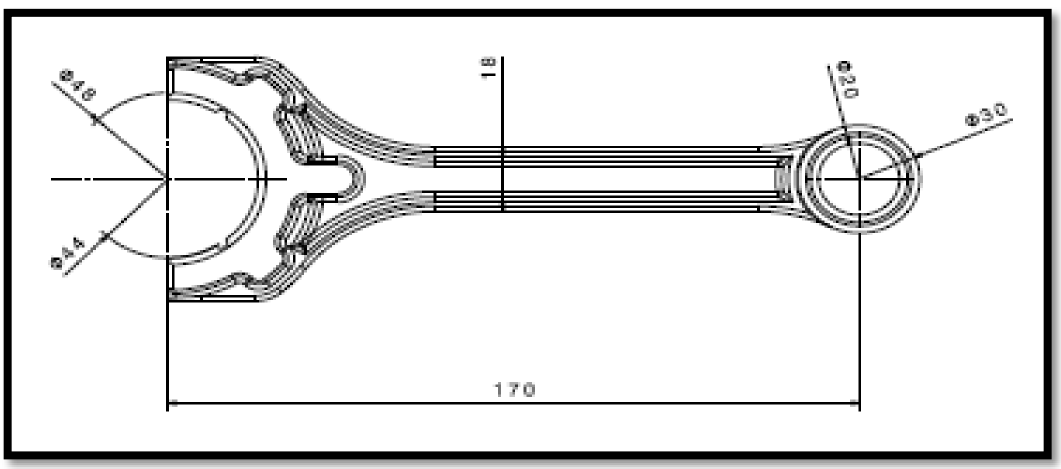



Figure 2.6 2D drawing of the connecting rod assembly

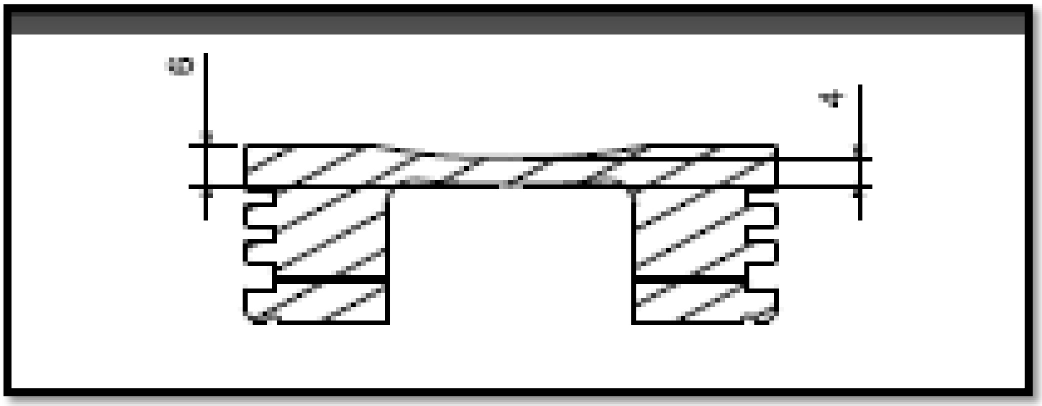


Figure 2.7 2D Piston head thickness

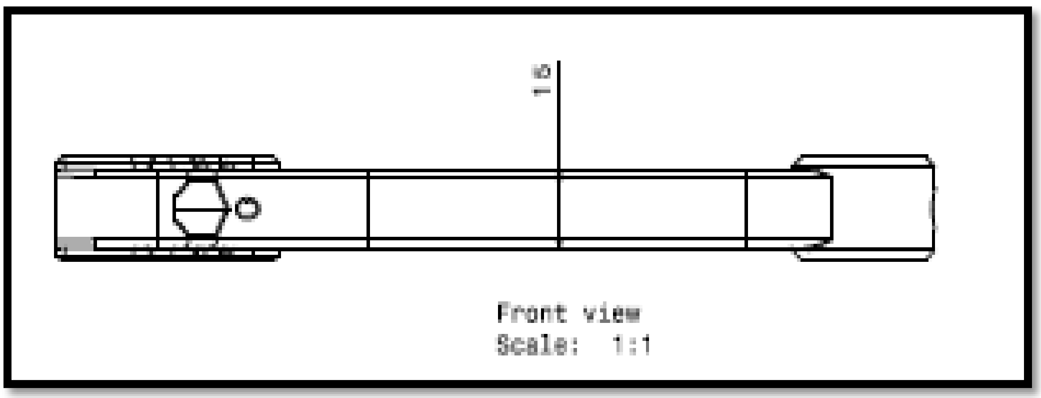



Figure 2.8 2D Assembly thickness

3 D views of connecting rod-

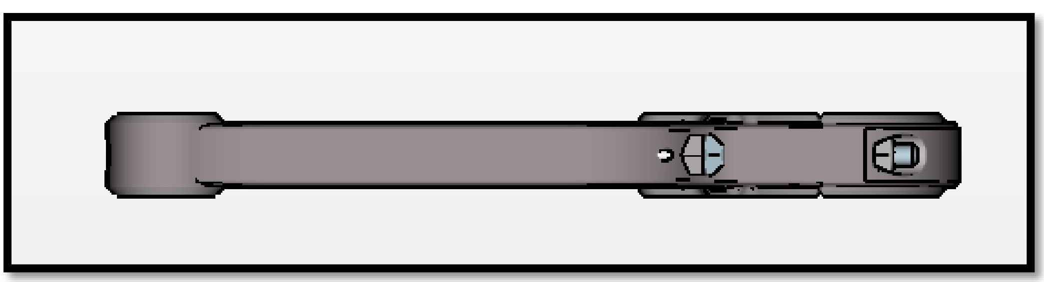


Figure 2.9 3D Side View

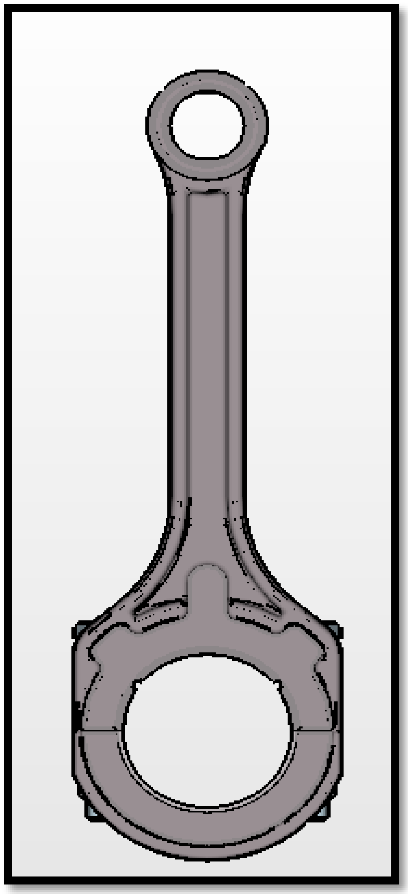


Figure 2.10 3D Front View

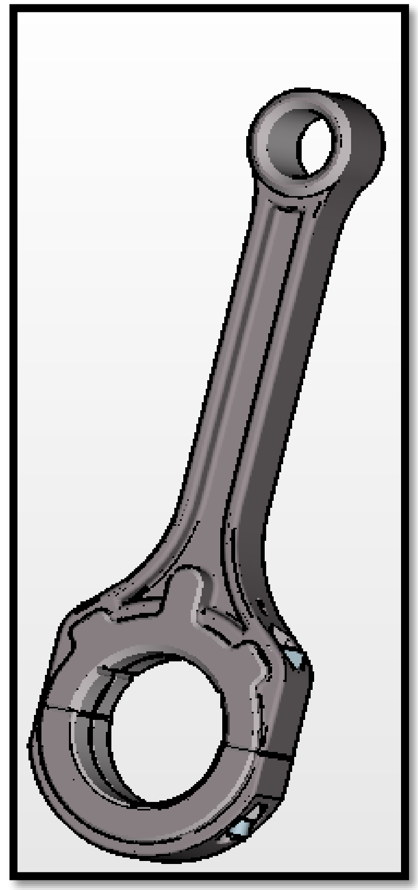


Figure 2.11 3D Isometric View

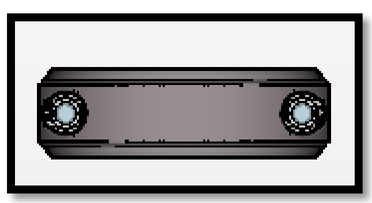


Figure 2.12 3D Top View

Figure 2.13 Crank Arm

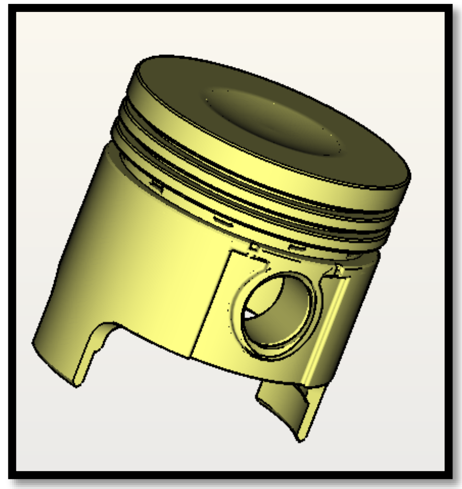


Figure 2-14 Piston

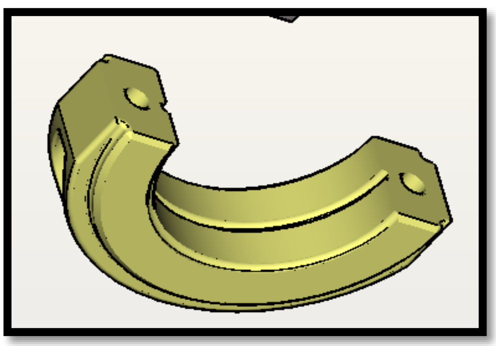


Figure 2.15 Crank Lock Ring

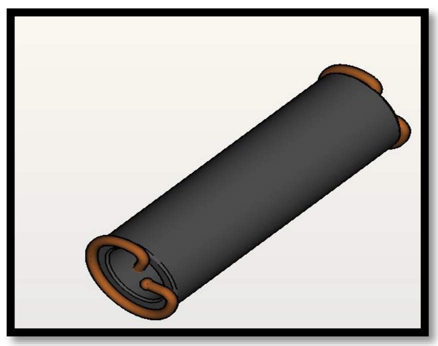


Figure 2.16 Piston Lock Pin

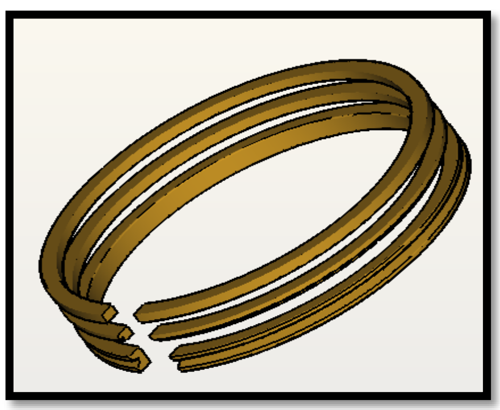


Figure 2.17 Oil Ring

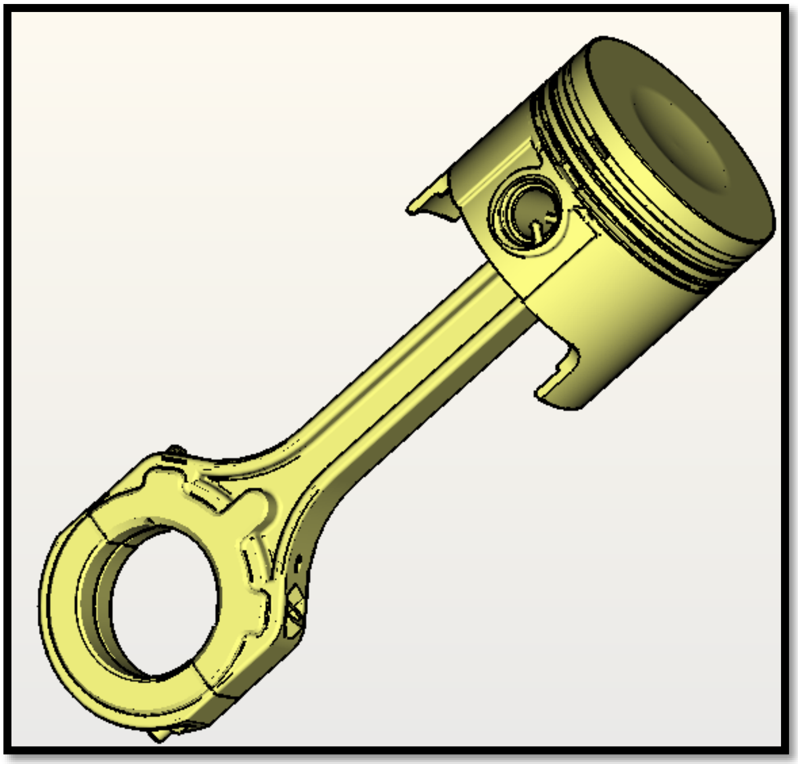


Figure 2.18 Piston- Connecting Rod Assembly

### *2.2.3 Temperature Effects*

Since most of the heat inside the CC is taken by the piston head, we do not see temperature effects as a major issue on the connecting rod.

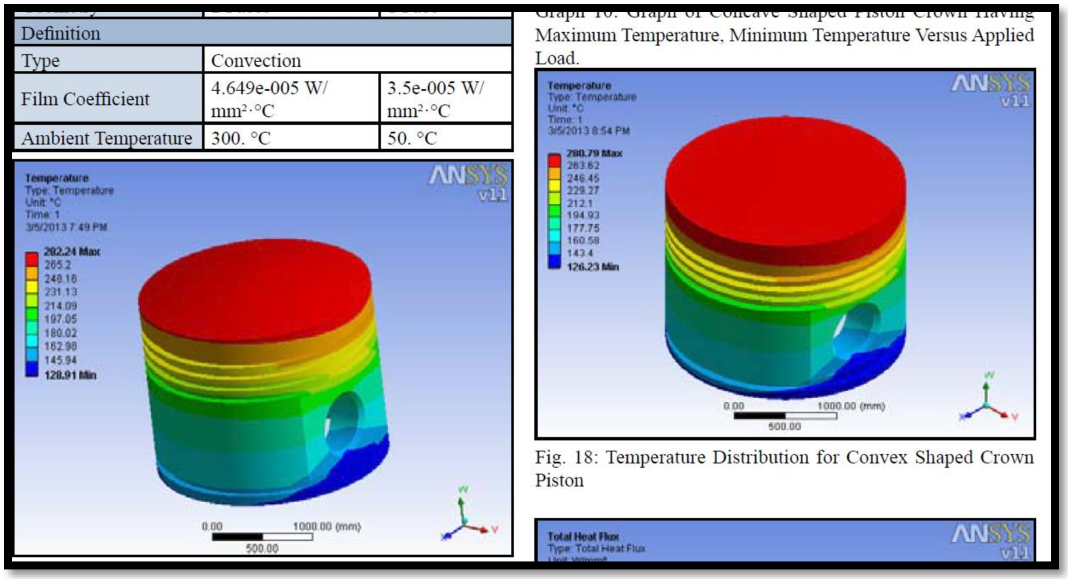



Figure 2.19 Piston head thermal stress plots [11].

Velivela Lakshmikanth, and Dr. Amar Nageswara Rao [11]- says that the temperature generated inside the CC is around 300 C for a 4 stroke IC, which is taken by the piston head. As we see in the picture the temperature effects are very high on the piston head and the temperature reduces to 50 C at the skirt of the piston (Piston skirt is the side portion of the piston which is in contact with the piston ring). By the time temperature effects reach CR, it continues to reduces, which is the reason temperature effects are neglected.

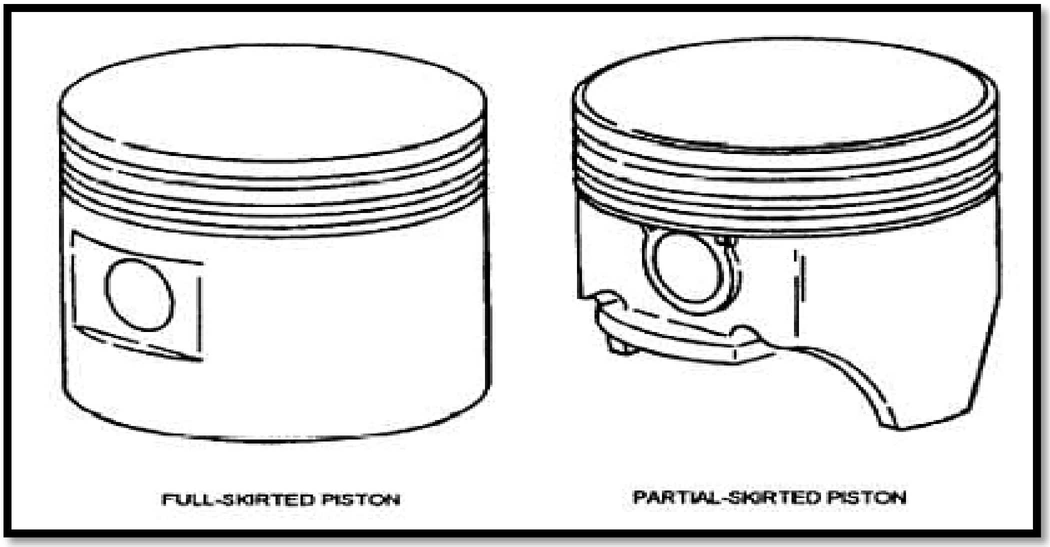



Figure 2.20 Piston body [12]

Bending stress are neglected since the crankshaft design is unavailable- Bending stresses are very important to consider since it causes lot of damages like fracture growth, failure due to wear. However, in this analysis due to the unavailability of the crankshaft design, the bending stresses which are caused due to rotational action of the CR are neglected. Bending stresses can only be calculated using crankshaft design.

Basic connecting rod design requirements-

- Max. Stress developed in connecting rod must be lower than the yield limit of the material.
- FOS must be 2 or above.
- Infinite fatigue life is preferred.
- Safety factor for fatigue must be 2 or above.

Chapter 3

BOUNDARY CONDITIONS

Axial stress developed and fixed constraints on the CR are the real time boundary conditions which are seen in Ford Eco Boost Mustang Engine.

Axial Stress- Axial stresses are developed due to the

- Combustion Chamber pressure (CC)
- Inertia Force

Combustion chamber pressure (CC) - High value of axial stresses is developed due to compressive pressure developed inside the combustion chamber due to the combustion of fuel [5].

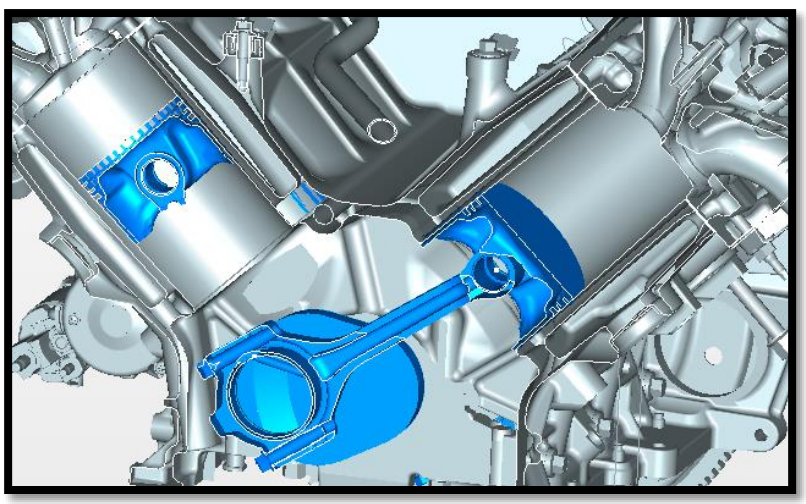


Figure 3.1 Piston- Combustion chamber and piston connecting rod.

## 3.1 Combustion Chamber Pressure Calculation (CC) [3]

Ford Eco Boost Mustang Engine specs-

Engine type- 2.3 L, 4 cylinder engine.

Bore X Stroke in mm- 87.55 X 94

Displacement- 140 cu.in

Horse Power- 310 @550

Torque- 320 @3000

Compression ratio- 9.5:1

Density of petrol $C_8H_{18}$= 732.22 Kg/m3

Temperature- 323.15 K

Mass = Density X Volume

= 737.22 E-9 X 140 E3

=0.1032 Kg.

r= Molecular weight of petrol 114.228 g/mole.

R= Universal gas constant.

rR= 8.314 E6/114228 = 72.786

Gas Equation-

PV= Mt rR

P= 0.1032 X 72.786 X 323.15 / 140

= 17. 73 MPa.

## 3.2 Inertia Forces Calculation

Inertial force: - Axial stress are developed due to the reciprocating action [5].

Inertia forces can be found out by;

Maximum Inertia force of reciprocating parts [3]

F in = Mr (Wmax)2 r (1+1/N)

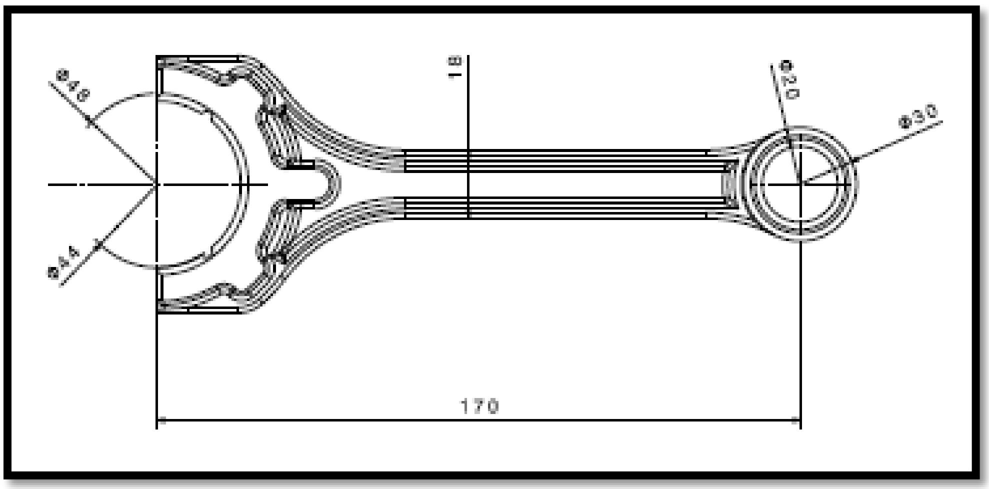



Figure 3-2 2D drawing of the connecting rod assembly

M= Mass of the fuel (C8H18)

R= radius of the crank end of CR

L= length of the CR

Wmax- Max. Angular speed

N= L/r

Horsepower= 310@5500 (93-octane fuel)

N= 5500 rpm (Max. operating rpm in a Ford Eco Boost Mustang)

Maximum angular speed Wmax = [2π Nmax ]/60.

$= [2\pi\times5500]/60$

$= 575.7$ rad/sec

Ratio of the length of connecting rod to the radius of crank

N= L/r =170/ 24 = 7.08

Density of Petrol $C_8H_{18}$ = 737.22 kg/m3

Mass= Density × Volume

= 737.22E-9 x149.5E3

= 0.11kg [16]

Maximum Inertia force of reciprocating parts

F in = Mr (Wmax) 2 r (COS θ + COS 2θ/N) (Or)

F in = Mr (Wmax)2 r (1+1/N)

= 0.11x (575.7 x 575.7) x (0.024) x (1+ (1/7.08))

= *1000 N*

3.3 Fixed Support

Cr is fixed at Z direction, this motion constraint due to presence of the engine block.

## Chapter 4

## MESH AND MESH SENSITIVITY

Solving a complex body to find the results of stress and fatigue life without using Finite element analysis is tedious and takes a lot of man hours and often results in human errors in solving complex equations. In 1943 an efficient way to solve complex problems related to a component was introduced by R. Courant [13]. He discretized the whole component into small elements, this process of breaking down the body is called meshing. This small elements are solved individually for solutions. Then solution of each individual element is summed up to get a final solution. One should understand that the obtained solution are not exact, but are approximate solutions which Engineers can trust.

Mesh- A very fine mesh was created at the critical areas like fillet region and edges of the CR. These are the sections in the CR where there is probability of max. stress concentration. Mesh connections are created in the assembly for connectivity while mesh operation is performed and make assembly a single model for analysis results.

Connections are created-

1. Between Piston and piston lock pin:

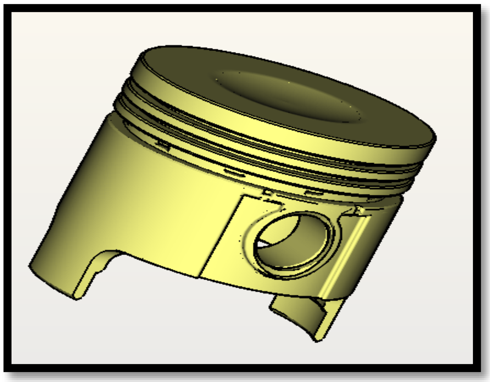


Figure 4.1 Piston.

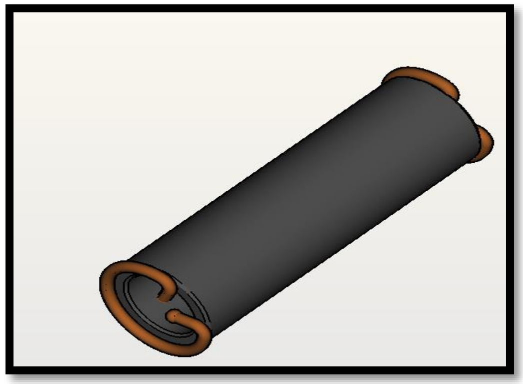
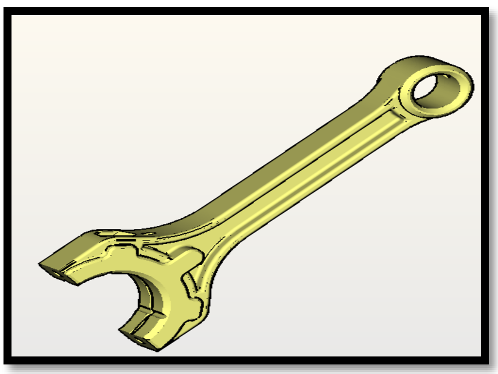


Figure 4.2 Wrist lock or piston lock pin

2. Between Piston lock pin and crack arm:



4.3 Connecting rod

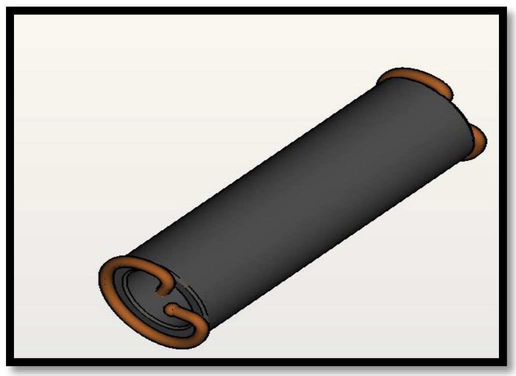


Figure 4.4 Wrist lock or piston lock pin

3. Between crank arm and crank lock ring:

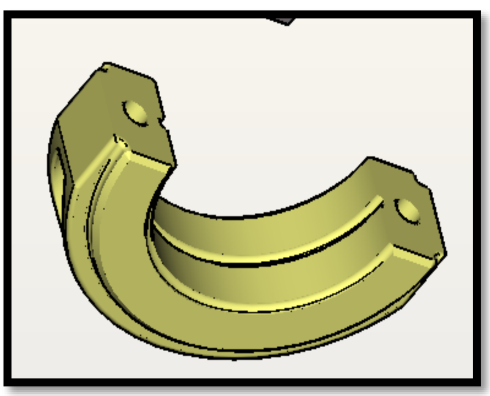


Figure 4.5 Crank lock ring

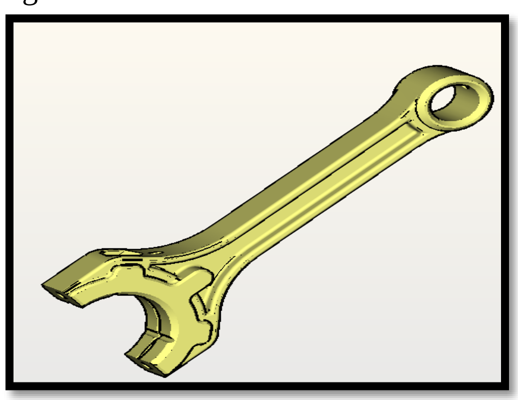


Figure 4.6 Connecting rod

Mesh sensitivity analysis

The purpose of conducting this analysis is to get accurate output solution. In this thesis, it is carried out to fin exact stress and fatigue plots. The relationship between input value and output values are understood using mesh sensitivity analysis. Output results were studied for different input element sizes from 8mm to 2 mm (element size).

## 4.1 Mesh Sensitivity Analysis for 8 mm.

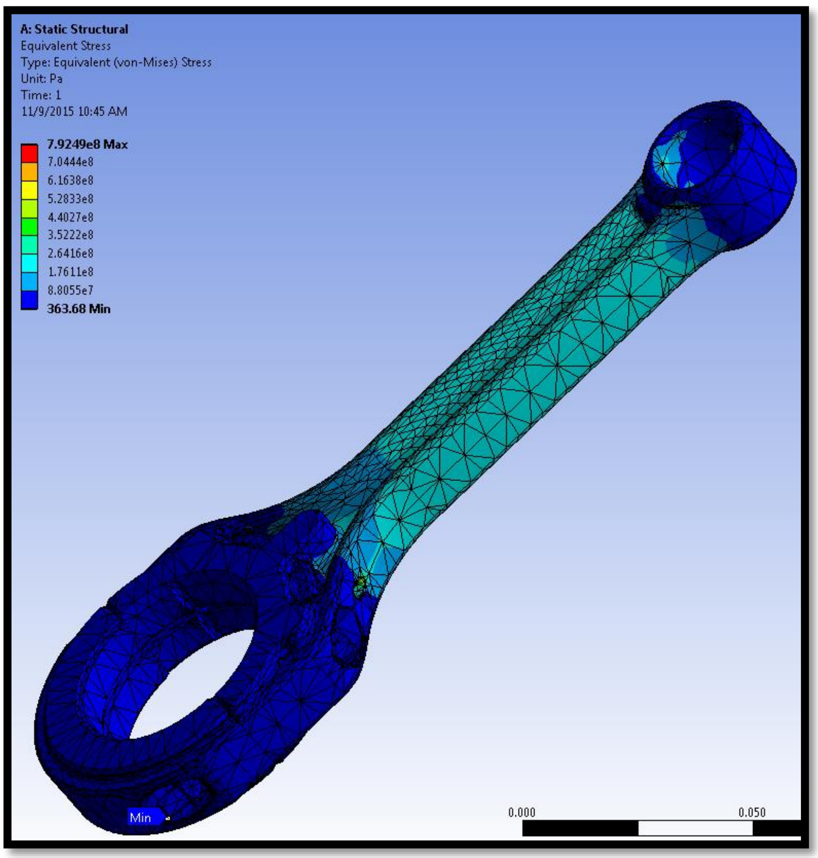



Figure 4-7 Von Misses stress plot.

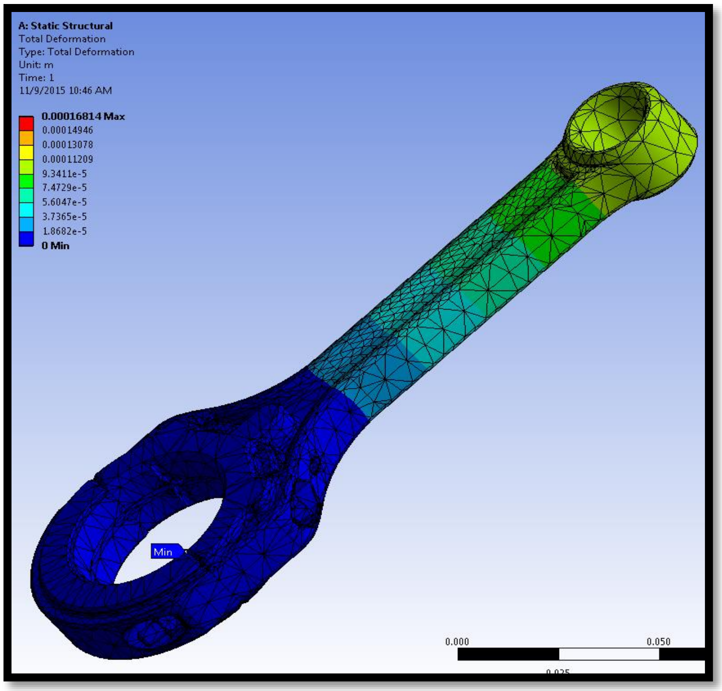



Figure 4.8 Deformation stress plot

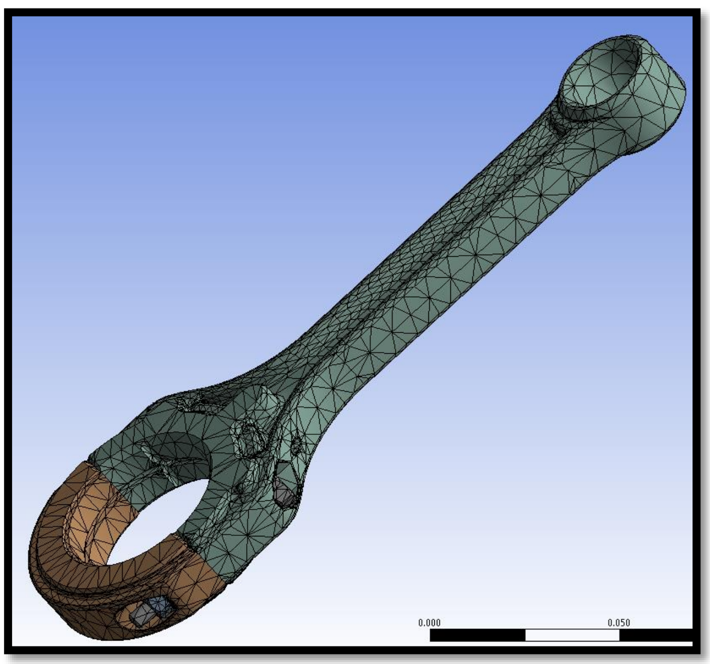



Figure 4.9 Connecting rod mesh

Table 4.3 Element Size 8 mm Results

| Element size in mm | Stress in MPa | Displacement in m | No of Nodes | No. of Elements |
|---|---|---|---|---|
| 8 | 792.49 | 0.0001654 | 87557 | 46372 |

## 4.2 Mesh Sensitivity Analysis for 5 mm


Figure 4.10 Connecting rod mesh

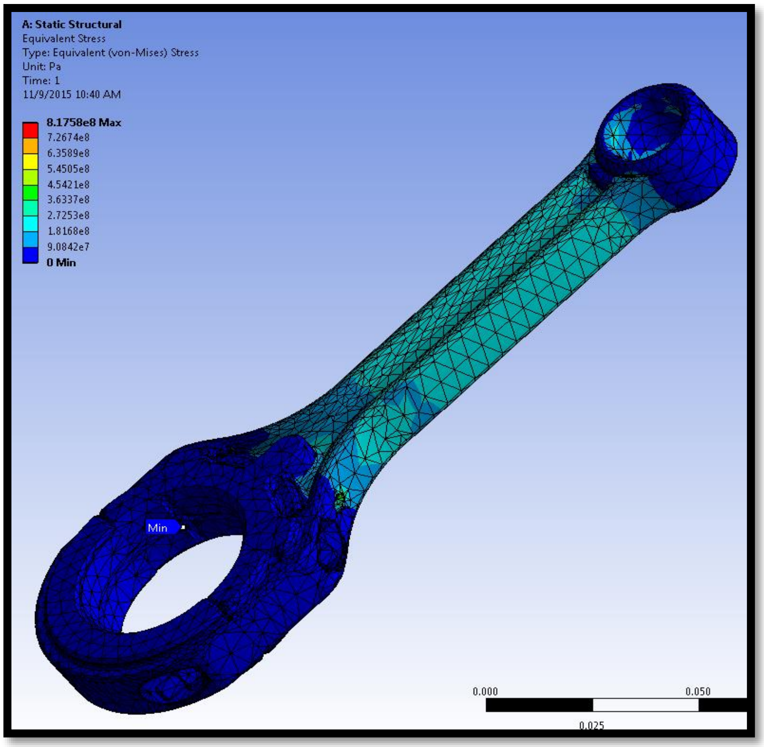



Figure 4.11 Von Misses stress plot.

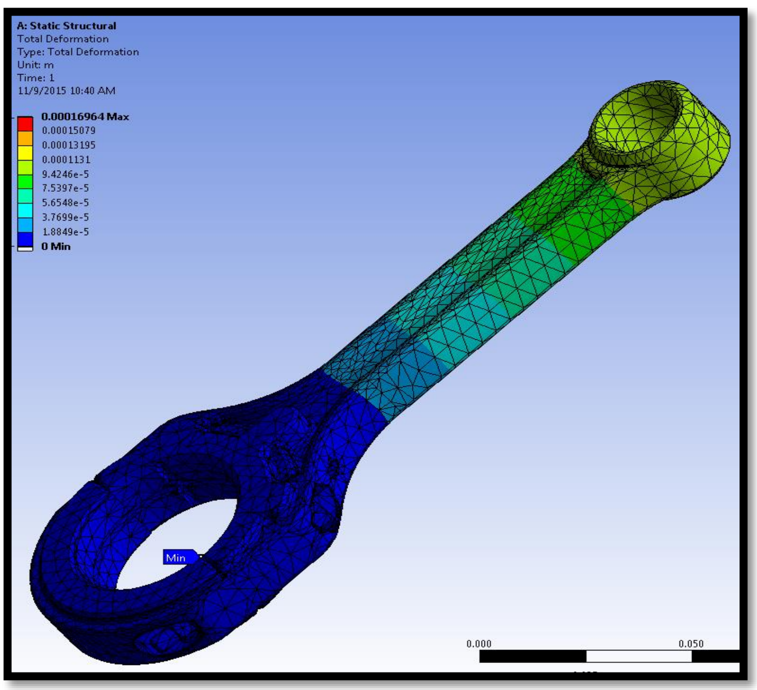



Figure 4.12 Deformation stress plot

Table 4.2 Element Size 5 mm Results

| Element size in mm | Stress in MPa | Displacement in m | No of Nodes | No. of Elements |
|---|---|---|---|---|
| 5 | 817.2 | 0.0001694 | 107761 | 57877 |

## 4.3 Mesh Sensitivity Analysis for 3 mm

It was observed that for element size 2 mm, the stress plots were similar to that of 5 mm element size. This analysis led to chooses element size of 3 mm.

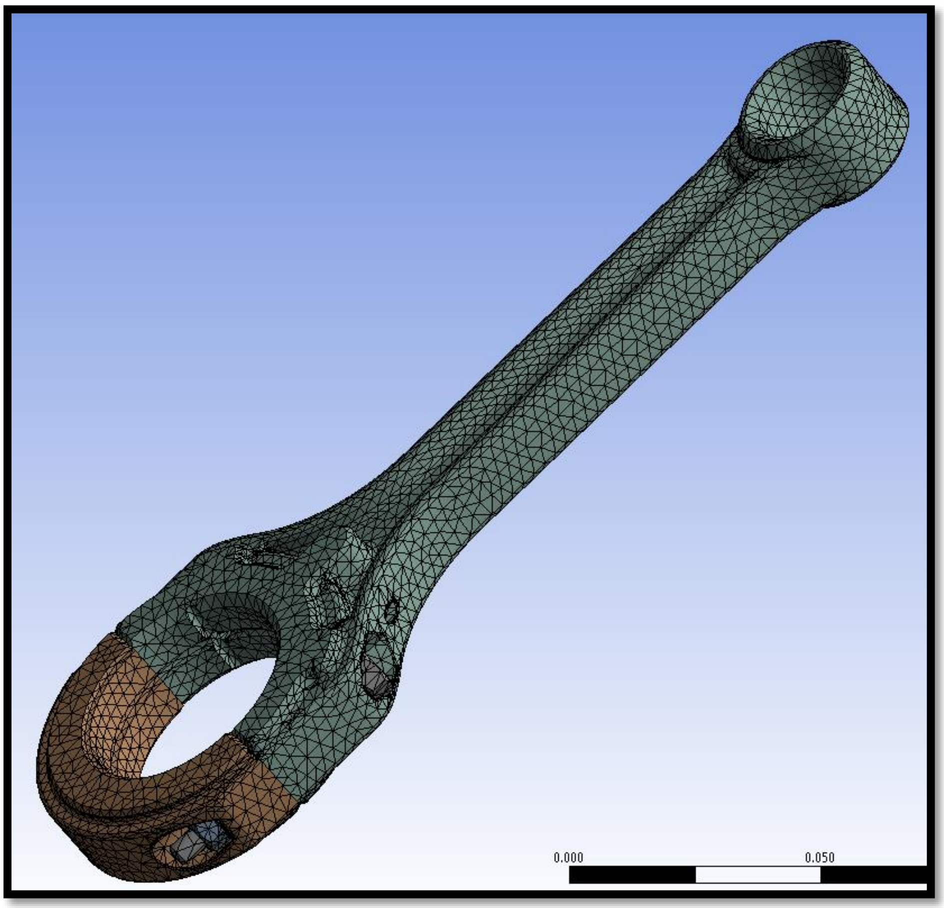



Figure 4.13 Connecting rod mesh

Table 4.3 Element Size 3 mm Results

| Element size in mm | Stress in MPa | Displacement in m | No of Nodes | No. of Elements |
|---|---|---|---|---|
| 3 | 770.23 | 0.0001655 | 191748 | 110631 |

# Chapter 5

# CAE ANALYSIS FOR FS AND AA

Static structural and fatigue analysis are carried out on the connecting rod. Here the analysis is done for FS and AA. BCs are applied, as inputs, to get stress and fatigue plots.

## 5.1 CAE analysis on a Forged steel connecting rod:

Boundary Conditions (BC) - These are the conditions or constraints, which are applied on the connecting rod, which is present inside the engine block of the Ford Eco Boost Mustang. BC is the pressure inside the CC, Inertia force due to the reciprocating action and fixed constraints on the CR.

- Piston connecting rod assembly- Forged Steel
- Pressure on piston head 17.73MPa (-Y direction axial loading).
- Force due to Inertia 1000 N (-Y direction axial loading).
- Z direction is fixed.
- Y direction free for reciprocating motion
- X direction free for rotational motion.

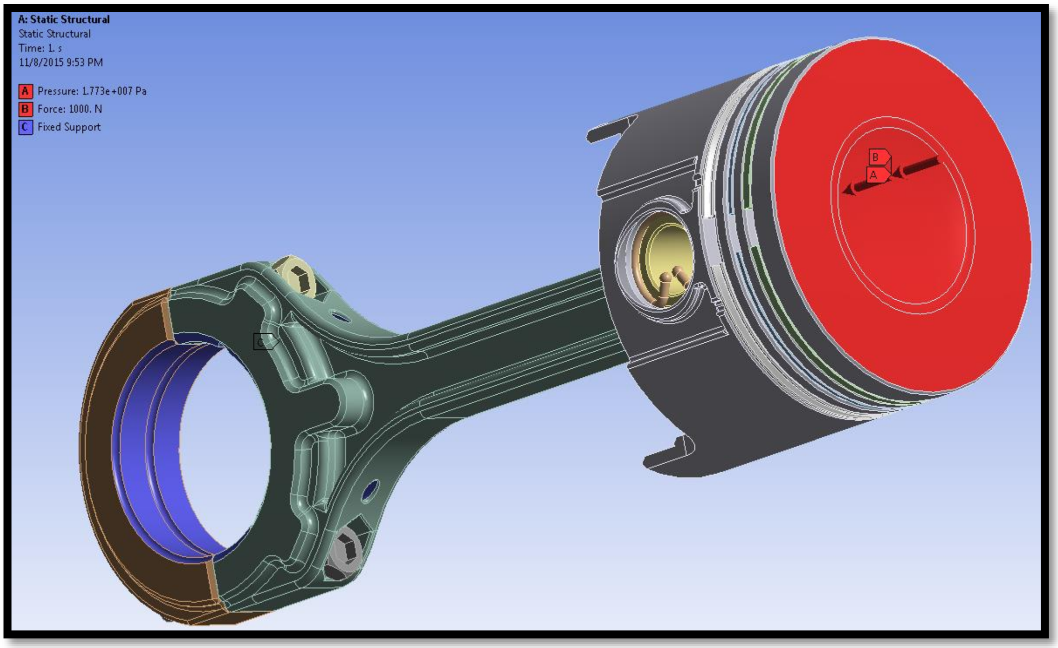



Figure 5.1 Boundary conditions applied on the CR assembly

Static structural analysis [14] - This type of analysis deals with steady loading conditions only and ignores effects of loads which changes over time, for example inertia and damping effects. However, inertia loading which are caused due to self-weight, reciprocating and rotational motion, can be considered.

Von misses stress, deformation, and factor of safety plots are obtained by conducting static structural analysis. Von misses stress plots are used in this analysis since they give detailed stress plots versus the yield limit and also often used since it give a detailed plot for all ductile materials in theory of plasticity.

Von misses stress plots

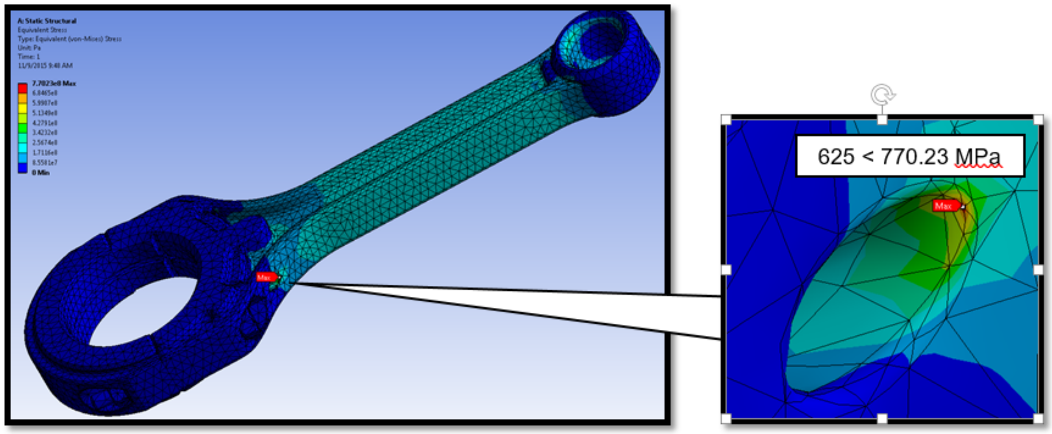



Figure 5.2 Von misses stress plot for –Y directional axial loading.

Maximum stress developed is at the fillet region of the CR for (–Y) direction axial loading. Maximum stress is 770 MPa at the fillet section which is higher than the yield of the material 625 MPa. [15]

Deformation plots-

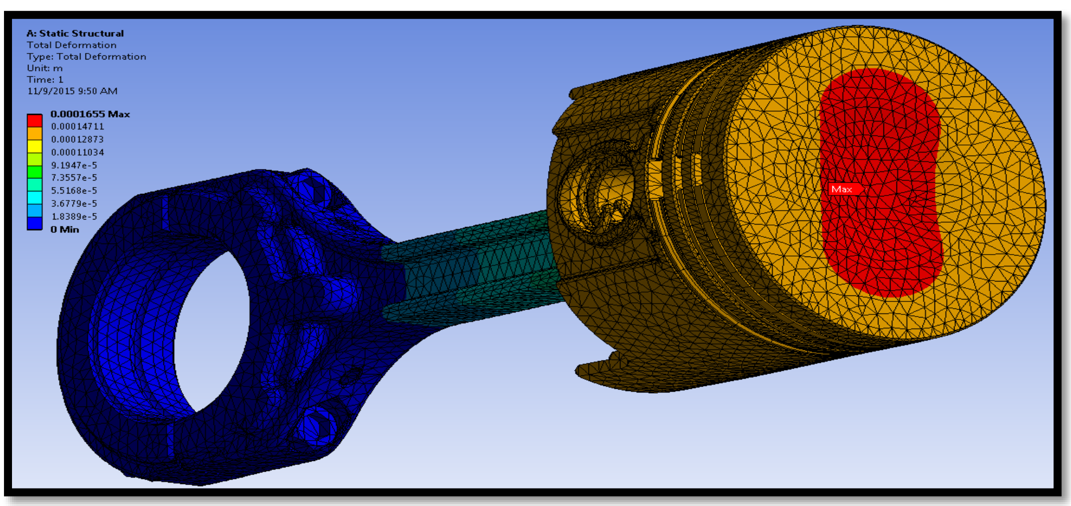



Figure 5.3 Deformation plot on piston head for –Y directional axial loading.

Maximum deformation occurs at the piston head since cross section area is less.[11].

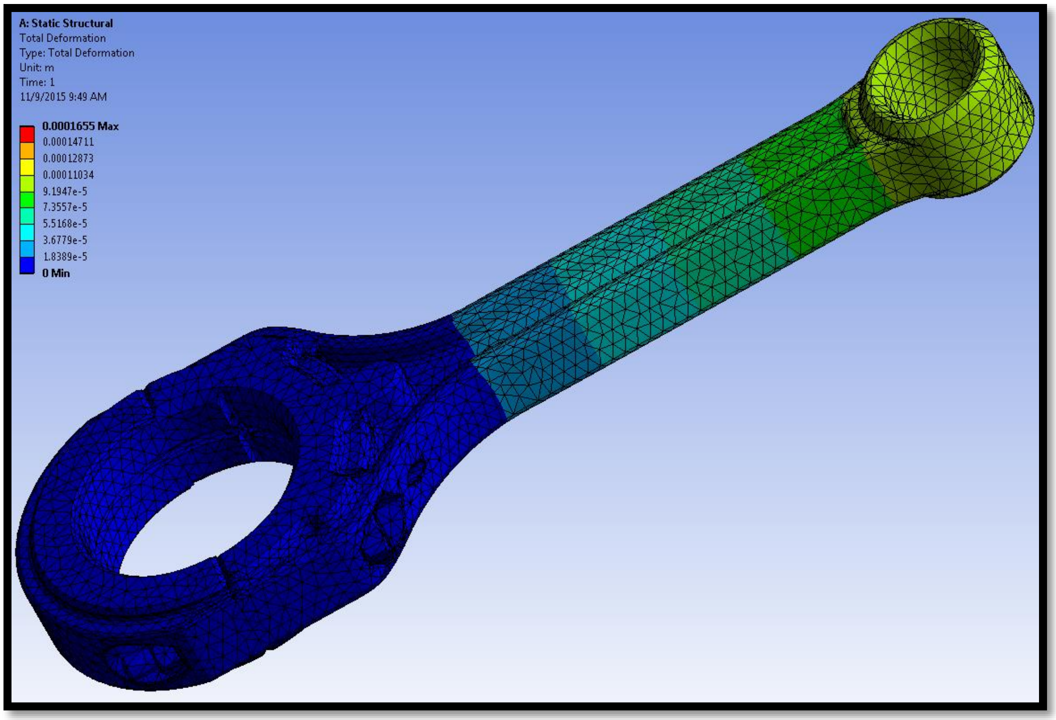



Figure 5-4 Deformation plot on connecting rod for –Y directional axial loading.

Maximum deformation, which occurs in the CR, is at the piston end [16]. We can see maximum deformation at the piston end because the area is very small for pressure distribution.

Factor of safety (FOS) plots- It is the ratio of the yield to the maximum stress developed. In general, practicing Engineers try to have FOS of 2 or above for connecting rod.

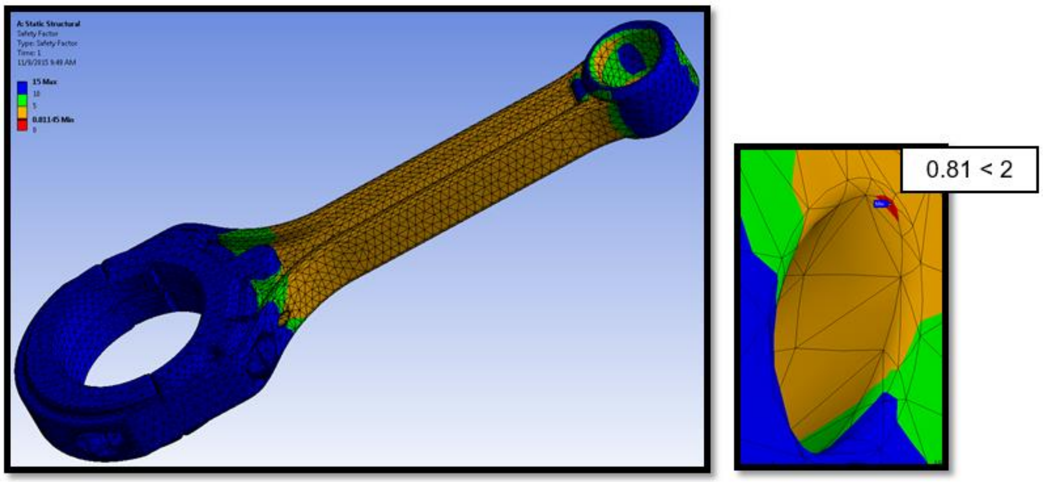



Figure 5.5 FOS plot for –Y directional axial loading

Minimum FOS occurs at the fillet section of the CR assembly; desired FOS is 2 or above.

Fatigue analysis: - When the connecting rod is applied repeated cyclic loads, like pressure and inertia force, the material begins to weaken, this is known as fatigue. When the material is subjected to repeat cyclic loading there will be progressive and localized structural damage [16]. The stress developed will be always less than the yield stress and ultimate stress, however due to repeated loading, the material will fail from generations of crack to brittle material like failure. This type of failure generation is very hard to identify since the connecting rod is not visible to naked eyes and it is inside the engine cylinder. This type of failure is called “throwing a rod” and the whole engine ceases, which leads to irreparable engine. According to survey it says 90% of the connecting rod failure is due to the fatigue. In this thesis, fatigue analysis is carried out to see if the connecting rod fulfills infinite life requirement, also if the connecting rod fails, further analysis is carried out to find value of the stress for which the life of the CR increases to infinite and giving FOS of value 2.

Fatigue analysis for forged steel connecting rod material

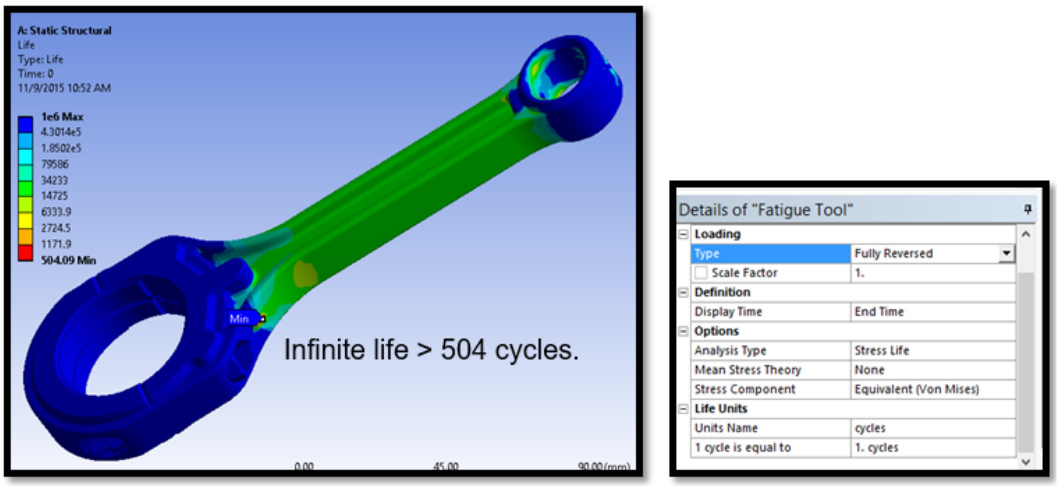



Figure 5. 6 Fatigue plot for –Y directional axial loading

1. Minimum life of the CR is 504 cycles only.
2. CR is at high risk of failure as the min. life of the component is 504 cycles only.
3. It is the responsibility of the Engineer to redesign the CR to give fatigue life of 10E6 cycles.
4. In general practice for steel material, CR is designed for infinite cycles.

Factor of safety/safety factor

1. Safety factor for fatigue is 0.1, risk of failure is very high.
2. General practice is to have safety factor of 2 or above.

3. Further analysis is carried out to increase the CR life to 10E6 and safety factor to 2 or above.

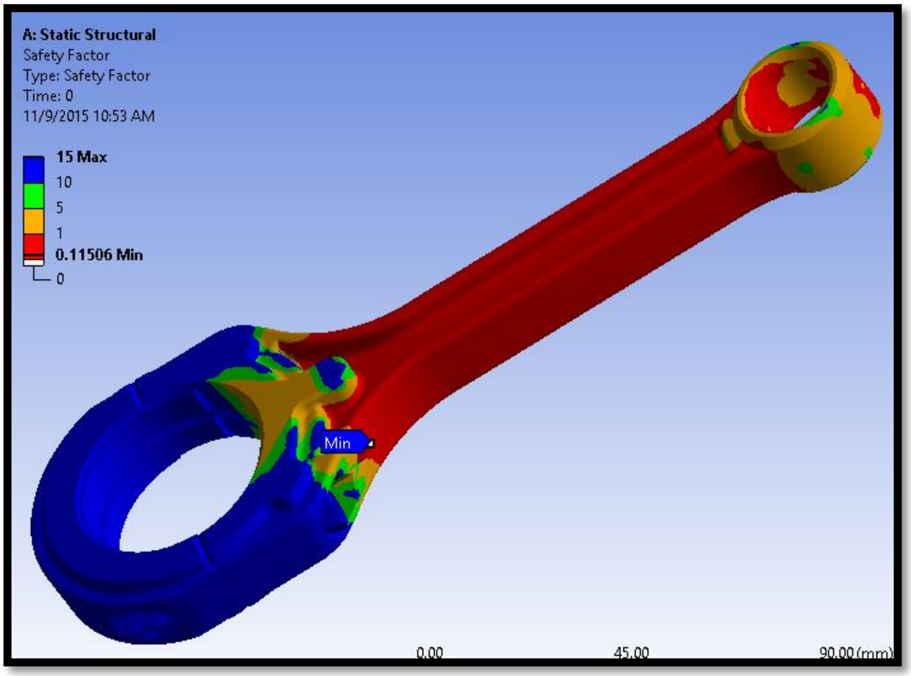



Figure 5.7 FOS fatigue plot for –Y directional axial loading

5.2 Fatigue redesign for forged steel connecting rod

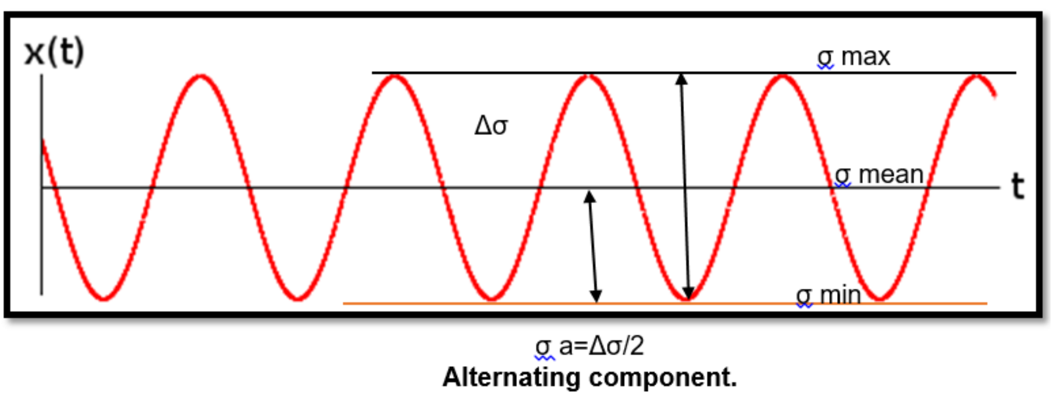



Figure 5.8 fully reversed case

Fully Reversed case. A case where there is tensile and compressive loading on the connecting rod are the same. According to Yogesh, CR will undergo a fully reversed case and further analysis is carried out considering the CR is under tension and compression loading [5].

- $\sigma$ max- Maximum alternating stress developed= +770.23 MPa
- $\sigma$ min – Minimum alternating stress developed= - 770.23 MPa
- $\sigma$ mean- mean of $\sigma$ min and $\sigma$ max= 0 MPa.
- $\Delta\sigma$ = Total value of stress developed= 770.23 + 770.23= 1.54 E3 MPa
- $\sigma$ a=$\Delta\sigma/2$= 770.23 MPa.
- Alternating component= $\sigma$ a= 770.23 MPa (max stress developed)
- Assuming the CR has maximum stress value of 770.23 MPa throughout its life cycle (Worst possible case).

Analysis is carried out to determine the value of stress to give infinite life and FOS of 2.

General procedure to find value allowable stress which gives material Infinite life and FOS of 2-[17]

A. Select material- Forged steel
B. Calculate Sm and Se'-

According to Dr. Cyders- Sm=0.75 X Ultimate limit of the material (Sut) for axial case and 0.9 X Sut for bending case.

Since we are dealing with axial case Sm= 0.75 X 827= 620.25 MPa.

Se' is the endurance limit of the material- Endurance limit is the value of the stress (below) which the material will have infinite life.

Se' for forged steel [32, pg 290]= 0.5 X S ut= 0.5 X 827= 413.5 MPa.

In real life scenario, the material will have lot of manufacturing defects so the corrected endurance limit has to be found by determining what the possible errors are.

C. Calculate the correction factor [18]

According to Shigleys Mechanical engineering design hand book - it is very unrealistic to consider the specimen to have an endurance limit same as the one calculated for lab specimen. These factors vary in real life compared to lab specimen-

Material- Composition, basis of failure.

Manufacturing- Method, heat treatment, fretting corrosion, surface condition, stress concentration.

Environment- Corrosion, temperature, stress state, relaxation times.

Design- Size, shape, life, stress state, speed, fatigue, galling.

Factors- load, size, temperature, reliability and surface finish must be considered.

This imperfection in the real world scenario is calculated using-

Correction Factors= C load X C size X C temp X C reliability X C surface finish

- C load- For axial loading= 0.85
- C size- No size effects for axial loading= 1.
- C temperature $kd = 0.975 + 0.432(10E\text{-}3)Tf - 0.115(10E\text{-}5)Tf^2 + 0.104(10E8)Tf^3 - 0.595(10E\text{-}12)Tf^4$, Tf= working temperature= 50 C or 122F

  C temperature- $kd = 1.01$
- C reliability [32 pg 301]= $ke = 1 - 0.08\, za$

  From the table considering 50% reliability= Za= 0, Ke= 1
- C surface finish [32 pg 295]= $ka = a\ Sut^b$

  C surface finish= a(Sut)^-b= 4.51X827^(-0.265)= 0.88.

Correction Factors= C load X C size X C temp X C reliability X C surface finish

=0.85 X 1X 1.01 X1 X 088.

Correction Factors = 0.748

D. Calculate corrected Endurance limit Se= Endurance limit X Correction.

Se= 413.5 X 0.748= 309.3 MPa.

E. Draw Stress vs Time (S-N) diagram

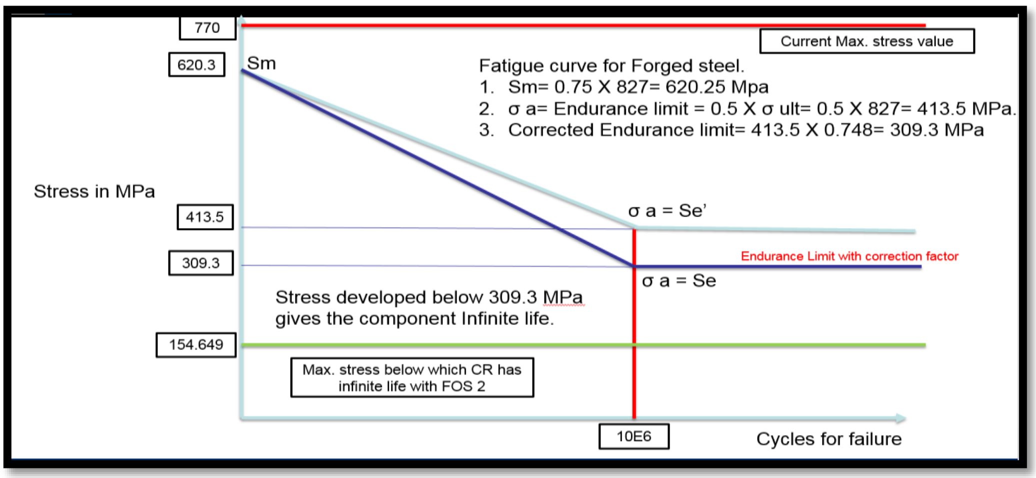



Figure 5.9 Stress vs time graph (S-N graph)

Stress versus cycles to failure graph is plotted. Sm, endurance limit, and corrected endurance limit is plugged in the graph. Stress value below 309.3 MPa gives the material infinite life.

F. Calculate σ a (Alternating stress) for 10E6 cycle life

Stress value below 309.3 MPa gives the material infinite life, however, it will not give the CR a FOS of 2.

G. Set acceptable σ a and FOS of 2

To get FOS of 2, divide the corrected endurance limit by

$\sigma$ a and FOS of 2 = 309.3/2= 154.649 MPa.

The above stress value gives the CR infinite life and FOS of 2 and infinite fatigue life.

This satisfies the design guide requirement.

Conclusion and Validation

1. Maximum working stress is 154.6 MPa, which is less than yield stress, which is 625 MPa.
2. FOS is 4 for static structural axial loading; meets the design guide requirement.
3. Working stress at fatigue is 154.6 MPa which is less than endurance limit, with correction factor is 309.3 MPa.
4. Safety Factor at fatigue is 2; meets the design guide requirement.
5. Maximum stress and poor fatigue cycles occurs at the fillet section of the CR, redesign at this area, by either deleting the fillet section or increase the thickness at that particular site, is highly recommended to reduce stress concentration.
6. CR life is now designed for Infinite cycles and meet the design guide requirement.

By considering all the above factors, a robust CR design can be designed.

## 5.3 CAE analysis for ALUMINUM 7075 connecting rod

Boundary conditions are applied on the ALUMINUM 7075 connecting rod to get the output stress and fatigue plots

- Piston connecting rod assembly- Aluminum alloy
- Pressure on piston head 17.7MPa (-Y direction axial loading).
- Force due to Inertia 1000 N (-Y direction axial loading).
- Z direction is fixed.
- X direction free for rotational.
- Y direction free for reciprocating.

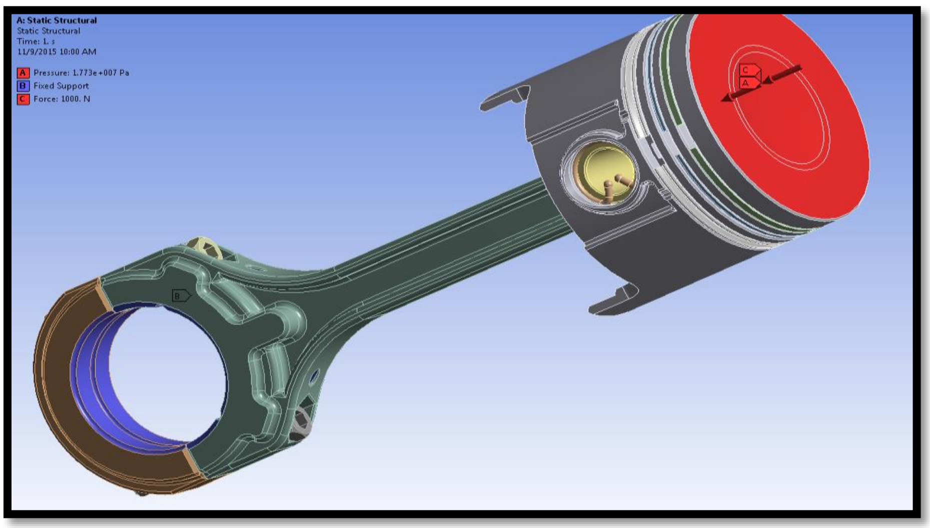



Figure 5.10 fully reversed case

Static stress analysis

By conducting static stress analysis, von misses plots, deformation plots, FOS and fatigue plots are obtained.

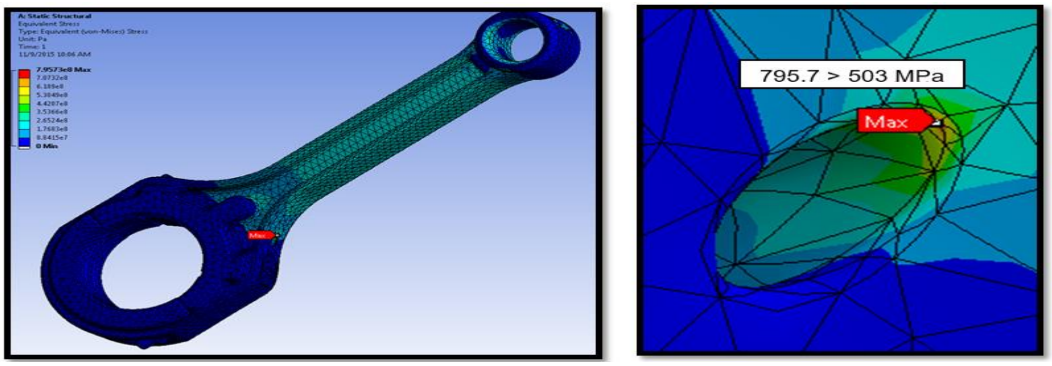



Figure 5.11 Von misses stress plot for –Y directional axial loading

Max. stress developed is at the fillet region of the CR for (–Y) direction axial loading. Maximum stress is 795.7 MPa at the fillet section, which is higher than yield of the material 503 MPa [15].

Deformation plots

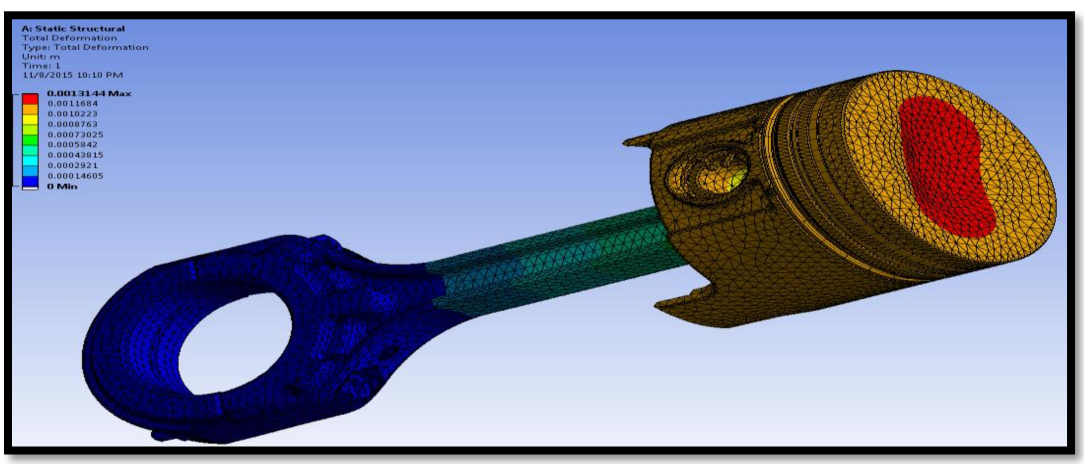



Figure 5.12 Deformation plot on piston head for –Y directional axial loading.

Maximum deformation occurs at the piston head since cross section area is less [11].

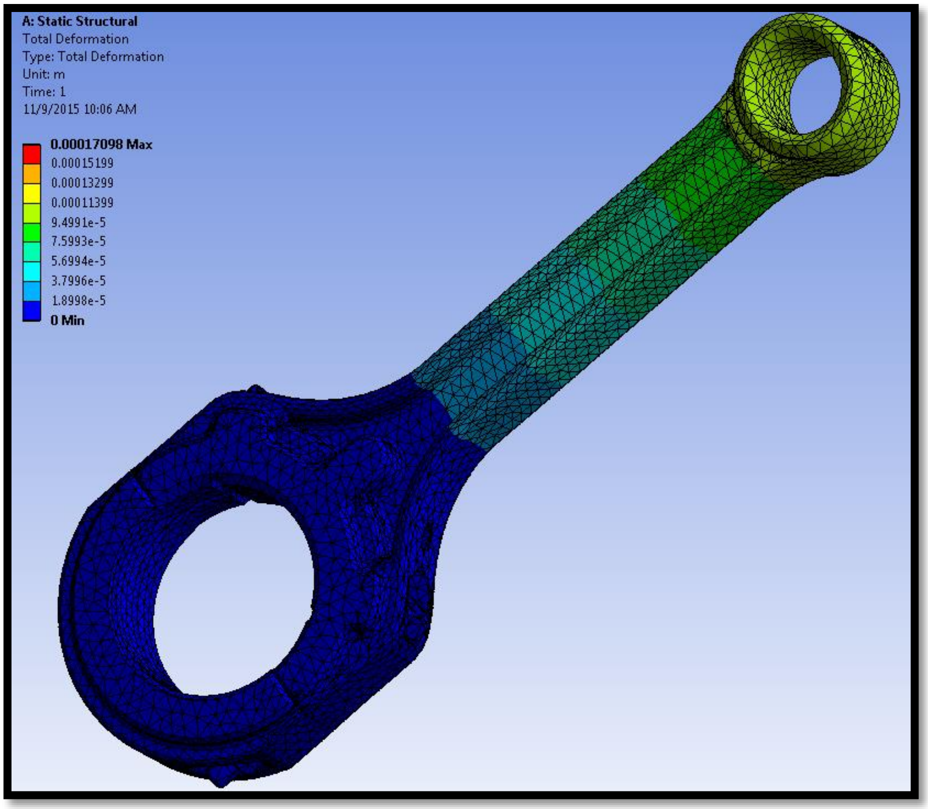



Figure 5.13 Deformation plot on connecting rod for –Y directional axial loading.

Maximum deformation occurs in the CR, is at the piston end [16]. One can see maximum deformation at the piston end because the cross section area is very small.

Factor of safety (FOS) plots- It is the ratio of the yield to the maximum stress developed. In general practice Engineers try to have FOS of 2 or above for connecting rod.

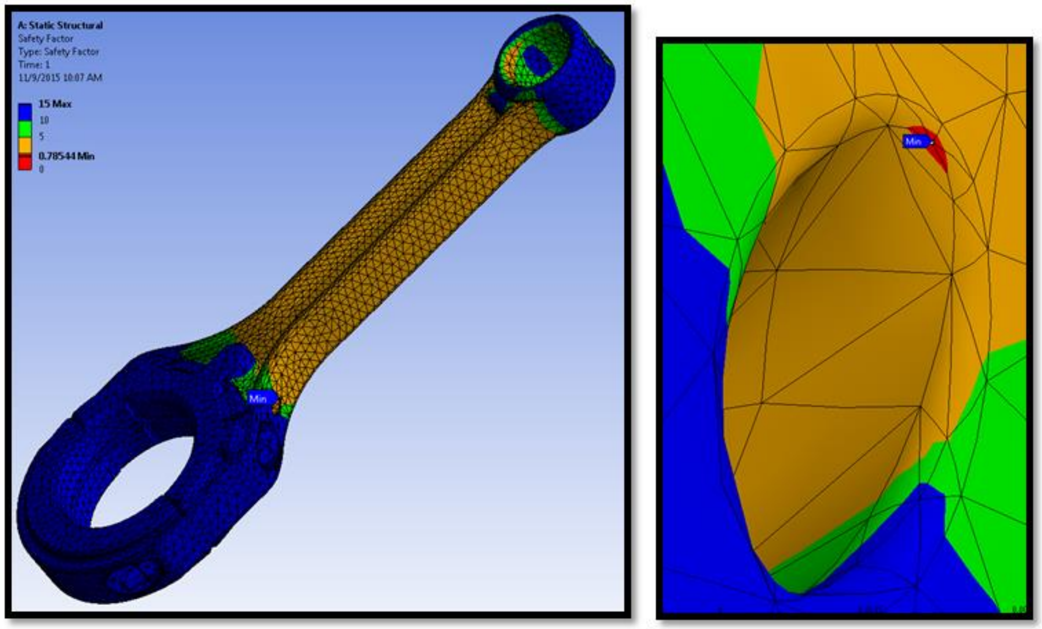



Figure 5.14 FOS plot for –Y directional axial loading

Minimum FOS occurs at the fillet section of the CR assembly, desired FOS is 2 or above.

Fatigue analysis for Aluminum 7075

Objective of the analysis is to maintain fatigue life of 10E8 and FOS of 2.

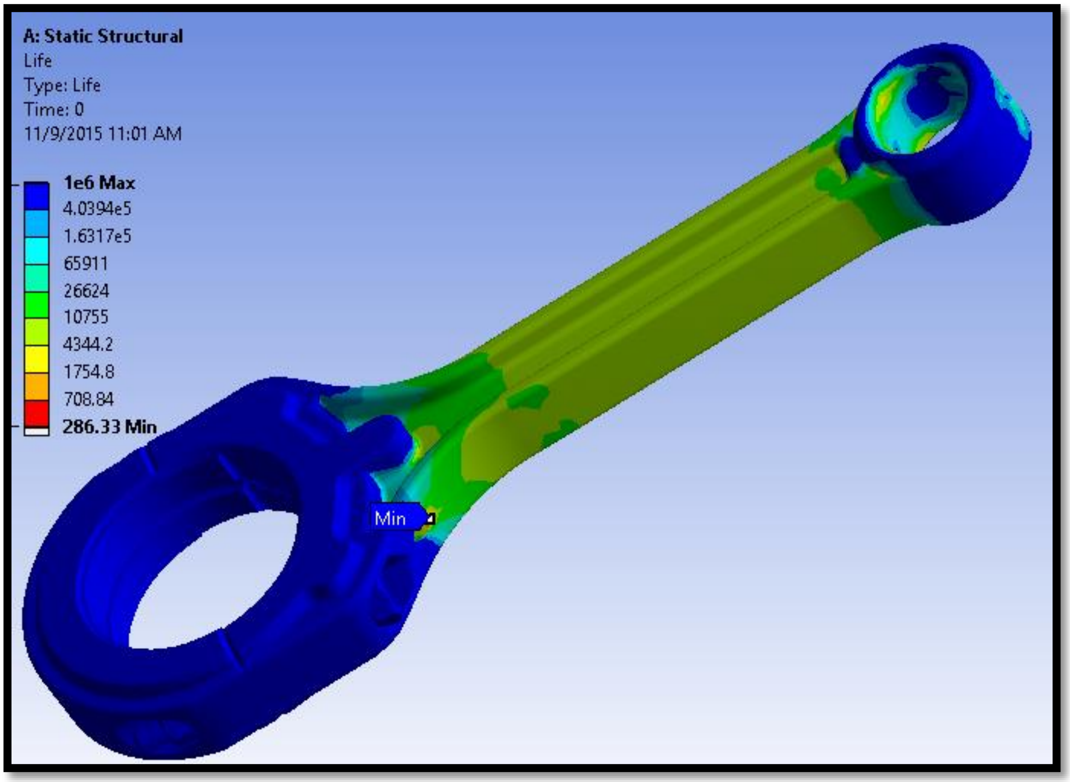



Figure 5.15 Fatigue plot for –Y directional axial loading

- CR is at high risk of failure as the min. life of the component is 286 cycles only.
- It is the responsibility of the Engineer to redesign the CR to give fatigue life of 10E8 for aluminum alloy.
- In general practice, CR is designed for a minimum of 10e8 cycles or infinite cycles.

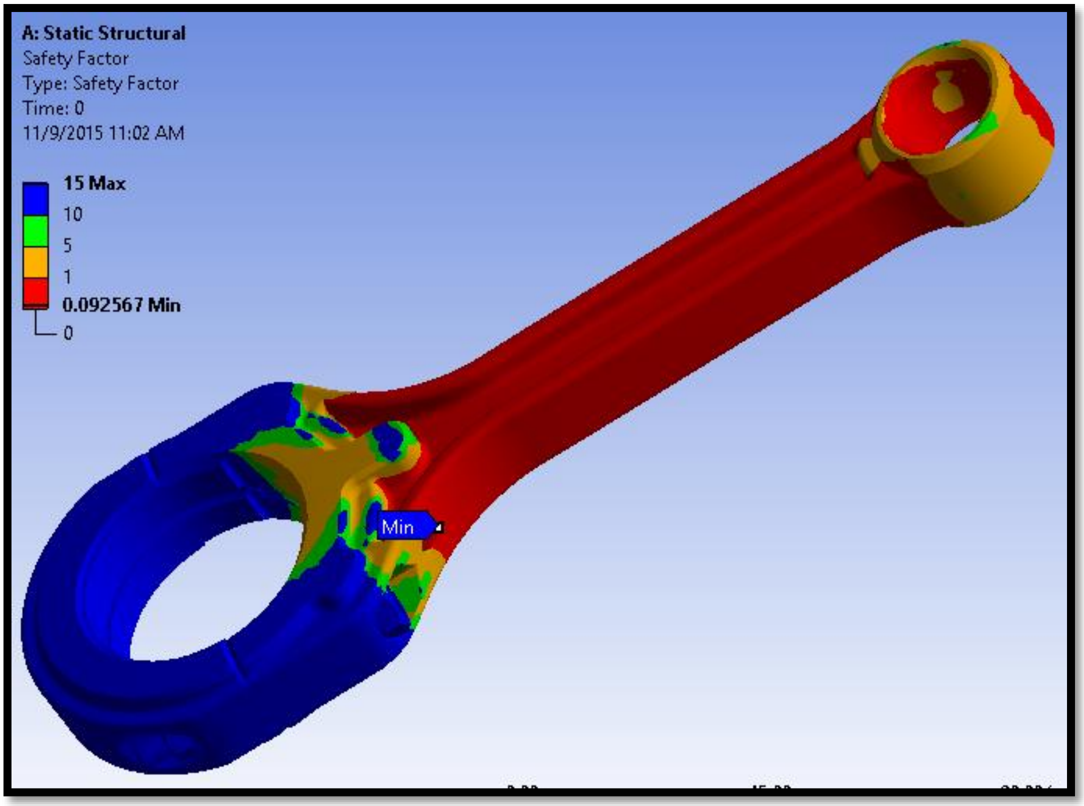



Figure 5.16 FOS fatigue plot for –Y directional axial loading

1. Safety factor for fatigue is 0.09 which is less than 1, risk of failure is very high.
2. General practice is to have safety factor of 2 or above.
3. Further analysis is carried out to increase the CR life to 10E8 and safety factor to 2 or above.

## 5.4 Fatigue redesign for forged steel connecting rod

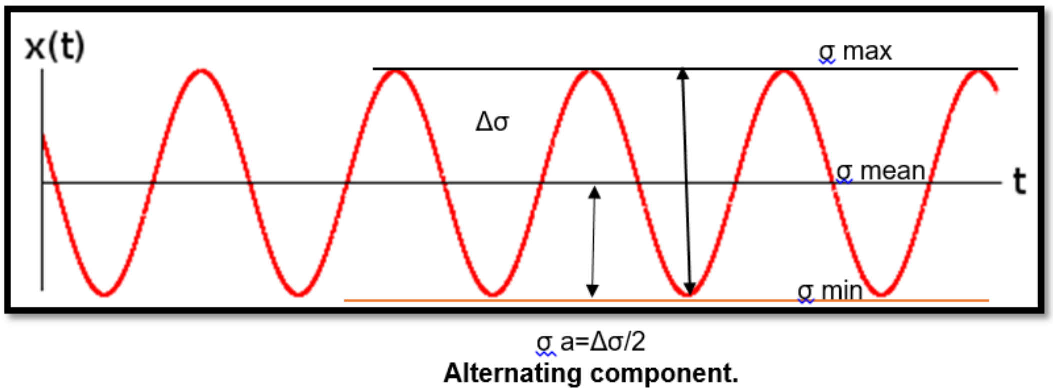



Figure 5.17 Fully reversed case

Fully Reversed case- A case where there is tensile and compressive loading on the connecting rod are the same. According to Yogesh, CR will undergo fully reversed case and further analysis is carried out considering the CR is under tension and compression loading [5].

- σ max- Maximum alternating stress developed= +795.7 MPa
- σ min – Minimum alternating stress developed= - 795.7 MPa
- σ mean- mean of σ min and σ max= 0 MPa.
- Δσ = Total value of stress developed= 770.23 + 770.23= 1.59 E3 MPa
- σ a=Δσ/2= 795.7 MPa.
- Alternating component= σ a= 795.7 MPa (max stress developed)
- Assuming the CR has maximum stress value of 795.7 MPa throughout its life cycle (Worst possible case).

- Analysis is carried out to determine the value of stress to give infinite life and FOS of 2.

General procedure to find value allowable stress which gives material 10E8 life and FOS of 2 [17]

A. Select material- Forged steel
B. Calculate Sm and Se'

According to DR Cyders- Sm=0.75 X Ultimate limit of the material (Sut) for axial case and 0.9 X Sut for bending case.

Since we are dealing with axial case Sm= 0.75 X 572= 429 MPa.

Se' is the endurance limit of the material- Endurance limit is the value of the stress in below which the material will have infinite life.

Se' for forged steel= 0.4 X S ut= 0.4 X 572= 228.8 MPa.

In real life scenario, the material will have lot of manufacturing defects so the corrected endurance limit has to be found out by finding out what are the possible errors that can be found.

C. Calculate the correction factor [18]

According to Shigleys Mechanical engineering design hand book- it is very unrealistic to consider the specimen to have an endurance limit same as the one calculated for lab specimen. These factors vary in real life compared to lab specimen-

Material- Composition, basis of failure.

Manufacturing- Method, heat treatment, fretting corrosion, surface condition, stress concentration.

Environment- Corrosion, temperature, stress state, relaxation times.

Design- Size, shape, life, stress state, speed, fatigue, galling.

Factors- load, size, temperature, reliability and surface finish must be considered.

This imperfection in the real world scenario is calculated using-

Correction Factors= C load X C size X C temp X C reliability X C surface finish

- C load- For axial loading= 0.85
- C size- No size effects for axial loading= 1

- C temperature $kd$ = 0.975 + 0.432(10E-3)$Tf$- 0.115(10E-5)$Tf$^2 + 0.104(10E8)$Tf$^3- 0.595(10E-12)$Tf$^4, Tf= working temperature= 50 C or 122 F

  C temperature- $kd$ = 1.01

- C reliability [32 pg 301]= $ke$= 1- 0.08 $za$

  From the table considering 50% reliability= Za= 0, Ke= 1

- C surface finish [32 pg 295]= $ka$= a Sut ^b

  C surface finish= a(Sut)^-b= 4.51X 572 ^(-0.265)= 0.99.

Correction Factors= C load X C size X C temp X C reliability X C surface finish

=0.85 X 1X 1.01 X1 X 0.99.

Correction Factors = 0.8415

D. Calculate corrected Endurance limit Se= Endurance limit X Correction factor

Se= 228.8 X 0.8415= 192.5 MPa.

E. Draw Stress vs Time (S-N) diagram

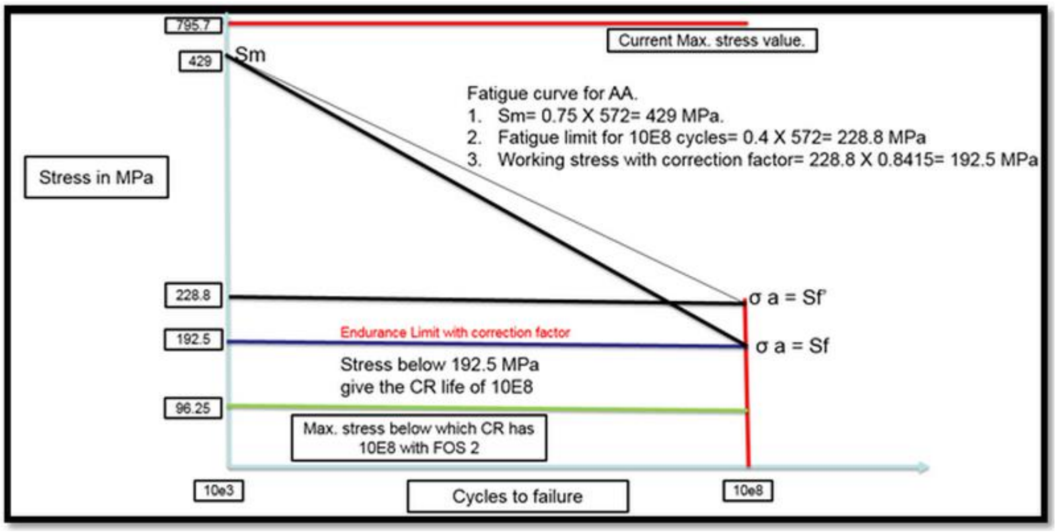



Figure 5.18 Stress vs time graph (S-N graph)

Stress versus cycles to failure graph is plotted. Sm, endurance limit and corrected endurance limit is plugged in the graph. Stress value below 192.5 MPa gives the material 10E8 life.

F. Calculate σ a( Alternating stress) for 10E6 cycle life

Stress value below 192.5 MPa gives the CR 10E8 cycles of life, however it will not give the CR a FOS of 2.

G. Set acceptable σ a and FOS of 2-

To get FOS of 2, divide the corrected endurance limit by 2

σ a and FOS of 2 = 192.5/2= 96.25 MPa.

The above stress value gives the CR 10E8 cycles of life and FOS of 2 and infinite fatigue life.

This satisfies the design guide requirement.

Conclusion and validation

1. Maximum working stress is 96.25 MPa, which is less than yield stress, which is 503 MPa.
2. FOS is 5.2 for static structural axial loading, meets the design guide requirement.
3. Maximum working stress at fatigue is 96.3 MPa which is less than the fatigue Limit with correction factor is 192 MPa.
4. Safety Factor at fatigue is 2, meets the design guide requirement.
5. Maximum stress and poor fatigue cycles occurs at the fillet section of the CR, redesign at this area, by either deleting the fillet section or increase the thickness at that particular site, is highly recommended to reduce stress concentration.
6. CR life is now designed for 10E8 cycles and meets design guide requirements.

By considering all the above factors a robust CR design can be designed.

Chapter 6

# RESULTS AND CONCLUSIONS

Table 6.1 Forged Steel v/s Aluminum 7075

| **Analysis results** | **Forged steel** | **Aluminum 7075** |
|---|---|---|
| Max design pressure | 17.73 MPa | 17.73 MPa |
| Max Inertia Force | 1000 N | 1000 N |
| Material Yield limit | 625 MPa | 503 MPa |
| Max. stress developed | 770.23 MPa | 795.7 MPa |
| Max deformation | 0.000165 m | 0.00017 m |
| FOS | 0.81 | 0.7 |
| Min. fatigue life | 504 cycles | 286 cycles |
| Safety factor | 0.1 | 0.09 |
| Endurance limit with correction factors | 413.5 MPa | 228.8 MPa |
| Max working stress proposal | 154.6 | 96.25 |
| Safety Factor | 2 | 2 |
| Design guide requirement | Met | Met |

From the table it can be observed that

1. AA weights three times less than FS; this material CR is mainly used in aerospace application.
2. FS has very high stress handling capacity without yielding.
3. Deformation is FS is less compared to AA.
4. Also with application of 17.7 MPa pressure and 1000 N inertia force, FS has better values of stress, deformation, FOS, and fatigue life, which is better than AA.
5. AA has no infinite life and fails at 10E8 cycles; FS has infinite fatigue life.
6. Also from manufacturing point of view-

- Manufacturing FS is easier when compared to CNC manufacturing of AA.
- Material thickness for AA is thicker when compared to FS, for same value of BC.
- As the thickness of the CR increases, CR comes in contact with the engine block and crankshaft.

7. By considering all the above factors, one can conclude that FS is better material than AA in terms of stress handling, manufacturability and cost.
8. FS is the best material to be used as a CR material for Ford Eco Boost Mustang.

References

[1] A. Fatemi, J. Williama, and F. Montazersadgh, "Fatigue Performance Evaluation of Forged Steel versus Ductile Cast Iron Crankshaft: A Comparative Study." in FIERF, August 2007, pp 1-31.

[2] "Connecting rod manufacturing." [Online]. Available: http://www.enginelabs.com/engine-tech/connecting-rod-tech-forged-and-billet-steel-rods/

[3] L. K. Vegi, and V.G. Vegi, "Design and Analysis of Connecting Rod using Forged steel," in International Journal of Scientific & Engineering Research, Volume 1, Issue 6, June 2013, pp. 2081-2090.

[4] "Connecting rod failures in 2014 GM vehicles." [Online]. Available: http://blog.caranddriver.com/maliboom-faulty-connecting-rod-bearing-causing-engine-failures-in-gm-four-cylinders/

[5] Y.B. Dupare, R. B. Tirpude, and A. Y. Bharadbhunje, "Fatigue Analysis in Connecting rod using ANSYS," in Scientific Journal Impact Factor, Volume 4, Issue 711, 2014, pp. 215-220.

[6] T.G. Thomas, S. Srikari, and M. L. J. Suman, "Design of Connecting Rod for Heavy Duty Applications Produced By Different Processes for Enhanced Fatigue Life," in SAS Tech Journal, Volume 10, Issue 1, May 2011, pp. 1-7.

[7] "Aluminum alloy connecting rods." [Online]. Available: http://rrconnectingrods.com/aluminum-connecting-rods/

[8] “Aluminum 7075 material properties.” [Online] Available: http://asm.matweb.com/search/SpecificMaterial.asp?bassnum=MA7075T6

[9] “Aluminum alloy and Structural steel connecting rod.” [Online]. Available: https://www.google.com/search?q=googgle+images&ie=utf-8&oe=utf-8#q=google+images

[10] “Mustang Eco boost engine specifications.” [Online] Available: https://lmr.com/products/2015-Mustang-Engine-Specs-23L-EcoBoost-4-Cylinder

[11] V. Lakshmikanth, and A. N. Rao, “Modelling and Analysis of I.C. Engine Piston Crown Using FEM Package Ansys,” in IJRMET, Volume 5, Issue 1, November 2014 – April 2015, pp. 51-59.

[12] “Piston skirt body.” [Online]. Available: https://www.google.com/search?q=googgle+images&ie=utf-8&oe=utf-8#q=google+images

[13] “Introduction to FEM.” [Online]. Available: http://www.sv.vt.edu/classes/MSE2094_NoteBook/97ClassProj/num/widas/history.html

[14] “Static structural analysis.” [Online]. Available: http://mostreal.sk/html/guide_55/g-str/GSTR2.htm.

[15] S. pal, and S. Kumar, “Design Evaluation and Optimization of Connecting Rod Parameters using FEM,” in IJEMR, Volume 2, Issue 6, December 2012, pp. 21-25.

[16] D. G. Gotiwale, and S. D. Ambekar, “Design of Connecting rod for Weigh Reduction using C70S6 Material,” in International Journal of Scientific & Engineering Research, Volume 5, Issue 5, May 2014, pp. 1253-1260.

[17] “Fully reversed fatigue case.” [Online]. Available: https://www.youtube.com/watch?v=bo4TdlQWSY4.

[18] R. G. Budynas, and J. K. Nisbett. (2015). Shigley’s Mechanical Engineering Design, Tenth Edition. New York, NY: McGraw-Hill Education.

[19] “Aluminum alloy connecting rod.” [Online]. Available: https://www.google.com/search?q=googgle+images&ie=utf-8&oe=utf-8#q=google+images

## Biographical Information

Shiva Prasad Keralapura Basavaraju was born in Keralapura, India. He received his Bachelor's degree in Mechanical Engineering from VTU, India in 2012. He completed his Master of Science degree in Mechanical Engineering at the University of Texas at Arlington in December 2015.

He worked at ROUSH as an Engineering Intern with fuel system Product Development team. He was responsible for designing fuel lines and brackets at ROUSH. This Internship motivated him to do research on CAE Analysis of automobile components. He started his research work on Ford Mustang connecting rod which he extended later as a thesis topic.