Formula Student Engine Design

A thesis
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by
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EXAMIANTION COMMITTEE

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Abstract

The Formula SAE ® Series competitions urge teams of university undergraduate students to devise, design, develop , fabricate, and compete with small, formula style vehicles. The purpose of Formula SAE competition is to assume that teams are working for design firm that is designing, fabricating, testing and demonstrating a prototype vehicle for the non-professional, weekend, competition market. The vehicle should have very high performance in term of acceleration, braking and handling and be sufficiently durable to complete all the events.

In static events 150points are allocated for the design event. Purchased items are not evaluated as student designed units. Team with maximum designed components and efficient one is awarded with maximum points and rest of the teams are given point accordingly on relative basis.

Most student teams participating in the Formula SAE competitions just buy the engine, fix it and drive the vehicle. The engine might be compact, powerful and well-toned but it cannot earn the maximum points allocated for the design in FS competition. NBR also participates in the Formula SAE Series Competitions and they are currently using Daihatsu Mira 660cc with three cylinder engine which has more weight and low torque. The engine is designed such that it has high acceleration due to high torque. To produce high torque and to keep weight minimum the engine is kept to be two cylinder. We keep bore to stroke ratio to be 0.8 that makes the stroke length more and allows more time for air/fuel mixture to burn that results in the increase of efficiency of the engine while Daihatsu Mira has a bore to stroke ratio of 0.86. The engine is being optimized by iterative process on SolidWorks and MATLAB.

Preface

NUST Bolts Racing (a team from SMME) participates in the Formula Student Competition organized by Society of Automotive Engineering (SAE). In this competition, undergraduate students have to develop, design and fabricate a small Formula 1 type vehicle by limiting themselves to the set of rules made by SAE. This report includes that how we designed the Formula Student Racecar Engine by limiting ourselves to the rules. This effort is supervised by Dr. Sami Ur Rahman Shah. This report provides the detailed exposition of the tasks and the results that we achieved while finalizing our engine design. The report starts with the background and introduction to our project. We then explained how we calculated the different parameters and then optimized these parameters using MATLAB and Solid Works.

Acknowledgement

We would never be able to complete this project without the help of Allah Almighty. We would like to express our deepest gratitude to Dr. Sami Ur Rahman Shah for his excellent guidance, caring and patience with us.

We would like to pay special thanks to School of Mechanical and Manufacturing Engineering for providing such an excellent environment to carry out our project.

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Chapter 1

1. Introduction

Formula Student is a competition organized by Society of Automotive Engineering (SAE) every year at different countries. The main objective of this competition is to provide undergraduate students the maximum design flexibility and the freedom to express their creativity in designing and fabricating a small Formula style vehicle in order to compete with the teams from the universities around the globe. The main objective of the vehicle design is that the vehicle should have high performance in terms of acceleration, handling, braking and must be durable to compete in all the events of the competition. The participating teams are evaluated in a series of static and dynamic events. The points table for these events is given below.

Static	
Presentation	75
Engineering design	100
Cost Analysis	100

Dynamic	
Acceleration	75
Skid-Pad	50
Auto Cross	150
Efficiency	100
Endurance	300

NUST Bolts Racing (NBR) is a team from SMME that participates in Formula SAE competition. Like most of the teams they bought an engine from the market, fixed it in the vehicle. By doing so, the engine might produce more power but it cannot earn the maximum points for engineering design. Because, the team designing the most of their vehicle itself gets the maximum points and other teams are rewarded points relative to the team getting the maximum points according to the design. There are 75 points for the acceleration event and 100 points are reserved for efficiency. NBR is currently using Daihatsu Mira 660cc engine. The rules of FSAE says that the total swept volume of the engine could not exceeds 610cc, so they are reducing the volume by destocking the engine, this would result in the decrease in acceleration and efficiency and hence the deduction in the marks for the acceleration and efficiency events. The Daihatsu Mira 660cc is a three cylinder engine and has low torque besides it is not suitable for Formula student vehicles. It cannot produce enough acceleration to earn maximum points.

It takes years to design and fabricate an engine. Team from Oxford Brooks University also participates in FSAE competition. They started to design the engine for their vehicle in year 2006 and in year 2014 the engine was first tested on dynamometer and they are still improving the engine. As a part of NBR, we decided to take first step and to carve a way for our upcoming juniors to design the first engine for NBR formula student vehicle.

When we speak of performance, the first thing that comes to mind is the acceleration. The engine that we have designed would be capable of producing a high acceleration due to high torque at low revolutions per minute (rpm). To increase the efficiency we kept the engine under square (i.e. bore/stroke ratio < 1). This means that stroke length is greater than that of Daihatsu Mira 660cc engine and thus it allows more time to burn air/fuel mixture and extracts more energy from the fuel which ultimately increases the efficiency of the engine.

Chapter 2

2. Operating Cycle of Otto Cycle

The commonly fuel available at Formula SAE site is gasoline. Thus, the engine would operate on Otto Cycle. In the rules of FSAE, there is a compulsion that the engine should operate on four stroke cycle. So, we kept it to be four strokes. The steps involved in the operating cycle of four stroke Otto Cycle engine are as follow

1. Intake Stroke:

In this stroke, intake valve opens and the piston comes downward and air/fuel mixture is drawn into the cylinder. When the piston reaches the bottom dead center the inlet valve closes.

2. Compression Stroke:

In this stroke, the piston moves up and the mixture is compressed and thus increase the pressure and temperature of the mixture.

3. Power Stroke:

At the end of the compression stroke, the spark plug fires and the compressed air/fuel mixture ignites. As the fuel burns, it results in the expansion and thus driving the piston downwards.

4. Exhaust stroke:

When the piston reaches at the bottom dead center after power stroke, the exhaust valve opens and piston moves up and drives the exhausted mixture out of the cylinder.

These four processes are explained with the help of diagram as follow

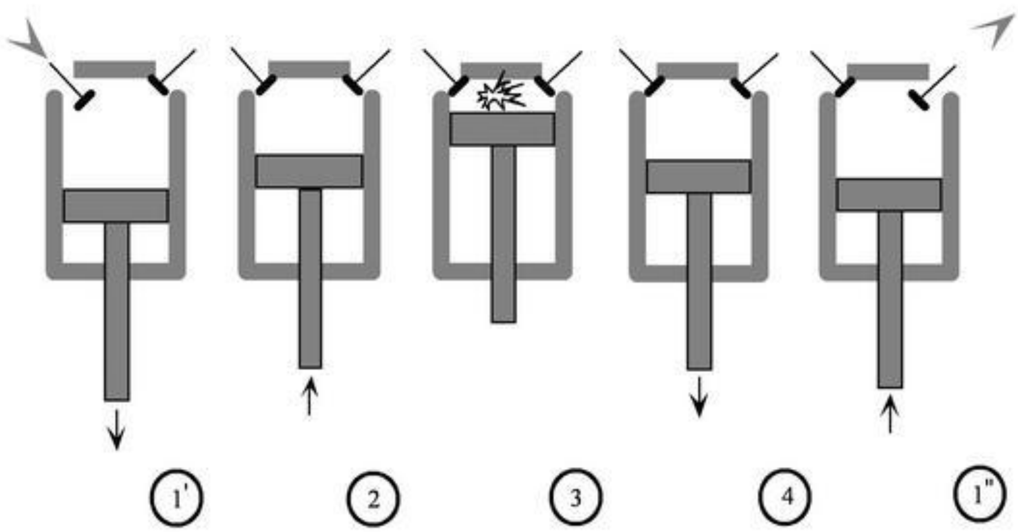

Figure 1: Four Stroke Engine

In the above diagram,

1' is the **Intake Stroke**

2 is the **Compression Stroke**

3 and 4 are the **Power Stroke**

1" is the **Exhaust Stroke**

The PV diagram of the cycle is shown below

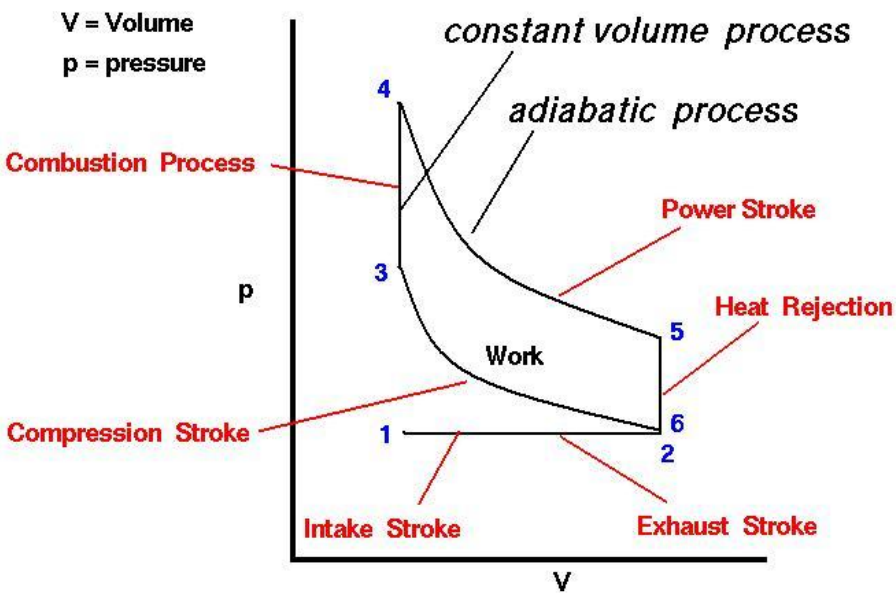

Figure 2 : Otto Cycle

Chapter 3

Literature Review

3.1. Piston

Piston is the one of the main component of the engine. It transfers power from the expanding gases in the cylinder to the crankshaft with the help of connecting rod. The connecting rod is connected between crank and the piston. When the linear motion of the piston is converted into rotational motion, the mechanical losses are experienced. The piston is the main reciprocating part of the engine and due to which it produces imbalance in the engine. This imbalance exhibits itself as the vibration that is very dangerous to the engine.

The energy supplied by the piston is not continuous and its nature is impulsive. This results in the uneven motion of the crankshaft.

Usually, the piston is made up of cast aluminum alloy for excellent thermal conductivity [1]. Thermal conductivity is the ability of any material to conduct and transfer the heat. Aluminum expands when subjected to heat, thus, proper clearance must be given between cylinder and the piston to provide proper movement of the piston between cylinder walls. Insufficient clearance can lead the piston to seize between the cylinder walls while excessive clearance can cause loss of compression.

Piston can be divided into the following parts

a. Piston head
b. Piston pin bore
c. Piston pin
d. Skirt
e. Ring groves
f. Ring lands
g. Piston rings

3.1.1 Piston head

It is the top surface of the piston which faces huge amount of forces and heat during the operation of the engine.

3.1.2 Piston pin bore

It is a through hole in the sides of the piston that is perpendicular to piston travel that has the piston pin.

3.1.3 Piston pin

A piston pin is the hollow shaft that connects the connecting rod and the piston.

3.1.4 Skirt

The skirt of the piston is the portion of the piston that is closest to the crankshaft that aligns the piston as the piston reciprocates between the cylinder walls. To reduce the piston weight and to provide clearance for the counterweights of rotating crankshafts some profiles are cut into the skirt.

3.2 Piston Rings

Ring groove

It is a recessed area which is located around the perimeter of the piston that is used to hold a piston ring.

Ring lands

These are two parallel surfaces of the ring groove which acts as the sealing surface for the piston rings.

A piston ring is expandable split ring which is used to seal the piston and the cylinder walls. The piston rings are usually made up of cast iron to maintain the integrity of its original shape under heat, load and other dynamic forces. The purpose of piston rings is to seal the combustion chamber (i.e. Cylinder and the piston), conducts heat from piston to the cylinder walls and returns the oil to the crankcase [2].

There are usually three piston rings

Compression ring

Wiper ring

Oil ring

3.2.1 Compression ring

It is located in the ring groove that is closest to the piston head. It is used to seal the combustion chamber in order to avoid any leakage during combustion. When sparkplug ignites the air-fuel mixture inside the cylinder, pressure rises due to combustion of gases forcing the piston downwards. These pressurized gases travel via the gap present between the cylinder wall and the piston and into the piston ring groove. Pressure due to the combustion gas pushes the piston ring against the cylinder wall to form a seal.

3.2.2 Wiper ring

This piston ring has a tapered face and is located in the ring groove between the compression and the oil ring. The wiper ring further seals the combustion chamber and wipes the cylinder wall cleans of excess oil. Combustion gases escaping the compression ring are ceased by the wiper ring.

3.3.3 Oil Ring

An oil ring is located in the ring groove which is closest to the crankcase. The oil ring wipes excess oil from the cylinder walls during piston motion. There are small openings through which excess oil is returned to the oil reservoir present in the engine block. The purpose of Piston rings is to seal the combustion chamber, to transfer heat to the cylinder wall and to control oil consumption.

The sealing of the combustion chamber take place through inherent and applied pressure. Inherent pressure is the internal force that stretches the piston ring. A significant force is required for inherent pressure in order to compress a piston ring to a smaller diameter. Inherent pressure is determined by the free piston ring gap. The distance between the two ends of a piston ring in an uncompressed state is known as free piston ring gap. The greater the free piston ring gap, the more force is required to compress the piston ring in the cylinder bore. To get an effective seal a piston ring must provide a positive radial fit between the cylinder wall and the running surface of the piston ring.

In addition to inherent pressure, a piston ring requires applied pressure to seal the combustion chamber. Applied pressure can be defined as the pressure applied from combustion gases to the piston ring in order to expand the piston rings. In some piston rings there is a chamfered edge present opposite the running surface of piston. This chamfered edge twists the piston ring when it is not affected by combustion gas pressures.

The piston must withstand pressure variations, thermal stress, and other loads as it acts as the movable end of the combustion chamber. Most pistons are usually made from die- or gravity-cast aluminum alloy. Cast aluminum alloy is lightweight and has low manufacturing costs [1]. The light weight of aluminum reduces the overall weight along with the force necessary to initiate and maintain piston acceleration. This permits the piston to use more of the force produced due to combustion of gases to power the application.

3.3 Connecting rod

The connecting rod is a link inside between the piston and the crankshaft and transfers power from the piston to the crankshaft and send it further to the transmission. There are different types of materials and methods used in the manufacturing of connecting rods. The most common material used for connecting rods are steel and aluminum [1]. The most common type of manufacturing processes for connecting rod are casting, forging and powdered metallurgy.

When an engine fails, the most common cause is connecting rod failure. It is under a tremendous amount of load and is needed to handle with special care to make sure that it does not collapse prematurely. The sharp edges of connecting rod are sanded smooth to reduce stress points on the connecting rod. The connecting rod is also hardened to increase its strength against cracking. For high-performance applications, usually the connecting rod is balanced so that it can be prevented from creating excessive wear.

The most common method to manufacture connecting rod for vehicle engines is by casting. This type of rod is the least expensive to manufacture. The cast rod has been used in nearly every type of engine, either it is a gasoline or diesel.

3.4 Crankshaft

The crankshaft converts reciprocating linear piston motion into rotation. For the purpose of converting the reciprocating motion into rotation, the crankshaft has crankpins and additional bearing surfaces whose axis is offset from that of the crank, to which the connecting rod from each cylinder attach.

Crankshaft is connected to a flywheel, to reduce the fluctuating characteristic of the four-stroke cycle and sometimes a damper at the opposite end to absorb the torsion vibrations caused along the length of the crankshaft due to the cylinders far away from the output end acting on the torsion elasticity of the material [3].

Usually we use three types of crank configuration i.e. 360°,180° and 270°.

- In an engine having crank configuration of 360°, both pistons rise and fall together. The firing order of this engine is such that cylinder 2 fires after 360° of cylinder 1, again after 360°, cylinder 1 fires again making total of 720°. This indicates the beginning of next four stroke cycle.
- In an engine having configuration of 180°, one piston is going down as other piston s rising up. This provides a good primary balance but results in irregular ignition pulses. This is because cylinder 2 fires after 180° of the cylinder 1 and cylinder 1 does not fire for another 540^o of the rotation.
- For 270^o configured engine, one piston has to follow 3/4 of the rotation behind the other piston. The firing order is such that the cylinder 2 fires after the 3/4th of the rotation i.e. (270^o of the rotation) of cylinder 1 and cylinder 1 fires again after one and a quarter of rotation (i.e. after 450^o).

This part of engine is usually made up of very heavy cast iron in most cases. The crankshaft's face must be made very strong to hold the stress of placing the crankshaft pulley and the stress produced from driving all of the components off of that single pulley.

3.5 Camshaft

One can say that camshaft is the “brain” of the engine. This is because it opens and closes the valve at just the right time during engine rotation, in order to obtain the maximum power and efficient cleanout of exhaust. The camshaft operates the distributor to electrically synchronize spark ignition. Camshafts do their job via eccentric "lobes" that activate the components of the valve train. The camshaft is produced from forging a single piece of steel, on which the lobes are grounded. On single-camshaft engines there are twice number the lobes than number of cylinders, in addition to a lobe for fuel pump actuation and a drive gear for the distributor. The camshaft is driven by the crankshaft, through a set of gears or a chain or belt. It takes two full rotations of the crankshaft to complete one rotation of the cam and to complete a four-stroke cycle. The camshaft operates the cam-followers that further operate the rest of the valve train. Cam-followers or lifters can be of several types. The most commonly used are hydraulic, mechanical and roller lifters. *Hydraulic lifters* are filled with oil and act as a shock absorber in order to eliminate clearance in the valve train. They don't require any periodic adjustment and they are quiet. *Mechanical lifters* require scheduled adjustment in order to get proper valve clearance. These are used in high-rpm applications. *Roller lifters* have a roller device at one end and they can either be hydraulic or mechanical [4].

There is a point during crank rotation when both the intake and exhaust valves are open simultaneously, this point is known as *Overlap*. This point comes at the end of the exhaust stroke when the exhaust valve is closing and the intake valve is opening.

Several factors are responsible for how much overlap is ideal for the engine. Small combustion chambers typically require small overlap. When the rpm increase, the intake valve opens for a small period of time. The same amount of air/fuel mixture must be pulled into the combustion chamber in short time, and the engine uses all the help it can get to fill the chamber.

The amount of time (in terms of degrees of rotation of the camshaft) that the lobe holds the valve off its seat is known as *duration*. Duration is the most important thing to consider when choosing a camshaft. The point at which the intake valve opens is critical to run an engine properly.

If intake valve opens too early, exhaust gases can get into the intake manifold. This causes soot growth on the intake runners, low engine vacuum and hence low power. If the valve opens too late, less air/fuel mixture gets into the combustion chamber and exhaust gases won't be as effectively removed.

If the exhaust valve closes too early then some exhaust gases can get trapped in the cylinder and reduces the efficiency of engine. If the valve closes too late an enormous amount of fuel/air mixture will escape into the exhaust manifold and the combustion chamber will not be optimized.

The commonly used material for the manufacture of cam shaft is chilled or forged cast ion [1].

Chapter 4

4. Engine Parameters

4.1.Bore and Stroke length of cylinder

The bore and stroke length can be calculated by the bore to stroke ratio. Generally the range for bore to stroke ratio is from 0.75 to 1.5 [2]. We keep this ratio to be 0.8 to have more combustion efficiency. With large stroke length the combustion time of air-fuel mixture increases. The under-square engine makes maximum torque at lower rpm. The total swept volume of both cylinders is 600cc so it makes 300cc for each cylinder. The calculations of bore and stroke are as follow

$$\frac{B}{S} = 0.8$$

$$B = 0.8 * S$$

We know that swept volume of each cylinder is, V_s = 300cc and

$$V_s = \frac{\pi}{4} B^2 \mathrm{S}$$

Therefore,

$$V_s = \frac{\pi}{4} (0.8 * S)^2 \mathrm{S}$$

$$V_s = \frac{\pi}{4} (0.8)^2 \mathrm{S}^3$$

$$300 = \frac{\pi}{4} (0.8)^2 \mathrm{S}^3$$

$$S^3 = \frac{4 * 300}{\pi * (0.8)^2}$$

$$S = 8.42\ cm$$

and

$$B = 0.8 * 8.42$$

$$B = 6.74\ cm$$

4.2.Crank Radius

Suppose the crank radius is a. When the piston moves one stroke then the crank moves $2a$. Thus, the crank radius is half of the stroke length. [3]

$$2a = S$$

$$a = \frac{S}{2}$$

$$a = \frac{8.42}{2}$$

$$a = 4.1cm$$

4.3.Connecting rod length calculations

Connecting rod is connected to the piston by means of the small end. We used *connecting rod ratio* to calculate the centre length *l* of the connecting rod. Connecting rod ratio is the ratio between crank radius and connecting rod centre length and is given by $\delta = \frac{a}{l}$.

δ lies between 0.23-0.33 [4] and we also found that the design height increases with the increase in the ratio. If the connecting rod is small then center of gravity of the vehicle will be low but there is a problem that the side load would increase on the piston which may result in the bursting of cylinder. So, we keep δ small and hence the connecting rod length large so that the side pressure on the cylinder would be small and hence more safety as the requirement of FSAE rules. We keep $\delta = 0.23$ and the connecting rod length is calculated as

$$\delta = \frac{a}{l}$$

$$l = \frac{a}{\delta}$$

$$l = \frac{4.1}{0.23}$$

$$l = 18.0\ cm$$

4.4.Compression ratio

The fundamental specification for many engines is *compression ratio.* It is defined as *the ratio between the volume of the cylinder and the combustion chamber when the piston is at the bottom, and the volume of the combustion chamber when the piston is at the top.* The fuel economy can be increased by using higher compression ratios. When the compression ratio is high, the air will be compressed more in the cylinder. [4] With the compression of air we get more powerful explosion from the fuel-air mixture. The higher compression ratio makes the engine to get more power from the fuel. So, higher compression ratio is desired in the engine. The higher compression ratio has its draw backs too. When the engine run with low octane fuel with higher compression ratio then it causes *knocking*. The engine knocks when the fuel-air combustion does not happen at the optimal time in the combustion stroke. With low octane fuel and higher compression ratios, heat density of the compressed air increases and causes the fuel to begin combustion prior to ignition from the spark plug which results into undesirable burn pattern and causes the engine to knock. The knocking appears to be a pinging sound and causes severe damage to the engine.

The compression ratios used in the petrol engines for racing is generally between 10- 14 and for simple petrol engines it compression ratio is between 6-10 to avoid engine knocking. The compression ratio for the petrol engines which are pressure charged are no more than 10.5. Since, our engine would be turbo charged, so we used compression ratio of 10. By using this compression ratio, engine knocking could be avoided and more energy would be utilized from the fuel. [4]

4.5.Revolution per minutes (RPM) of engine

The mean piston speed at maximum rpm is usually kept between 20m/s to 25m/s of petrol engine. Our desired maximum engine speed is 7000-7500rpm. Using this as reference maximum engine rpm is calculated as follow.

$$V_p = \frac{2NS}{60}$$

where N is rpm, S is stroke length and V_p is mean Piston Velocity at peak engine rpm [3]

$$N = \frac{30V_p}{S}$$

$$N = \frac{30 * 20}{0.084194}$$

$$N = 7126\ rpm$$

So, calculated speed is within the desired speed.

Chapter 5

5. Kinematic and Kinetic calculations

5.1 Kinematic calculation for engine

Up till now we have calculated the different parameters of the engine, they are stated below

$B\ =\ \ 6.74\ cm$

$S\ =\ 8.42\ cm$

$l\ =\ 18.0\ cm$

$N =\ 7126\ rpm$

In order to calculate angular velocity ω we will use the following formula

$$\omega = \frac{2 \times \pi \times N}{60}$$

$$\omega = \frac{2 \times \pi \times 7126}{60}$$

$$\omega =\ 746\ rad/s$$

The position of the piston at any time interval is given by

$$S_x = R + \frac{R^2}{4l} - Rcos\theta + \frac{R^2 cos2\theta}{4l}$$

where, $\theta = w \times t$ is the crank angle [3] . The position of piston at different time intervals is shown below with the help of a graph obtained on MATLAB

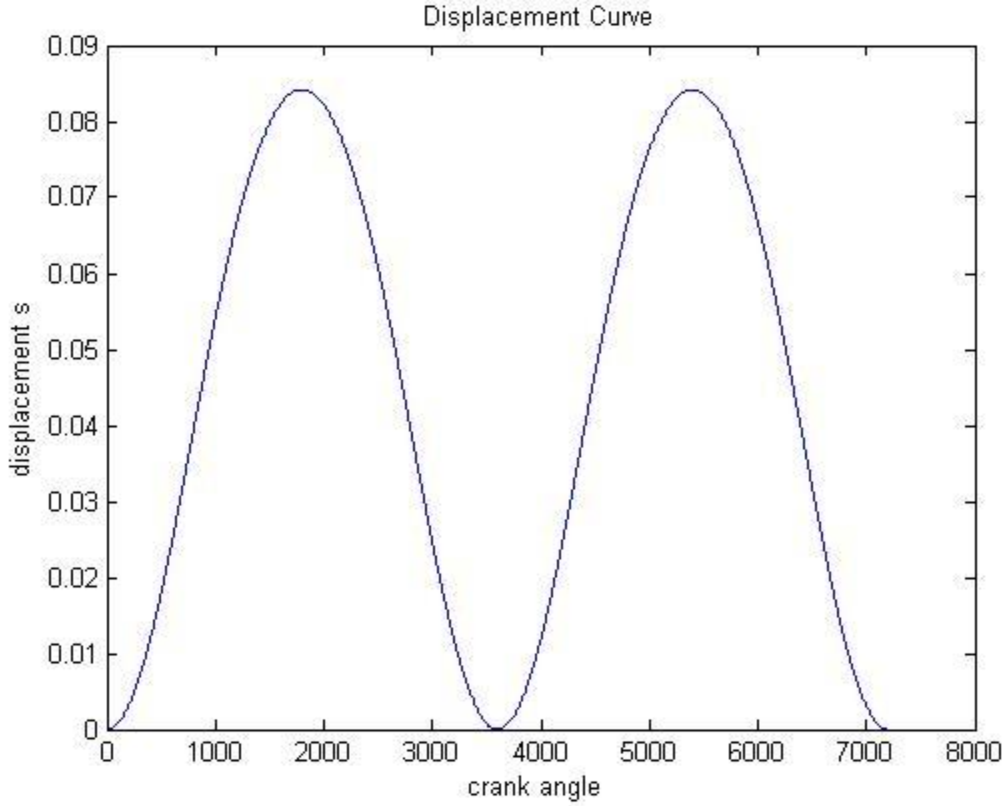

Figure 3: Displacement of piston with respect to crank angle

Velocity of the piston is given by the following formula [5]

$$V_x = R \times w((\sin(\theta) + \frac{\mathrm{R}}{2\mathrm{l}} \times \sin(2\theta))$$

where, $\theta = w \times t$ is the crank angle. The crank angle is changing at every time interval. So, we get a plot of velocity on MATLAB shown below

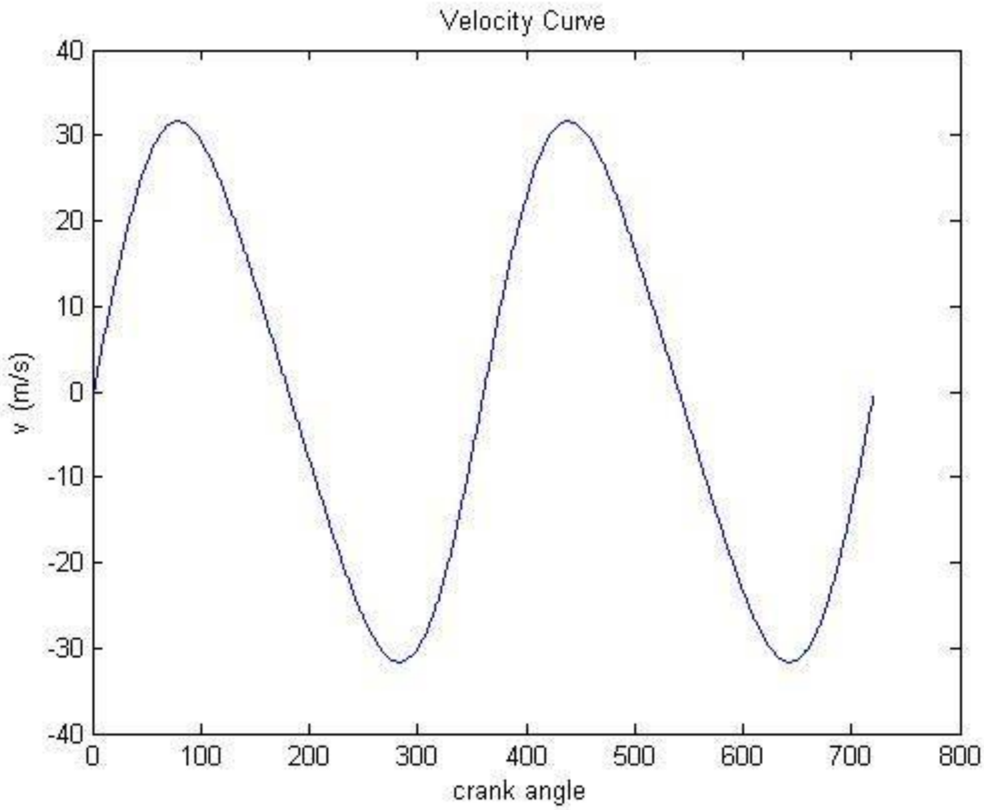

Figure 4 Velocity of Piston with respect to crank angle

The **acceleration** of the piston at any crank angle is given by

$$A_x = R \times w^2((\cos(\theta) + \frac{R}{l}\cos(2\theta))$$

where, $\theta = w \times t$ is the crank angle [6]. The crank angle is changing at every time interval. So, we get a plot of acceleration against crank angle on MATLAB shown below.

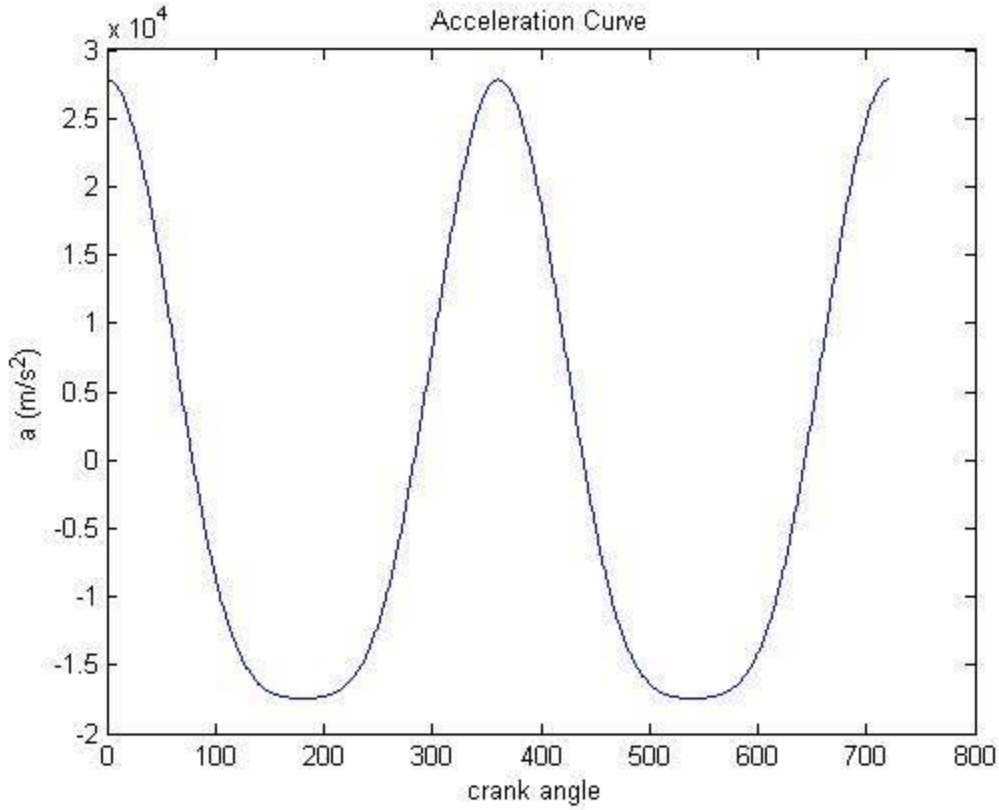

Figure 5 Acceleration of the piston with respect to crank angle

5.2 Kinetic Calculations

5.2.1 Gas forces

The gas forces are a function of crank angle. The equation for analytical calculation of gas force is given as

$$F_g = P_h \times \left(\frac{S_h + S_c}{S_x + S_c}\right)^n - P_{np}$$

Where,

P_{np} is the pressure taken on other side of the piston. For four stroke engine, it is 0.1 MPa

P_h is initial pressure

S_h is the working stroke of the piston

S_c is the motion which is consistent with the height of cylinder

and n is the index of politropata process whose value depends upon the crank angle θ.

$\boldsymbol{\theta}$	$\boldsymbol{n}$
0-180	0
180-360	1.375
360-540	1.25
540-720	0

and

$S_x = \left(1 + \frac{1}{\delta}\right) - \left(\cos(\theta) + \frac{1}{\delta}\cos(\beta)\right) \times \frac{B}{2}$ is the current section of stroke length and it depends on the angle of deflection of connecting rod β.

The relation between change in angle of the crank and gas force is shown with the help of the graph that is plotted with the help of MATLAB.

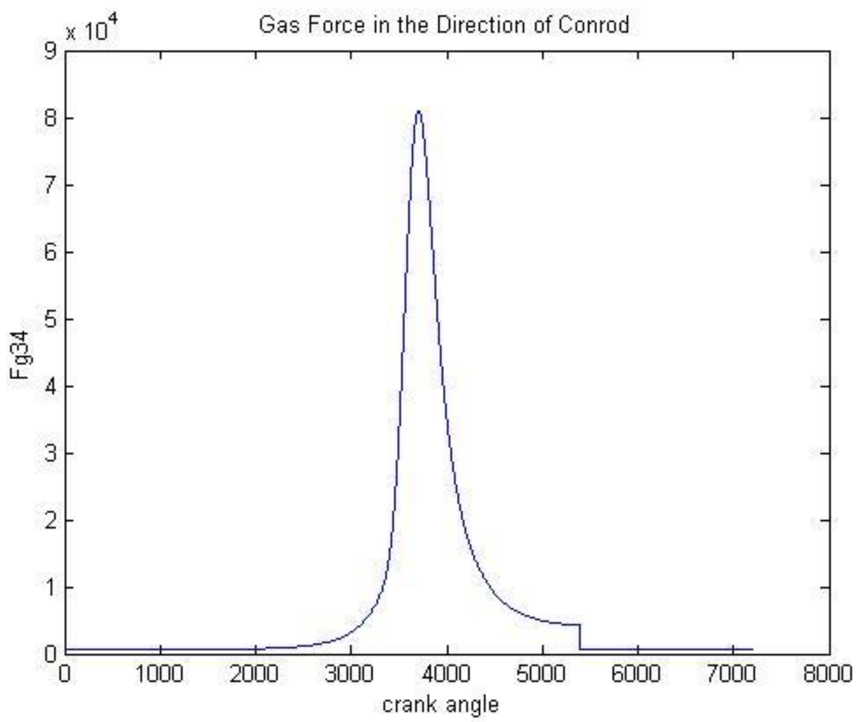

Figure 6: Gas force at different crank angle

5.2.2 Inertia Forces

The inertial forces of mass having linear motion are expressed as the function of crank angle and change in deflection of connecting rod.

$$F_j = -m_j \times \omega^2 \times \frac{B}{2} \times \left(\frac{\cos(\theta + \beta)}{\cos(\beta)} + \delta \times \frac{\cos^2(\theta)}{\cos^2(\beta)} \right)$$

where, ω is the angular velocity and m_j is the mass of particle having linear motion

$$F_R = -m_R \times \omega^2 \times \frac{B}{2}$$

where m_R is the mass of particle having rotational motion.

We have to calculate the mass that have linear and rotational motion. The masses having linear motion are the mass of pistonm_p and the mass of the rod m_d aligned to the piston pin which is the 25% of the mass of connecting rod. While masses having rotational motion are mass of crank m_c and mass of connecting rod which is aligned with the axis of crank m_{d2}.

$$m_j = m_p + m_d$$

$$m_R = m_c + m_{d2}$$

In our case,

$$m_p = 0.35\ kg$$

Mass of connecting rod is 0.50 kg therefore

$$m_d = 0.112\ kg$$

and

$$m_{d2} = 0.338\ kg$$

$$m_c = 0.55\ kg$$

Thus,

$$m_j = 0.462\ kg$$

$$m_R = 0.888\ kg$$

The inertial forces are changing with every rotation of crank and is shown with the help of graph plotted in MATLAB.

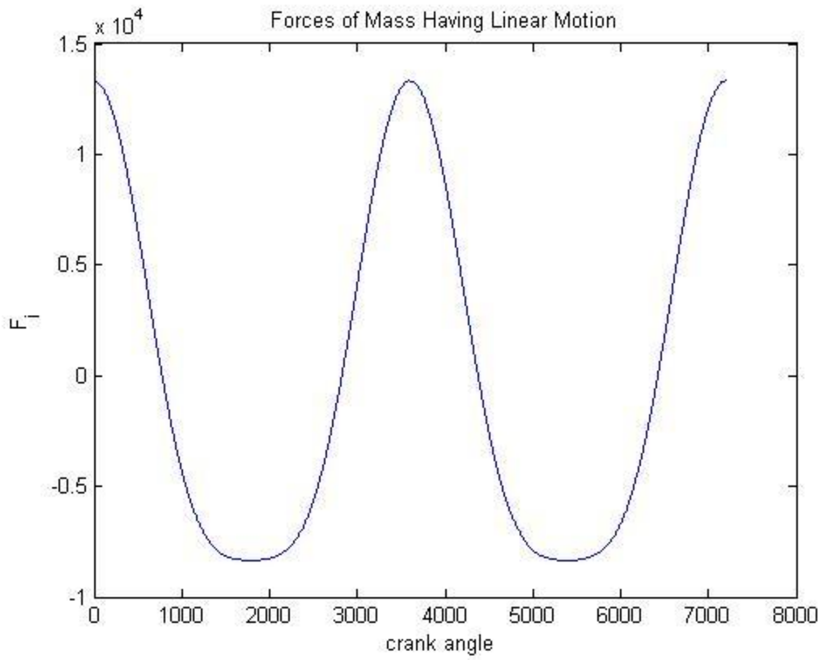

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Figure 7: Force of masses having linear motion at different angles of crank

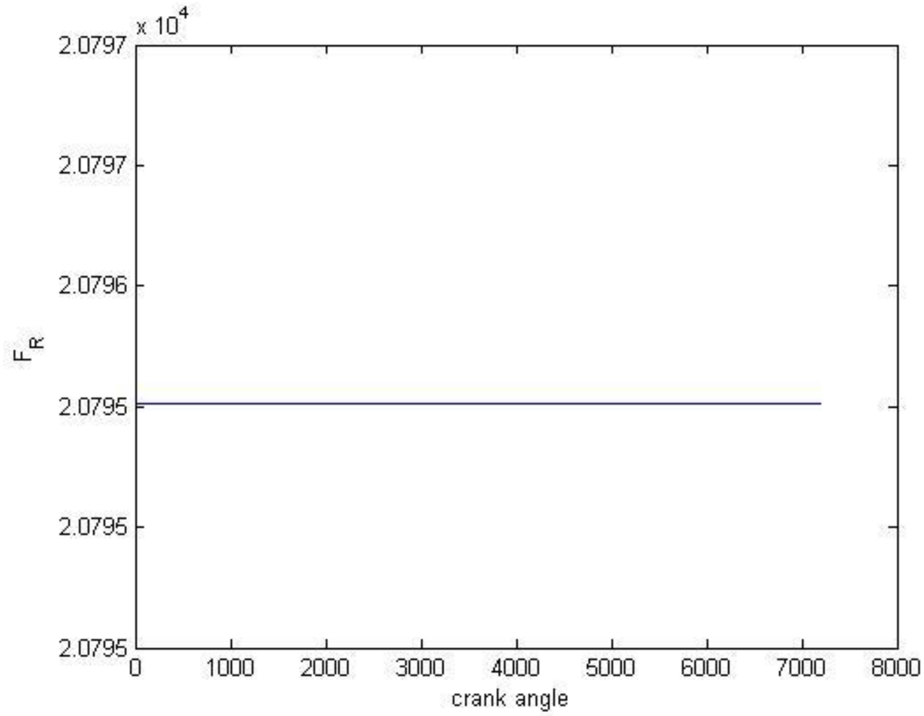


Figure 8: Forces of masses having rotational motion

5.2.3 Forces on Crank-Connecting rod mechanism

The total force on piston at any point of the rotation of crank is given by

$$F_z = F_j + F_g$$

The total force on the piston is shown with the help of graph generated on MATLAB.

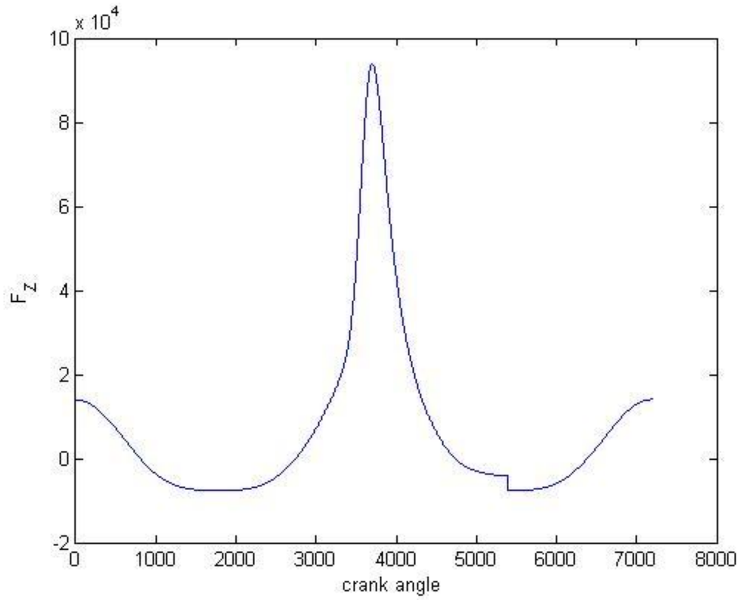


Figure 9: Total Force on the piston at different crank positions

The forces that are acting on the crank and connecting rod mechanism are [3]

- Static Force F_s
- Force acting along the axis of the rod F_a
- Tangential force F_T

These forces are calculated as follow

$$F_s = F_z \tan(\beta)$$

$$F_a = F_z \times \frac{1}{\cos(\beta)}$$

$$F_T = F_z \times \frac{\sin(\theta + \beta)}{\cos(\beta)}$$

We can see all these forces at any rotation with the help of the plot generated in MATLAB.

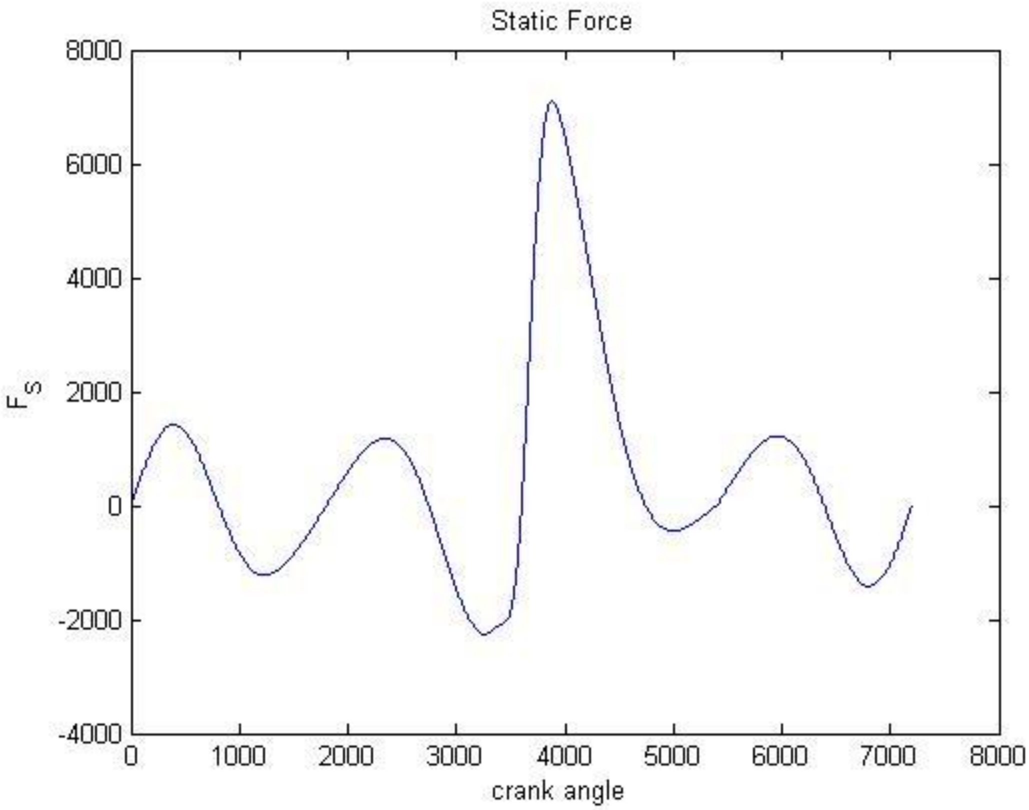


Figure 10: Static force on crank-connecting rod mechanism

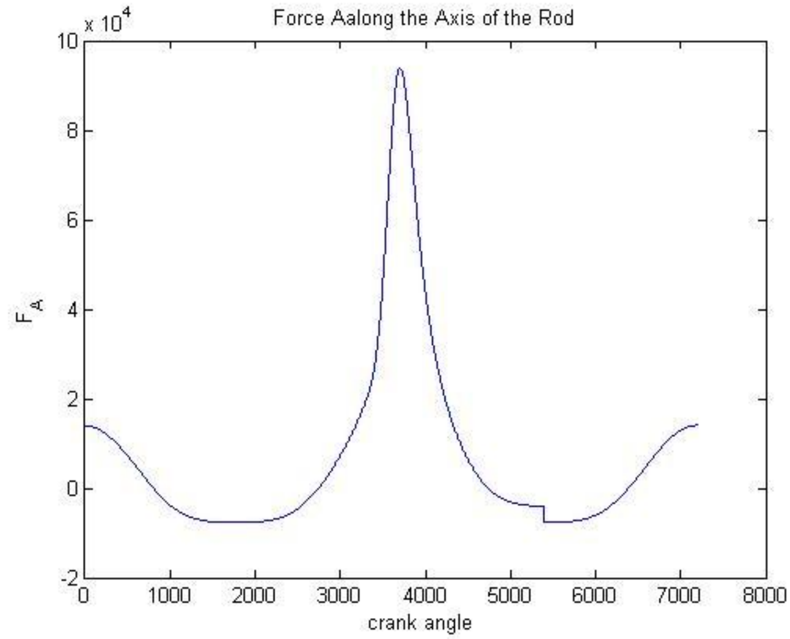


Figure 11:Force acting along the axis of connecting rod at different crank angles

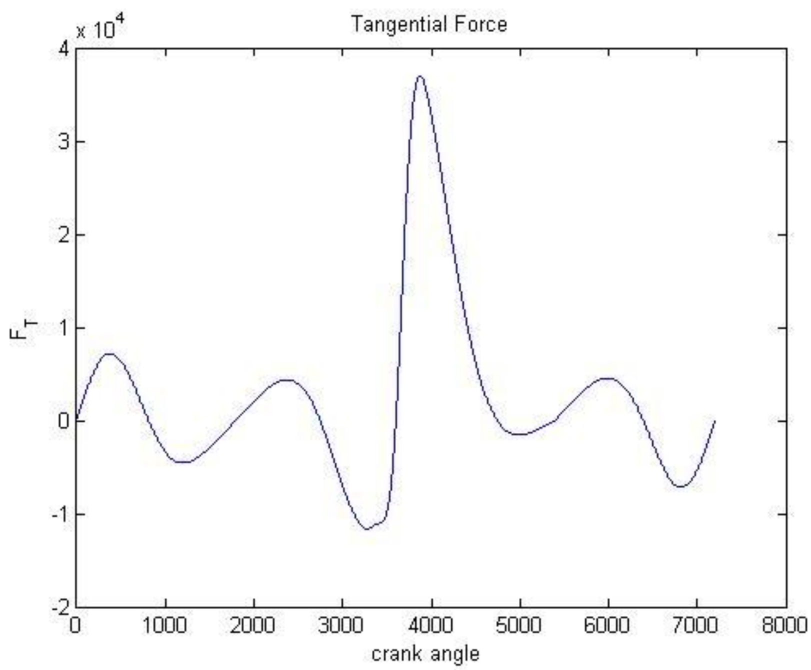


Figure 12:Tangential force on crank-connecting rod mechanism

MATLAB Simulation to calculate pressure and temperature:

In order to calculate pressure and temperature at every point during the process, we used cold air standard assumptions. The calculations are given below.

Intake Stroke

At the intake, the temperature is taken as the standard temperature for petrol i.e. 298 K. Similarly, the pressure is taken as the standard atmospheric pressure but due to the turbocharger we are multiplying it by 2 i.e. 202650 Pa. The temperature and pressure remains constant during the intake

$$T_1 = 323\ K$$

$$P_1 = 202650\ Pa$$

In order to calculate the mass of air/fuel mixture entrapped in the cylinder, we use the following relation [7].

$$m_{ent} = \frac{P_1 \times V_1}{(R \times T_1)}$$

The crank angle is changing and the piston is continuing its up and down motion so the mass entrapped in the cylinder is changing at every point. So, we created a MATLAB code in order to find the relationship between mass entrapped in the cylinder and the crank angle as shown in the figure below.

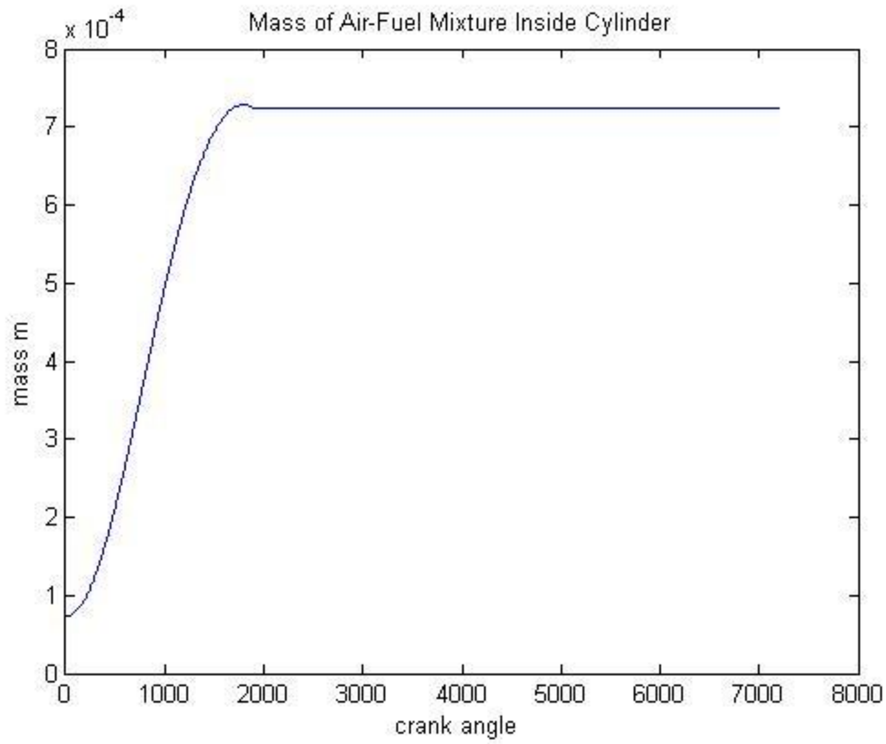


Figure 13:Intake of fuel/air mixture with respect to crank angle

The mass of air/fuel mixture burnt at every rotation of crank during the cycle is shown in the figure below

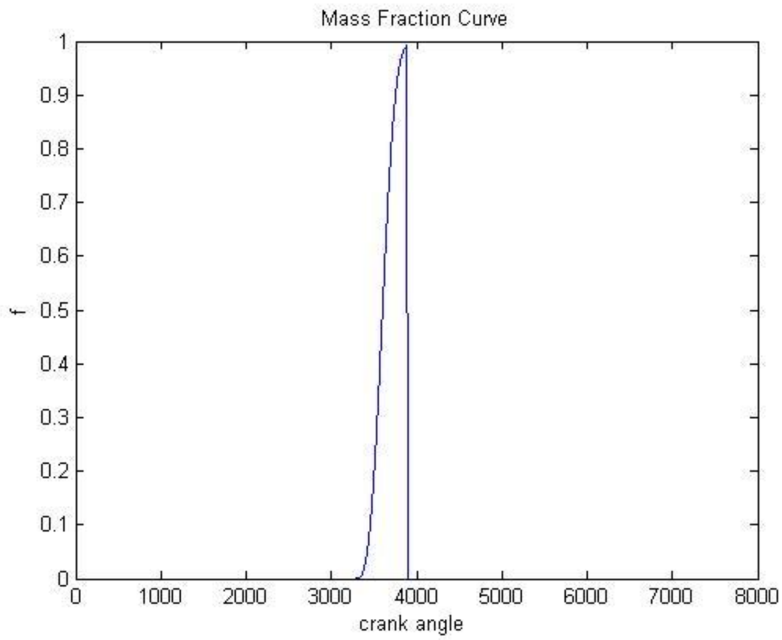


Figure 14:Mass of fuel/air mixture burnt at different position of crank

Compression Stroke

During compression stroke the temperature and the pressure are calculated with the help of Otto cycle

$$T_2 = T_1 \times \left(\frac{V_1}{V_2}\right)^{\gamma-1}$$

and Pressure can be calculated as

$$P_2 = \frac{m_{ent} \times R \times T_2}{V_2}$$

During the compression there will be the loss of heat. This loss of heat is given by the relation as follow [8]

$$h_{loss} = 130 \times {V_2}^a \times (\frac{P_2}{1 \times 10^5})^b \times {T_2}^c \times \left(v_p + 1.4\right)^b$$

where,

v_p is the mean piston velocity

a is constant and has value 0.8

b is constant having value -0.4

c is constant having value -0.06

The heat transfer from the walls of the cylinder during the compression stroke is given by

$$dqp = -h_{loss} \times S \times B \times \pi \times (T_2 - T_{CW}) \times \frac{60}{N \times 1000}$$

Since the angle of crank is changing, therefore for every change in angle h_{loss} and dqp are shown with the help of graphs generated by using MATLAB.

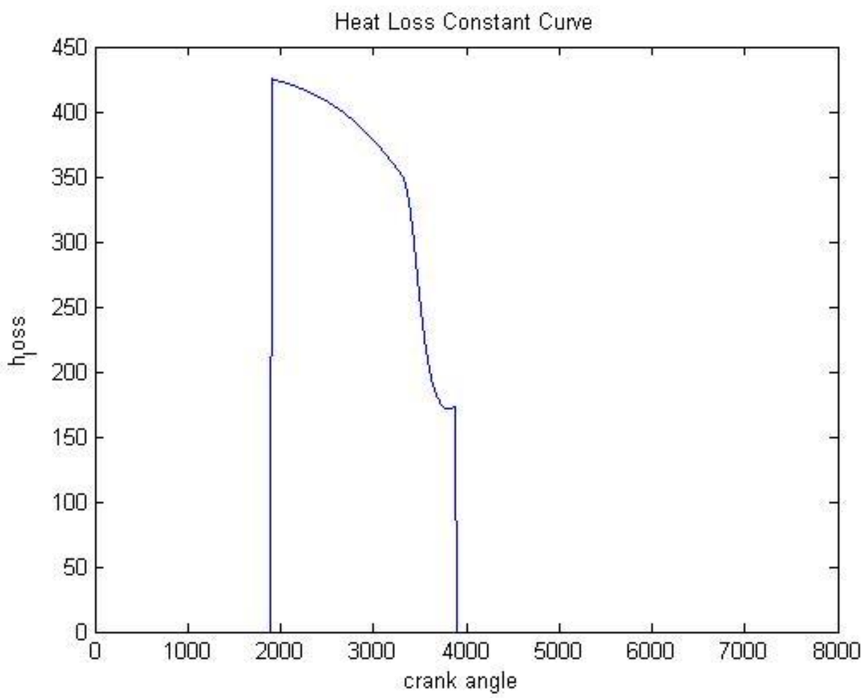


Figure 15:Heat loss constant according to Hohenburg with respect to crank angle

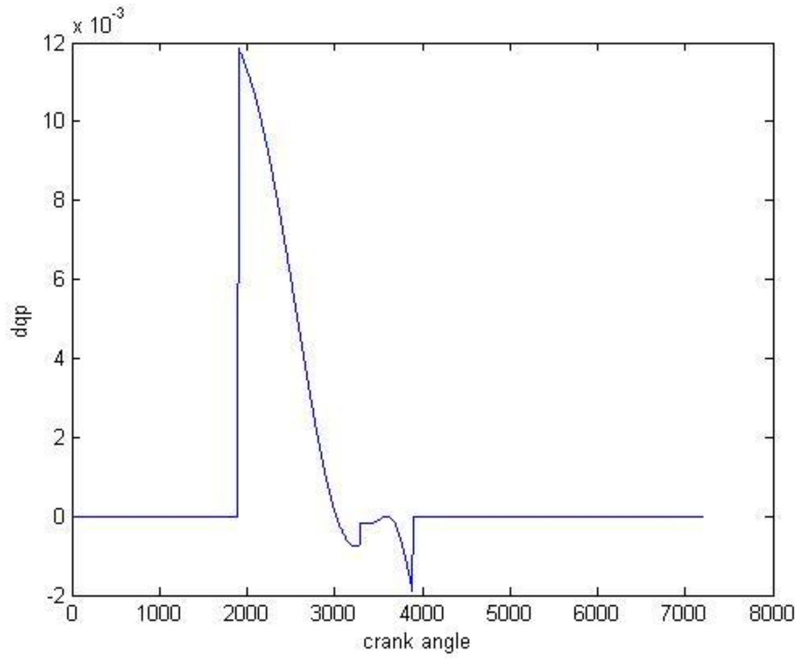


Figure 16:Change in heat during the cycle

Power Stroke

When piston reaches the top dead center at the end of the compression stroke, the spark plug fires and the air/fuel mixture ignites which increases the pressure. This increased pressure pushes the piston towards bottom dead center. The working fluid expands isentropically and work is done by the system on the piston. The pressure and temperature changes with the addition of heat and expansion of the working fluid which can be calculated as follow

$$T_3 = T_2 \times \left(\frac{V_2}{V_1}\right)^{\gamma-1}$$

$$P_3 = \frac{m_{ent} \times R \times T_3}{V_3}$$

Exhaust Stroke

The heat is removed in a constant volume process to an idealized sink source as a result of which the pressure drops to the state 1 (i.e. pressure at intake). As the piston moves up towards the top dead center, the exhaust valve open and the burnt mixture and gases leave the cylinder. The pressure and temperature during this stroke are given as follow

$$T_4 = \mathrm{cons}tant$$

$$P_4 = P_1$$

Pressure and Temperature curve

The pressure and temperature changes throughout the cycle. The following curves indicates the pressure and temperature at every angle of rotation. These curves are obtained from MATLAB

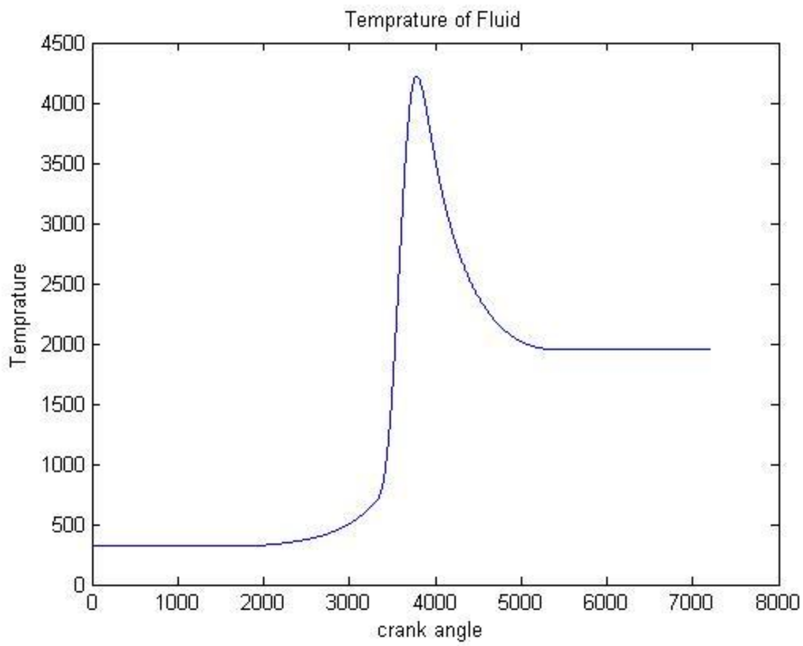


Figure 17:Temperature of the fuel/air mixture during the cycle

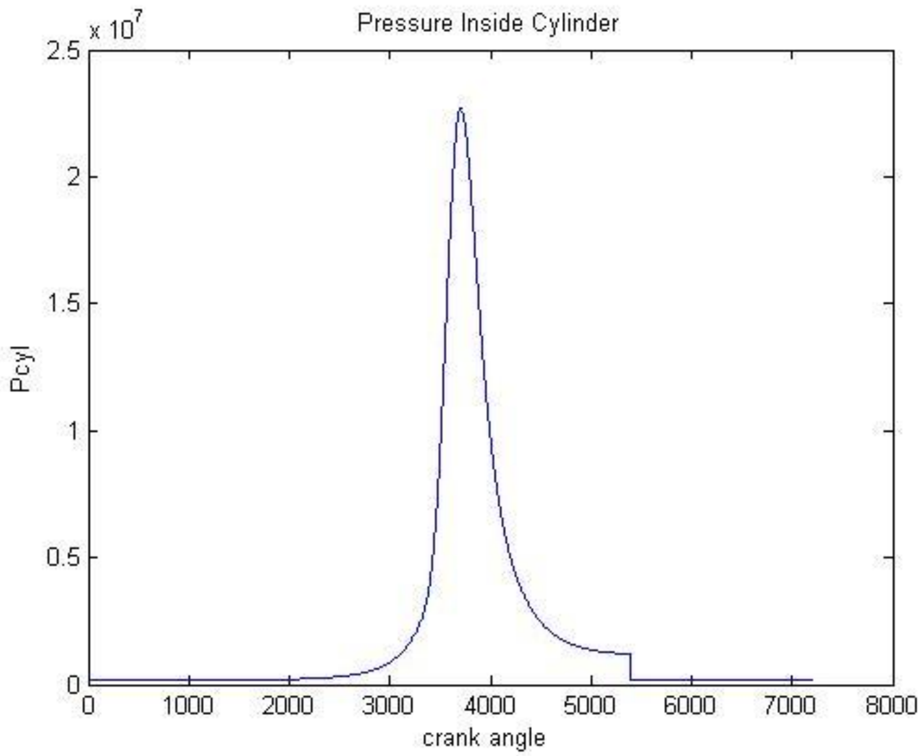


Figure 18: Pressure change during the Cycle

Engine output

Engine output has two things torque and power. Torque is the measured quantity of the engine output while power is dependent on the torque produced by the engine. The power is that how fast the vehicle is moving while the torque tells that how quickly that speed can be reached. The torque can be measured even when the vehicle is not moving but to measure the power there must be some movement. The formula of engine torque is given by [9]

$$T = (P - P_{atm}) \times \frac{\pi}{4} \times B^2 \times \frac{S}{2} \times \sin{(\theta)}$$

The torque at every crank rotation is shown below with the help of the graph

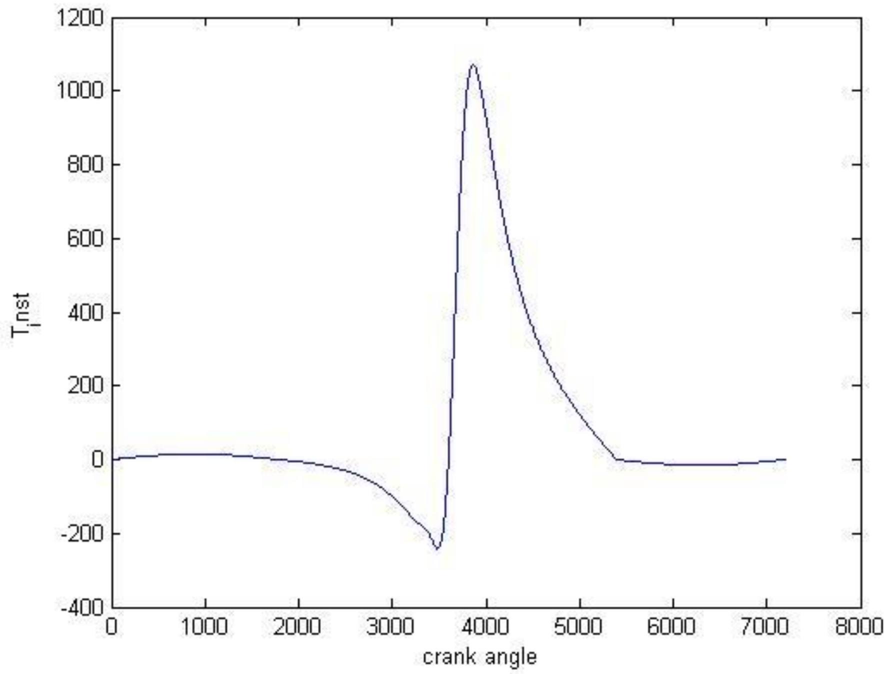


Figure 19: Torque vs rotation of the crank

The total torque is the area under the graph i.e. 85.92 N-m. Out of which 10 % is the frictional torque and rest is the actual torque.

$$T_f = 0.10 \times 85.92 = 8.59$$

$$T_a = T - T_f$$

$$T_a = 85.92 - 8.59$$

$$T_a = 77.33\ Nm$$

In order to calculate the power we will use the following formula

$$P_t = T_a \times \omega$$

$$\omega = 746\ rad/s$$

$$P_t = 746 \times 77.33$$

$$P_t = 57688.3\ W$$

$$P_t = 77.36\ hp$$

The horsepower for 1 cylinder is 77.36 hp. For twin cylinder engine the power would be 154.72 hp.

Chapter 6

Engine Design

6.1 Design of Piston

For the design of piston, there are some parameters to decide. These parameters lies between the ranges shown below in the table [10]. For the design of the piston we used iterative process in Solid Works and Solid works simulation in order to determine these parameters. We took one value put it in the CAD model and then run static analysis study in Solid Works simulation. The value of parameters where stresses were minimal we used those values to get our final piston design.

The following table shows the parameters their values and the values which we used.

Name of parameter	Symbol	Expected Range	Our range(mm)	Used value(mm)
Thickness of piston crown	t	(0.05-0.10)B	6.15-15.23	11
Height of piston	H	(0.8-1.3)B	53.92-87.62	65
Distance from the front to the axis of the piston plane	h1	(0.45-0.47)B	30.33-31.678	31
Diameter of thickening of piston pin	D	(0.3-0.5)B	20.22-33.7	27
Size	B	(0.3-0.5)B	20.22-33.7	20
Wall thickness of leading part	t2	1.5-4.5	1.5-4.5	2.5
Thickness of sealing part	S	(0.03-0.8)B	2.02-5.39	3.5
Distance from the front to the first channel	E	(0.06-0.12)B	4.06-8.08	6
Wall thickness between channel	t2	(0.03-0.05)B	2.022-3.37	2.5

Number of oil hole	N	6-12	6-12	8
Diameter hole for oil	Dm	(0.03-0.05)d6	0.45-0.94	0.7
Hole diameter of piston pin	d6	(0.22-0.28)B	14.83-18.87	16
Height of the leading part	h6	(0.6-0.8)B	40.44-53.92	52
Dome height				5

6.2 Design of Piston pin

Piston pin is designed following the approach used for the piston. i.e. by iterative method in Solid works simulation. The table for parameters is shown below [10]

Name of parameter	**Symbol**	**Expected Range**	**Our Range**	**Value used**
Outer diameter of the bolt	d6	(0.22-0.28)B	14.83-18.87	16
Inner diameter of the bolt	dg	(0.65-0.75)B	9.638-14.15	12
Length of the bolt	l	(0.88-0.93)B	59.31-62.68	61
Length of the upper head of the rod	lm	(0.28-0.38)B	18.87-25.61	24

6.3 Design of Piston Rings

The parameters used to design piston rings are enlisted below [10].

Name of parameter	**Symbol**	**Expected Range**	**Our Range**	**Value used**
Radial thickness of the gurmmet	t_y	(0.039-0.045)B	2.63-3.03	3.0
Radial thickness of the oil collecting ring	t_m	(0.038-0.043)B	2.56-2.90	2.8

6.4 Design of Connecting Rod

When the engine is in running state, the connecting rod varies in size and direction gas and inertia forces. This is why we used stainless steel so that it can resist fatigue and avoid failure. The parameters used are given below. By using these parameters and running a static study in Solid Works simulation, we found that there is minimal stress in the connecting rod [11].

Name of parameter	Symbol	Expected Range	Our Range	Value used
Inner diameter of the upper head without sleeve	d6	(0.22-0.28)B	14.83-18.87	16
Outer diameter of the upper head	dz	(1.25-1.65)d6	18.54-31.12	26
Length of the upper head	lm	(0.33-0.45)B	22.24-30.33	28
Minimal radial thickness of the upper head	hz	(0.16-0.27)d6	2.37-5.1	4
Minimal height of the profile	hm,min	(0.50-0.55)dz	9.27-17.13	16.24
Diameter of the connecting rod neck	dmu	(0.56-0.75)B	37.74-50.55	48
Thickness of the bearing shells	tn	(0.03-0.05)dmu	1.13-2.52	2.4
Distance between connecting rod bolts	C	(1.30-1.75)dmu	49.1-88.41	87
Length of the low head	lk	(0.45-0.95)dmu	16.98-48.02	46

Chapter 7

Analysis on Solid Works

7.1 Analysis on piston

The analysis is done on the Solid Works Simulation in order to get the optimum result for the engine components. For the first iteration different values from the above mentioned parameters were taken and the CAD model was designed on Solid Works then in the Solid Works Simulation the static study was run to check whether these parameters are correct or not.
The report is generated with the help of Solid Works.

7.1.1Model Information

Document Name and Reference	Treated As	Volumetric Properties
Cut-Extrude3 	Solid Body	Mass:0.310899 kg Volume:0.000111035 m^3 Density:2800 kg/m^3 Weight:3.04681 N

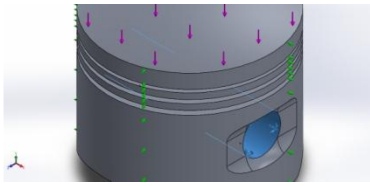
7.1.2 Study Properties

Study name	Static 1
Analysis type	Static
Mesh type	Solid Mesh
Thermal Effect:	On
Thermal option	Include temperature loads
Zero strain temperature	298 Kelvin
Include fluid pressure effects from SOLIDWORKS Flow Simulation	Off
Solver type	FFEPlus
Incompatible bonding options	Automatic
Large displacement	Off
Compute free body forces	On
Friction	Off

7.1.3 Units

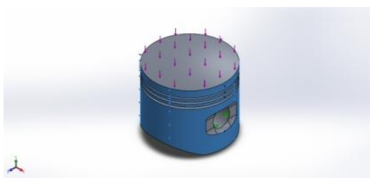
Unit system:	SI (MKS)
Length/Displacement	mm
Temperature	Kelvin
Angular velocity	Rad/sec
Pressure/Stress	N/m^2

7.1.4 Loads and Fixtures

Fixture name	Fixture Image	Fixture Details
Fixed-1		**Entities:** **2 face(s)** **Type:** **Fixed Geometry**

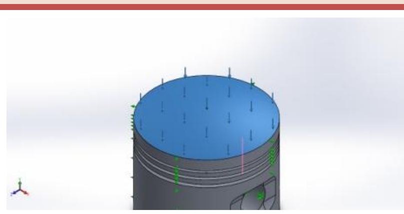
Resultant Forces

Components	X	Y	Z	Resultant
Reaction force(N)	**24.5149**	**93000**	**2.88352**	**93000**
Reaction Moment(N.m)	**0**	**0**	**0**	**0**

Fixture name	Fixture Image	Fixture Details
On Cylindrical Faces-1		**Entities:** **4 face(s)** **Type:** **On Cylindrical Faces** **Translation:** **0, ---, ---** **Units:** **mm**

Resultant Forces

Components	X	Y	Z	Resultant
Reaction force(N)	**-25.5179**	**-0.171222**	**-2.67424**	**25.6582**
Reaction Moment(N.m)	**0**	**0**	**0**	**0**

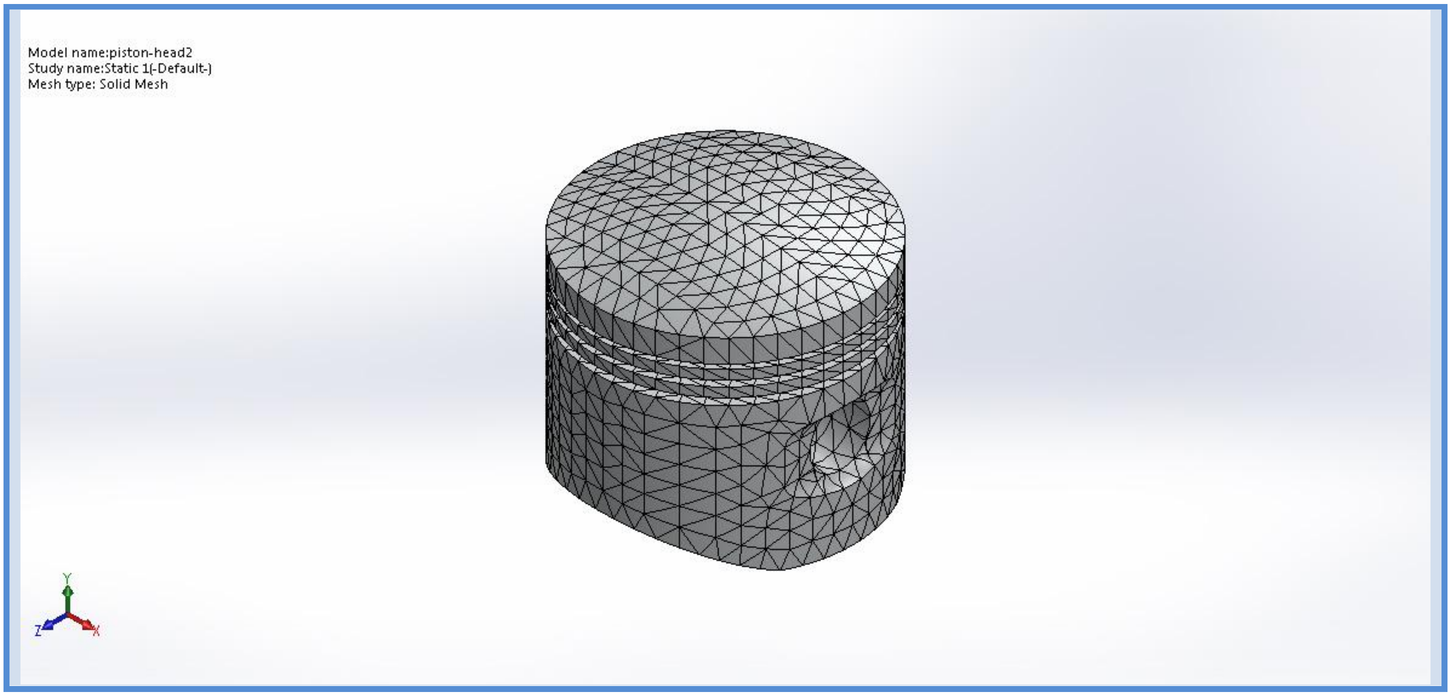
Load name	Load Image	Load Details
Force-1		**Entities:** **1 face(s)** **Reference:** **Edge< 1 >** **Type:** **Apply force** **Values:** **---, ---, 93000 N**

7.1.5 Mesh information

Mesh type	Solid Mesh
Mesher Used:	Standard mesh
Automatic Transition:	Off
Include Mesh Auto Loops:	Off
Jacobian points	4 Points
Element Size	4.80787 mm
Tolerance	0.240393 mm
Mesh Quality	High

7.1.5.1 Mesh information - Details

Total Nodes	16690
Total Elements	9986
Maximum Aspect Ratio	15.09
% of elements with Aspect Ratio < 3	94.1
% of elements with Aspect Ratio > 10	0.0401
% of distorted elements(Jacobian)	0
Time to complete mesh(hh;mm;ss):	00:00:02
Computer name:	Tayab-PC



7.1.6 Resultant Forces

7.1.6.1 Reaction forces

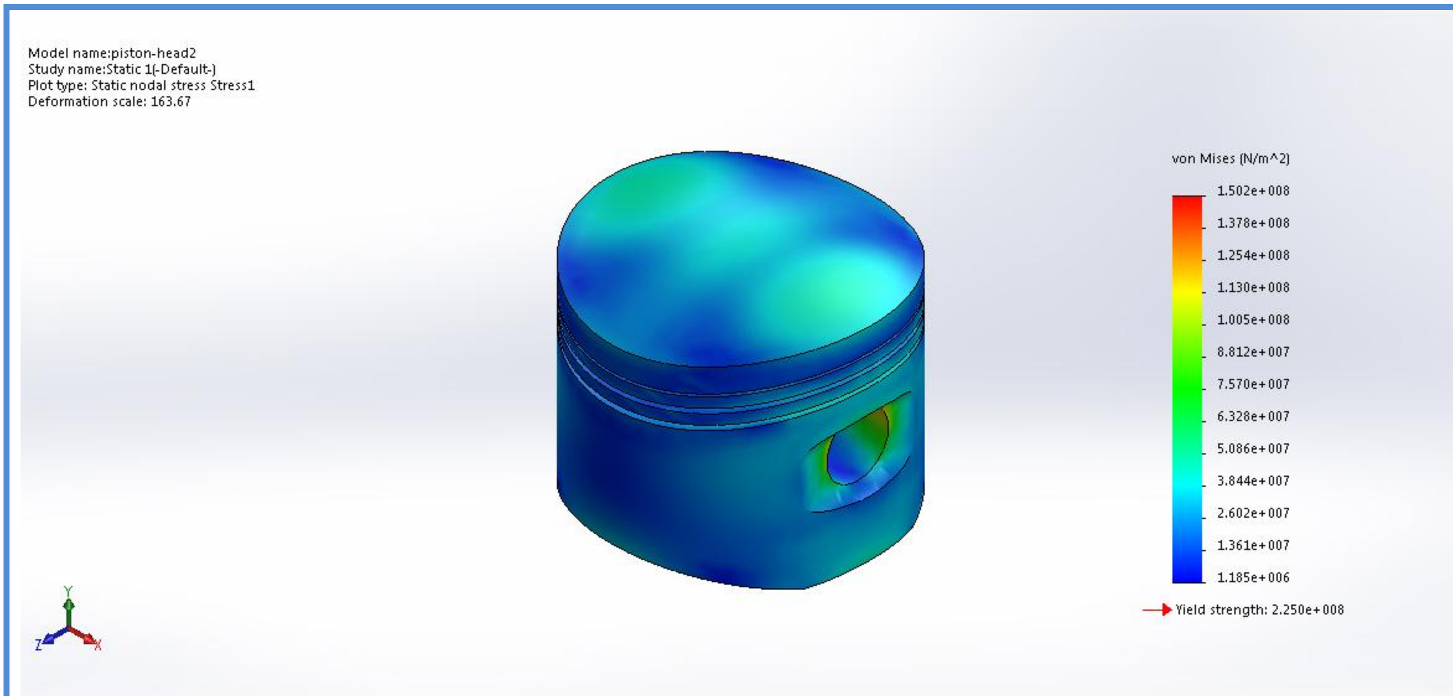
Selection set	Units	Sum X	Sum Y	Sum Z	Resultant
Entire Model	N	-1.00109	92999.8	0.212135	92999.8

7.1.6.2 Reaction Moments

Selection set	Units	Sum X	Sum Y	Sum Z	Resultant
Entire Model	N.m	0	0	0	0

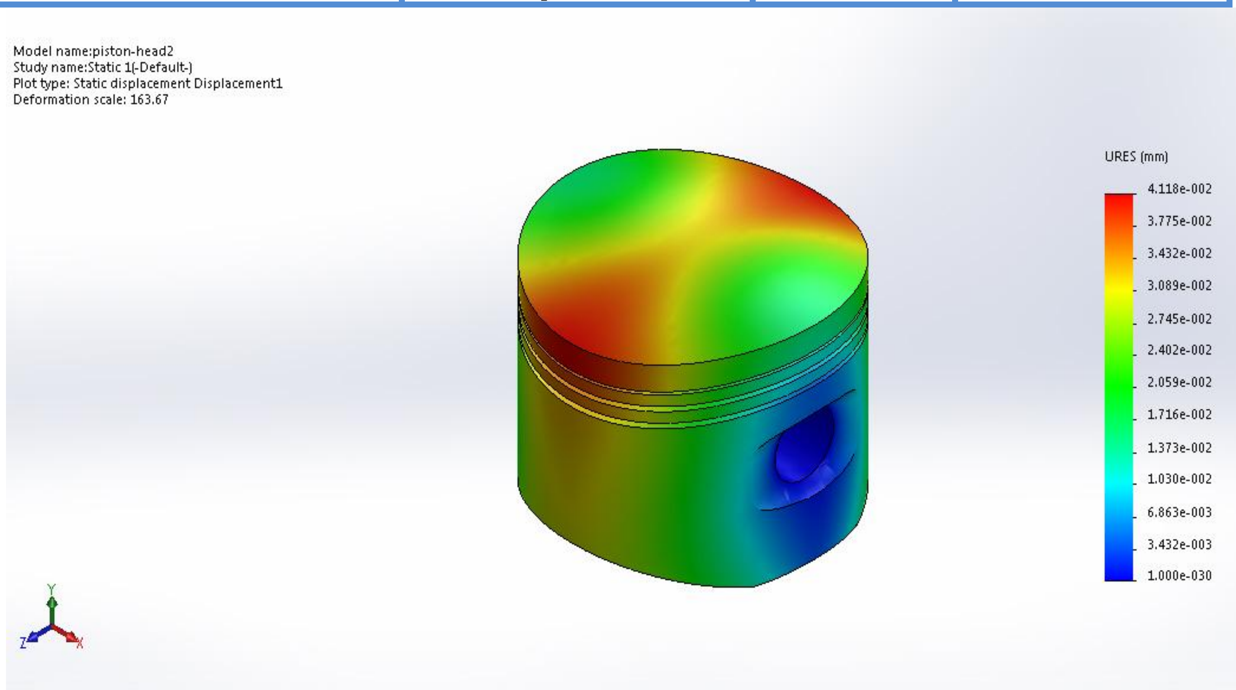
7.1.7 Study Results

Name	Type	Min	Max
Stress1	VON: von Mises Stress	1.18536e+006 N/m^2 Node: 13114	1.50221e+008 N/m^2 Node: 10753



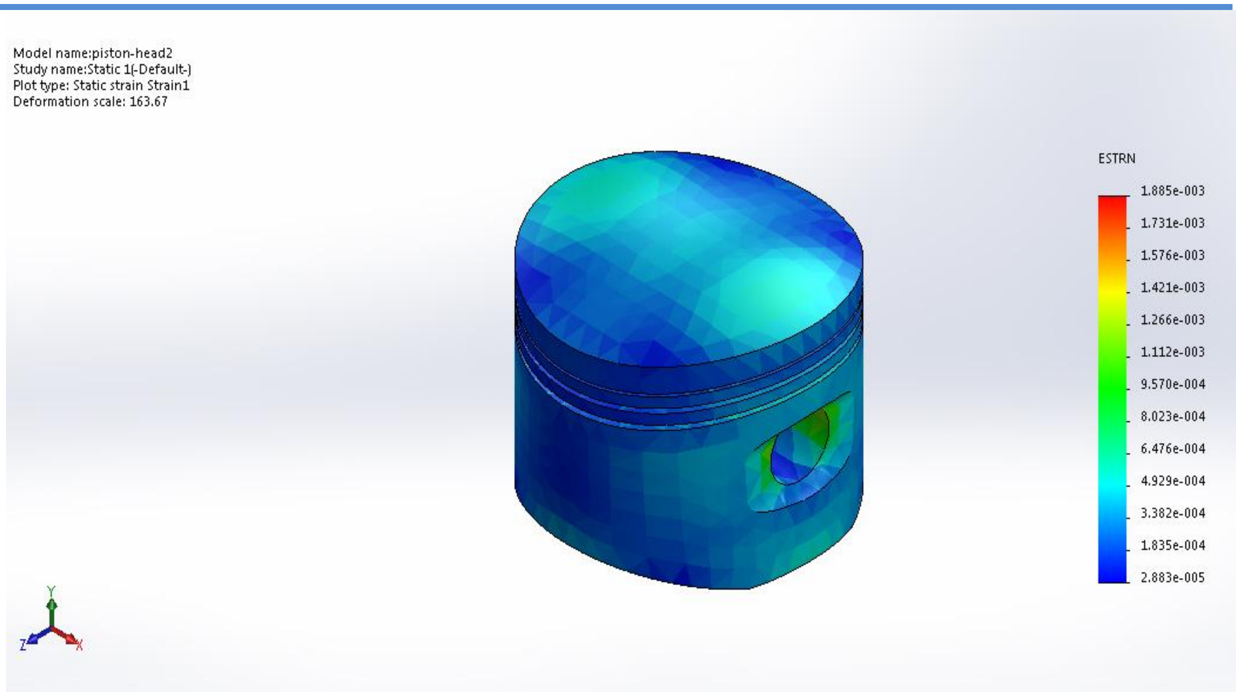
piston-head-Static 1-Stress-Stress1

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0 mm Node: 27	0.0411807 mm Node: 7689



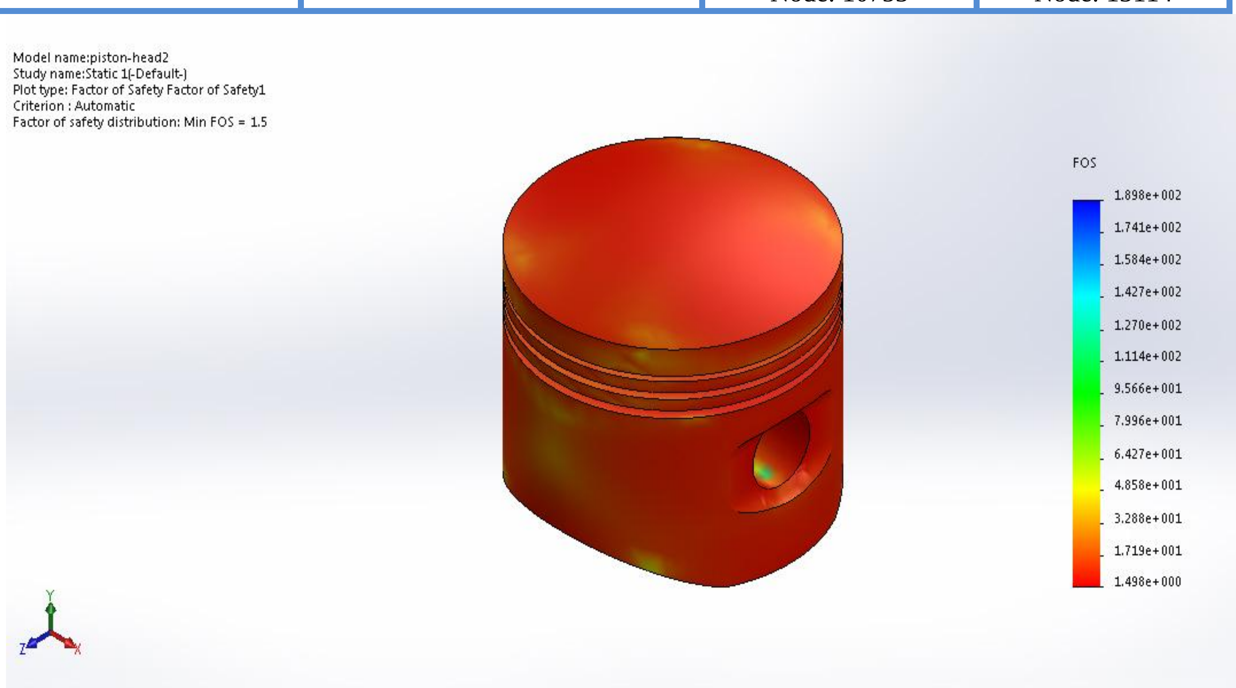
piston-head-Static 1-Displacement-Displacement1

Name	Type	Min	Max
Strain1	ESTRN: Equivalent Strain	2.88301e-005 Element: 5886	0.0018852 Element: 4311



piston-head-Static 1-Strain-Strain1

Name	Type	Min	Max
Factor of Safety1	Automatic	1.49779 Node: 10753	189.816 Node: 13114



piston-head-Static 1-Factor of Safety-Factor of Safety1

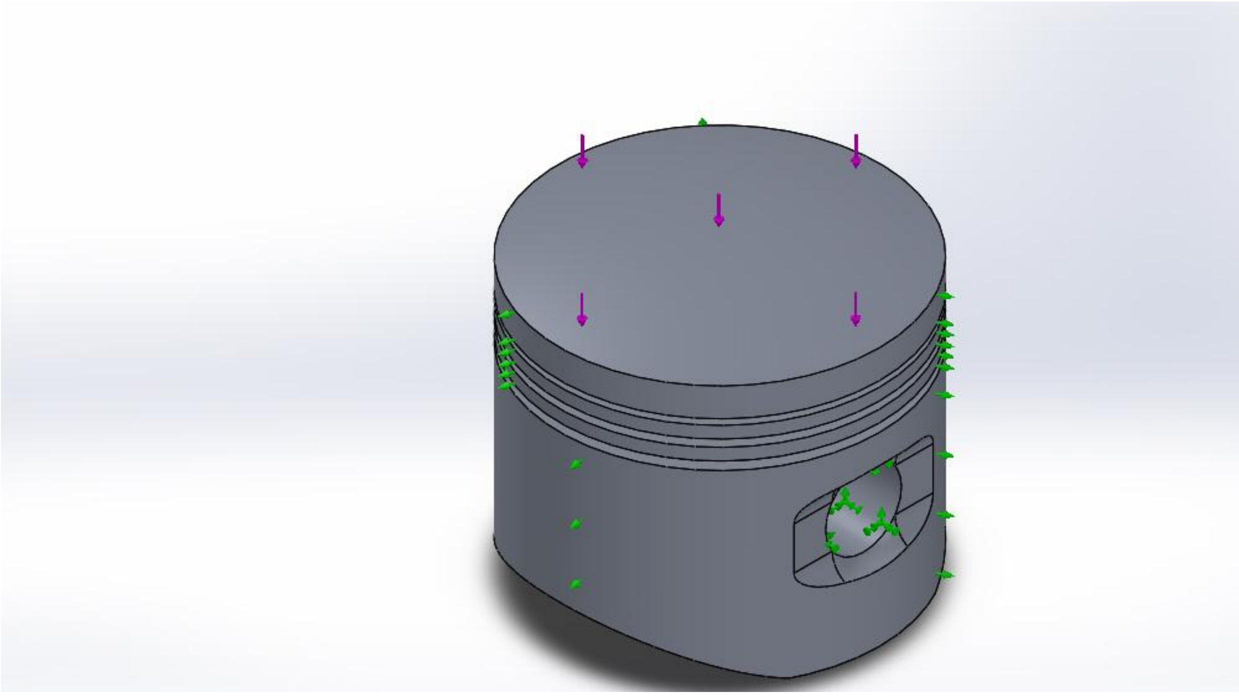


Image-1

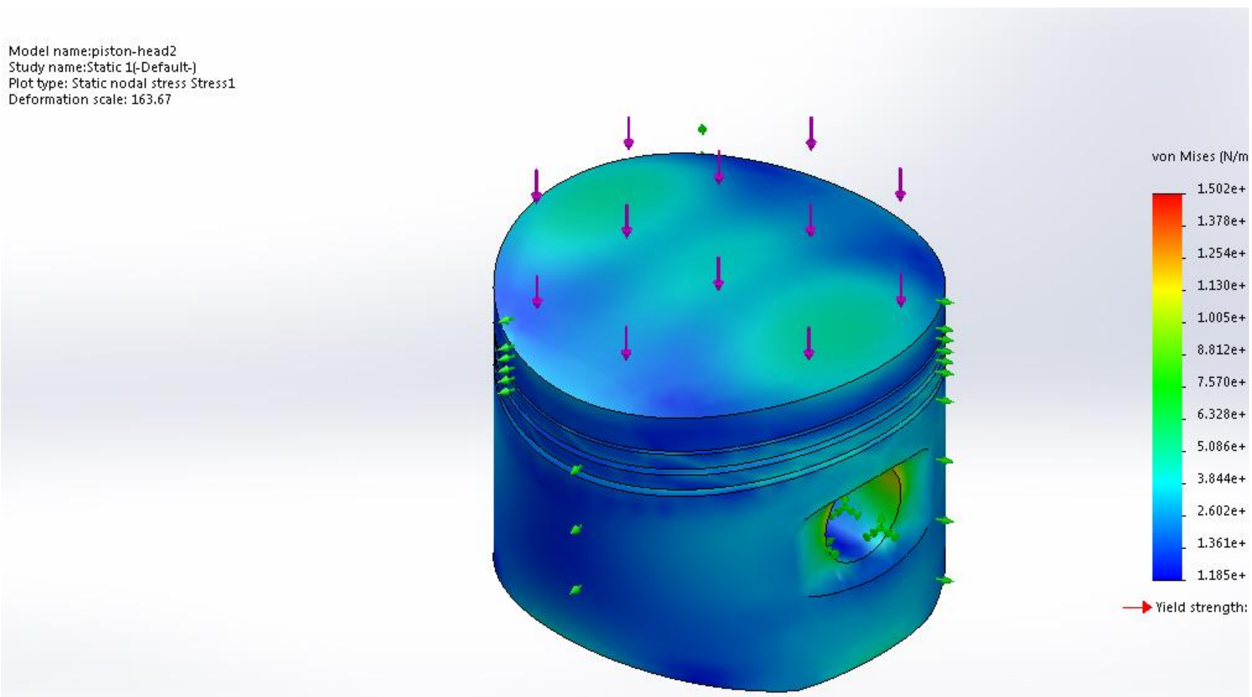


Image-2

7.2 Analysis on connecting rod

Like piston head the CAD model is drawn on the SolidWorks and Finite Element analysis is done on the SolidWorks Simulation.

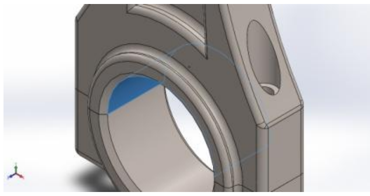
7.2.1 Study Properties

Study name	Static 1
Analysis type	Static
Mesh type	Solid Mesh
Thermal Effect:	On
Thermal option	Include temperature loads
Zero strain temperature	298 Kelvin
Include fluid pressure effects from SOLIDWORKS Flow Simulation	Off
Solver type	FFEPlus
Inplane Effect:	Off
Soft Spring:	Off
Inertial Relief:	Off
Incompatible bonding options	Automatic
Large displacement	Off
Compute free body forces	On
Friction	Off
Use Adaptive Method:	Off
Result folder	SOLIDWORKS document (c:\users\malik\appdata\local\temp)

7.2.2 Units

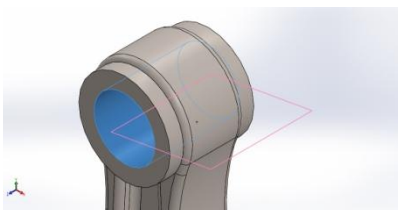
Unit system:	SI (MKS)
Length/Displacement	mm
Temperature	Kelvin
Angular velocity	Rad/sec
Pressure/Stress	N/m^2

7.2.3 Loads and Fixtures

Fixture name	Fixture Image	Fixture Details
Fixed-1		**Entities:** 1 face(s) **Type:** Fixed Geometry

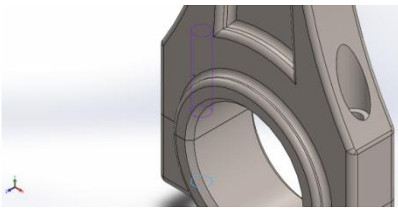
Resultant Forces

Components	X	Y	Z	Resultant
Reaction force(N)	**-2604.72**	**14772.1**	**-0.000111461**	**15000**
Reaction Moment(N.m)	**0**	**0**	**0**	**0**

Load name	Load Image	Load Details
Force-1		**Entities:** 1 face(s), 1 plane(s) **Reference:** Plane1 **Type:** Apply force **Values:** ---, ---, -15000 N

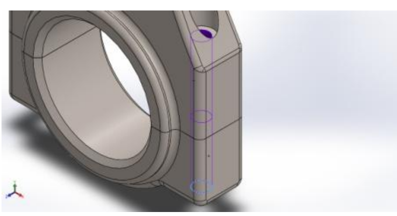
7.2.4 Connector Definitions

Pin/Bolt/Bearing Connector

Model Reference	Connector Details	Strength Details
 Counterbore Screw-1	Entities: **1 edge(s), 1 face(s)** Type: **Bolt(Head/Nut diameter)(Counterbore screw)** Head diameter: **7.5 mm** Nominal shank diameter: **5** Preload (Torque): **2** Young's modulus: **2.1e+011** Poisson's ratio: **0.28** Preload units: **N.m**	**No Data**

Connector Forces

Type	X-Component	Y-Component	Z-Component	Resultant
Axial Force (N)	**0**	**-1696.4**	**0**	**1696.4**
Shear Force (N)	**-0.50754**	**0**	**-0.017293**	**0.50783**
Bending moment (N.m)	**-0.00049663**	**0**	**-0.011046**	**0.011057**

Model Reference	Connector Details	Strength Details
 Counterbore Screw-2	Entities: **1 edge(s), 2 face(s)** Type: **Bolt(Head/Nut diameter)(Counterbore screw)** Head diameter: **7.5 mm** Nominal shank diameter: **5** Preload (Torque): **2** Young's modulus: **2.1e+011** Poisson's ratio: **0.28** Preload units: **N.m**	**No Data**

Connector Forces

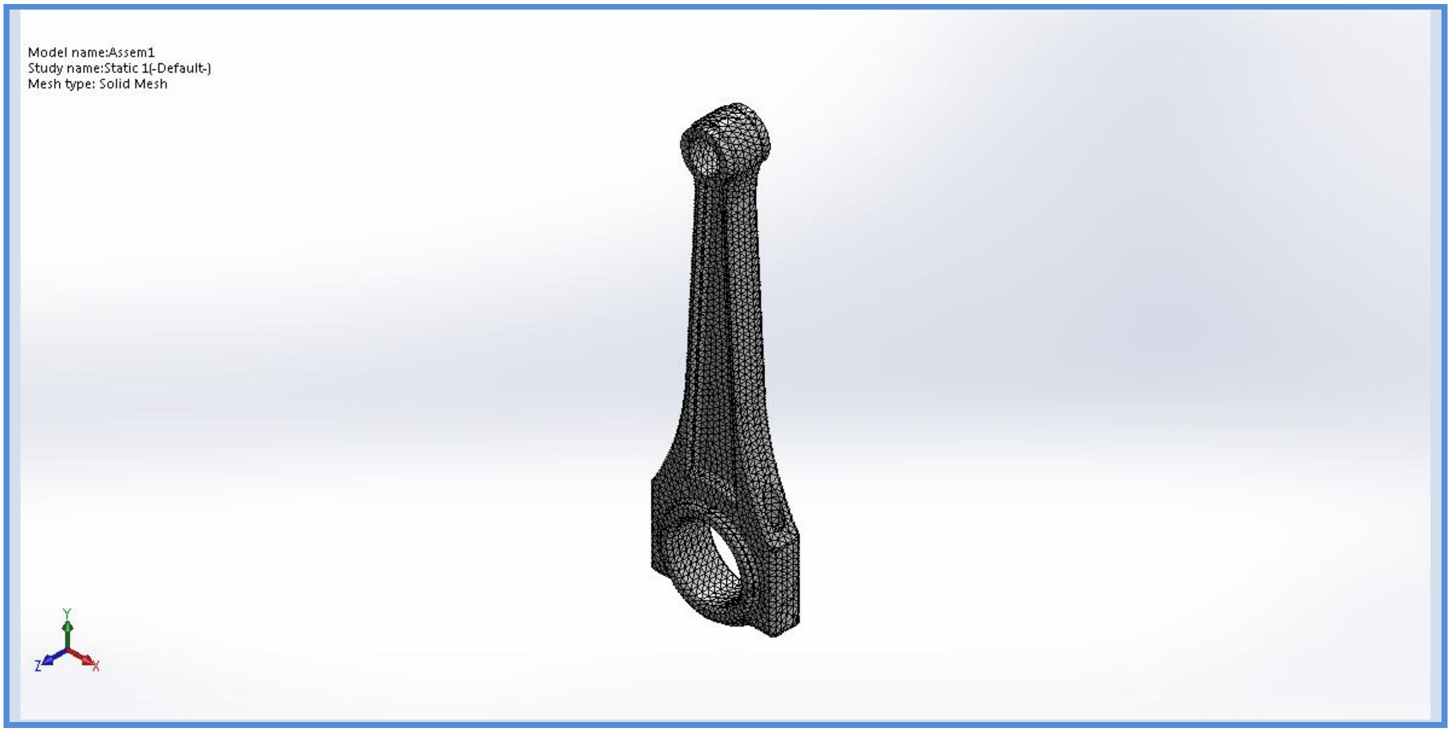
Type	X-Component	Y-Component	Z-Component	Resultant
Axial Force (N)	**0**	**-1666.4**	**0**	**1666.4**
Shear Force (N)	**0.010598**	**0**	**7.6882e-007**	**0.010598**
Bending moment (N.m)	**-4.7411e-007**	**0**	**-0.00015134**	**0.00015134**

7.2.6 Mesh information

Mesh type	Solid Mesh
Mesher Used:	Standard mesh
Automatic Transition:	Off
Include Mesh Auto Loops:	Off
Jacobian points	4 Points
Element Size	2.98968 mm
Tolerance	0.149484 mm
Mesh Quality	High
Remesh failed parts with incompatible mesh	Off

7.2.6.1 Mesh information - Details

Total Nodes	47194
Total Elements	29420
Maximum Aspect Ratio	27.637
% of elements with Aspect Ratio < 3	97.7
% of elements with Aspect Ratio > 10	0.0238
% of distorted elements(Jacobian)	0
Time to complete mesh(hh;mm;ss):	00:00:04
Computer name:	Tayab-PC



7.2.7 Resultant Forces

7.2.7.1 Reaction forces

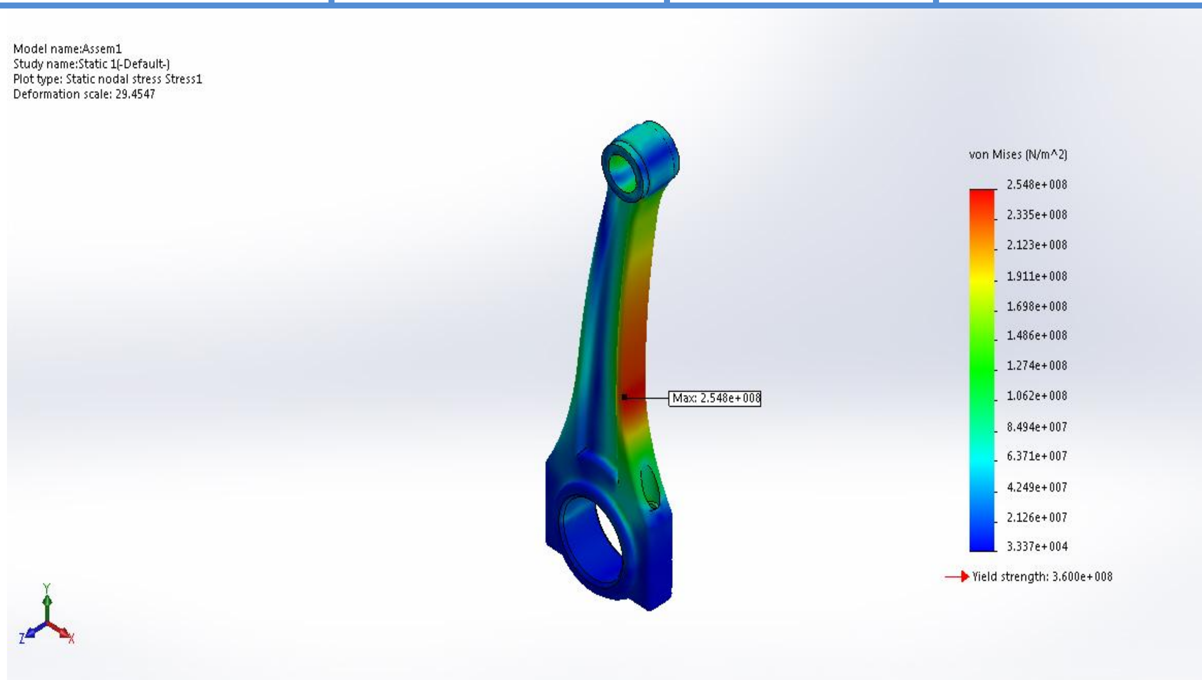
Selection set	Units	Sum X	Sum Y	Sum Z	Resultant
Entire Model	N	-2604.72	14772.1	-0.000111461	15000

7.2.7.2 Reaction Moments

Selection set	Units	Sum X	Sum Y	Sum Z	Resultant
Entire Model	N.m	0	0	0	0

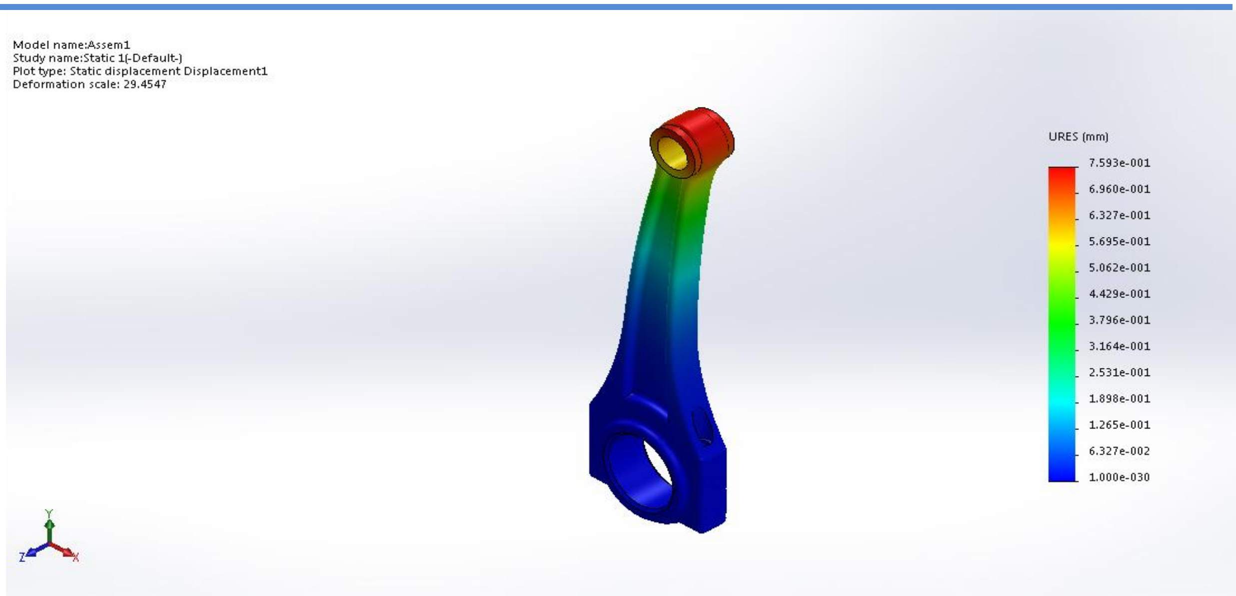
7.2.8 Study Results

Name	Type	Min	Max
Stress1	VON: von Mises Stress	33371.5 N/m^2 Node: 38698	2.54757e+008 N/m^2 Node: 1425



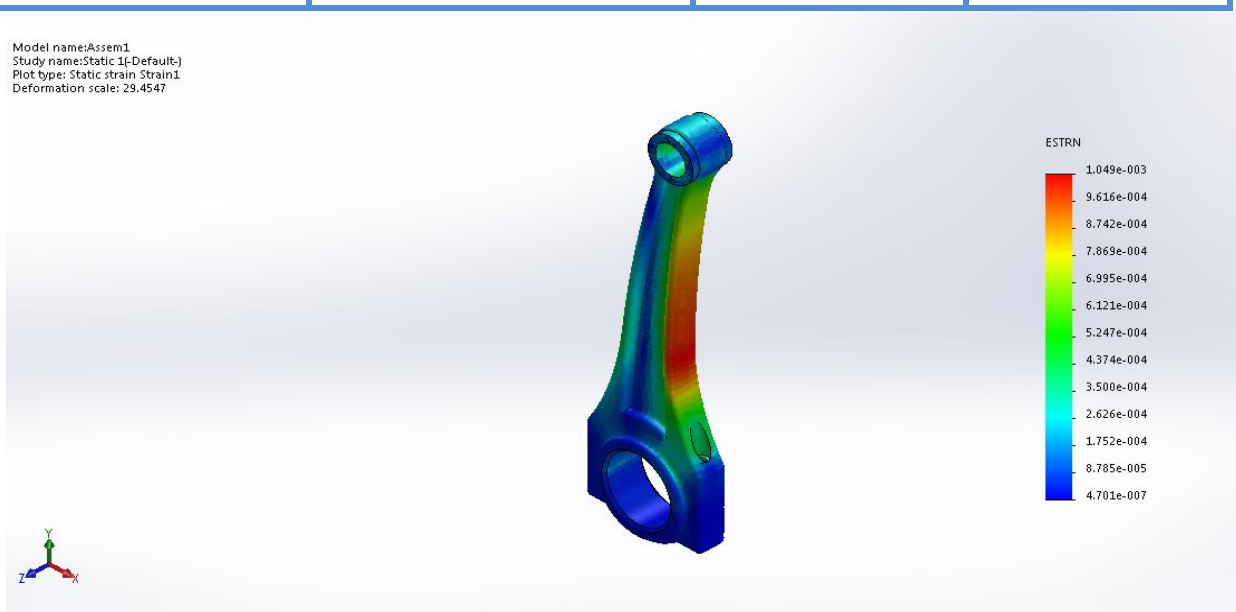
Assem1-Static 1-Stress-Stress1

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0 mm Node: 190	0.759295 mm Node: 33712



Connecting rod -Static 1-Displacement-Displacement1

Name	Type	Min	Max
Strain1	ESTRN: Equivalent Strain	4.70147e-007 Element: 26616	0.00104898 Element: 2877



Connecting rod-Static 1-Strain-Strain1

Name	Type	Min	Max
Factor of Safety1	Automatic	1.41311 Node: 1425	10787.6 Node: 38698

Connecting rod-Static 1-Factor of Safety-Factor of Safety1

Works Cited

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[7] R. v. Basshuysen, Modern Engine Technology from A to Z, Warrendale, USA: SAE International, 2006.

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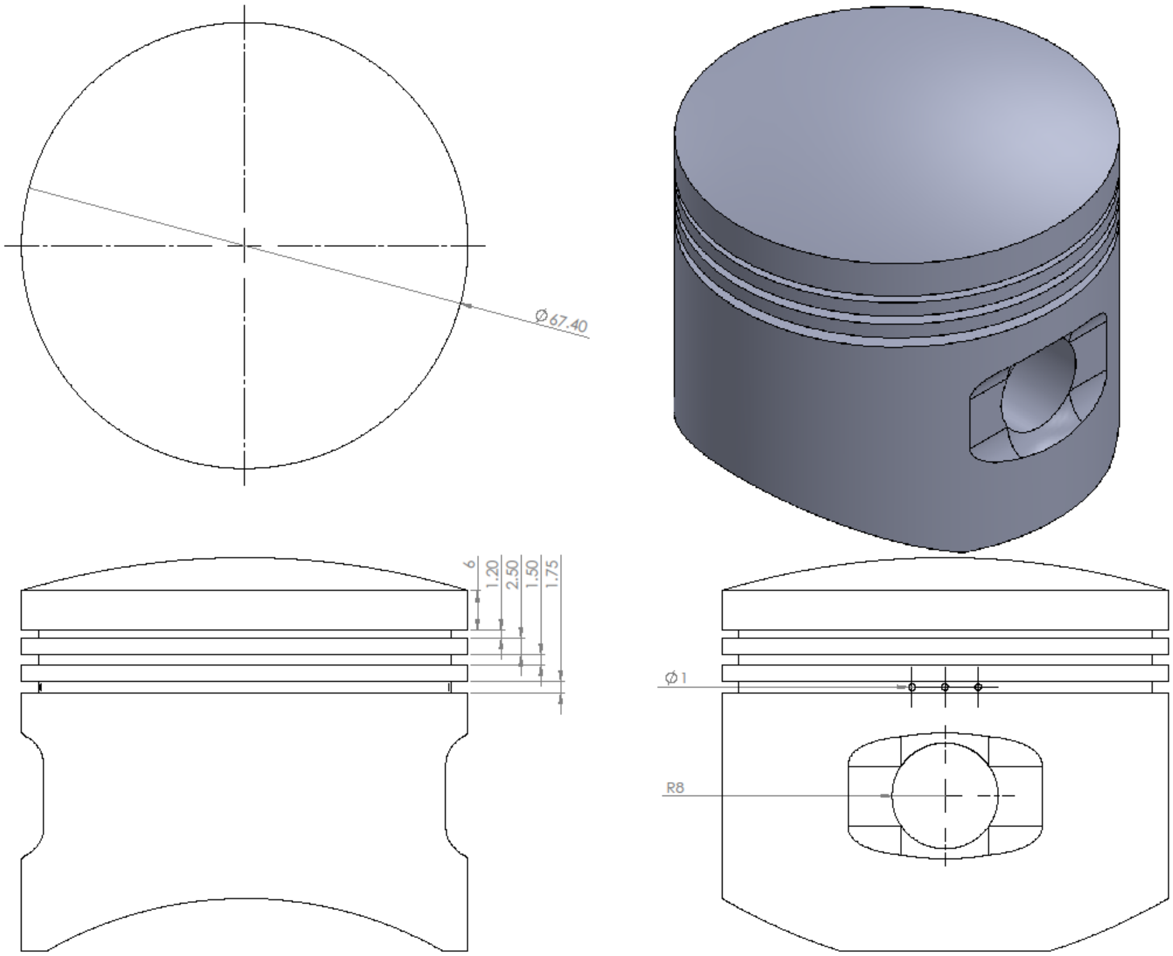
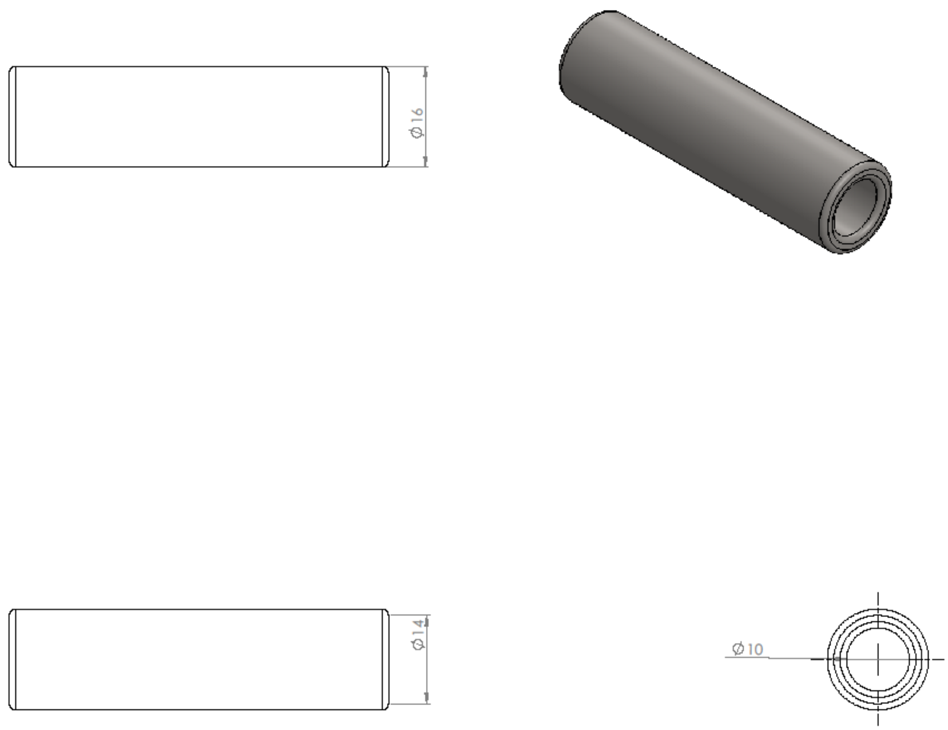
APPENDIX I: CAD Drawing

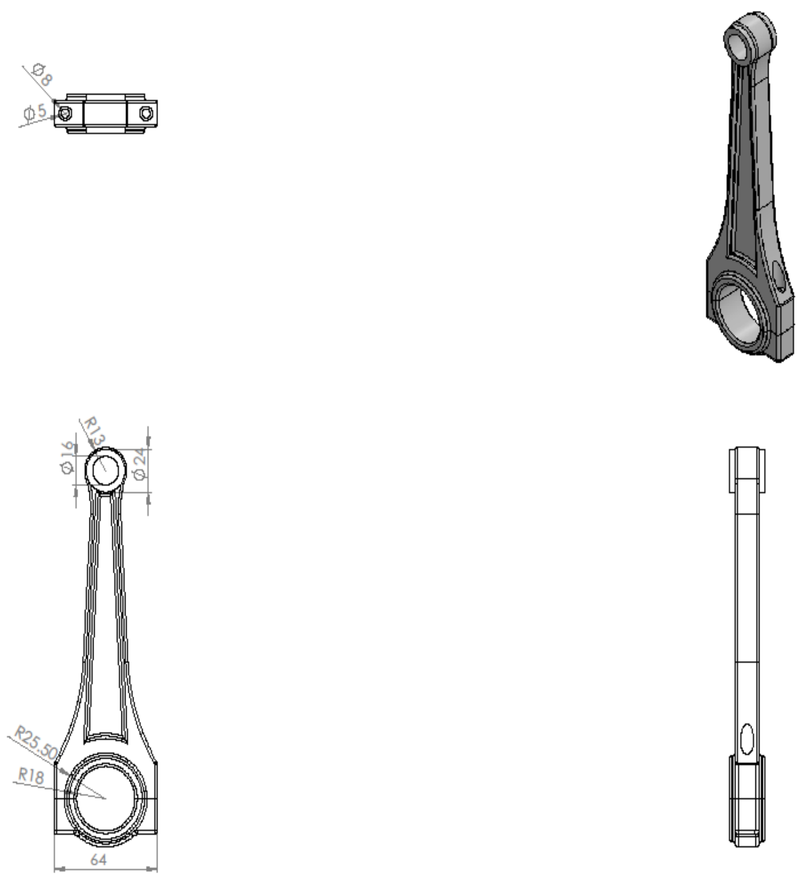
Piston head



Connecting pin



Connecting rod



APPENDIX II: Material properties

Connecting rod and Connecting pin
Material used: Alloy Steel

Property	Value	Unit
Elastic Modulus	2.1e+011	N/m^2
Poisson Ratio	0.28	N/A
Shear Modulus	7.9e+010	N/m^2
Density	7700	Kg/m^3
Tensile Strength	723825600	N/m^2
Yield Strength	620422000	N/m^2
Thermal Expansion Coefficient	1.3e-005	/K
Thermal Conductivity	50	W/(m·K)
Specific Heat	460	J/(kg·K)

Piston Head

Material used: 201.0-T6 Insulated Mold Cast Aluminum

Property	Value	Unit
Elastic Modulus	7.0999997e+010	N/m^2
Poisson Ratio	0.33	N/A
Shear Modulus	2.3e+010	N/m^2
Density	2800	Kg/m^3
Tensile Strength	359000000	N/m^2
Yield Strength	349000000	N/m^2
Thermal Expansion Coefficient	3.5e-005	/K
Thermal Conductivity	121	W/(m·K)
Specific Heat	963	J/(kg·K)