Downsizing of Internal Combustion Engine from 660cc to 610cc

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A thesis submitted in partial fulfillment of the requirements for the degree of Bachelors of Engineering in Mechanical Engineering

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DECLARATION

We certify that this research work titled "*Downsizing of Internal Combustion Engine 660cc to 610cc*" is our work. The work has not been presented elsewhere for assessment. The material that has been used from other sources has been properly acknowledged/referred.

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Many great minds worked together to accomplish this project

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Last but not the least; we are indebted to our parents, without whom we would not have been able to achieve our goal.

ABSTRACT

Part of a bigger program, Formula Students International competition vehicle, the main objective of the project was to provide a locally made 610cc engine for formula student vehicle done my modifying a Daihatsu KF-VE 660cc engine’s rotating assembly and removing 50cc. By de-stroking along the way compression ratio of the engine was also adjusted and set to an appropriate value for turbo-charging purposes in future mainly with a purpose of avoiding knocking of engine at elevated cylinder pressure.

To accomplish this reverse engineering was employed and measurements were made of stock engine crank shaft followed by CAD modeling, then new crankshaft was designed with reduced radius, modification in connecting rods was also made ie a shift from I type to H type & an increase in length so to cater for the imbalance introduced bob weights were also adjusted in the new crankshaft. Once the modelling was done extensive analysis were done using ANSYS. Once we were done with analysis we proceeded with prototype manufacturing. Two prototypes were made before final manufacturing so we had all the clearance set in the engine. Both connecting rods & crank shaft are machined from AISI 4140 due to our high strength requirement. Once completed this will be the first ever downsized engine to be installed in a formula student’s vehicle and will be a very affordable counter solution to the expense CBR 600 engines used for the purpose.

Keywords: Engine, Downsizing, Crankshaft, Connecting rod, CC

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ABBREVIATIONS USED

COG	Centre of Gravity
HP	Horse Power
CAD	Computer Aided Design
TDC	Top Dead Centre
BDC	Bottom Dead Centre
COM	Centre of Mass
AISI	American Institute of Steel and Iron
TIG	Tungsten Inert Gas Welding
CNC	Computerized Numeric Control
SAE	Society of Automotive Engineers
CR	Compression Ratio

CHAPTER: 1

INTRODUCTION

1.1 Project Objective

Formula student is the international student engineering racing vehicle competition in which students from different universities participate with their own built small scale racing cars on formula style. This competition has its own set of rules and regulations which are strictly monitored in the cars that are participating in the competition. As per the rules of the competition, the car must have a piston engine with displacement no more tha 610 cc. The goal of the project was to develop a low cost engine for Formula Students competition by reducing the volume of engine from 660cc to 610cc by modifying engine's rotating assembly. This will provide an alternate solution to an existing approach of using an expensive CBR 600cc engine. Following is a section from the Formula SAE rulebook stating competition engine requirement of 610cc or less

IC1.1 Engine Limitation

IC1.1.1 The engine(s) used to power the car must be a piston engine(s) using a four-stroke primary heat cycle with a displacement not exceeding 610 cc per cycle. Hybrid powertrains, such as those using electric motors running off stored energy, are prohibited.

Note: All waste/rejected heat from the primary heat cycle may be used. The method of conversion is not limited to the four-stroke cycle.

IC1.1.2 The engine can be modified within the restrictions of the rules.

IC1.1.3 If more than one engine is used, the total displacement cannot exceed 610 cc and the air for all engines must pass through a single air intake restrictor (see IC1.6 "Intake System Restrictor.")

Figure 0.1

For the said purpose we purchased a locally available Daihatsu KF-VE 660cc engine mostly used in Daihatsu Mira car, we then reverse engineered the stock crankshaft to obtain base dimensions and then later modified these as per our calculations and clearance available in the engine block. Compression ratio of the engine was also reduced so that engine can be turbocharged with introduction of engine knocking.

1.2 Formula Students Competition:

Formula Students Competition is a student design competition organized by SAE International (formerly Society of Automotive Engineers). Student teams from around the world design, build, test and race a small scale formula style race car. It is the world's most established educational motorsports competition. Backed by industry and high profile engineers, the competition aims to inspire and develop enterprising and innovative young engineers. Universities from across the globe are challenged to design and build a single seat racing car in order to compete in static and dynamic events, which demonstrate their understanding and test performance of the vehicle.

1.2.1 The Challenge

The concept behind Formula SAE is that a fictional manufacturing company has contracted a design team to develop a small Formula-style race car. The prototype race car is to be evaluated for its potential as a production item. The target marketing group for the race car is the non-professional weekend autocross racer. Each student team designs, builds and tests a prototype based on a series of rules whose purpose is both to ensure onsite event operations and promote clever problem solving.

1.2.2 The Mission

Formula SAE promotes careers and excellence in engineering as it encompasses all aspects of the automotive industry including research, design, manufacturing, testing, developing, marketing, management and finances. Formula SAE takes students out of the classroom and allows them to apply textbook theories to real work experiences. The format of the event is such that it provides an ideal opportunity for the students to demonstrate and improve their capabilities to deliver a complex and integrated product in the demanding environment of a motorsport competition.

Formula SAE is

- A high performance engineering project that is extremely valued by universities and usually forms part of a degree level project
- Viewed by the motorsports industry as the standard for engineering graduates to meet, transitioning them from university to the workplace
- The kite-mark for real world engineering experiences

Formula SAE is the leading competition for undergraduates who want hands on experience in an innovative racing team.

1.2.3 Key Facts

Today, the competition has expanded and includes a number of spin off events. In the United States there are two locations: California and Michigan; Michigan being the largest event and longest running.

Internationally, as part of the Official Formula SAE Series utilizing the Formula SAE Rules Copyright under signed agreement are:
Formula SAE Australasia
Formula SAE Brazil
Formula SAE Italy
Formula Student (UK)
Formula Student Austria
Formula Student Germany
Formula SAE Japan

In 2006, Dartmouth College (partnered with SAE International) created the fuel alternative event Formula Hybrid using Formula SAE cars with gasoline-electric hybrid power plants.

1.2.4 Judging

Teams are judged by experienced industry specialist. Each team goes through the following rigorous testing process of their car.

1.2.4.1 Static events

- Design, Cost and sustainability, and Business Presentation Judging
- Technical and Safety Scrutinizing
- Tilt Test
- Brake and Noise Test

1.2.4.2 Dynamic events

- Skid Pad
- Sprint
- Acceleration
- Endurance
- Fuel Economy

1.3 Society of Automotive Engineers

SAE International, initially established as the Society of Automotive Engineers, is a U.S.-based, globally active professional association and standard organization for engineering professionals in various industries. SAE International coordinates the development of technical standards based on best practices identified and described by SAE committees and task forces. Task forces are composed of engineering professionals from relevant fields. SAE International has more than 138,000 members globally. Aside from its standardization efforts, SAE International also devotes resources to projects and programs in STEM education, professional certification, and collegiate design competitions.

CHAPTER: 2

LITERATURE REVIEW

To accomplish the task, an extensive literature review was required to be done in order have exemplary pre-requisite knowledge of subject, concepts and methodology. For this we studied and consulted numerous automotives and mechanical engineering design books as well as different journals and research papers were taken into consideration and studied thoroughly.

2.1 Engine

An engine is a machine designed to convert one form of energy into useful mechanical motion. In heat engines (internal and external combustion engines), fuel is burnt to release thermal energy which then creates motion. Electric motors convert electrical energy to mechanical motion. Pneumatic motors use compressed air.

2.2 Internal Combustion Engine

Internal Combustion Engines, Commonly known as IC Engines are those in which the combustion of a fuel takes place inside a combustion chamber (cylinder). The products of combustion (high temperature and high pressure gasses) form the working fluid. These gasses apply direct force (conversion of chemical energy to thermal energy and then to mechanical movement) to some component of the engine, typically pistons, forcing the component to move a particular distance and bring about consequent useful movements.

2.2.1 Classification of IC Engines

There are various parameter that help classify an Internal Combustion engine.

1. On the basis of thermodynamic cycle:
   i) Otto Cycle (Constant volume combustion)
   ii) Diesel Cycle (Constant pressure combustion)
   iii) Dual cycle
   iv) Brayton cycle

2. Ignition Method:
   i) Spark Ignition (Otto cycle)
   ii) Compression Ignition (Diesel cycle)

3. Working Cycle:
   i) Two stroke
   ii) Four stroke
   iii) Six stroke

4. Basic Engine Design:
   i) Reciprocating Engine
   ii) Rotary Engine

5. Type of Fuel:
   i) Light Oil – Petrol, Kerosene
   ii) Heavy Oil – Diesel
   iii) Gas – LPG, CNG, Hydrogen
   iv) Bi-fuel – Gas and Liquid fuel

6. Cooling System:
   i) Air Cooling

ii) Liquid Cooling

7. Fuel Injection:
   i) Single Point Injection (throttle body injection, Central fuel injection, pressure carburetor
   ii) Continuous Injection
      a) Single point
      b) Multi point
   iii) Central Port Injection
   iv) Multiport Fuel Injection
   v) Direct Ignition
   vi) Swirl Ignition

8. Air Intake Method:
   i) Naturally aspirated
   ii) Turbocharged and Supercharged

9. Number of Cylinders:
   i) Single Cylinder
   ii) Multi Cylinder

10. Cylinder Arrangement:
   i) In-line arrangement
   ii) V arrangement
   iii) Opposed Cylinder arrangement
   iv) Opposite Piston arrangement
   v) Vertical arrangement
   vi) Horizontal arrangement
   vii) Radial arrangement

11. Speed of Engine;

i) Slow Speed
ii) Medium Speed
iii) High Speed

12. Valve Mechanism:
    i) Overhead valve
    ii) Side valve

13. Field of Application:
    i) Stationary Engines (Power generation)
    ii) Marine Engines (Ships)
    iii) Aero Engines (Aircrafts)
    iv) Automotive Engines (Land transport)
    v) Locomotive Engines (Trains)

2.3 Components of an engine and important concepts

2.3.1 Piston

A piston is a cylindrical component in an engine, contained by the cylinders, used to transfer the forces of the expanding gases via a connecting rod. It has a reciprocating motion and is attached to the small end of a connecting rod. The piston has piston rings that help to make the combustion chamber gas tight.

Figure 2.1 Pistons

2.3.2 Piston Rings

Piston rings are split rings present on the curved surface of the piston. They have three functions described as follows:

I. To ensure that the combustion chamber is sealed and no gases are transferred to the crank
II. To aid in heat transfer from the piston to the cylinder wall
III. To regulate engine oil consumption.

Presence of three rings indicate that the engine is a four stroke engine and the presence of two rings indicate a two stroke engine.

Figure 2.2 Piston RIngs

2.3.3 Piston Pin

The Piston pin or Wrist pin connects the piston to the connecting rod and provides a bearing for the connecting rod to rotate about as the piston moves.

2.3.4 Connecting Rod

The connecting rod connects the piston to the crankshaft. It can rotate at both ends so that its angle can change as the piston moves and the crankshaft rotates. It is made of special steel alloys or aluminum alloys as they are constantly under compressive, tensile and bending stresses. They help convert the reciprocating motion of the piston to the rotating motion of the crankshaft.
There are two types of connecting rod

2.3.4.1 H beam Connecting Rod

Figure 2.3 H beam connecting rod

2.3.4.2 I Beam Connecting Rod

Figure 2.4 I Beam connecting rod

2.3.5 Crankshaft

A crankshaft is a mechanical part inside the engine that translates the reciprocating motion of the piston into rotational motion. In order to accomplish this the crankshaft has crankpins whose axis is offset from that of the crank, and to which the big ends of the connecting rods from each cylinder are attached.

Figure 2.5 Common crankshaft

2.3.6 Crank Pin Journal / Rod Journal

Crank Pin journals are the offset pins to which the big ends of the connecting rods are attached. In an inline engine, one crank journal serves one connecting rod. In a V-engine, one crank journal serves two connecting rods whereas in a radial engine, one journal serves more than two connecting rods.

2.3.7 Main Journal

Main journals are present on the central axis of the crankshaft. They do not have connecting rods attached to them. They serve to keep the crankshaft intact via connections on the counter weights.

2.3.8 Stroke

Stroke is the distance that a piston travels inside the cylinder as it moves from the top dead center to the bottom dead center. The stroke and bore together determine the total engine displacement.

2.3.9 Bore

Bore is the diameter of the cylinders present inside the engine. The stroke and bore together determine the total engine displacement.

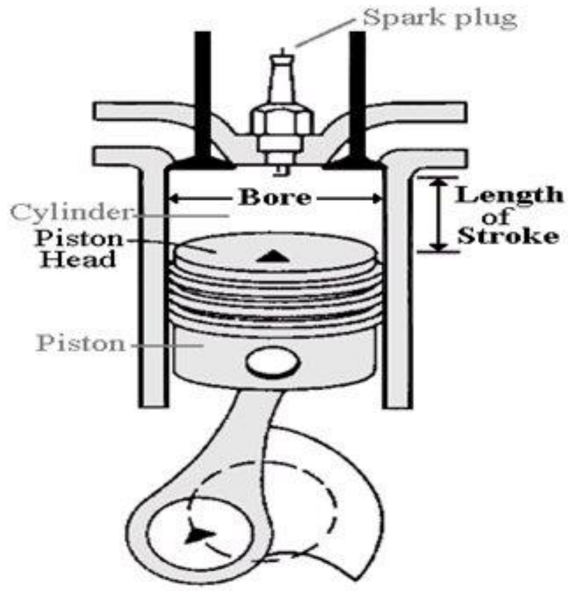


Figure 2.6 Engine bore and Stroke

2.3.10 Engine Displacement

Engine Displacement is the volume of an engine's cylinders and a general indicator of the engines size and power. It is normally expressed in liters, cubic centimeters or cubic inches.

2.3.11 Swept Volume

Swept volume of one cylinder is the volume that a piston sweeps in a single movement from the top dead centre to the bottom dead centre. It is normally expressed in liters, cubic centimeters or cubic inches.

2.3.12 Clearance Volume

Clearance volume is the volume of gas left inside the cylinder at the end of the discharge stroke. At this point, the piston is at the top dead centre and the space between the piston and the cylinder head constitutes the clearance volume. It is generally expressed as a percentage of the swept volume.

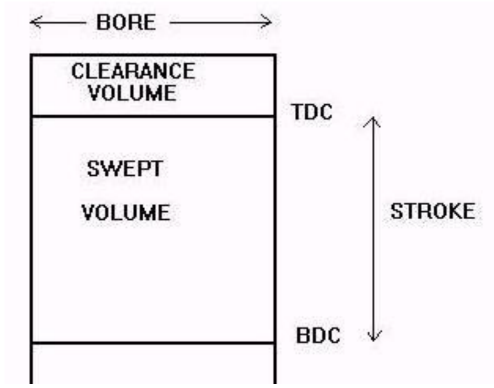


Figure 2.7 Swept and Clearance Volume

2.3.13 Compression Ratio

The compression ratio of an engine is the ratio of its maximum volume (total volume) to its minimum volume (TDC position).

2.3.14 Top Dead Centre (TDC)

Top dead centre is the position of the piston when it is furthest from the crankshaft. The connecting rod is vertical and only the clearance volume remains inside the cylinder. It is the highest point of the piston travel inside the cylinder.

2.3.15 Bottom Dead Centre (BDC)

Bottom dead centre is the position of the piston when it is nearest to the crankshaft. It is the lowest point of the piston travel inside the cylinder.

2.3.16 Cylinder

It is one of the most important parts of the engine, in which the piston moves to and fro in order to develop power. The engine cylinder has to withstand a high pressure (more than 50 bar) and temperatures (more than 2000 °C). Thus the material for the engine cylinder should be such that it can retain sufficient strength at such a high temperature and pressure. For ordinary engines, the cylinder is made of ordinary cast iron. But for heavy duty engines, it is made of steel alloys or aluminum alloys.

2.3.17 Cylinder Head

It is fitted on one end of the cylinder and acts as a cover to close the cylinder bore. Generally, the cylinder head contains inlet and exit valves for admitting the fresh charge and exhausting the burnt gases. In petrol engines, the cylinder head also contains a spark plug for igniting the fuel-air mixture, towards the end of compression stroke. But in diesel engines, the cylinder head contains nozzles for injecting the fuel into the cylinder. The cylinder head is cast as one piece and bolted to one end of the cylinder. The cylinder block and cylinder head are made from the same material. A copper or asbestos gasket is provided between the engine cylinder and cylinder head to make an air-tight joint.

2.3.18 Intake and Exhaust Valves

The intake and exhaust valves open at the proper time to let in air and fuel and to let out exhaust. Note that both valves are closed during compression and combustion so that the combustion chamber is sealed.

2.3.19 Spark Plugs

The spark plug supplies the spark that ignites the air/fuel mixture so that combustion can occur. The spark must happen at just the right moment for things to work properly.

2.3.20 Crankcase

It is a cast iron case that holds the cylinders and crankshaft of an IC engine. It also serves as a sump for the lubricating oil.

2.3.21 Flywheel

It is a big wheel mounted on the crankshaft whose function is to maintain its constant speed. It stores energy during the power stroke and then returns it during the other stroke.

Figure 2.8 Flywheel

2.3.22 Engine Balancing

Counter weights and dampers are placed on the crank shaft to counter the many imbalance forces that are created due to the motion of the pistons and the connecting rods. A crankshaft has to be balanced to reduce vibrations and increase performance and reliability.

2.3.22.1 Internal Balancing

A crankshaft is said to be internally balanced when the counter weights alone are able to balance the crankshaft.

2.3.22.2External balancing

A crankshaft is said to be externally balanced when the counterweights are too light and extra material needs to be added to balance it.

2.3.23 Bob Weight

In balancing, bob weight plays a very important role. Bob weight is indicative of the mass present on a single crank pin. It includes both the rotating and the reciprocating weight portions of the entire assembly.

2.3.24 Knocking

Knocking occurs when combustion of the air/fuel mixture does not start off correctly in response to ignition. When the combustion does not occur at the precise position of the stroke, an explosion occurs, creating a metallic pinging sound and increasing the cylinder pressure drastically. This is extremely damaging to the engine.

2.3.25 Yield Strength

Yield strength or yield point is the stress at which plastic deformation occurs. This means that the material does not return to its original shape after removal of load.

CHAPTER: 3

RESEARCH AND FINALISED SOLUTIONS

3.1 Selection of Material

Crankshafts and connecting rods are usually made of materials that are easily shaped, machined, heat treated and have adequate strength, toughness, hardness and high fatigue strength. Medium carbon steel alloys are used that contain iron and a small percentage of carbon (0.25 to 0.45) along with other several other alloying elements such as manganese, chromium, molybdenum, nickel, silicon, cobalt, vanadium, aluminum or titanium.

3.1.1 Effects of alloying elements

- **Carbon**

  Carbon is the single most important alloying element in steel and its presence is necessary to make steel. It increases hardness, toughness and wear resistance.

- **Manganese**

  Manganese helps in increasing the steels hardness, machinability and yield and reduces brittleness by combing with Sulphur and Phosphorus.

- **Molybdenum**

  Molybdenum increases the hardness penetration of steel and increases high temperature tensile strength.

- **Chromium**

  Chromium has a tendency to increase hardness penetration. Chromium also increases toughness of steel and its wear resistance. It also makes it stain and corrosion resistant.

- **Silicon**

  It increases the strength, toughness and hardness penetration of steel.

- **Sulphur**

  Sulphur helps in improving the machinability of steel.

- **Phosphorus**

  Phosphorus increases strength and hardness but at the expense of ductility and impact to toughness. It increases resistance to corrosion and improves machinability.

Almost all materials that are used to make crankshafts and connecting rods have the above mentioned materials in them to enhance their properties.

3.1.2 Our material

Common grades that have these elements and are used for the said purpose are AISI 4340, AISI 4330, AISI 4140, AISI 4130 and AISI 5140. However the best material for lightweight applications is AISI 4130 and AISI 4140. Upon market survey we decide to use **AISI 4140**, as it easily available and is also one of the best materials for the purpose.

AISI 4140

The following table shows the chemical composition of AISI 4140 alloy steel.

Element	**Content (%)**
Iron, Fe	96.785 - 97.77
Chromium, Cr	0.80 - 1.10
Manganese, Mn	0.75 - 1.0
Carbon, C	0.380 - 0.430
Silicon, Si	0.15 - 0.30
Molybdenum, Mo	0.15 - 0.25
Sulfur, S	0.040
Phosphorous, P	0.035

Mechanical Properties

The following table outlines the mechanical properties of AISI 4140 alloy steel.

Properties	**Metric**
Tensile strength	850 MPa
Yield strength	710 MPa
Bulk modulus (typical for steel)	140 GPa
Shear modulus (typical for steel)	80 GPa
Elastic modulus	190-210 GPa
Poisson's ratio	0.27-0.30
Elongation at break (in 50 mm)	25.70%
Hardness, Brinell	197
Hardness, Knoop (converted from Brinell hardness)	219

3.2 First Approach– Reduction of bore

This section describes the first approach that the group took towards the process of de-stroking the engine.

Reducing the bore

One of the methods of downsizing an engine was to reduce the bore of engine by inserting a sleeve in the cylinder. This would result in decrease of swept volume. The calculation of sleeve diameter is as under:

New Bore by keeping same stroke:

Swept Volume = $\frac{\pi}{4} \times D^2 \times h$

New Swept volume = 610 cm^3

Putting stroke = 70.4 mm

$610 \times 10^{-6} = \frac{\pi}{4} \times D^2 \times 70.4 \times 10^{-3}$

$D^2 = 36.77 \times 10^{-4}\ m^2$

D = 0.0606 m

D = 60.6 mm

Original bore was = 63 mm

End to end distance between the valves in diagonal = 60.5 mm

Our sleeve's diameter = 60.6 mm

This means clearance of 1 mm, which is not sufficient

So it was not possible to go with this approach.

3.3 Second Approach– Reducing crank radius

Second possible method of downsizing an engine is to reduce the stroke of the engine. To reduce the stroke, we have to come up with a solution, so that the piston travels less linear distance inside the cylinder. For this we had to decrease the crank radius. The crank radius is the distance between the main journal and pin journal and it is half of the stroke of engine.

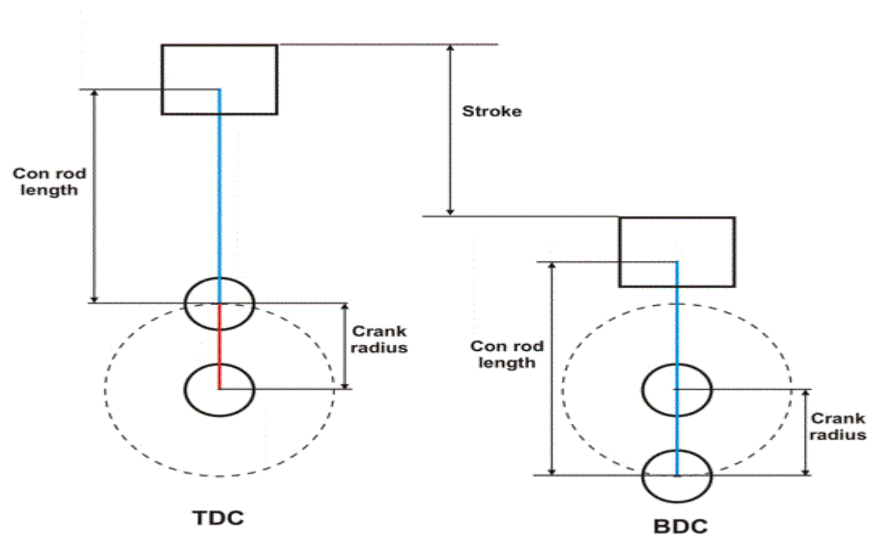


Figure 3.1 Crank radius and stroke

$$\text{Crank radius} = \frac{\text{Stroke}}{2}$$

This approach of downsizing is also called De-stroking. De-stroking simply means to decrease the stroke of a piston inside a cylinder. The original travel from TDC to BDC is reduced by introducing a new TDC and BDC. Consequently, the swept volume of the piston is also reduced which when added up with the volume from all cylinders, results in a lesser total displacement of the engine. To lessen the stroke, the team decided to alter the crank radius.

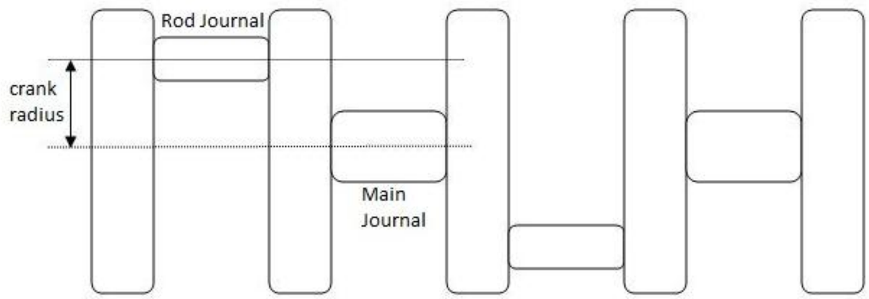


Figure 3.2 Crank radius

3.3.1 Method for reducing the crank radius – Welding more material

To reduce the crank radius, we decided to decrease the axial distance between crank pin and main journal.

The crankshaft is one complete component and not made up of separate components. Hence the only way to change the radius was to add more material onto the rod journal by welding and machine it again to obtain the, dimensionally same, rod journal at a lesser distance from the crankshaft axis of rotation. In simpler words, we decided to move the rod journal inwards towards the axis of rotation.

The material was to be TIG welded onto the rod journal. Then on a CNC Lathe machine, the crankshaft was to be turned to obtain the new rod journal positions.

3.3.1.1 Disadvantage of welding

Connecting rods and crankshafts are components that are subjected to high fatigue stresses. They are constantly under high pressure and in constant motion. Thus the material should be structurally strong to endure the infinite number of stress cycles.

The welded steel components have very low endurance limit and they are prone to fatigue failure. Microscopic flaws remain that are not visible to the naked eye and can have severe effects on the integrity of the weld. Fatigue induced cracks can propagate and cause both life and financial damage.

So taking these factors in to consideration, we decided to abandon this approach and move towards the alternative solution.

3.4 Third Approach Designing of new crankshaft

Another method to de-stroke the engine was to make entirely new crankshaft and connecting rods. This was a more complicated task as it involved the designing of the components from scratch, keeping in mind the added effects of turbo charging.

3.4.1 Altering the connecting rods

De-stroking does not end with just re-designing the crankshaft. By decreasing the stroke due to decrease in crank radius, the compression ratio of the engine is also changed. The new TDC is now lower and hence increases the clearance volume. To correct this, new connecting rods were to be made; keeping in mind the effect turbo charging has on compression ratio. Returning the engine to its original compression ratio is not appropriate as it may induce knocking under high pressure turbo charging which is dangerous for engine life.

The length of the connecting rods was to be increased and the cross section was to be changed keeping in mind the increased pressure in the combustion chamber due to turbo charging.

Our Connecting Rod

Following factors were taken into consideration while designing the new connecting rods.

- Elevated combustion pressure in the cylinder due to turbo charging.
- Altered compression ratio required for turbo charging
- Increase in moment of inertia of connecting rod cross section to prevent buckling

3.4.1.1 Cross Section of the Connecting Rod

The new cross section of the connecting rod required the determination of the maximum force that would act on the rod. The following formula was used to calculate the force

$$\text{Pressure} = \frac{\text{Force}}{\text{Area}}$$

The total pressure in the combustion chamber increases to 130bar or 13 MPa after turbo charging.

The area of the piston was calculated as follows:

$$\text{Area} = \pi r^2$$

'r' is the radius of the piston face and it is equal to half of the stroke = $\frac{63\text{mm}}{2}$ =31.5mm or 0.0315m

Area= 3.1176 x 10^{-3} m^2

The new maximum pressure in the cylinder is 130 bar and this point of maximum pressure is achieved just after ignition.

Hence,

$$130 \times 1.0135 \times 10^5 = \frac{F}{3.1176 \times 10^{-3}}$$

F = 41.065 kN

The normal factor of safety incorporated in such situations is 1.5

$$1.5 = \frac{Force}{41.065}$$

Force = 61.598kN

The new cross section of the connecting rod should withstand a force of 61.598kN.

Considering these conditions, a lot of materials were taken into consideration based on yield strength and endurance limit. The material chosen for this purpose is AISI 4140 which is molybdenum chromium steel

The chosen material, AISI 4140 steel, has yield strength of 327.5MPa. This means that at 327.5MPa, the material undergoes plastic deformation.

The cross sectional area that would withstand the maximum force was then calculated as shown:

$$\text{pressure} = \frac{\text{force}}{\text{area}}$$

$$327.5 \times 10^6 = \frac{61.598 \times 10^3}{\text{New Area}}$$

New Area = 1.88 x 10^6

The picture shows the H section clearly visible with the optimum area calculated above.

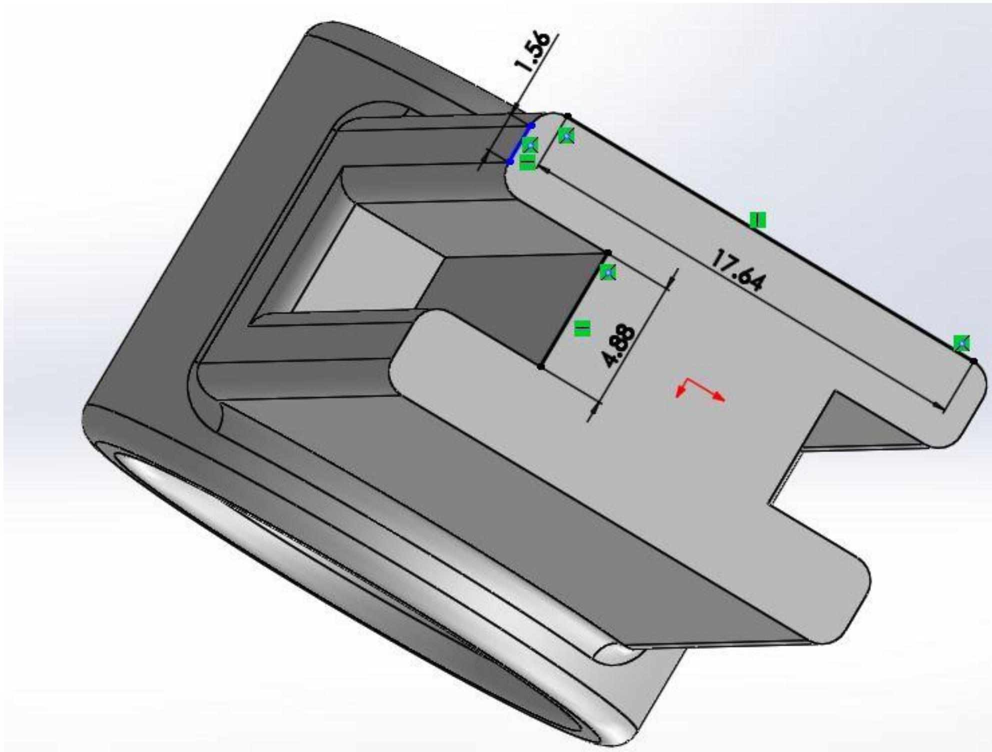


Figure 3.3 Connecting rod cross section

The new section properties are as under

Figure 3.4 Section Properties

Area = 188.002 mm^2

Moment of inertia about the indicated axis is found as

$$I = \frac{1}{12} \times 3.56 \times 19.64^3 + \frac{1}{12} \times 4.88 \times 9.87^3 + \frac{1}{12} \times 3.56 \times 19.64^3$$

$\mathrm{I} = 4885.932\ \mathrm{mm}^4$

This moment of inertia of section of H type connecting rod is approximately 2.6 times greater than the original stock connecting rod which had I type section.

According to the Euler's formula for buckling, the critical buckling load is given by

$$P_{cr} = \frac{\pi^2 EI}{L^2}$$

So it is clear from the above equation that by increasing the moment of inertia, the crictial buckling load increases and the coloumn is more resistant to buckling.

3.4.1.2 Shape of the connecting rod.

The shape of the connecting rod was another critical parameter to decide. There are two standard shapes of connecting rod. One is with I shaped section and other is with H shaped section. Both the shapes have their own pros and cons.

I Beam rods

An I-Beam rod can handle high compressive loads while also providing good tensile strength. But the thickness and strength of the steel in the rod limit what it can safely handle. So performance I-Beam rods are typically made of a higher grade of steel (4340 or 300M), and often have a thicker cross-section in critical areas to increase strength.

Figure 3.5 I beam Connecting Rod

H Beam Connecting Rod

H-Beam rods have a completely different design. An H-Beam rod has two large, flat sides that are perpendicular to the piston pin and crankshaft journal, with a thin center section in the middle. This makes the H-Beam design very stiff so it can handle higher compressive loads without bending.

A study was done to investigate the optimum shape for a connecting rod in turbo charged engines. It was to decide what shape is most resistive to buckling when force is applied.

The H-beam has wider flanges and resists bending, torsion and shear well compared to an I beam of equal dimensions. Thus an H-beam is generally stronger and able to withstand more horse power. So H-beam rods perfectly satisfy our criteria of turbo charging.

Figure 3.6 H Beam connecting rod

3.4.1.3 Length of the connecting rod

The new length of the connecting rod depended on the new compression ratio of the engine. The compression ratio of the stock engine is 10.8. However there are many tradeoffs involved when an engine is turbo charged. Air is pumped into the cylinders by the turbo charger which increases the pressure well over the normal pressure. In addition, the piston also increases pressure. This results in very high temperatures inducing ignition before the firing of spark plugs, called knocking. This is disastrous for the pistons and the connecting rods. Thus the compression ratio of such engines needs to be reduced. Normally the compression ratio of turbo charged engines is kept between 8 and

9. We have kept the new compression ratio at 8.5. Following is how the new length was determined:

Dimensions:

Original Bore = 63mm

Original stroke = 70.4mm

Swept Volume (one cylinder) = $\pi r^2 h$ = 219.48cm^3

Number of cylinders = 3

Total Volume = 219.48 x 3 = 658.5cm^3

Original crank radius = $\frac{stroke}{2}$ = 35.2mm

New crank radius (after iterations) = 32.5mm

New Stroke = 65mm

New Swept Volume = $\pi r^2 h$ = 202.64cm^3

Compression Ratio = C_r = $\frac{\text{swept volume + clearance volume}}{\text{clearance volume}}$

Swept Volume = V_s, Clearance Volume = V_c

Finding out the length of cylinder that made up the clearance volume:

10.8 V_c = 219.48 + V_c

9.8 V_c = 219.48

V_c = 22.45cm^3

$V_c = \pi r^2 h$

$h = \frac{V_c}{\pi r^2}$

h = 7.2mm

Finding out the length of cylinder that made up the new clearance volume:

$8.5\ V_c = 202.64 + V_c$

$7.5\ V_c = 202.64$

$V_c = 27.02cm^3$

$V_c = \pi r^2 h$

$h = \frac{V_c}{\pi r^2}$

h = 8.6mm

The original connecting rod was 117.47mm in length.

The difference in stroke length = 70.4 – 65 = 5.4mm

The TDC now moved to a new location i.e. $7.2 + \frac{5.4}{2} = 9.9$mm from the top of the cylinder.

The new TDC should be 8.6mm from the top of the cylinder.

Hence the length that should be added to the connecting rod = 9.9 – 8.6 = 1.3mm

New Length of connecting rod = 118.77mm

3.4.1.4 CAD models of connecting rod

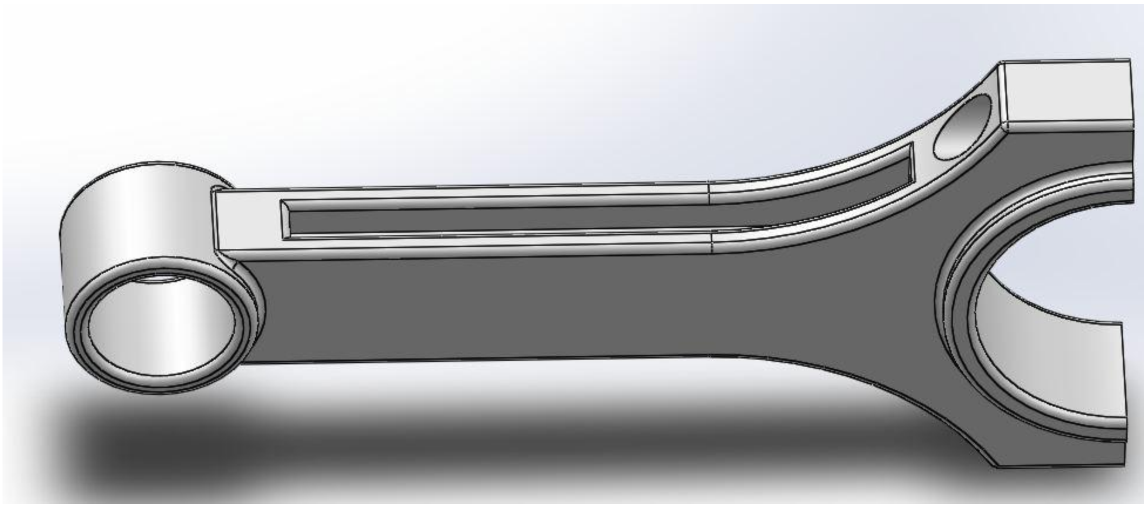


Figure 3.7 Newly Designed Connecting Rod

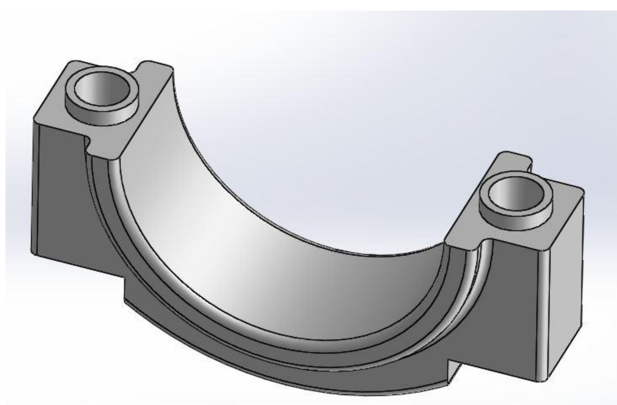


Figure 3.8 Connecting Rod big end cap

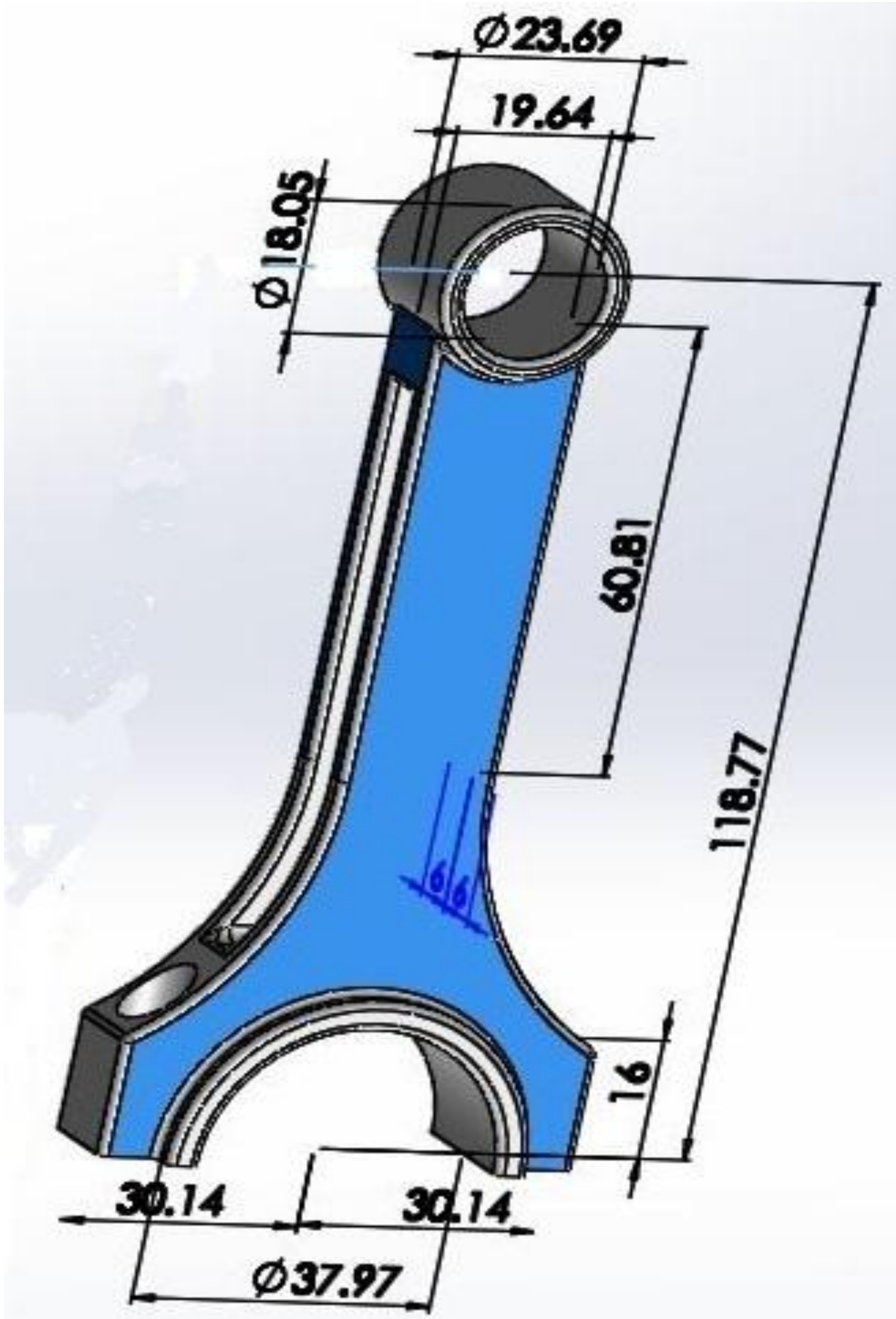


Figure 3.9 new con rod dimensions

3.4.1.5 Connecting Rod drawings

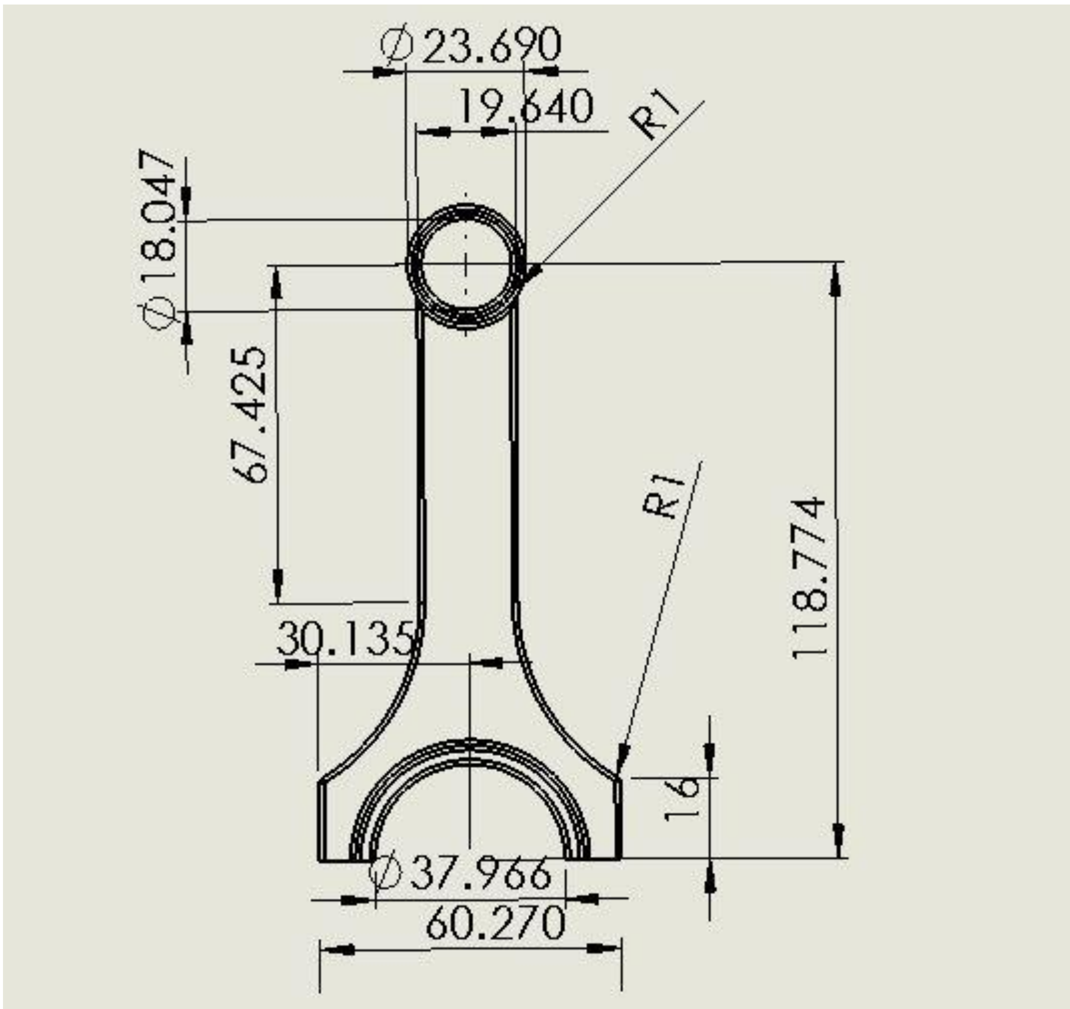


Figure 3.10 Connecting rod drawing

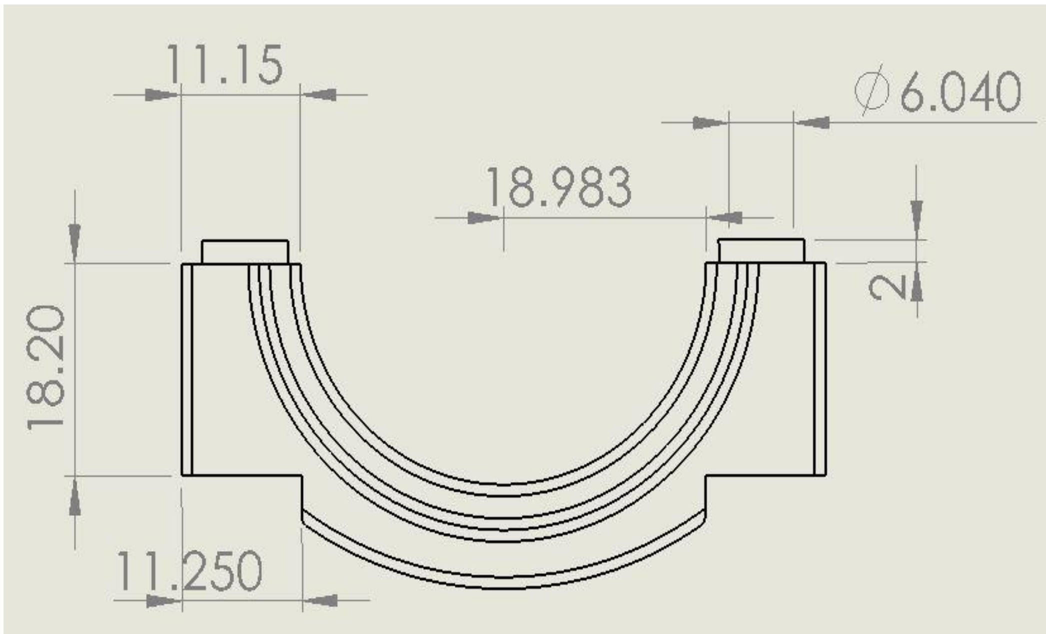


Figure 3.11 Big end cap drawing

3.4.1.6 Stress Analysis of connecting Rod

The stress analysis of connecting rod was done on ANSYS 14.5 in static structural module. The analysis was done for the harshest operating condition such as maximum gas pressure just after combustion. Maximum deformation and equivalent Von Mises Stress was calculated. The maximum stress was lying within the endurance limit i.e. 50% of yield strength. This means that connecting rod is safe for unlimited cycles and fatigue failure would not occur.

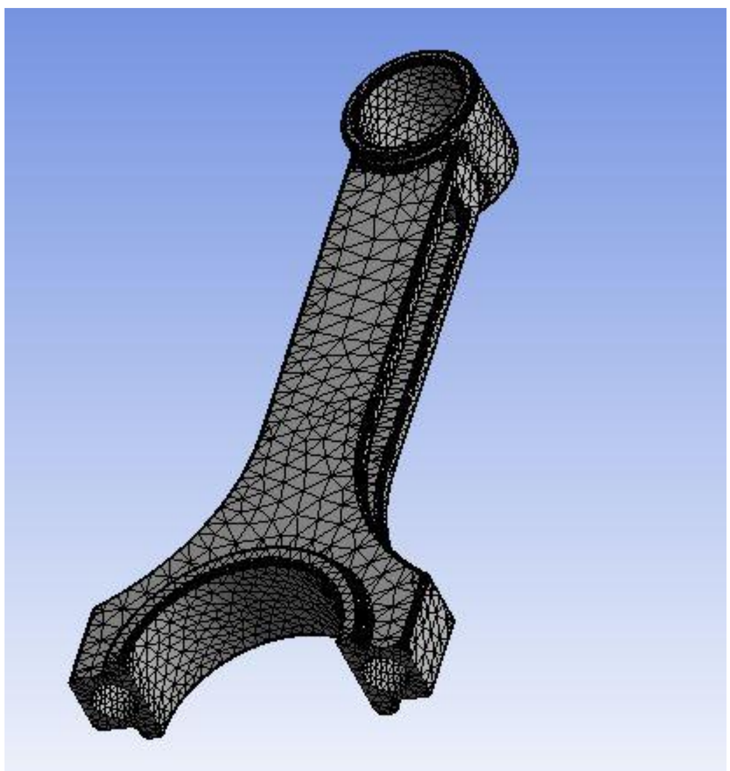


Figure 3.12 Connecting rod mesh

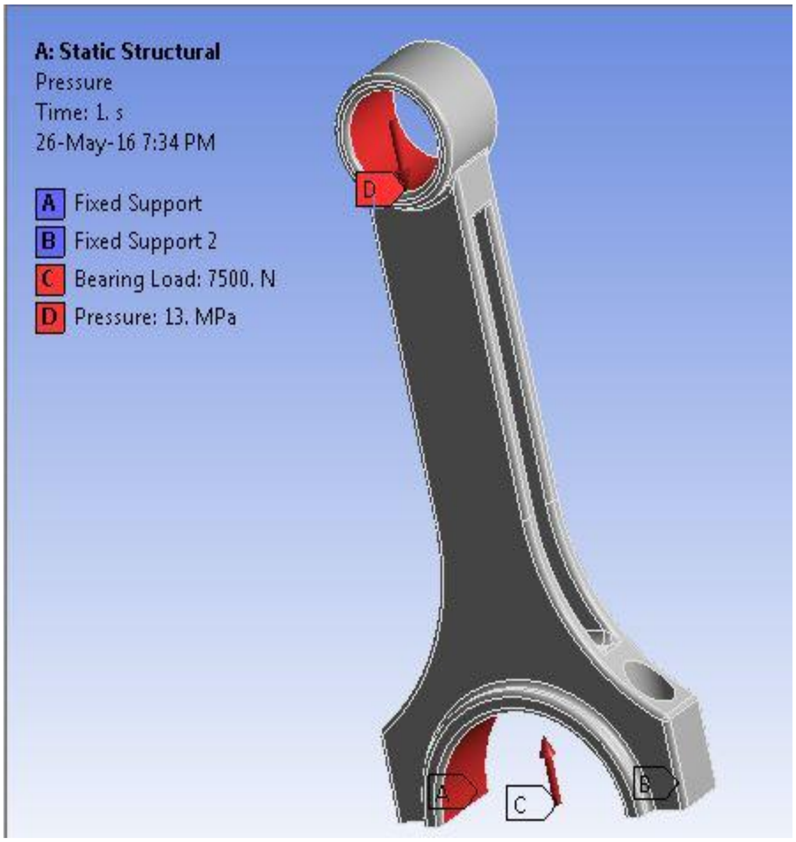


Figure 3-9 Connecting Rod Forces

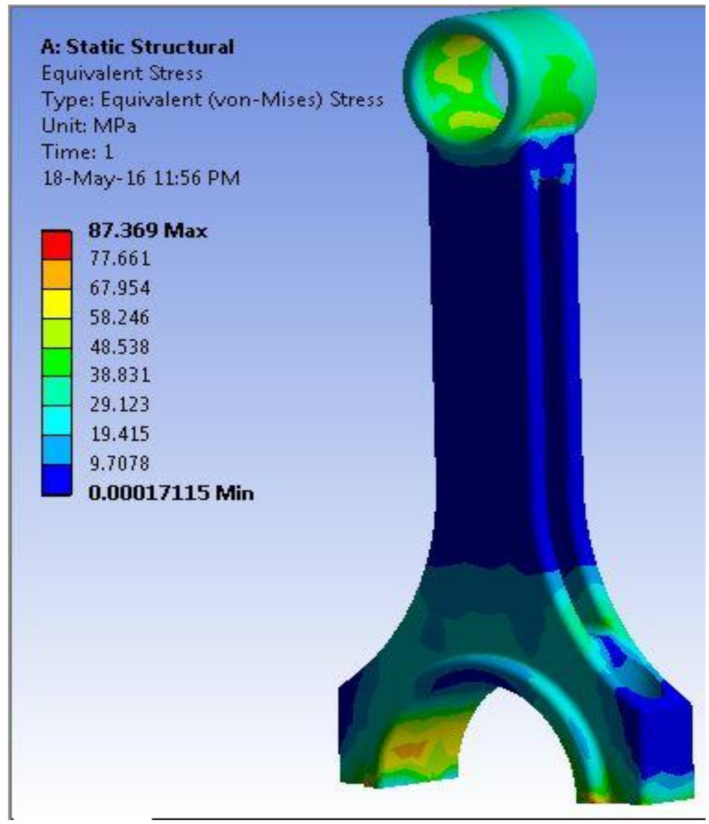


Figure 3-10 Connecting Rod Stresses

3.4.1.7 Manufacturing of Connecting Rods

The connecting rods were manufactured on CNC milling machine. The CAD models were converted into igs format and then there G codes and M codes were generated. These codes were fed into CNC milling machine and the connecting rods were manufactured up to accuracy of 0.01 mm.

Figure 3-11 Connecting Rod manufacturing

Figure 3-12 CNC milling

3.4.2 New Crankshaft

There were number of factors which were taken into consideration while designing the new crank shaft

- Overall dimensions of the counterweights and the shaft as a whole to be kept as close as possible to the original crankshaft. This is important as the new crankshaft has to fit and move in the space available in the crank case.
- Change of weight in crankshaft counter weights due to increased weight of other components
- Keeping the distance between rod journals equal to that in the stock crankshaft.

3.4.2.1 Main Journals and Rod Journals

Using Vernier calipers the rod journals and main journals from the original crankshaft were measured. The journals in the new crankshaft are made of the exact same dimensions and their mutual distances are kept equal to ensure that they coincide with the cylinders perfectly.

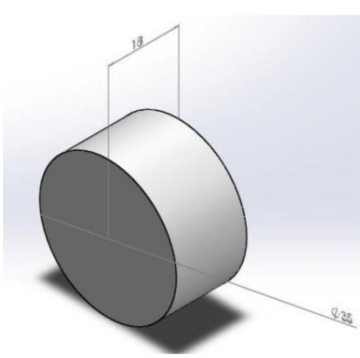


Figure 3-17 Rod Journal

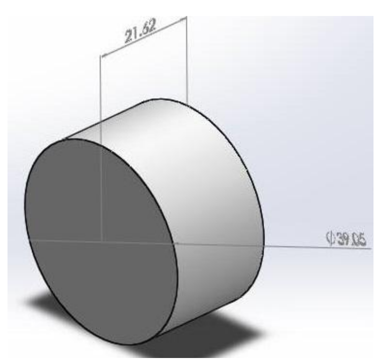


Figure 3-18 Main Journal

3.4.2.2 New counterweights

Counterweights in a crankshaft serve to reduce vibrations. They nullify the reactions caused by the rotations and help achieve higher RPMs smoothly. Due to change in the mass and length of the connecting rod, new counterweights were made. However the overall dimensions (maximum diameter of the circle of rotation and maximum length of the crankshaft with journals and counterweights attached) were kept into consideration at all times. Following figure shows how one counterweight is represented.

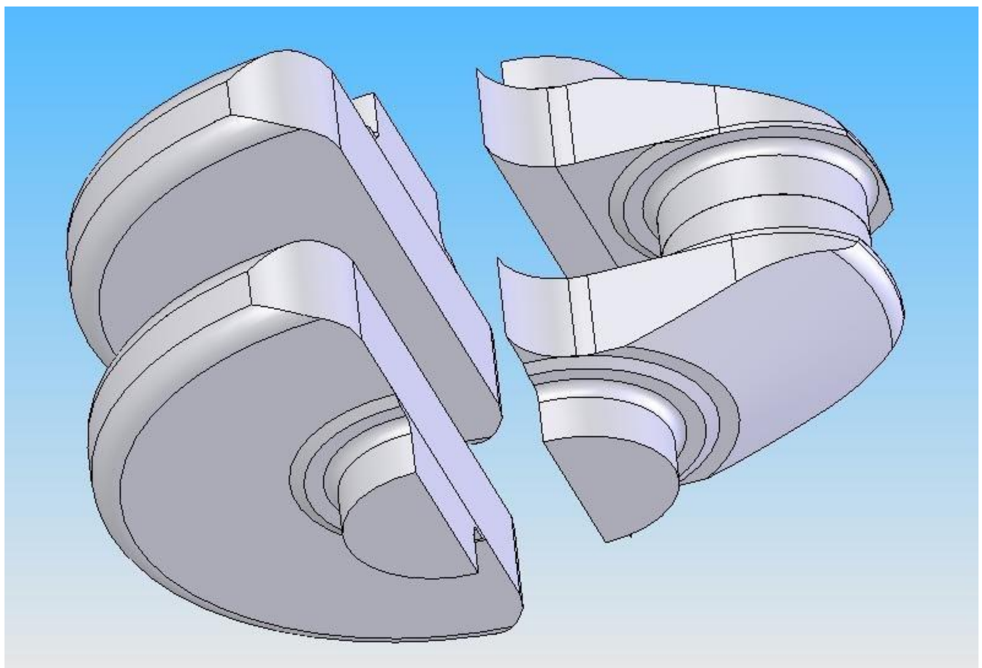


Figure 3.19 Counterweight Representation

In the figure above, the part on the left is the counterweight and the part on the right is the crank arm.

To calculate the exact counterweight, the moments on either side of the rotational axis (central axis of the main journals) should balance each other. Counterweight determination is different than balancing.

Following formula was used to calculate the counterweights:

$$CW \times A = (CA \times B) + (BE + (0.5 \times SE) \times C$$

Where,

CW	=	weight of the pair of counterweights
A	=	distance b/w main journal axis and COG of counterweights
CA	=	weight of the crank arm.
B	=	distance b/w main journal axis and COG of crank arm.
BE	=	big end weight of the connecting rod.
SE	=	small end weight of the connecting rod
C	=	distance b/w main journal axis and axis of con-rod journal.

a. Counterweight corresponding to the middle cylinder was analyzed for these measurements as follows:

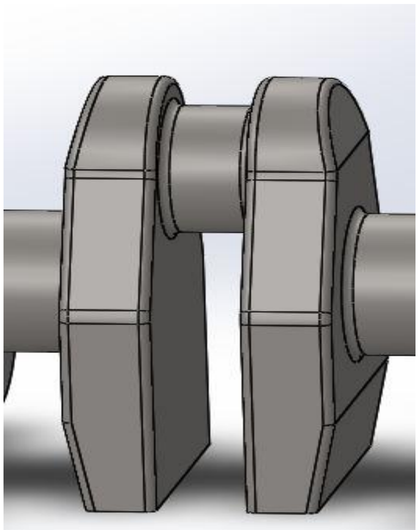
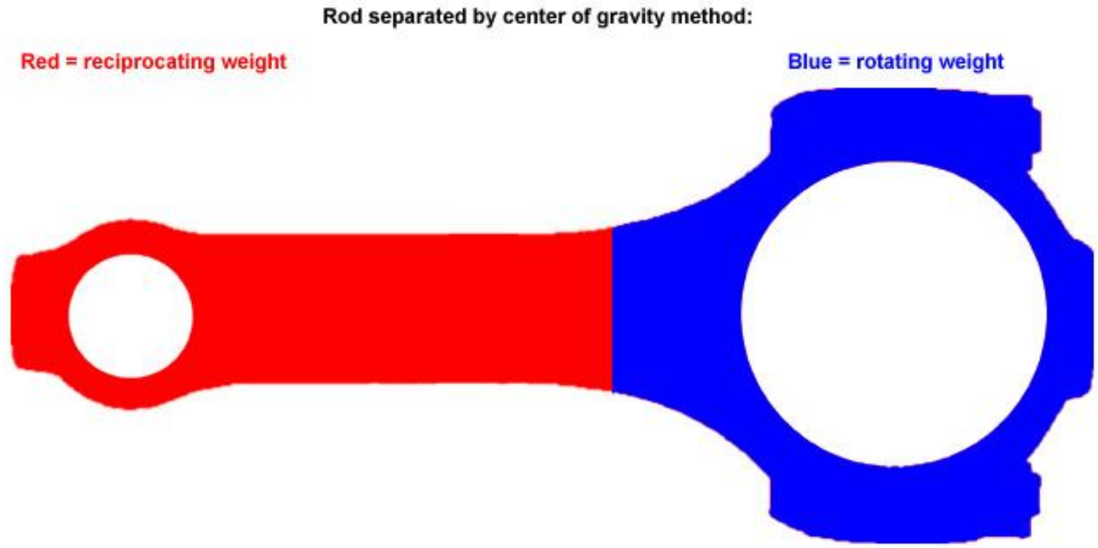


Figure 3.20 Middle cylinder counter weight

Big end of a connecting rod is also called the rotating end. Small end of a connecting rod is also called the reciprocating end.



Now

BE = Big end of rod (including fasteners) + Bearings + Oil

BE = 227.38 +30 + 4 = 261.38g

Weight of small end was calculated as follows:

SE = Piston + Wrist Pin + Small end of rod

SE = 136 + 52 + 112.38 = 300.38g

CA = 831.29g (from SolidWorks Mass Properties)

B = 23.58mm (from SolidWorks Mass Properties)

C = 32.5mm (the new crank radius)

CW x A = (CA x B) + (BE + (0.5 x SE) x C

Putting values in above equation

CW x A = (831.29 x 23.58) + (261.38 + (0.5 x 300.38)) x 32.5

CW x A = 32977.83 g.mm

The combination of counterweight and its centre of gravity should be such that its product is equal to 32977.83 g.mm

.

Trials were done by adding different amount of weights to the counterweights and then computing the new weight with its centre of gravity. After numerous iterations following combination was finalized as it gave the closest results.

Counterweight = **CW = 1175.88 gm**

A = 27.65 mm

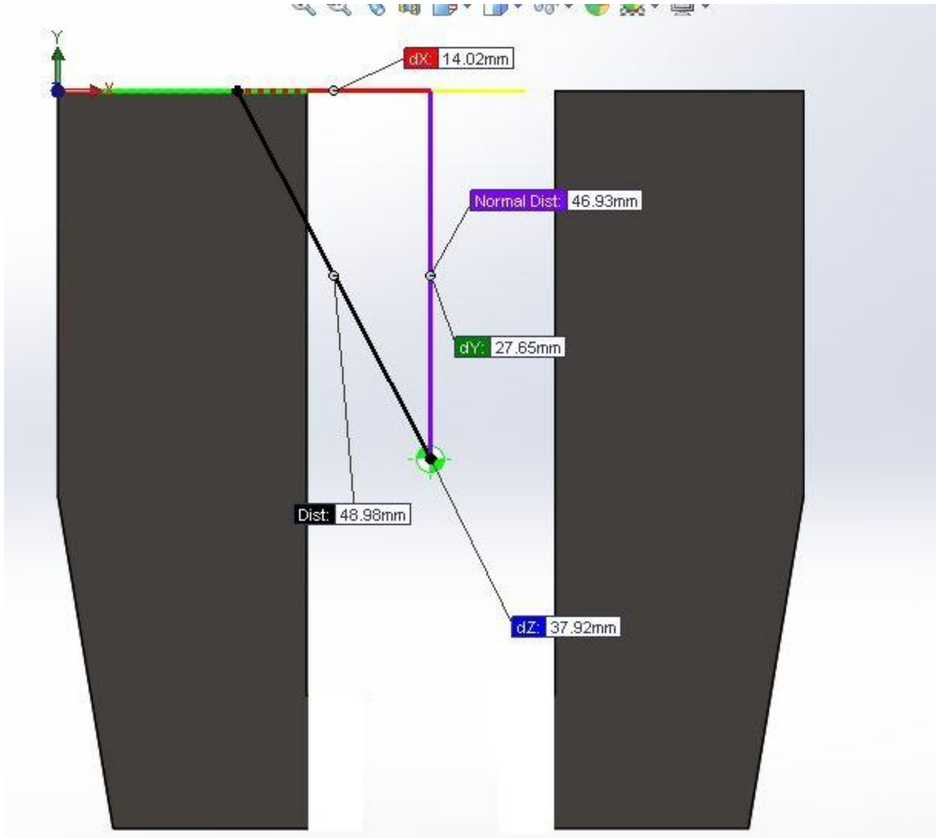
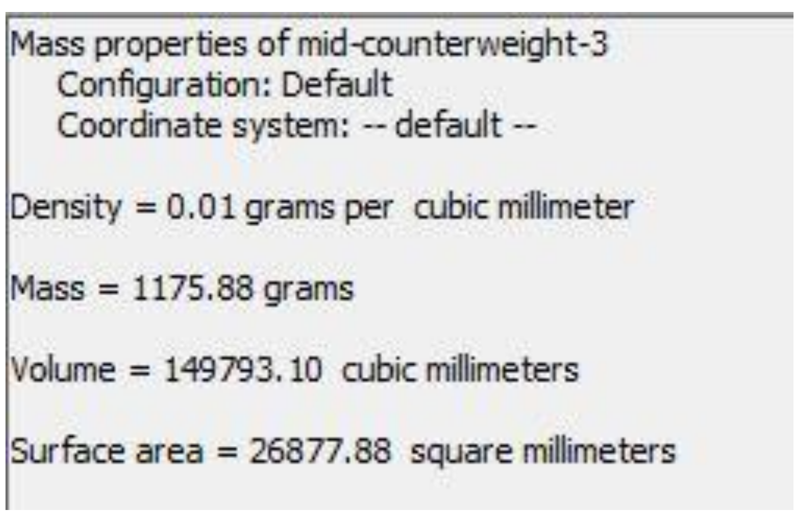


Figure 3.2113 Middle counter weight geometry calculations



Mass properties of mid-counterweight-3
Configuration: Default
Coordinate system: -- default --

Density = 0.01 grams per cubic millimeter

Mass = 1175.88 grams

Volume = 149793.10 cubic millimeters

Surface area = 26877.88 square millimeters

Figure 3.22 Middle counter weight mass properties.

(SOLIDWORKS)

Following is the finalized shape of the counterweights.

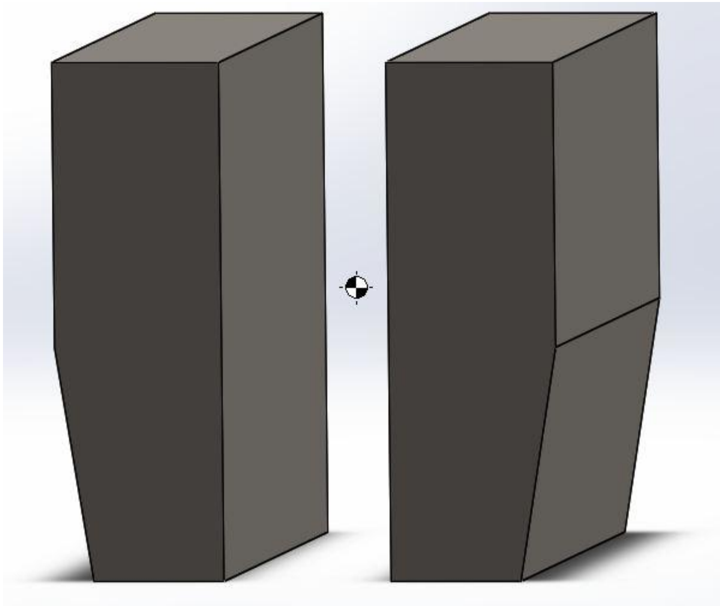


Figure 3.14 Finalized counter weight

b. Counterweight corresponding to the first and third cylinder were analyzed for these measurements as follows:

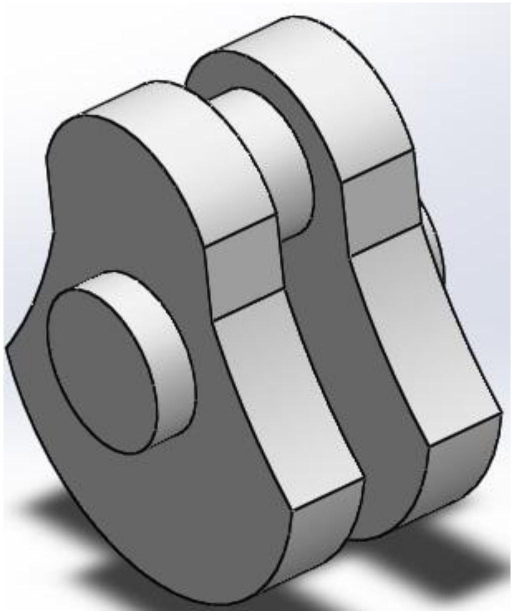


Figure 3.24 Middle counter weight

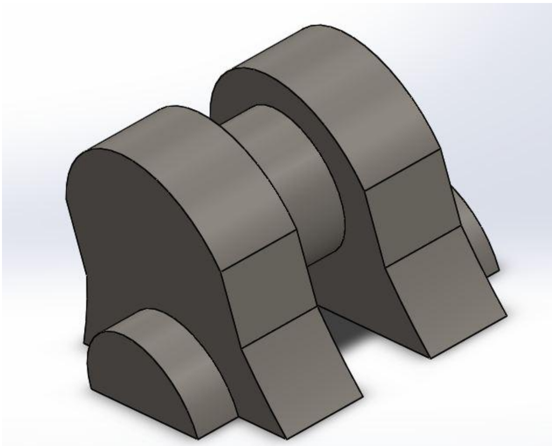


Figure 3.25 Crank arm

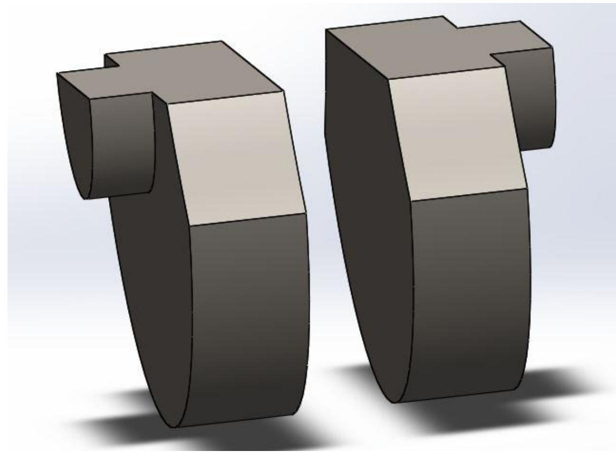


Figure 3.26 Counter weight

C = 32.5mm (the new crank radius)

CW x A = (CA x B) + (BE + (0.5 x SE) x C

Submitting values in above equation,

CW x A = (1241.65 x 25.06) + (261.38 + (0.5 x 300.38)) x 32.5

CW x A = 31115.749 + 13376.025

CW x A = 44491.774

The combination of counterweight and its centre of gravity should be such that its product is equal to 44491.774g.mm.

Trials were done by adding different amount of weights to the counterweights and then computing the new weight with its centre of gravity. After numerous iterations following combination was finalized as it gave the closest results.

Counterweight = CW =1685.31g

A = 26.58

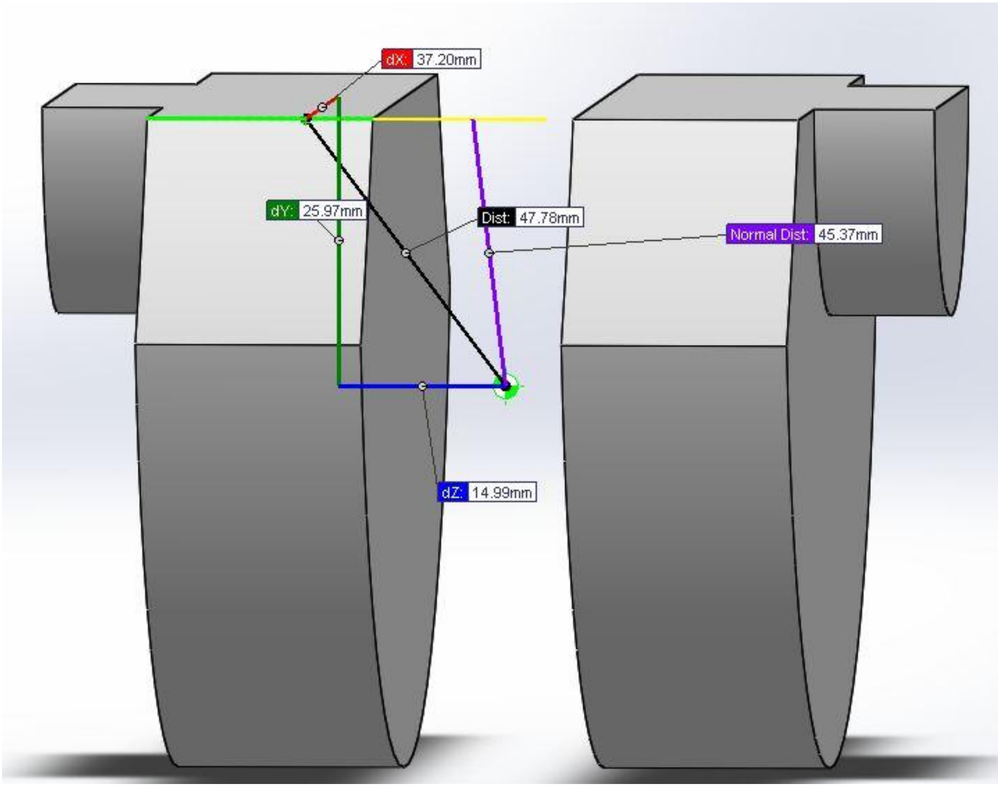


Figure 3.27 Middle counter weight geometry (SOLIDWORKS)

Mass properties of Crank3
Configuration: Default
Coordinate system: -- default --

Density = 0.01 grams per cubic millimeter

Mass = 1627.93 grams

Volume = 207379.20 cubic millimeters

Surface area = 32057.55 square millimeters

Center of mass: (millimeters)
X = 0.00
Y = -25.97
Z = 107.86

Figure 3.28 Middle counter weight mass properties

Following is the finalized shape of the counterweights.

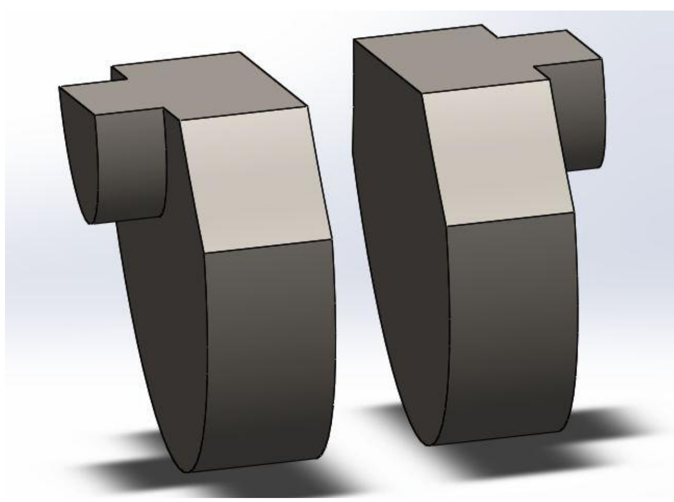


Figure 3.29 Finalized counter weight

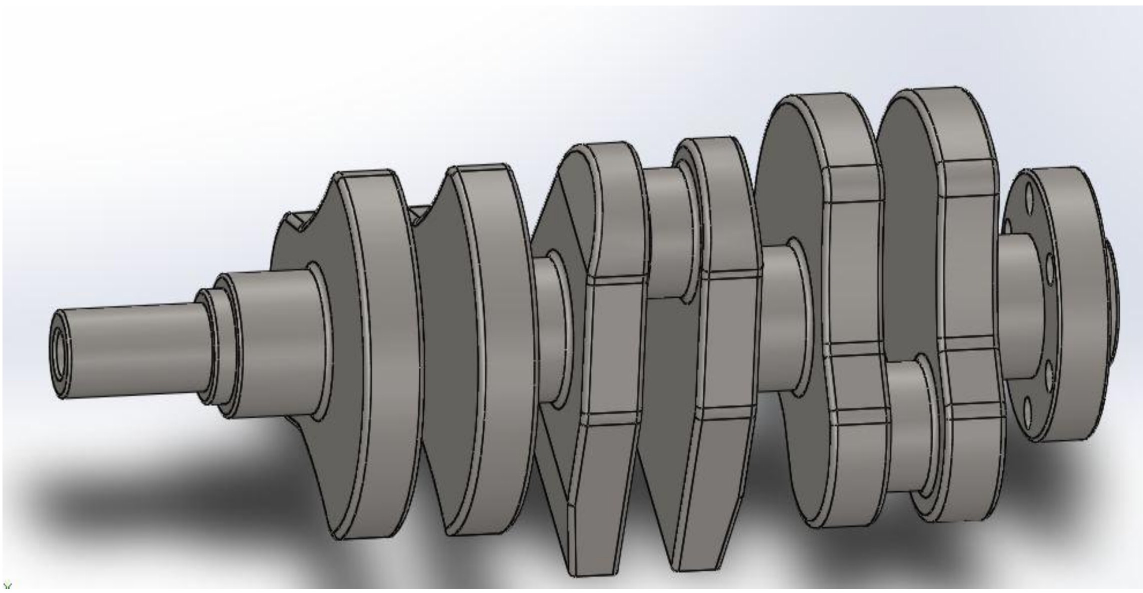


Figure 3.30 Finalized crank shaft

3.4.2.3 Crank Shaft drawings

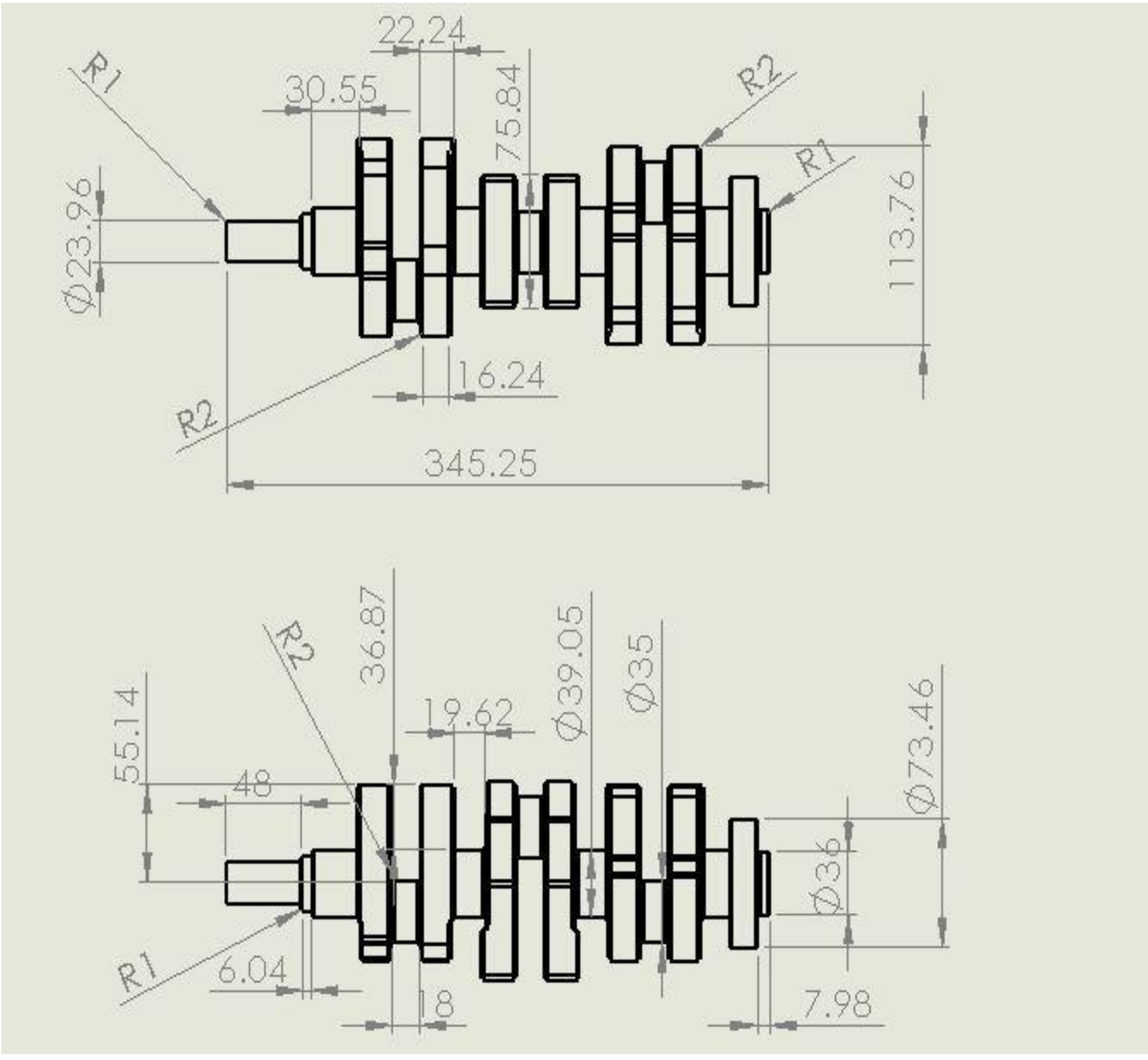


Figure 3.31 Crank Shaft drawing

3.4.2.4 Stress Analysis of Crank Shaft

The crank shaft is the most critical component in engine and it is subjected to repeated stresses. So its stress analysis was the most critical part of design.

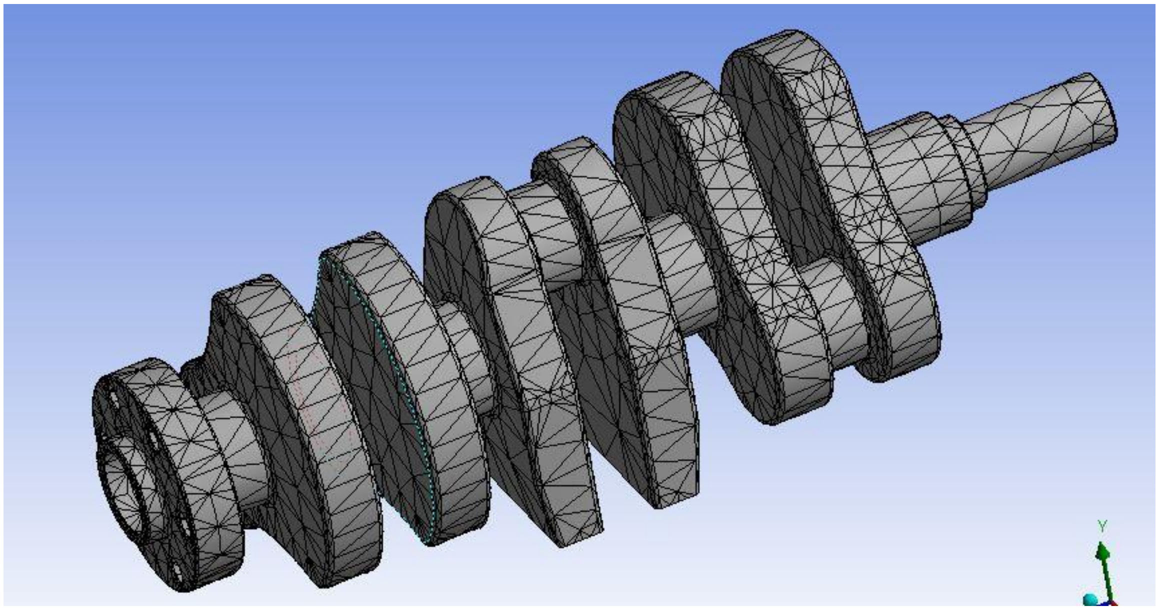


Figure 3.32 Crankshaft Mesh

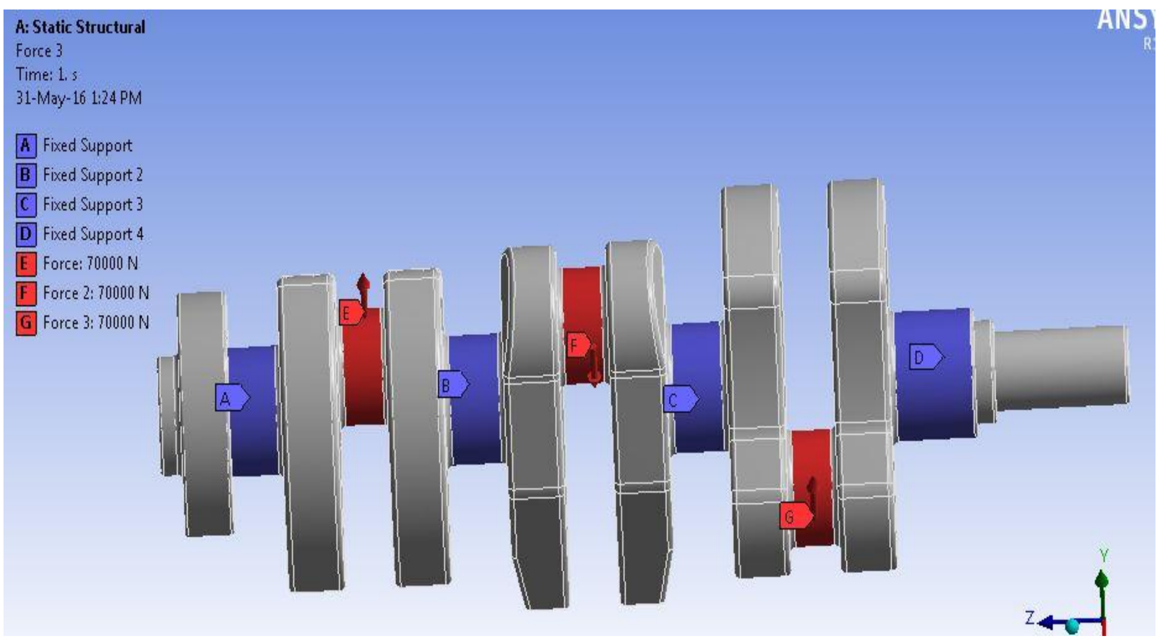


Figure 3.33 Crank shaft forces

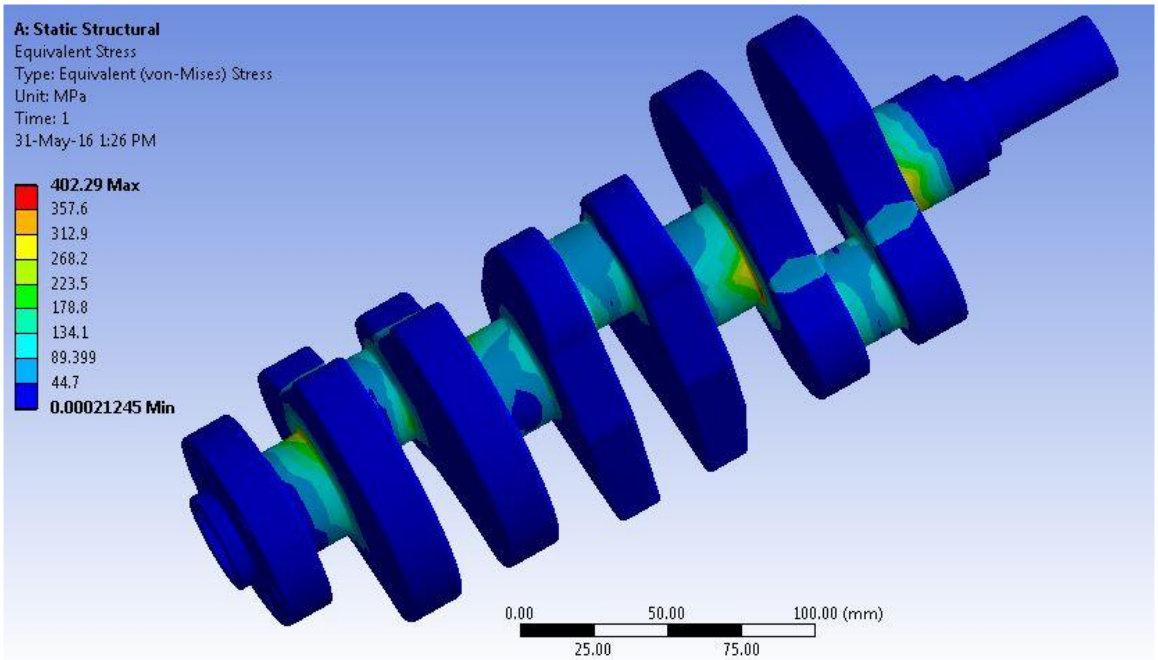


Figure 3.15 Crank Shaft stresses

3.4.2.5 Manufacturing of Crankshaft

Due to certain manufacturing limitations such as non availability of rotary indexer as well as non availability of 4 axis CNC, We decided to develop a 3D printed prototype of Teflon for the demonstration and to get a working prototype

Figure 3.35 3D printed crank shaft

3.5 Balancing of the crankshaft

After designing and machining of the crankshaft, it has to be dynamically balanced on a balancing machine. Despite careful considerations while designing, a crankshaft should be balanced on a dynamic balancing machine to eliminate all chances of unwanted vibrations and to ensure smooth engine running.

The crankshaft with the connecting rods attached cannot be mounted on the balancing machine. Instead equivalent bob weights are calculated which are attached to the rod journals to simulate the effect of the connecting rods and the pistons.

Figure 3.36 Crank shaft balancing machine

3.4.1 Calculating Bob Weight

Following formula is used to calculate the bob weight for a crankshaft.

Bob weight = Rotating weight + 0.5(Reciprocating weight)

3.4.1.1 Rotating weight

Big end of rod = 208.42g

Bolts x 2 = 9.48 x 2 = 18.96g

Bearing = 30g

Oil = 4g

Pin = 135.95g

Total = 397.33g

3.4.1.2 Reciprocating weight

Piston = 136g

Wrist pin = 52g

Small end of rod = 112.38g

Total = 300.38g

The weights of all components used above are simulated results from SolidWorks. Material density and component volume was used to calculate the weights.

3.4.1.3 Bob weight

Bob weight = Rotating weight + 0.5(Reciprocating weight)

Bob weight = 547.52g

547.52g would be attached at each rod journal in order to balance it on a balancing machine. Consequently, material can be removed from the counterweights if they are too heavy. In case the counterweights are light, holes are drilled in the counterweights and heavier material (lead or Mallory) is poured into them, making the counterweights heavier.

CHAPTER: 4

RESULTS ACHIEVED

4.1 Final Assembly

The newly manufactured connecting rods and the crankshaft prototype were assembled in the Daihatsu Mira Engine Block and the entire assembly passed all the clearances and tolerances and was giving smooth operation according to the new swept volume calculations.

Figure 4-1 Final Assembly

4.2 Compression Ratio Verification

The new crankshaft and connecting rod were designed to give us the compression ratio of 8.5.

Stock compression ratio = 10.8

So, clearance volume per cylinder = $\frac{219.33}{10.8}$ = 10.8 cc

In stock engine there was no clearance volume in cylinder and all the clearance volume was present in cylinder head; so the clearance volume in the cylinder head becomes known to us

$V_{c\text{-}head}$ = 20.30 cc

In engine with new crank and connecting rods,

$V_{c\text{-}cyl}$ = 7.1 cc as per new volume

Total clearance volume in new engine = $V_{c\text{-}cyl}$ = $V_{c\text{-}head}$ + $V_{c\text{-}cyl}$ = 27.4cc

New Swept Volume per cylinder $= \frac{610}{3} =$ 203.33 cc

The new compression ratio comes out to be = $\frac{203.33+27.4}{27.4} = 8.42$

The design compression ratio was 8.5, so there is a difference of 0.08.

REFRENCES

[1] R. S. Khurmi, J.K. Gupta *A Textbook of Machine Design*, 2005.

[2] Pravardhan S. Shenoy, *Dynamic Load Analysis and Optimization of Connecting Rod*, The University of Toledo: 2004.

[3] Travis M. Blais, *Analysis of Connecting Rod Bearing Design Trends Using a Mode-Based Elastohydrodynamic Lubrication Model*, *KATE GLEASON COLLEGE OF ENGINEERING*, 2013.

[4] Polski Rejestr Statków *CALCULATION OF CRANKSHAFTS FOR DIESEL ENGINES*, 2012.

[5] S.M.Sorte, S.M.Sheikh, *Stress Analysis And Design Optimization Of Crankpin*, International Journal of Science and Modern Engineering (IJISME): 2013.

[6] R.L. Norton, *Design of Machinery*, *2nd Endition*. 1999.

[7] M.Y.K.Niazi, H.Qamar, H.Qazi, *DE-STROKING OF A 660CC KEI CLASS ENGINE, TO FORMULA SAE SPECIFICATIONS, SMME, 2015.*