DESIGN AND MANUFACTURING OF A FUEL-EFFICIENT INTERNAL COMBUSTION ENGINE

A Final Year Project Report

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of the Requirements for the Degree of
Bachelors of Mechanical Engineering

by

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ABSTRACT

In today's day and age, where there is a scarcity of fossil fuels, increase in pollution, and global warming, fuel efficient vehicles have become a need of the hour. Fuel efficiency also brings along a cleaner and greener environment. Unfortunately, fuel efficiency is a concept rarely employed in Pakistan's automotive industry. All of the Motorbikes currently manufactured in Pakistan incorporate a carburetor as a mechanism for fuel delivery instead of Electronic Fuel Injection (EFI) which has proven to be more fuel efficient than carburetor. Our goal is to design an engine based solely on the principle of fuel efficiency, an engine which can be easily manufactured and incorporated in currently used motorbikes. The engine would incorporate Atkinson cycle as the basic heat engine cycle. Fuel injection to engine would be done electronically through an EFI kit. The engine will introduce Atkinson Cycle and Electronic Fuel Injection in Motorbike industry of Pakistan. All in all, this project aims to promote and support less fossil fuel consumption to achieve a greener future.

ACKNOWLEDGMENTS

The authors are thankful to the faculty supervisor, Assistant Professor **Dr. Samiur-Rahman Shah**, whose continuous guidance and encouragement over the entire course of the project has helped us overcome obstacles faced during the various stages of the project. We believe his high expectations and regular involvement in the project are the reasons for the quality of the work that has been produced till this day.

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ABBREVIATIONS

AFR	Air-fuel ratio
EFI	Electronic Fuel Injection
ICE	Internal Combustion Engine
ECU	Electronic Control Unit [1]
GDI	Gasoline Direct Injection
MPFI	Multiport Fuel injection

NOMENCLATURE

V_a	Volume flow rate of air
V_f	Volume flow rate of fuel
P_e	Power of engine
P_{tr}	Traction power
τ_e	Torque of engine
Q_{LHV}	Lower Heating Value

CHAPTER 1: INTRODUCTION

An internal combustion engine is a reciprocating engine in which the chemical energy stored inside the fuel is converted to thermal energy during combustion which is turn converted into useful mechanical work [1]. The concept of internal combustion engines was developed in late nineteenth century. Since then internal combustion engines have proved to be a foundation rock for success in many commercial technologies. Internal combustion engines can provide power in a bandwidth of 0.01 kW to $20x10^3$ kW based on the displacement of engine. Internal combustion engines have taken place as the primary mover technology in early twentieth century. With the advent of ICEs dominancy in automotive filed, it is only imperative that work should be done towards increasing the efficiency of an internal combustion engine.

1.1 Motivation of Work

Since internal combustion engines constitute a majority of the automotive industry now a days, it is essential that the rapid rate at which the fossil fuels are being consumed must be kept in check. For this purpose, there is a dire need to work on the efficiency of vehicles and devise ways to save fuel and create a greener tomorrow. In Pakistan specially, there is a great need to work on fuel efficiency as well as emission control of vehicles. No consideration is made regarding the consumption of fuel in our automotive industry due to which the county might face a major fuel crisis in future. To avert such a situation, it is very important that major work should be done on improving the efficiencies of the engines used in our automotive industry.

Pakistan is the 5th largest motorcycle market with 41% population going to work daily on a motorbike. According to a survey from Pakistan Bureau of Statistics PBS, 7500 new motorcycles are hitting the roads of Pakistan daily. The survey further states that the production of motorcycles jumped a record 22.34% in first four months of fiscal year 2017-18. Motorcycle sales are booming in Pakistan at a rate of over 20%.

Such a huge industry with no consideration towards fuel economy in motorbikes demanded certain measures to be taken. This is why the selected project relates to enhancement of fuel efficiency in smaller engines as they constitute a greater part of Pakistani automotive industry. Outmoded engines with same engine design as decades ago are being currently manufactured and sold in Pakistan. There is no innovation being introduced in the engine design of motorbikes which require for revolutionary measures.

1.2 Problem Statement

Design and Manufacture a fuel-efficient engine while incorporating Atkinson cycle and Electronic Fuel Injection.

1.3 Objectives of Project

The main objective of the project is to achieve maximum possible fuel efficiency for given requirements. The size of discussed engine shall be small comparable to the size of engine in daily use motorbikes. We shall use Atkinson cycle for this purpose which is explained in detail in Chapter 2: Literature Review. Electronic Fuel Injection is used by incorporating Speeduino No2C V0.4.0 and custom made wire harness.

The aim is to test the engine with a concept prototype car designed specifically to achieve maximum fuel- efficiency. The said concept car has a highly aerodynamics design along with low rolling resistance tires to ensure minimum frictional losses.

The project is aimed to serve as the first step towards consideration of fuel efficiency in motorbike industry in Pakistan and will make way for further improvements in coming future.

CHAPTER 2: LITERATUIRE REVIEW

In a combustion engine, a fluid follows a certain cycle of operations influenced by a thermodynamic principle. Thus, the basic operating cycle is known as thermodynamic heat engine cycle. An internal combustion engine is one in which the combustion process occurs in the working fluid that is undergoing the thermodynamics cycle. Such an example of the internal combustion engine is the reciprocating engine [2] in which a piston moves up and down in a cylinder to convert thermal energy obtained from combustion of fuel into mechanical energy.

2.1 Two Stroke Vs Four Stroke Engine

2.1.1 Two Stroke Engine

For a reciprocating two stroke engine, the combustion cycle is completed in one revolution of crankshaft or two strokes of the piston.

i. [3]Compression – From mid to end of upstroke of piston
ii. Power– Start of downstroke to end of downstroke
iii. Exhaust and Intake – End of downstroke to mid of upstroke

[4]In general, in a two stroke engine, the first stroke is a combination of intake and

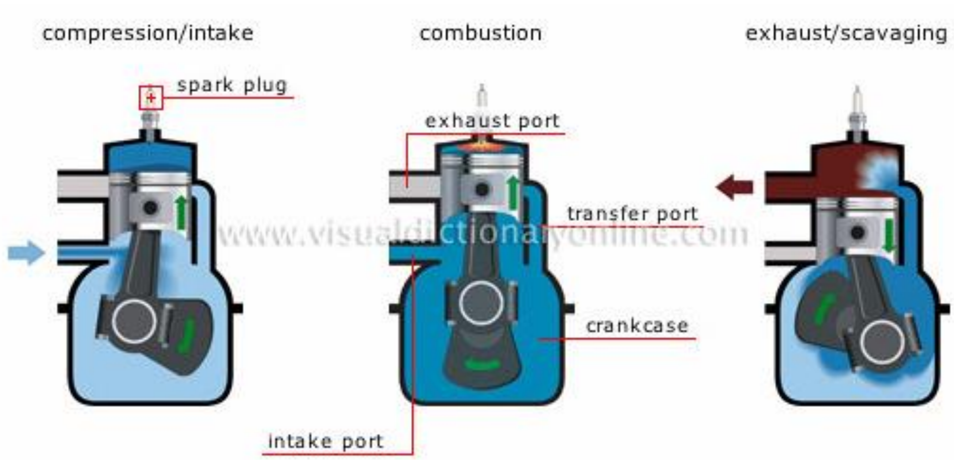


Figure 1: Two Stroke Engine

compression stroke. The second stroke is a combination of power and exhaust stroke.

2.1.2 Four Stroke Engine

In a traditional reciprocating four stroke engine, the combustion cycle occurs over four strokes of the piston or two complete revolutions of crankshaft.

i. Intake – Downstroke
ii. Compression – Upstroke
iii. Power – Downstroke
iv. Exhaust – Upstroke

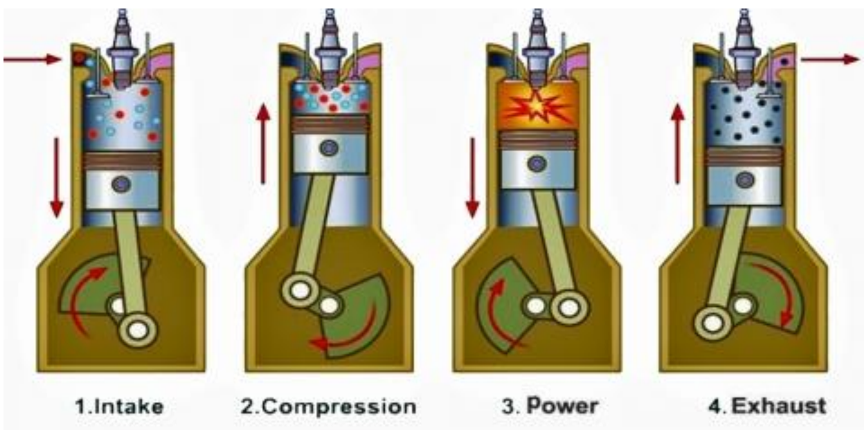


Figure 2: Four Stroke Engine

2.1.3 Comparison

Two Stroke Engine	Four Stroke Engine
Combustion cycle completed in one revolution of crankshaft	Combustion cycle completed in two revolutions of crankshaft
Relatively high specific power output	Relatively low specific power output
Uses port for inlet and outlet of fuel	Uses valves for inlet and outlet of fuel

Less fuel-efficient	More fuel-efficient
Poor scavenging and cylinder filling	Comparatively better scavenging and cylinder filling
Subjected to greater wear and tear	Less wear and tear
Poor emissions control	Comparatively better emissions control
Poor lubrication and lubricating oil consumption	Comparatively better lubrication

Table 1: Engine Comparison

The use of two stroke engines has declined in the near past. They are also less efficient compared to the four stroke engines so we decided to go with the four stroke engine as fuel-efficiency is the main objective of our project.

2.2 Fuel Efficiency

Fuel efficiency of a vehicle can be increased through various measures all of which have certain requirements for their implication. Both transmission and engine account towards the overall efficiency of the vehicle. Fuel efficiency can be increased through:

i. VVT (Variable Valve Timing)
ii. Cylinder Deactivation
iii. Deceleration fuel cut-off, DFCO
iv. Turbocharging/Supercharging
v. Continuous Variable Transmission
vi. Regenerative braking system/Hybrid system
    a. Mechanical Energy Recovery system
    b. Electronic Energy recovery system

vii. Atkinson Cycle

viii. Electronic Fuel Injection

Since our concern is only with the measures involving the engine efficiency so transmission efficiency will not be dealt with herein.

Variable Valve Timing or VVT controls the flow of air-fuel charge and exhaust gases by controlling the time for which the engine valves open and close. VVT adjusts this timing sequence of valves according to the operating conditions of the engine so as to optimize the working of the engine. This improves the efficiency of the engine over a wide range of engine rpm. Using VVT has its drawbacks also. It requires a hydraulic pump for its operation that adds complexity specifically in a smaller engine such as ours. It also operates for wide range of rpm whereas we are concerned with a specific range of rpm for which the engine produces maximum torque and operates at maximum efficiency. It also adds to the manufacturing cost. In lieu these drawbacks, Variable Valve Timing is dropped as an option to increase fuel efficiency.

Cylinder Deactivation is used in engines with more than one cylinder. It cuts off the fuel supply to the cylinder and restricts the engine valves from opening on the cylinder in order to save fuel when the engine requirements are at a minimum. Since the designed engine is a single cylinder engine so it will not be incorporating cylinder deactivation.

Turbocharged engine works on the principle of forced induction. The exhaust gases from the engine are used to operate a turbine which in turn runs a compressor. This compressor pressurizes the intake air to above atmospheric pressure. This reduces the pumping losses and engine efficiency is increased. Using turbocharger requires a high rpm as threshold for operation. Since the designed engine will be operating at rather low rpm so a turbocharger is not required.

Deceleration Fuel Cut-off, DFCO increases fuel efficiency by cutting off the fuel supply when the vehicle decelerates. The fuel supply is connected again as the driver pushes on the throttle pedal. Since the *BURN and COAST* technique is being used to achieve maximum fuel efficiency as explained in Chapter 3, DFCO will not be used.

2.3 Thermodynamic Cycles

2.3.1 Otto Cycle

Otto cycle is an ideal thermodynamic cycle that is used to describe the functionality of a typical spark ignition engine. It is the most commonly used thermodynamics cycle in

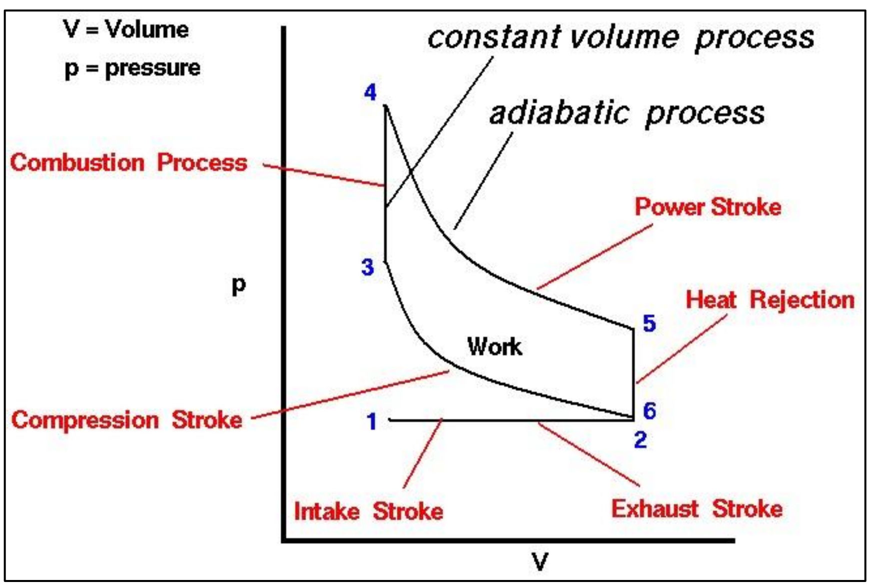


Figure 3: Otto Cycle

automobiles. Otto cycle basically describes the effect of change in temperature, pressure, volume and removal and addition of heat applied on a given mass of a gas.

2.3.2 Atkinson Cycle

Atkinson cycle is a thermodynamic cycle designed specifically for better fuel-efficiency or reduced fuel consumption. Atkinson cycle operates similar to an Otto cycle except that it incorporates an expansion ratio greater than the compression ratio. This is achieved by early or delayed closure of intake valve. Opening of the intake valve over the initial stages of compression stroke reduces the effective amount of air-fuel charge filled in the cylinder. Thus, the compression ratio is reduced while retaining full expansion during the power stroke and hence utilizing less fuel to achieve same amount of work on crankshaft. Achieving atmospheric pressure in the combustion chamber at the end of power stroke means that all available energy from combustion process has been obtained. For given amount of air-fuel charge, a greater expansion ratio means that more heat energy is being converted to useful mechanical energy. Although Atkinson cycle provides a greater engine efficiency it has a disadvantage of reduced power density. This is because the engine is filled with less amount of air-fuel charge during the intake and compression stroke as compared to similar Otto-cycle engine.

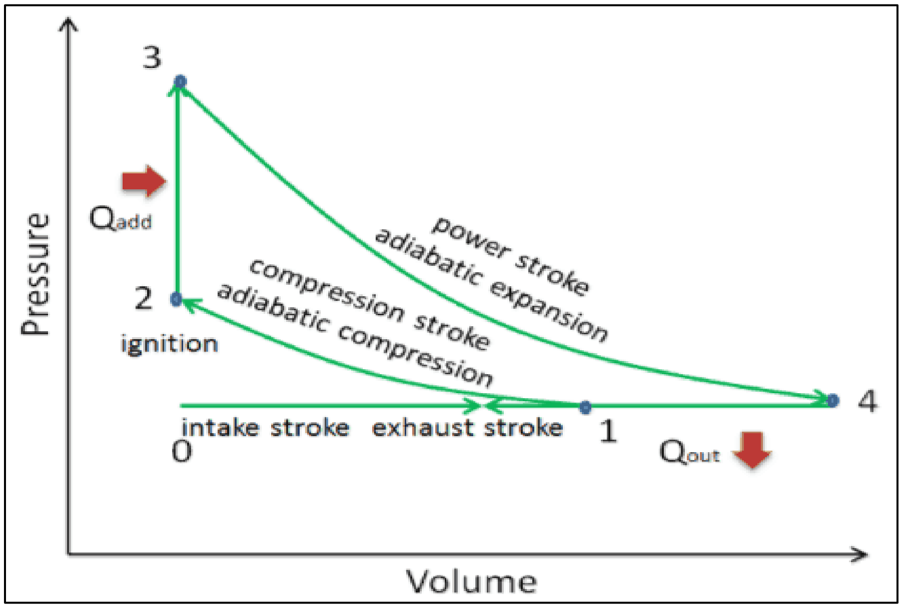


Figure 4: Atkinson Cycle

Since Atkinson cycle provides greater fuel efficiency due to a greater expansion ratio compared to the compression ratio so the designed engine will be based on the Atkinson Cycle for Spark Ignition Engines.

2.4 Fuel Injection

2.4.1 Carburettor

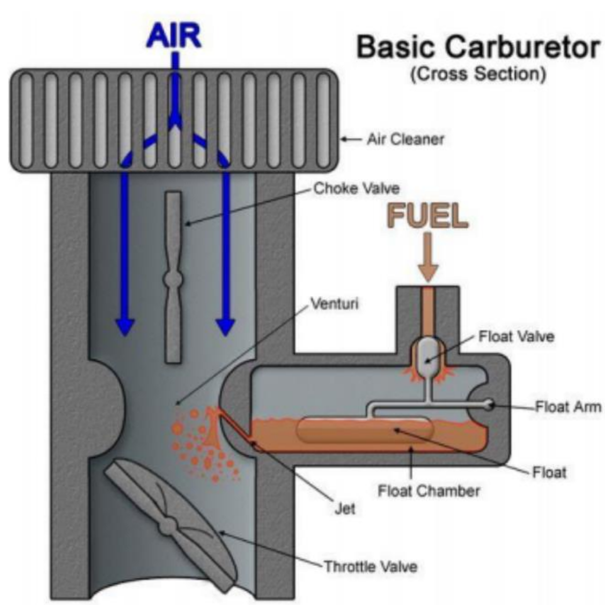


Figure 5: Carburetor Working

First method of fuel delivery is using carburetor. This methos is most widely used in old cars. It is still being used in cheap motorbikes. This method requires no electronic devic or computer, it is completely mechanical device. One problem in carburetors is that the tuning of a vehicle to obtain very high fuel efficiency is very difficult. One other problem is that they cannot vary the air-fuel mixture based on requirement and based on different situation and moreover they need to be continuously adjusted. This results in decreased reliability and more maintenance. Following figure shows the working of carburetor based on pressure differences:

2.4.2 Electronic Fuel Injection

The other method of fuel delivery is Electronically controlled fuel-injection or commonly known as Electronic Fuel Injection. The fuel efficiency is increased by using this method since the timing of injection, timing of ignition, air-fuel ratio, all can be varied easily and extremely precisely through programing based on different requirements and different load conditions. This also makes it very reliable system. The working principle is that sensors all over the engine give computer the situation/loading condition of engine. The computer then decides the timing of ignition and amount of fuel to be sprayed very precisely.

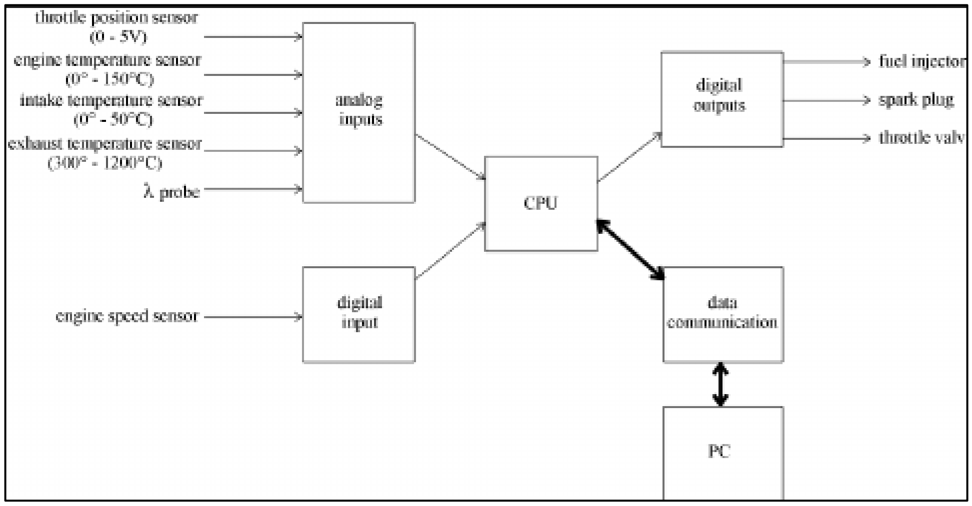


Figure 6: Working of EFI

2.5 Engine Components

The variation in volume for the working fluid is brought about by the reciprocating action of the piston inside the cylinder. The crank transforms this alternating movement into a rotational movement. The intake and exhaust events take place via the intake and exhaust valves. These valves are opened and closed at the right moment by the camshaft.

2.5.1 Piston

Piston is one of the kernel parts of the engine. The main purpose of the piston is to transfer the force generated by expanding gas due to combustion to the crankshaft through a connecting rod. Since the linear movement of piston needs to be converted to rotary motion of the crankshaft, hence mechanical losses are induced as a consequence. This reduces the overall efficiency of the engine.

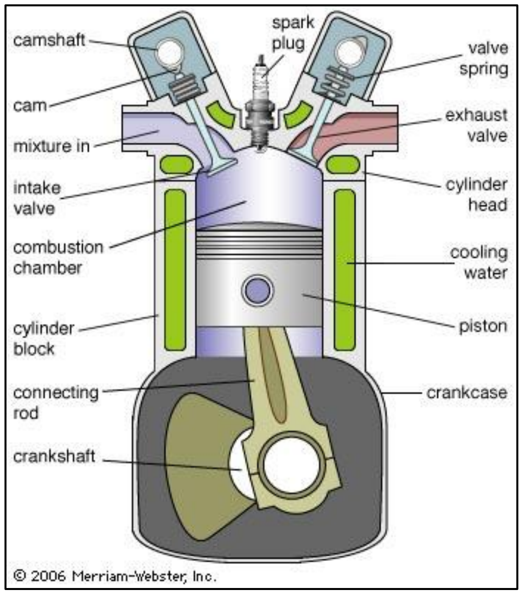


Figure 7: Spark Ignition Engine

Pistons are commonly made of cast aluminum alloy due to its excellent thermal conductivity and light weight. Since aluminum expands upon heating, proper clearance must be provided between the piston and the cylinder walls to prevent greater wear and tear. Excessive clearance can induce a loss in compression ratio whereas insufficient clearance can cause the seizure of piston in cylinder.

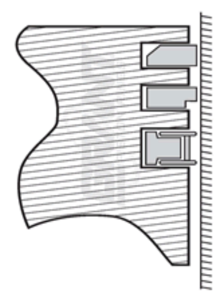
Piston Component	Function
Piston Head	Piston head is the upper part of the piston which is subjected to forces due to expanding gas in cylinder.
Piston Pin Bore	It is a through hole in the piston perpendicular to its movement that contains the piston pin.

Piston Pin	[5]It is the hollow shaft that connects the small end of the connecting rod to the piston.
Piston Skirt	[6]It is the circular portion of the piston below the piston rings that maintains the alignment of the piston as it moves through the cylinder.
Piston Rings	[6]These rings assure gas tight seal between the cylinder and the piston and also provide for lubrication of the cylinder walls to reduce wear and tear.

Table 2: Piston Components

2.5.2 Piston Rings

i. Top Compression Ring is located in the first ring groove and is used primarily to provide sealing between the cylinder walls and piston. It is also used to transfer heat from piston **to cylinder walls.**



ii. Intermediate Ring assists the top ring in sealing of combustion gases and heat transfer from piston walls. It also shears off the oil layer left by the oil ring and hence supports in oil control.

iii. Oil Ring is responsible for lubrication of cylinder walls. It controls the oil that is splashed onto the cylinder walls from connecting rod bearing throw-off.

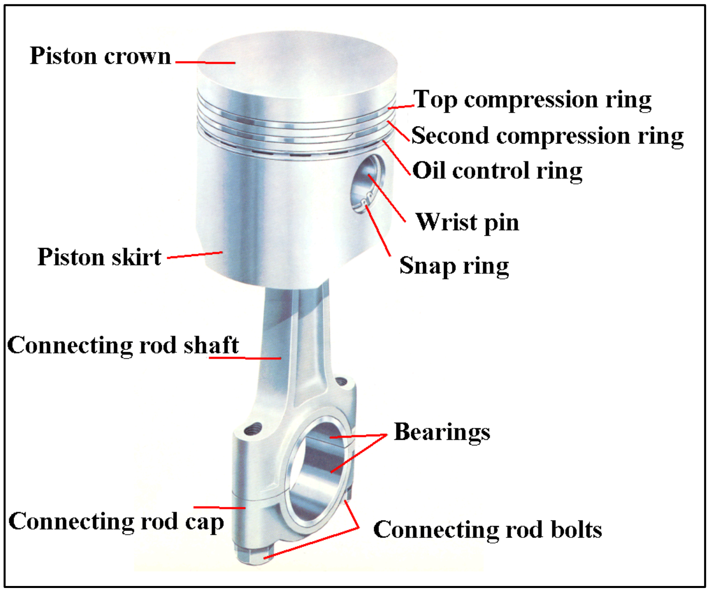


Figure 8: **Pisotn nad Connecting Rod**

2.5.3 Connecting Rod

Connecting rod connects the piston to the crankshaft. It is responsible for transmission of power from cylinder to crankshaft and from there to transmission. Connecting rods are commonly manufactured using casting, forging and powdered metallurgy. The most common type of connecting rods are steel and alumium. Shpae of connecting rod is imperial in its peformance and strength. Connecting rod has a long shank, a small end and a big end. The cross-section of shank can be rectangular, cirtcular, tubular, H-section or I-section based on the engine operationg conditions.

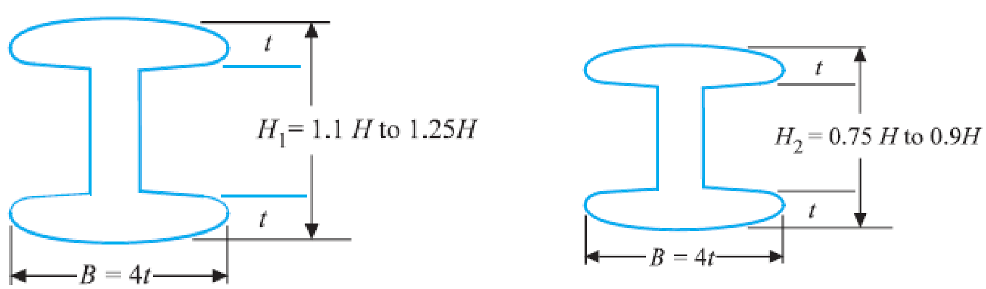


Figure 9: Big End and Small End

For the designed engine, I-section is chosen the cross-section of the shank because it can withstand high gas pressure and is light. I-section also allows keeping the inertial forces as low as possible. The bearings at two ends of connecting rod are either pressure lubricated or splash lubricated. The bearing at the big end is commonly splash lubricated while the bearing at the small end is pressure lubricated.

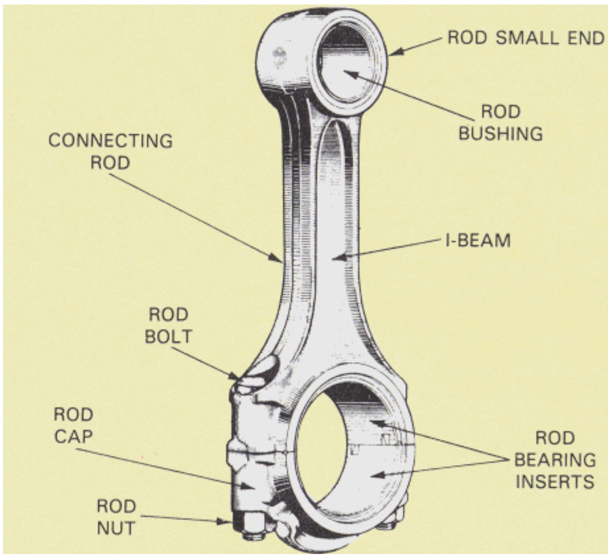


Figure 10: Connecting Rod

Forces acting on Connecting Rod:-

i. Force on piston due to gas pressure and inertia of reciprocating parts.
ii. Force due to inertial bending forces.
iii. Force due to friction of piston rings and piston.
iv. Force due to friction of piston pin bearing and crankpin bearing.

2.5.4 Crankshaft

Crankshaft is a major component of engine which converts the linear movement of piston into rotary motion and transmits it to the transmission via a flywheel. Crankshaft consists of crankpins and bearing journals which have an axis offset from crank where the big end of the connecting rod is connected. A flywheel is attached between crankshaft and transmission to reduce the pulsating motion and provide a rather smooth output to the transmission. The engine's crankshaft is made of very heavy cast iron in most cases. The snout of the crankshaft is made very strong to withstand the stress of placing the crankshaft pulley and the stress induced by driving other components via the same pulley.

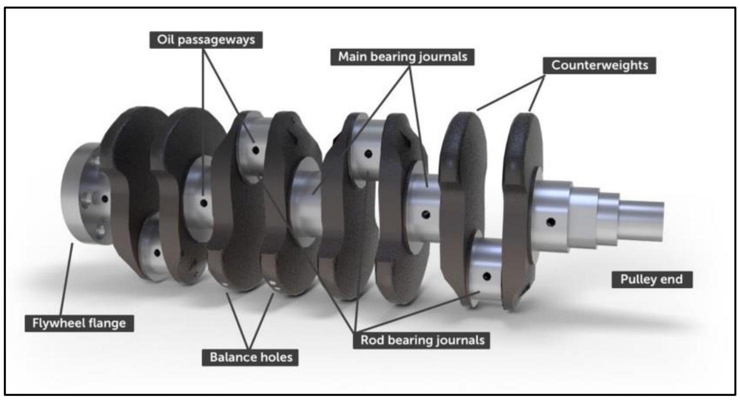


Figure 11: Crankshaft

2.5.5 Camshaft

Camshaft is often known as the ‘brains’ of the engine. This is because the camshaft opens and closes the engine valves at the right time to ensure maximum scavenging and cylinder filling is obtained and hence maximum power is obtained for given operating conditions of engine. Eccentric lobes in the camshaft actuate the components of the valve train. Crankshaft drives the camshaft through a set of gears or through chain. The camshaft always rotates at half of crank rpm, taking two full rotations of the crankshaft to complete one rotation of the cam, to complete a four-stroke cycle. For the designed engine, Single Overhead Camshaft (SOHC) is used with two cam lobes to regulate the opening and closing of intake and exhaust valves

Figure 12: Camshaft

2.5.6 Valve Overlap

In real engines, to ensure that the valve is fully open during a stroke and to ensure a greater volumetric efficiency, the valves are open for more than 180°. [7]The exhaust valve opens before BDC and closes after TDC. The intake valve opens before TDC and closes after BDC. At TDC there is valve overlap where both intake and exhaust valves are open.

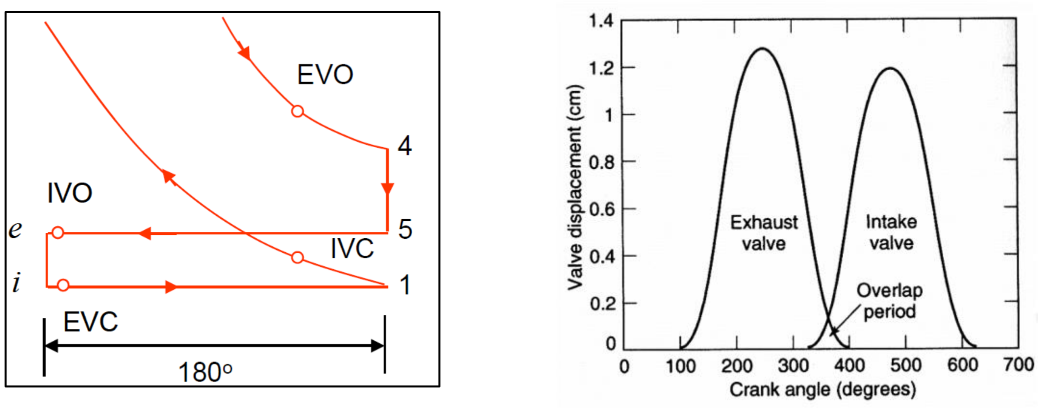


Figure 13: Valve Overlap

2.5.7 Cylinder Head

In an internal combustion engine, the cylinder head sits above the cylinders on top of the cylinder block. It closes in the top of the cylinder, forming the combustion chamber. This joint is sealed by a head gasket. In most engines, the head also provides space for the passages that feed air and fuel to the cylinder, and that allow the exhaust to escape. The head can also be a place to mount the valves, spark plugs, and fuel injectors. Different designs of cylinder heads that exist are:-

- Flat head
- Hemi-spherical
- Pent-roof

The **hemi-spherical** cylinder head does not allow for multiple intake and exhaust valves since the valve heads must be parallel for camshaft to operate.

The **flat head** cylinder design has low combustion efficiency as compared to others since the heat flow is towards the cylinder walls and greater losses are induced.

The **pent-roof** cylinder head allows for multiple intake and exhaust valves to be incorporated while keeping the valve heads parallel for operation. Pent-roof design also allows for greater ratio of A_v/A_p. The inclined valves allows for greater turbulence efficiency as it enhances squish and tumble.

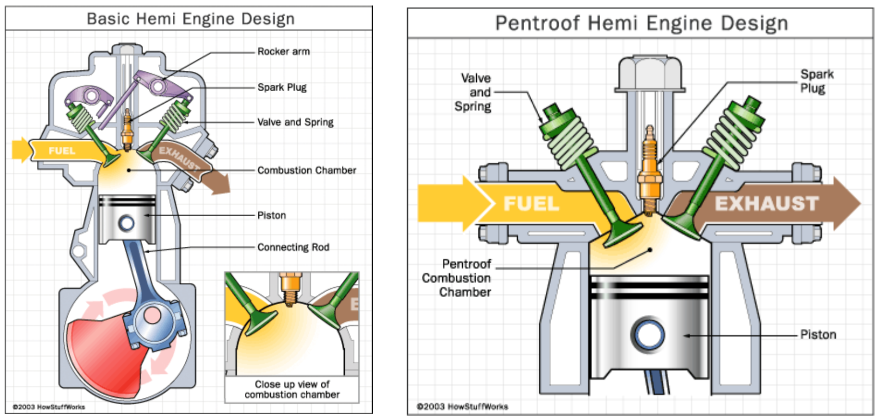


Figure 14: Hemi Design vs Pent-roof Design

Squish:-

Squish is achieved by making the outer edges of cylinder head to be planar with the piston as it reaches the Top Dead Center (TDC). A burst of turbulence is created that is directed towards the center of combustion chamber. This allows for better mixing of the air-fuel charge and better flame propagation.

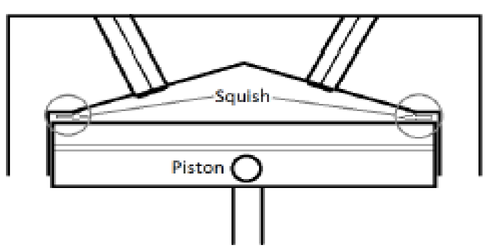


Figure 15: Squish

Tumble:-

Tumble is based on radial airflow entering the cylinder centerline. Tumble creates a homogeneous mixture that allows for better flame propagation resulting in an efficient air- fuel mixture.

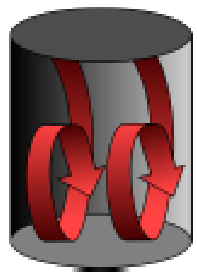


Figure 16: Tumble

BURN AND COAST

In order to achieve maximum possible efficiency from the designed engine, *BURN and COAST* driving technique is used. In burn and coast, the driver accelerates under Wide Open Throttle (WOT) for burn period and then the fuel supply is cut-off to the engine or alternatively the engine is turned off for the coast period. The vehicle cruises for this period without utilizing fuel hence increasing the fuel economy of the vehicle. Once after the vehicle slows down past the threshold speed for burn, the engine is tuned on again and the driver accelerates again for burn period repeating the cycle. To aid for the cruising of the vehicle, low rolling resistance tires from Michelin are used that have a rolling friction of 2kg/ton i.e 0.002. Also, the wheel hubs are mounted with ceramic bearings which also have low rolling resistance and hence allow for greater fuel-economy. The effectiveness of burn and coast technique is discussed in Chapter 3.

CHAPTER 3: METHODOLOGY

3.1 Concept Development

The aim of the project was to design an engine which will be able to achieve maximum possible fuel-economy, or least possible fuel-consumption by a vehicle. In order to initiate the designing of the engine, it is first needed to determine the requirements of engine. For this, a prototype model based on rules of Shell Eco Marathon Asia was developed. This model would have extremely low drag forces and ultra-lightweight body and chassis. The prototype model consists of fully enclosed, Aluminum chassis, covered by highly aerodynamic Carbon-fiber Shell. This designed model had following properties. For comparison, sedan car specifications of most fuel-economic car available in market (Toyota Prius Hybrid) is also given

Design Parameters	Prototype Model	Toyota Prius
Drag Co-efficient (C_d)	***0.14***	***0.25***
Frontal Area (A_f)	***0.58 m²***	***2.22 m²***
Rolling Friction Co-efficient (μ)	***0.002***	***0.015***

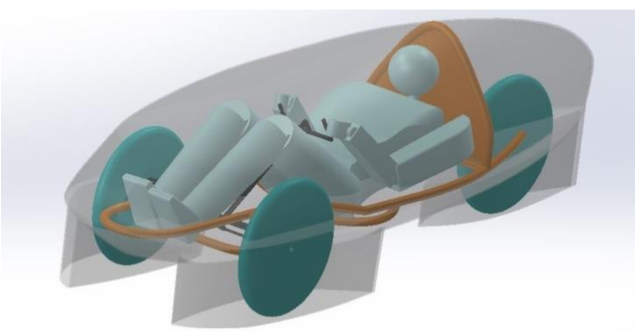


Figure 18: Isometric View

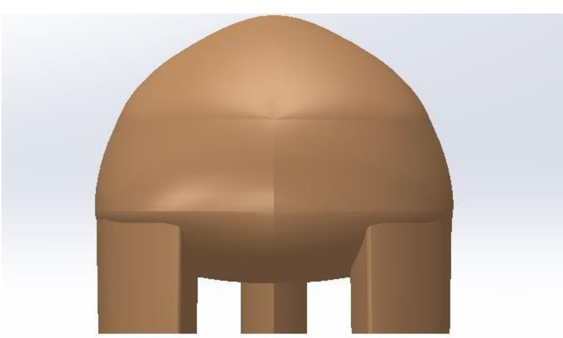


Figure 19: Front View

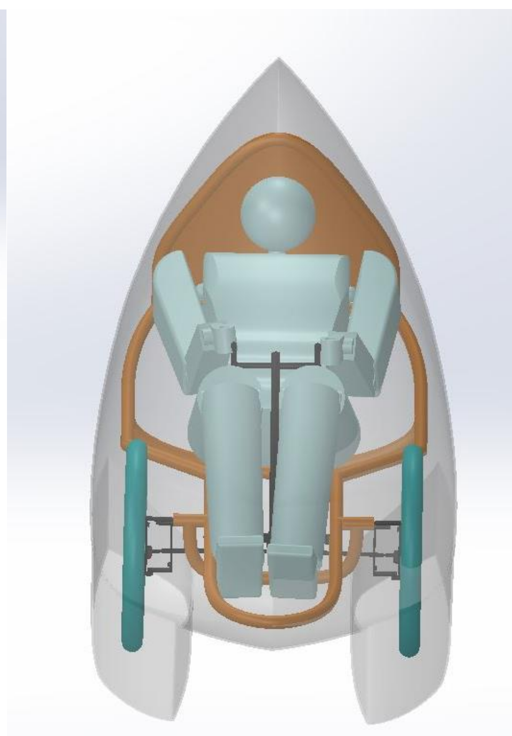


Figure 17: Front Through View

3.2 Design Methodology:

For the designing of the engine, the whole engine was divided into 4 small parts which are mentioned as following:

1) Rotating Assembly
2) Engine Block and Crankcase
3) Lubrication and Cooling
4) Cylinder head, spark and fuel

The Rotating Assembly consists of following components:

a) Piston
b) Piston Rings
c) Connecting Rod
d) Camshaft
e) Crankshaft
    a. Primary Balancing
    b. Secondary Balancing

The designing process of any engine begins with its requirement. Where is the engine going to be used and what power it is required to produce? Therefore, a model of already made vehicle should be used or a prototype model needs to be created. For ultimate fuel-efficiency, it was decided that a prototype model of highly fuel-efficient car would be made. This would not only give us estimate of aerodynamic drag forces but also the mass and weight of the car based on different material types of car chassis. For fuel-efficiency, the weight of the car needs to be as low as possible while also considering the financial constraints. It was decided that a carbon fiber shell would be made enclosing an Aluminum chassis. As compared to steel and carbon fiber, aluminum is lighter than steel and less costly than carbon fiber chassis. After the design of shell and chassis was made, estimates of mass of chassis was found using “evaluate” feature of SolidWorks while the drag coefficient and frontal area of vehicle was found using flow simulation over the vehicle via software “Autodesk Flow Design”

The mass of vehicle including driver came out to be 70 kg and with the added mass 70 kg of the driver, the total estimated mass is 140 kg

The drag coefficient came out to be 0.14 and the frontal area came out to be 0.58 m^2

The Aerodynamic Drag force is calculated as:

$$F_D = \frac{1}{2} \rho C_d A V^2$$

The F_D comes out to be 1.27604 N

One other resistive force on vehicle is also rolling resistance. It is resistive force acting on the tire due to their deformation on contact with the road surface. (Hysteresis loss).

The rolling resistance is measure as:

$$F_{RR} = \mu m g$$

The μ depends on the material and geometry of tires. Therefore, to make a car highly fuel efficient, tires with extremely low Coefficient of rolling resistance are used. Michelin tires with Coefficient of rolling resistance $\mu = 0.002$ are the most efficient tires available.

TYRE SIZE DIMENSION	Section width mm	Overall Diameter mm
44 - 406	44	500

Pressure max 500kPa

Figure 20: Michelin Low Rolling Resistance Tire Specifications

The specifications of these tires are as follows:

The Rolling Resistance is:

$$F_{RR} = 2.7468\ N$$

Therefore, the Total Resistive force is:

$$F_R = F_{RR} + F_D$$

$$F_R = 8.2651\ N$$

This is the total drag force (resistive force) we have to overcome in order to maintain velocity at 30 km/h. To accelerate, we need to overcome inertial forces, which is calculated as:

$$F_I = ma$$

The acceleration is calculated for increase of velocity from 10 km/h to 40 km/h as per requirement in 10 seconds.

$$a = \frac{v_f - v_i}{t}$$

$$a = 0.74\ m/s^2$$

For this acceleration, the inertial force comes out to be:

$$F_I = ma$$

$$F_I = 103.7036\ N$$

Therefore, the total force the tire has to give as output is:

$$F_{TR} = F_I + F_R$$

$$F_{TR} = 111.9687\ N$$

3.2.1 Power Requirement

The Power delivered by tire (tractive power) is:

$$P_{TR} = F_{TR} \times v$$

$$P_{TR} = 0.933\ kW$$

Brake Power is output power of engine after taking in consideration the internal frictional forces of engine.

$$P_{BHP} = P_{IHP} - P_{FHP}$$

Considering the mechanical efficiency of gears, chain and CVT, as 0.98 each, the brake engine power is calculated as:

$$P_{BHP} = \frac{P_{TR}}{\eta}$$

$$P_{BHP} = 0.991\ kW$$

Redline RPM is assumed to be 6500 rpm after multiple iterations so that mean piston velocity does not become very high.

We know from our study that maximum torque of most engines occur at 40% of the redline RPM and 70% of the load. The 40% of the redline RPM is

$$N_E = 2600\ rpm$$

Torque is calculated as:

$$T_E = \frac{P_E}{\omega_E}$$

$$T_E = 3.639\ Nm$$

Now it is known that the engine is required to produce 3.639 Nm of torque, the capacity and dimensions of cylinder of engine can be determined. Assuming Break Mean Effective Pressure (BMEP) of 8 bars, we get the engine capacity as:

$$V_S = \frac{T \times 4\pi}{BMEP}$$

$$V_S = 57.42\ cm^3$$

For higher fuel-efficiency, and over-square geometry of cylinder is used, that is, stroke is greater in length than bore [9]. This also accommodates for Atkinson cycle.

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

Assuming B = 0.8S, we get:

$$B = 38.7\ mm$$

And,

$$S = 48.4\ mm$$

The corresponding maximum power at redline RPM of 6500 comes out to be,

$$P_{max} = 3.32\ bhp$$

After the Power produced from the engine has been determined, we now need to calculate the mass flow rate of air and mass flow rate of fuel.

[8]The Redline RPM is the RPM beyond which engine damage may occur due to extreme velocity and extreme inertial forces of engine components

Redline RPM corresponding to the maximum mean piston velocity is [10]:

$$N_R = \frac{v_p \times 30}{S}$$

For Redline RPM of 6500, and Stroke of 48.4 mm, the mean piston velocity becomes:

$$v_p = 10.48\, m/s$$

This mean piston velocity is considerably low than modern day petrol engine cars. This indicates that the engine will encounter less friction loss due to piston sliding against the cylinder wall.

3.2.2 Fuel Consumption

The volume flow rate of air, V_a for engine with V_S = 57.42 cc is calculated as ;

$$\dot{V}_a = \frac{V_s \times \eta_v \times N}{120}$$

$$\dot{V}_a = 1.057477 \times 10^{-3}\ \ m^3/s$$

Thus mass flow rate for air is:

$$\dot{m}_a = \frac{P_i \times \dot{V}_a}{R \times T_i}$$

$$\dot{m}_a = 1.776 \times 10^{-3} kg/s$$

If stoichiometric mixture is to be used, air to fuel ratio must be 14.7. For this value of air-fuel ratio, the mass flow rate of fuel is

$$\dot{m}_f = 8.0108 \times 10^{-5} kg/s$$

Volume flow rate of fuel: $\dot{V}_f = \frac{\dot{m}_f}{\rho_f}$

$$\dot{V}_f = 0.37747\ l/\,hr$$

The fuel consumption under acceleration in km/l is:

$$FC = \frac{v}{V_f}$$

$$FC = 79.5\ km/l$$

In order to increase the fuel-economy, driving style called ‘burn and coast’ is used, in which acceleration is done under Wide Open Throttle (WOT) and then the fuel is cut-off so the car cruises without using ay fuel.

In order to calculate the fuel consumption for burn and coast driving, following method is used. Fuel Consumption during burn (acceleration) is known. Now for coast (cut-off fuel) the fuel consumption is calculated from the drag forces. The drag force is the only force that acts on the car, therefore, the deceleration due to drag force is:

$$F_D = ma$$

$$a = -0.05903\ m/s^2$$

For this acceleration, distance covered for vehicle to go from 40 km/h to 20 km/h is:

$$s = ut + \frac{1}{2}at^2$$

$$s = 784.2\ m$$

The fuel-consumption calculated is

$$FC = 828.2\ km/l$$

This efficiency is for an engine having normal otto cycle and stoichiometric air to fuel ratio. If we incorporate Atkinson cycle and lean air-fuel ratio, the fuel consumption can further be reduced [11].

The Volumetric flow rate of air for Atkinson cycle is:

$$\dot{V}_a = \frac{V_s \times \eta_v \times N}{120}$$

$$\dot{V}_a = 9.623 \times 10^{-4}\ m^3/s$$

And mass flow rate of air is:

$$\dot{m}_a = \frac{P_i \times \dot{V}_a}{R \times T_i}$$

$$\dot{m}_a = 1.072 \times 10^{-3}\ kg/s$$

If lean air-fuel mixture is to be used, air to fuel ratio could be extended up to 15.5. For this value of air-fuel ratio, the mass flow rate of fuel is

$$\dot{m}_f = 6.913 \times 10^{-5}\ kg/s$$

Volume flow rate of fuel: $\dot{V}_f = \frac{\dot{m}_f}{\rho_f}$

$$\dot{V}_f = 0.3257\ l/\ hr$$

The fuel consumption under acceleration in km/l is:

$$FC = \frac{v}{V_f}$$

$$FC = 92.1\ km/l$$

Since the acceleration and deceleration remains same, the distance covered under burn and coast is also the same. Therefore, the fuel consumption for lean air-fuel mixture with Atkinson cycle is:

$$FC = 959.4\ km/l$$

3.2.3 Cooling System

Now, since we have calculated the amount of fuel to be used, we can now do calculation on cooling of engine.

We know that the amount of fuel burned per second is $6.913 \times 10^{-5}\ kg/s$, therefore the heat supplied by that fuel is:

$$\dot{Q}_f = \dot{m}_f \ \times Q_{LHV}$$

Q_{LHV} of fuel we are going to use is 42,900 kj/kg

$$\dot{Q}_f = 2.96\ kW$$

We know that 30% of heat is rejected through exhaust and 25% is converted into work. The Remaining 35% is transferred to surrounded through the cooling system. Therefore

$$\dot{Q}_f = 1.031\ kW$$

Of heat needs to be dissipated by the cooling system.

Since the cooling system will be air-cooled, fins would be required to dissipate the heat.

Flat rectangular fins are chosen because of ease of manufacturability and cost.

$$a = \sqrt{\frac{hp}{kA}}$$

$$a = 17.41$$

The fin efficiency is:

$$\eta_{fin} = \frac{\tan(aL)}{aL} = 0.717$$

$$A_{unfin} = (0.09)(0.065)(2) = 0.0117\ m^2$$

$$A_{fin} = [(0.015)(0.02)(2) + (0.02)(0.065)(2)] \times n_{fin}$$

$$A_{fin} = 3.2(10^{-3}) \times n_{fin}$$

$$\dot{Q}_{fin} = (0.717)(40)(\ 3.2(10^{-3}) \times n_{fin})(260 - 35)$$

$$\dot{Q}_{fin} = 20.64 \times n_{fin}$$

$$n_{fin} = 48\ fins$$

3.2.4 Cylinder and Piston Design

Since the bore and stroke of the piston is known, the length of cylinder is taken as:

$$L_c = 1.15S = 55.6\ mm$$

And the Outer Diameter, D_o and inner diameter, D_i of cylinder are:

$$D_i = 39\ mm$$

$$D_o = 42\ mm$$

The piston head thickness (t_h), given by the Grashoff's formula is:

$$t_h = \sqrt{\frac{3pD^2}{16\sigma_t}}$$

Now under consideration of heat transfer, the thickness of the piston should be such so that heat can be effectively and quickly transferred to the cylinder walls. This thickness is given by:

$$t_h = \frac{H}{12.56k(T_C - T_E)}$$

The heat flowing through the piston head (H) may be determined from the following formula:

$$H = C \times HCV \times m \times B.P.$$

The dimensions of piston are:

[8]Piston head thickness: $t_h = 5.4\ mm$

1st land: $t_1 = 4.7\ mm$

2nd land: $t_2 = 2.2\ mm$

3rd land: $t_3 = 1.2\ mm$

Radial thickness of ring: $r_1 = 1.45\ mm$

Axial thickness of ring: $a_1 = 1.3\ mm$

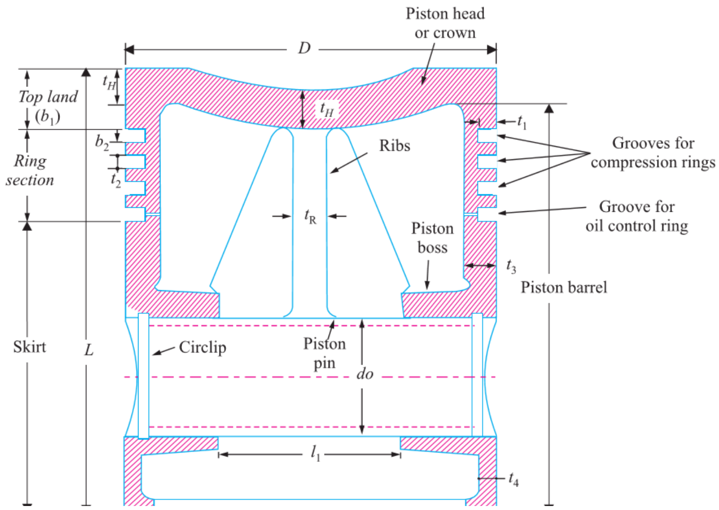


Figure 21: Cross-Section of Piston

3.2.5 Piston Pin

Piston pin is usually made of hardened steel alloy containing chromium, nickel, molybdenum or vanadium. The load on piston is given by:

$$Piston\ laod = \frac{\pi D^2}{4} \times p$$

Now, the load on piston pin due to bearing load is given by:

$$Bearing\ load = bearing\ pressure\ \times bearing\ area$$

$$Bearing\ load = p_{b1} \times d_o \times l_1$$

Equating the bearing load with piston load, we get

$$d_o = 18.24\ mm$$

The internal diameter of piston pin is taken as: $d_i = 0.6d_o$

$$d_i = 10.95\ mm$$

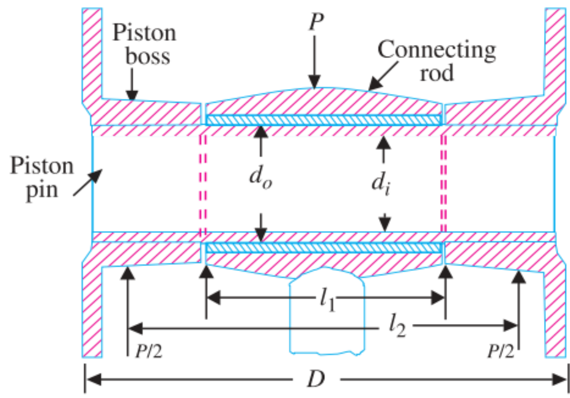


Figure 22: Cross Section Piston and Small End of Connecting Rod

From figure 5, we can find the length between supports (piston boss),

$$l_2 = l_1 + \frac{D - l_1}{2} = \frac{l_1 + D}{2}$$

$$l_2 = 28.06\ mm$$

Maximum Bending stress at the center of the pin is given by:

$$M = \frac{P.D}{8}$$

$$M = 51212.42\ \ N\ mm$$

It is also known that maximum allowed bending stress is:

$$M = \frac{\pi}{32}\left[\frac{(d_o)^4 - (d_i)^4}{d_o}\right]\sigma_b$$

Therefore,

$$\sigma_b = 98.76\ MPa$$

Which is less than permissible bending stress of 140 MPa for heat treated alloy steel

3.2.6 Connecting Rod

The length of connecting rod depends on l/r ratio, where r is half the length of stroke. Smaller l/r ratio will increase the side thrust force on the cylinder wall, therfore increasing the wear of the cylinder liner. Larger l/r ratio will decrease the side thrust but will increase the height of engine and weight of reciprocating mass. Therefore a compromise is made and generally l/r ratio of 4 to

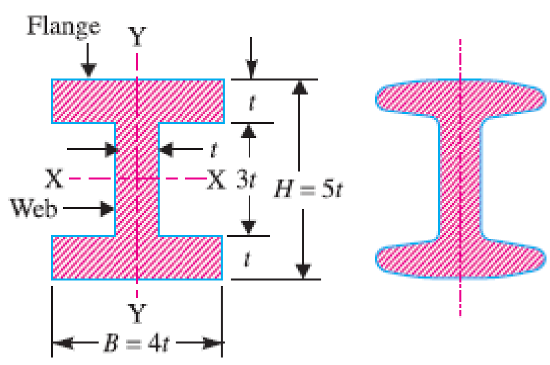


Figure 23: Cross-Section of Connecting Rod

5 is chosen. The material to be used for connecting rod is alloy steel.

Connecting rod with I-section cross section was chosen because of following reasons:

- Lightness
- To keep inertial forces as low as possible.
- To withstand high gas pressure.

Dimensions are determined by considering buckling about X-axis

Length: $L = 104.75\ mm$

$n = 4.3$

Thickness: $t = 2.6\ mm$

Width of section: $B = 4t = 10.4\ mm$

Height of section: $H = 5t = 13\ mm$

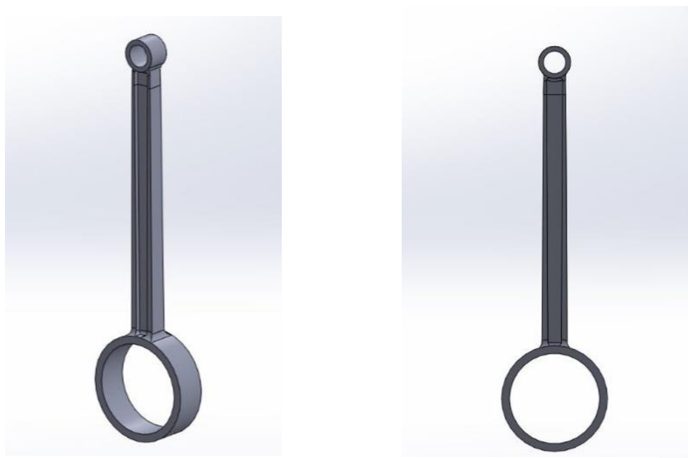


Figure 24: Designed Connecting Rod

The Designed Connecting Rod is shown below:

3.2.7 Cylinder Head

Based on our requirement of efficiency and requirement of multiple valves per intake and per exhaust, pent-roof cylinder head was chosen.

The Pent-roof cylinder head was designed such that it enhances squish and tumble for better combustion and flame propagation.

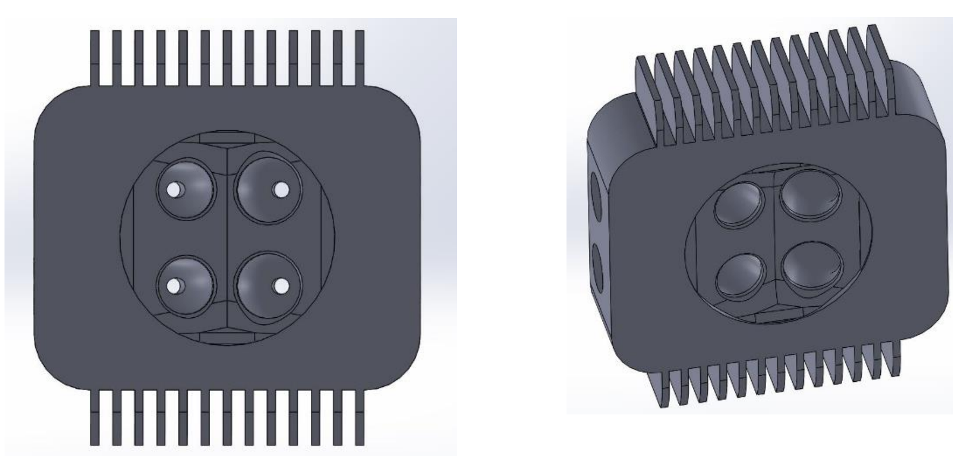
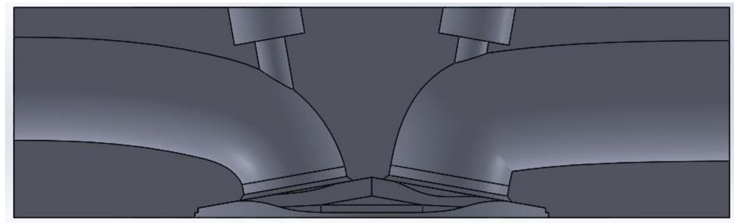
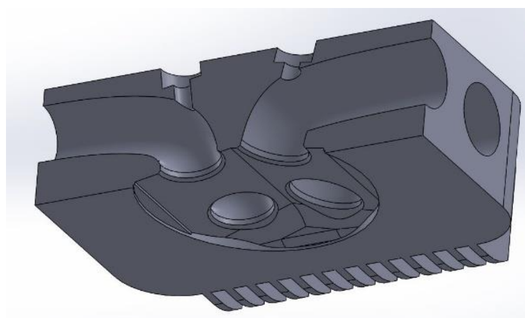


Figure 25: Designed Pent-roof Cylinder Head



3.2.8 Valves

The volumetric efficiency has a huge impact on the fuel efficiency of any engine. The volumetric efficiency depends on many factors, few of which are geometry of valve, number of valve, size of valve and orientation of valve. It can be shown that 2 smaller valves have greater valve area than one bigger valve. We will see which orientation provides grater valve flow as this this the purpose of valve. The curtain area for one is:



$$A_C = 2\pi rl$$

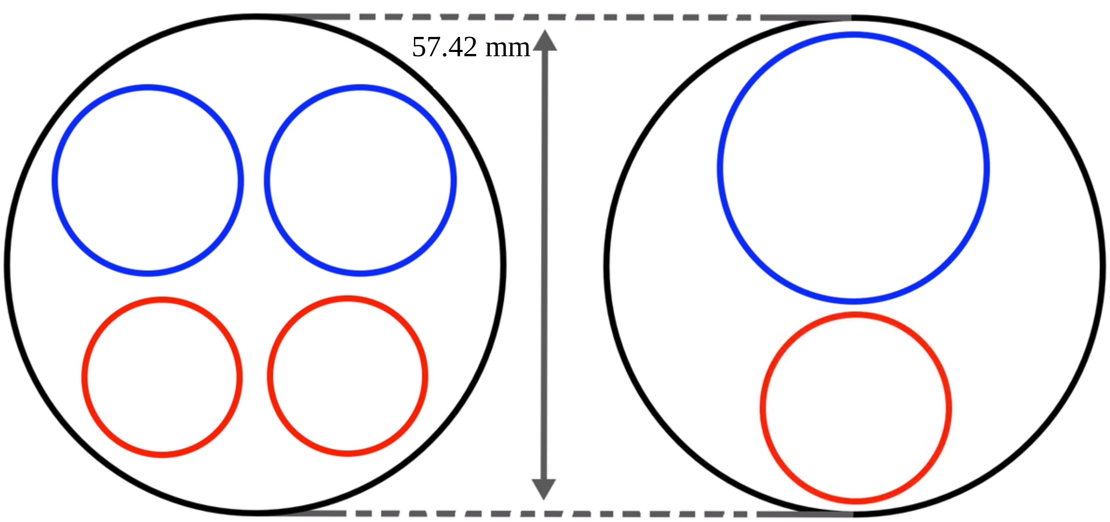


Figure 26: 4 Valves per cylinder vs 2 Valves per cylinder

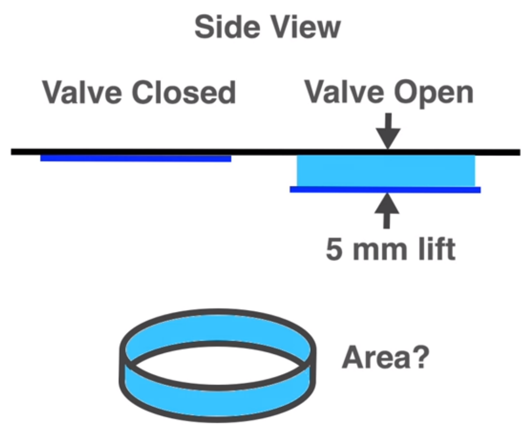
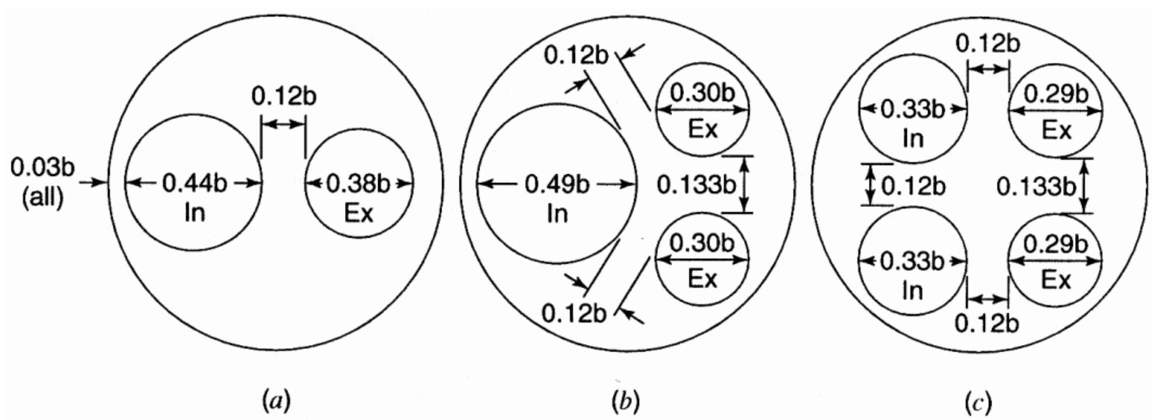


Figure 27: Valve lift side view



Valve diameter ratios for a flat cylinder head (b: bore, In: intake, Ex: exhaust)

Formula for diameter of intake valve for one intake valve per cylinder is:

$$d_i = 0.44B$$

$$d_i = 0.44(38.7) = 17.028\,mm$$

Therfore, Valve area becomes:

$$A_C = \pi d_i l$$

$$A_C = \pi(17.028)(5)$$

$$A_C = 267.47\ mm^2$$

Formula for diameter of intake valve of one intake valve for two intake valves per cylinder is:

$$d_i = 0.33B$$

$$d_i = 0.33(38.7) = 12.771\ mm$$

Therfore, Valve area becomes:

$$A_C = \pi d_i l$$

$$A_C = \pi(12.771)(5)$$

$$A_C = 200.61\ mm^2$$

But since there are two intake valves, the total intake valve area becomes:

$$A_C = 401.22\ mm^2$$

Which is significantly greater than single big valve. There is an increment of 50% Valve area. Similar method can be implied to show that 2 exhaust valve per cylinder provides greater valve area than one bigger exhaust valve.

Figure 28: Designed Intake Valve

The intake valves are taken to be bigger in size than exhaust valve because of the difference in purpose. Intake valves have to encounter suction of air from tube to cylinder, while exhaust valves have to encounter pressurized air to flow outside from cylinder to exhaust port/tube.

Since the required air flow is:

$$\dot{V}_a = 9.623 \times 10^{-4}\ m^3/s$$

The velocity of air corresponding to two intake valves with diameter of 12.771 mm each is calculated as:

$$\dot{V}_a = Av$$

$$v = \frac{9.623 \times 10^{-4}\, m^3/s}{401.22\, mm^2}$$

$$v = 2.4\, m/s$$

3.2.9 Spark

To make an engine fuel efficient, combustion efficiency must be improved. Usually it is greater than 90%, but it can be further increased by used of multiple spark ignition system and delayed multiple spark system. One such method is DSI (Dual & Sequential Ignition)

CHAPTER 4: RESULTS AND DISCUSSIONS

The aim of the project was to design a highly fuel-efficient engine by incorporating Atkinson cycle and to observe the differences between normal Otto cycle and Atkinson cycle. From Calculations done in previous parts, it is concluded that highly fuel-efficient Atkinson cycle engine will have following specifications:

Redline RPM: $N_r = 6500\ rpm$

Maximum Torque RPM $N_e = 2600\ rpm$

Maximum Engine torque: $\tau_e = \frac{P_e}{\omega_e} = 3.64\ Nm$

Maximum Power: $P_r = 3.32\ bhp\ @6500\ rpm$

Bore: B = 38.7mm

Stroke: S = 48.4mm

Swept Volume: $V_s = 57.42$cc

The fuel consumption is calculated for Otto cycle and Atkinson cycle as basic engine cycle and values are compared.

Otto Cycle Stoichiometric air-fuel mixture gives a theoretical fuel consumption of:

$$FC_{Otto-14.7} = 79.5\ km/l$$

Atkinson Cycle lean air-fuel mixture gives a fuel consumption of:

$$FC_{Atkinson-14.7} = 92.1 km/l$$

This is an increment of approximately 16% in fuel economy which is considered very high. The final design of engine is as follows

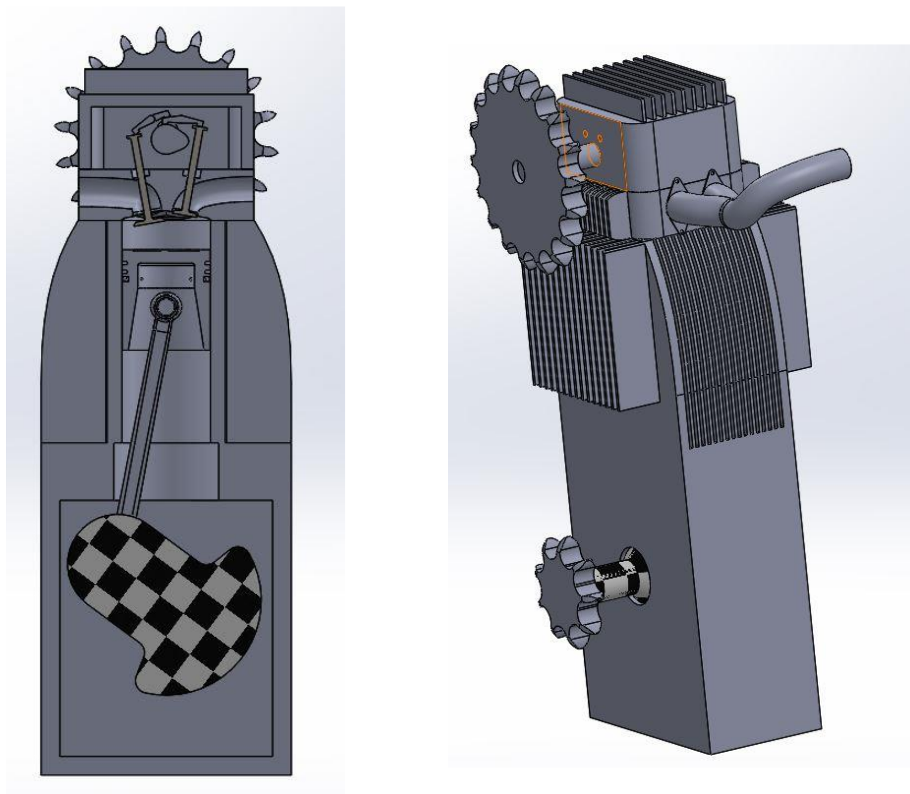


Figure 29: Engine Design

4.1 Stress Analysis

Stress Analysis is done to calculate the maximum stress applied due to combustion of gases in combustion chamber. The results show that the maximum Von Mises stress is 4.22e07 N/m^2 whereas the yield strength is 6.204e07 N/m^2. This means that the maximum applied stress on the cylinder walls is well below the yielding value of walls and hence the designed cylinder will easily withstand the stresses produced inside the cylinder.

Von Mises Stresses

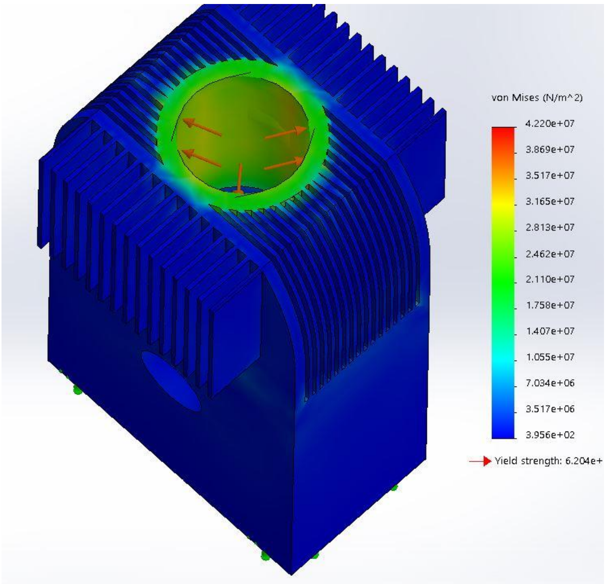


Figure 30: Stress Analysis

Displacement

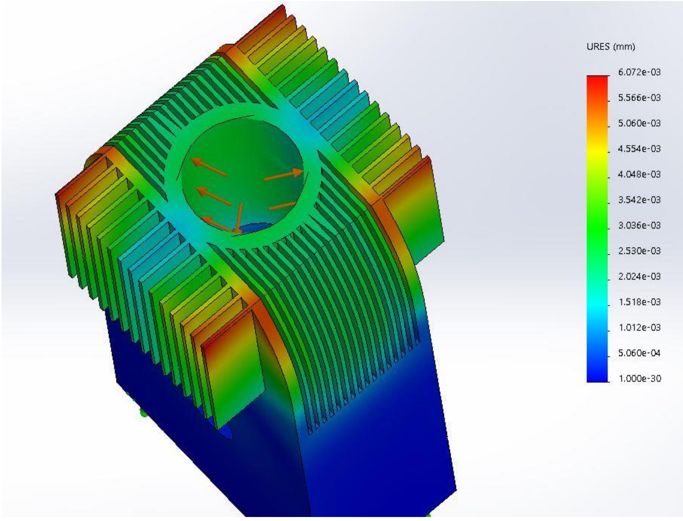


Figure 32: Stress Analysis

4.2 Manufacturing

A complete design of an engine incorporates the lubrication system and the cooling system. Designing of such systems is extensive and out of scope of this project since it focuses solely on fuel economy. Plus, the cost factor makes it impossible to manufacture the redundant components of the engine that do not play a part in increasing the fuel economy. Thus, a pre-built naturally aspirated engine with a working lubrication system was procured and modified to our requirements. The procured Honda CD 70 engine had following specifications:

Bore: B = 47 mm

Stroke: S = 41.4 mm

Swept Volume: $V_s = 71.8$ cc

Before manufacturing the designed camshaft and crankshaft to convert the operating cycle from Otto to Atkinson, this procured engine was modified to decrease its displacement to 58.6 cm3. This was done since the designed engine has displacement of 57cm3 and relatively fair comparison is possible between theoretical fuel consumption and achieved fuel consumption after conversion to Atkinson Cycle.

4.2.1 Modifications

The cylinder bore was reduced by inserting a brass liner of thickness 3mm into the cylinder. Consequently, the piston was machined from outer side to adjust into the engine cylinder. Minute grooves were machined on top of the brass liner to cater for proper opening and closing of intake and exhaust valves.

The designed crankshaft increase the stroke of piston by 3mm. This is achieved by displacing the connecting rod pin to outer periphery by 1.5mm.

Original bore: 47mm New bore: 41mm

Original Stroke: 41.4mm New Stroke: 44.4 mm

New displacement:

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

$$V_S = 58.6\ cc$$

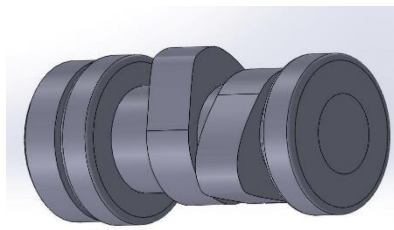
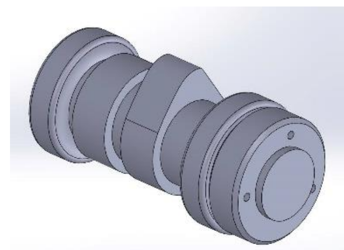
The new achieved parameters for the engine are comparable to original design parameters. Now the basic operating Otto cycle is converted to Atkinson cycle by incorporating the machined camshaft that alters the timing of the closing of intake valve and hence makes the expansion ratio greater than the compression ratio. The process of conversion from Otto cycle to Atkinson cycle is explained below:

4.2.2 Camshaft:

The Camshaft of the engine was designed and manufactured based on calculations that the expansion stroke has to be greater than the compression stroke. Therefore, the designed camshaft has profile for the intake valve which allows the intake valve to be closed before the piston reaches the bottom dead center. The experiment was performed where the crank was rotated and the crank angle where the intake valve closed was noted for the modified camshaft. The crank angle came out to be 155 deg after the intake valve opened.

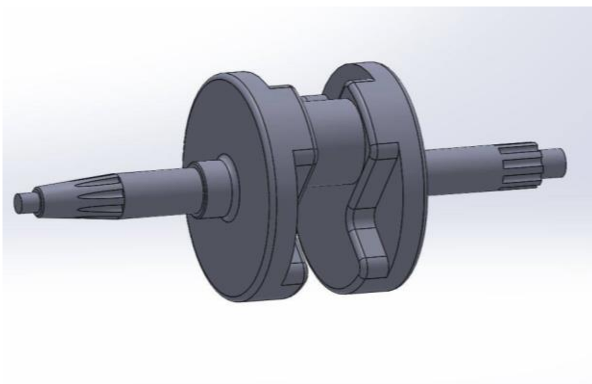
4.2.2.1 Comparison Table:

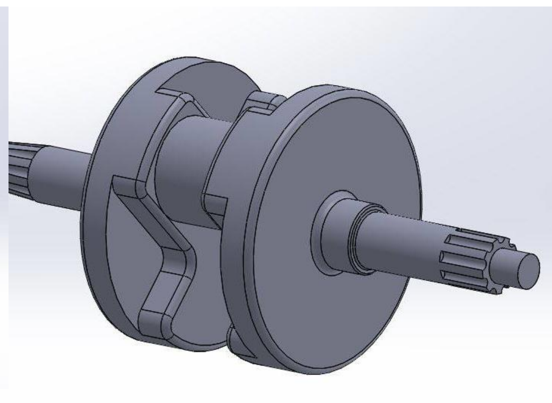
Assuming when the piston is at the Top Dead Centre and beginning the intake stroke, the crank angle is 0 deg of crankshaft.

	Old Camshaft	New Camshaft
Design	 	 
Intake Valve Opens	0 (deg)	0 (deg)
Intake Valve Closes	180	155
Exhaust Valve Opens	540	540
Exhaust Valve Closes	720/0	720/0

4.2.3 Crankshaft:

The new crankshaft would have much lighter flywheel and lighter counter weight owing to the fact that the capacity of current 72 cc engine is reduced to 58.6 cc. This in result produces less torque and thus require smaller mass of flywheel and balancing weights. To accommodate larger stroke, the distance of crank pin and center rod crankshaft is increased. Increased by 1.5 mm so stroke increases by 3 mm. To accommodate more stroke for same engine, a 1.5 mm gasket is introduced between cylinder and cylinder head. Gasket is made of steel to accommodate for the high pressure and stresses of cylinder while having same heat expansion of material.





4.2.4 Connecting Rod:

We had the possibility to use either of connecting rod design, I-Beam or H-Beam. From literature we know that I-Beam is used in cases where Torque and Power is Low but RPM is high whereas the H-Beam design is used where the torque and power production is very high at low RPMs. Owing to the fact that our 58.6 cc engine produces less torque than the original 72 cc engine and that the engine will run at less RPMs than 72 cc counterpart because of over squared geometry of cylinder, the connecting should have lesser cross section area. This will not only meet the requirement of the engine but it will also reduce the weight of oscillating component, connecting rod and will result in less vibrations and more fuel economy.

4.3 Experimental Results:

To measure the fuel consumption of both, unmodified and new modified engine, we devised an experiment. We filled a burette with fuel (gasoline) and measured the temperature of the fuel. Then we started the engine and increased the RPM. We were measuring the RPM by infra-red tachometer with a black strip attached to the Magnet. The throttle was increased until a preset value 2600 RPM was achieved. At this RPM a stopwatch was started and an initial value of burette was taken. After 2 minute, the final value of burette was taken and the difference between the initial and the final gave us the amount of fuel used for specific RPM. Same procedure was repeated for the new modified engine.

The results are calculated as:

4.3.1 Unmodified 58.6 cc Engine (Otto Cycle):

RPM = 2600 RPM

Fuel used = 13.1 ml

Fuel usage = 6.55 ml/min = 0.393 l/hr

$$FC = \frac{v}{V_f}$$

$$FC = \frac{30\ km/h}{0.393\ l/hr}$$

$$FC = 76.33\ km/l$$

4.3.2 Modified 58.6 cc Engine (Atkinson Cycle):

RPM = 2600 RPM

Fuel used = 11.8 ml

Fuel usage = 5.9 ml/min = 0.354 l/hr

$$FC = \frac{v}{V_f}$$

$$FC = \frac{30\ km/h}{0.354\ l/hr}$$

$$FC = 84.74\ km/l$$

CHAPTER 5: CONCLUSION AND RECOMMENDATIONS

Internal Combustion Engine is one of the most important inventions of the last century. It has been developed in the late 1800s and from there on it has had a significant impact on our society. It has been and will remain for foreseeable future a vital and active area of engineer research.

The aim of the project was to design a highly fuel-efficient engine by incorporating Atkinson cycle and to observe the differences between normal Otto cycle and Atkinson cycle. All necessary calculations concerning its basic components are taken into consideration. A concept prototype car has been developed to test the designed engine for maximum achievable engine. The developed concept is based on the guidelines set out by a global competition.

Electronic Fuel Injection is currently being used in cars but not in motorbikes except some heavy bikes. This project tests the use of EFI in a small engine comparable to the ones in motorbikes. Successful testing means that this concept can be introduced in local motorbike industry to employ EFI to enhance the efficiency of motorbikes being locally manufactured.

In addition, the most proper materials which have to be used have been determined.

It has been taken into consideration that the chosen materials must resist on the maximum forces, moments and stresses that occur when the engine is operating.

From the calculations and results it can be concluded that fuel efficiency is dependent on a lot of factors which includes air drag, rolling resistances and inertial mass. Decreasing these forces will improve the fuel economy. Air drag is reduced by decreasing the drag coefficient or decreasing the frontal area or by decreasing the velocity of the vehicle. Rolling resistance is reduced by decreasing the coefficient of rolling resistance or by decreasing the mass of the vehicle.

It can also be shown from the calculations that Atkinson cycle improves the fuel-economy by approximately 15% as compared to normal Otto cycle. It must be noted that engine size remains same but only the compression stroke is reduced effectively. The engine can further be tuned using EFI, that is by programming the ECU.

EFI is the future of fuel injection since carburetors are only effective as an educated guess. It does not provide complete control over the amount of fuel being injected into the engine. EFI allows for control over amount of fuel being injected into the engine and can be tuned for optimized working of engine under any operating conditions.

5.1 Future Recommendations

Since designing n engine is an extensive process, there are a lot of changes that can be made to the current design to make it better in regards of fuel economy. These changes can be incorporated in future projects building on the current design to achieve even better results:

1) The cylinder head can be modified in such a way to accommodate for more valves per cylinder as designed in phase 1. This will increase the fuel efficiency but increase the cost. Such design could not be incorporated in our final product because of cost and complexity considerations.

2) One other way in which better fuel economy can be achieved is by using Variable Valve Timing and Lift control. In such method, the valve timing can be changed from efficiency to performance and/or vice versa according to the change in input from the driver. Since our project design started from scratch, such complex design alterations were not possible to incorporate. Future teams building on our design can do their working to incorporate this feature.
3) Third way of improvement is to use lean air-fuel mixture. This would increase the fuel efficiency but it would require the use of lean catalytic converter as lean air-fuel mixture is responsible fo0r production of NOx gases. We have not used this technique as we were not using the catalytic converter for local motorbike industry thus it would have violated the Euro II exhaust gas regulations.

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APPENDIX I: DESIGN CALCULATIONS

Power Calculations:

Aerodynamic Drag:

$$F_D = \frac{1}{2}\rho C_d A V^2$$

Where,

F_D = Aerodynamic Drag Force

ρ = Density of air = 1.225 kg/m^3

C_d = Drag Coefficient = 0.14

A = Frontal Area of Vehicle = 0.58 m^2

V = Velocity of Vehicle = 30 km/h

The F_D comes out to be 1.27604 N

Rolling Resistance:

The rolling resistance is measure as:

$$F_{RR} = \mu m g$$

Where,

μ = Coefficient of rolling resistance [10]

m = total mass of vehicle

g = gravitational acceleration

$$F_{RR} = \mu mg$$

$$F_{RR} = 0.002 \times 140 \times 9.81$$

$$F_{RR} = 2.7468\, N$$

Total Resistive Force:

$$F_R = F_{RR} + F_D$$

$$F_R = 2{,}7468 + 5.5183$$

$$F_R = 8.2651\, N$$

Acceleration and Inertial Forces:

$$a = \frac{v_f - v_i}{t}$$

$$a = \frac{40 - 10\, km/h}{10\, s} \times \frac{1000\, m}{3600\, s}$$

$$a = 0.74\, m/s^2$$

$$F_I = ma$$

$$F_I = 140\, kg \times 0.8333\, m/s^2$$

$$F_I = 103.7036\, N$$

Traction Force:

$$F_{TR} = F_I + F_R$$

$$F_{TR} = 103.7036 + 8.2651$$

$$F_{TR} = 111.9687\, N$$

Tractive Power:

$$P_{TR} = F_{TR} \times v$$

$$P_{TR} = 111.9687\ N\ \times 8.333\ m/s$$

$$P_{TR} = 0.933\ kW$$

Brake Horse Power:

$$P_{BHP} = \frac{P_{TR}}{\eta}$$

$$P_{BHP} = \frac{0.933}{0.98^3}$$

$$P_{BHP} = 0.991\ kW$$

Maximum Torque RPM:

$$N_E = 0.4 \times 6500$$

$$N_E = 2600\ rpm$$

Maximum Torque:

$$T_E = \frac{P_E}{\omega_E}$$

$$T_E = \frac{P_E}{2\pi N_E/60}$$

$$T_E = \frac{0.991\ kW}{272.2713\ rad\ s^{-1}}$$

$$T_E = 3.639\ Nm$$

Engine Capacity:

$$V_S = \frac{T \times 4\pi}{BMEP}$$

$$V_S = \frac{3.639 \times 4\pi}{8 \times 10^5}$$

$$V_S = 57.42\ cm^3$$

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

$$B^3 = \frac{0.8 \times 4V_S}{\pi}$$

$$B = 38.7\ mm$$

$$S = 48.4\ mm$$

Mean Piston Velocity:

$$N_R = \frac{v_p \times 30}{S}$$

$$v_p = \frac{S \times N_R}{30}$$

$$v_p = \frac{48.4 \times 6500}{30 \times 1000}$$

$$v_p = 10.48\ m/s$$

Fuel Consumption:

Accelerating Stoichiometric Otto Cycle Fuel Economy:

Volumetric flow rate of air is calculated as:

$$\dot{V}_a = \frac{V_s \times \eta_v \times N}{120}$$

$$\dot{V}_a = \frac{57.42 \times 0.85 \times 2600}{120}$$

$$\dot{V}_a = 1.057477 \times 10^{-3}\ m^3/s$$

Thus mass flow rate for air is:

$$\dot{m}_a = \frac{P_i \times \dot{V}_a}{R \times T_i}$$

$$\dot{m}_a = \frac{1{,}00{,}000 \times 1.057477 \times 10^{-3}}{285 \times 313}$$

$$\dot{m}_a = 1.776 \times 10^{-3} kg/s$$

If stoichiometric mixture is to be used, air to fuel ratio must be 14.7. For this value of air-fuel ratio, the mass flow rate of fuel is

$$\dot{m}_f = 8.0108 \times 10^{-5} kg/s$$

Volume flow rate of fuel: $\dot{V}_f = \frac{\dot{m}_f}{\rho_f}$

$$\dot{V}_f = 0.37747\ l/\ hr$$

The fuel consumption under acceleration in km/l is:

$$FC = \frac{v}{V_f}$$

$$FC = 79.5\ km/l$$

Burn and Coast Stoichiometric Otto Cycle Fuel Economy:

The deceleration due to drag force is:

$$F_D = ma$$

$$a = -0.05903\ m/s^2$$

For this acceleration, distance covered for vehicle to go from 40 km/h to 20 km/h is:

$$s = ut + \frac{1}{2}at^2$$

$$s = u\left(\frac{v-u}{a}\right) + \frac{1}{2}a\left(\frac{v-u}{a}\right)^2$$

$$s = 40\left(\frac{20-40}{-0.05903}\right) + \frac{1}{2}\left(\frac{(20-40)^2}{-0.05903}\right)$$

$$s = 784.2\ m$$

The fuel-consumption calculated is

$$FC = 828.2\ km/l$$

Accelerating Lean Atkinson Cycle Fuel Economy:

The Volumetric flow rate of air for Atkinson cycle is:

$$\dot{V}_a = \frac{V_s \times \eta_v \times N}{120}$$

$$\dot{V}_a = \frac{57.42 \times 0.85 \times 2600}{120}$$

$$\dot{V}_a = 9.623 \times 10^{-4}\ \ m^3/s$$

And mass flow rate of air is:

$$\dot{m}_a = \frac{P_i \times \dot{V}_a}{R \times T_i}$$

$$\dot{m}_a = \frac{100000 \times 9.623 \times 10^{-4}}{285 \times 313}$$

$$\dot{m}_a = 1.072 \times 10^{-3}\ \ kg/s$$

If lean air-fuel mixture is to be used, air to fuel ratio could be extended upto 15.5. For this value of air-fuel ratio, the mass flow rate of fuel is

$$\dot{m}_f = 6.913 \times 10^{-5}\ kg/s$$

Volume flow rate of fuel: $\dot{V}_f = \frac{\dot{m}_f}{\rho_f}$

$$\dot{V}_f = 0.3257\ l/\ hr$$

The fuel consumption under acceleration in km/l is:

$$FC = \frac{v}{V_f}$$

$$FC = 92.1\ km/l$$

Burn and Coast Lean Atkinson Cycle Fuel Economy:

Since the acceleration and deceleration remains same, the distance covered under burn and coast is also the same. Therefore, the fuel consumption for lean air-fuel mixture with Atkinson cycle is:

$$FC = 959.4\ km/l$$

Cooling System:

Energy Input to System:

$$\dot{Q}_f = \dot{m}_f \times Q_{LHV}$$

Q_{LHV} of fuel we are going to use is 42,900 kj/kg

$$\dot{Q}_f = 6.913 \times 10^{-5} \times 42{,}900$$

$$\dot{Q}_f = 2.96\ kW$$

Fin Efficiency:

Flat rectangular fins are chosen because of ease of manufacturability and cost.

$$a = \sqrt{\frac{hp}{kA}}$$

Where,

h = heat transfer coefficient = 40 W/m^{2}ºC

l = length of fin = 65 mm

w = width of fin = 90 mm

t = thickness of fin = 1.5 mm

p = perimeter = 2(l + t) = 2(65+1.5)/1000 = 0.133 m

k = Thermal Conductivity = 180 W/mºC

A = area of fin = (l × t) = (1.5 × 65)/1,000,000 = 0.0000975 m^2

Thus a is evaluated as:

$$a = \sqrt{\frac{40 \times 0.133}{180 \times 0.0000975}}$$

$$a = 17.41$$

The fin efficiency is:

$$\eta_{fin} = \frac{\tan(aL)}{aL} = 0.717 = 71.7\%$$

Cylinder and Piston Design:

Piston head thickness:

$$t_h = \sqrt{\frac{3pD^2}{16\sigma_t}}$$

Where,

p = maximum gas pressure or explosion pressure

D = Cylinder bore or outside Diameter of piston

σ_t = Permissible bending (tensile) stress for the material of the piston.

Now under consideration of heat transfer, the thickness of the piston should be such so that heat can be effectively and quickly transferred to the cylinder walls. This thickness is given by:

$$t_h = \frac{H}{12.56k(T_C - T_E)}$$

Where,

H = Heat flowing through the piston head

k = Heat conductivity factor

T_C = Temperature at the center of the piston head

T_E = Temperature at the edges of the piston head

The heat flowing through the piston head (H) may be determined from the following formula:

$$H = C \times HCV \times m \times B.P.$$

Where,

C = Percentage of heat absorbed by the engine

HCV = Higher Calorific Value of fuel

m = mass flow rate of fuel per brake power

B.P. = Brake power per cylinder of engine

Piston Pin:

Piston pin outside diameter:

$$Piston\ laod = \frac{\pi D^2}{4} \times p$$

Now, the load on piston pin due to bearing load is given by:

$$Bearing\ load = bearing\ pressure\ \times bearing\ area$$

$$Bearing\ load = p_{b1} \times d_o \times l_1$$

Where,

p_{b1} = Bearing pressure at small end of connecting rod bushing. For bronze bushing, it is taken as 25 N/mm^2

d_o = Piston pin outside diameter

l_1 = Piston pin length in small end of connecting rod bush = 0.45D = 17.415 mm

Equating the bearing load with piston load, we get

$$d_o = \frac{\pi D^2}{4} \times p \times \frac{1}{p_{b1} \times l_1}$$

$$d_o = \frac{\pi (38.7)^2}{4} \times 6.75 \times \frac{1}{25 \times 17.415}$$

$$d_o = 18.24\ mm$$

Piston pin inner diameter:

internal diameter of piston pin is taken as: $d_i = 0.6d_o$

$$d_i = 0.6d_o \; mm$$

$$d_i = 0.6(18.24) mm$$

$$d_i = 10.95 \; mm$$

Length Between Supports:

$$l_2 = l_1 + \frac{D - l_1}{2} = \frac{l_1 + D}{2}$$

$$l_2 = \frac{17.415 + 38.7}{2}$$

$$l_2 = 28.06 \; mm$$

Maximum Bending Stress:

$$M = \frac{P.D}{8}$$

$$M = \frac{10586.55 \times 38.7}{8}$$

$$M = 51212.42 \;\; N \, mm$$

It is also known that maximum allowed bending stress is:

$$M = Z \times \sigma_b$$

$$M = \frac{\pi}{32}\left[\frac{(d_o)^4 - (d_i)^4}{d_o}\right]\sigma_b$$

Therefore,

$$51212.42 \; = \frac{\pi}{32}\left[\frac{(18.24)^4 - (10.944)^4}{18.24}\right]\sigma_b$$

$$\sigma_b = 98.76\ MPa$$

Valves:

Curtain Area/Valve Area:

$$A_C = 2\pi rl$$

Where,

A_C = Curtain Area

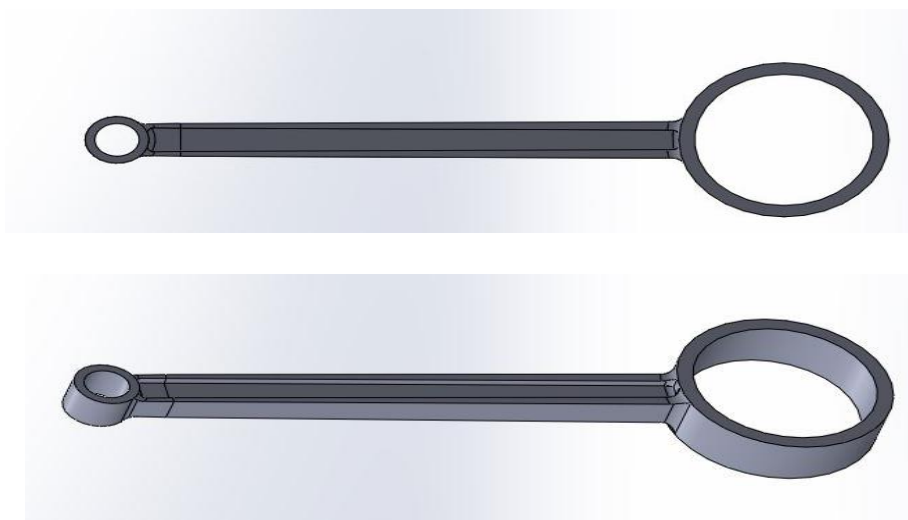
r = radius of valve head

l = valve lift

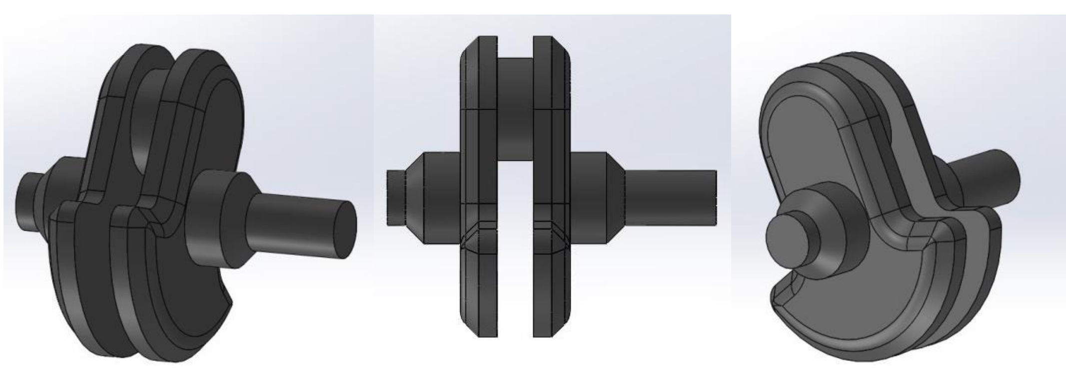
APPENDIX II: CAD DRAWINGS

1. Piston

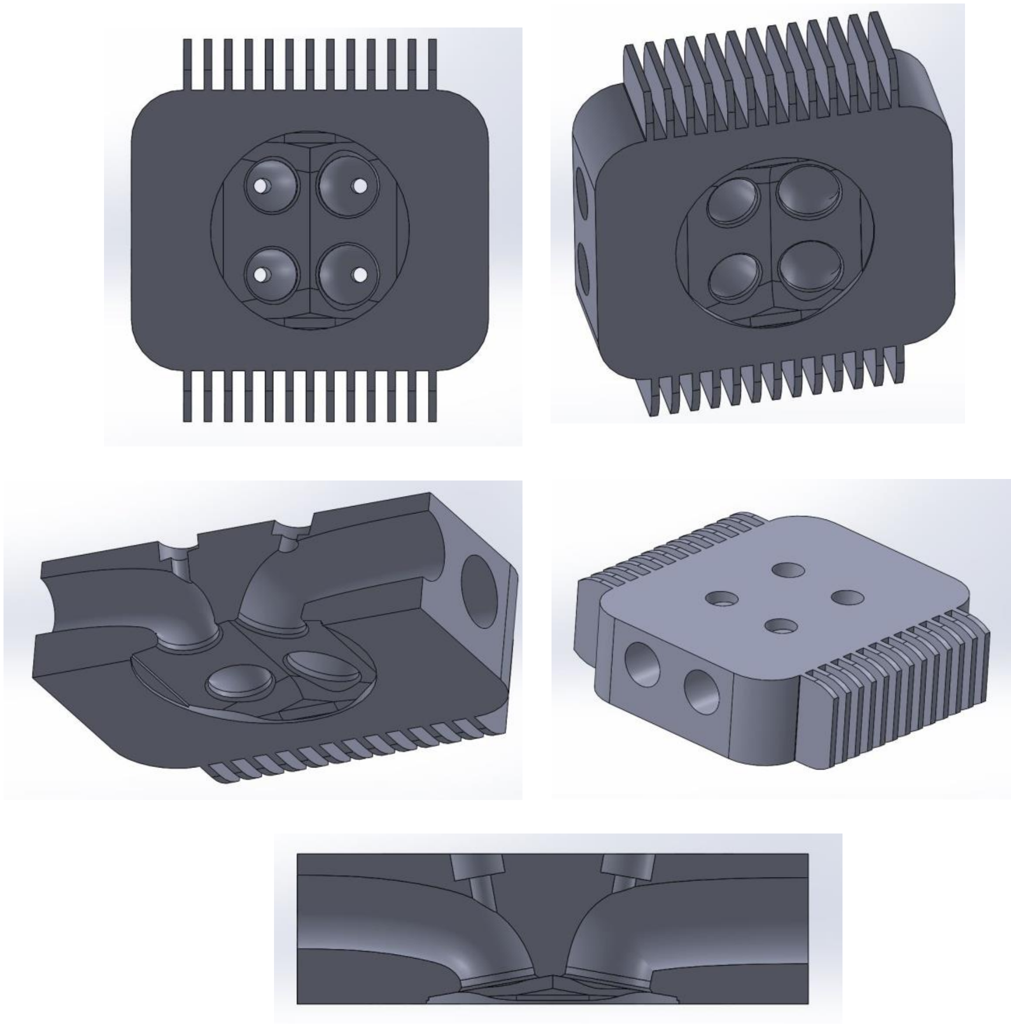
2. Connecting Rod



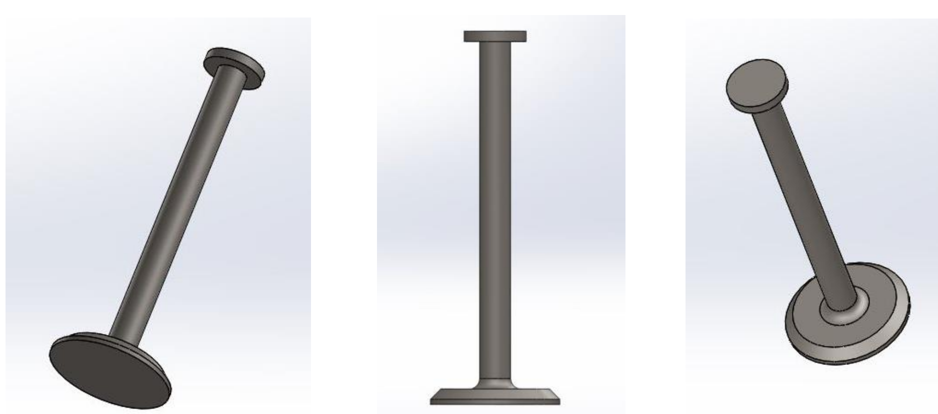
3. Crankshaft



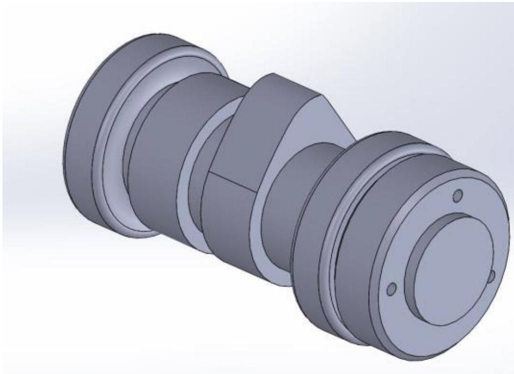
4. Cylinder Head

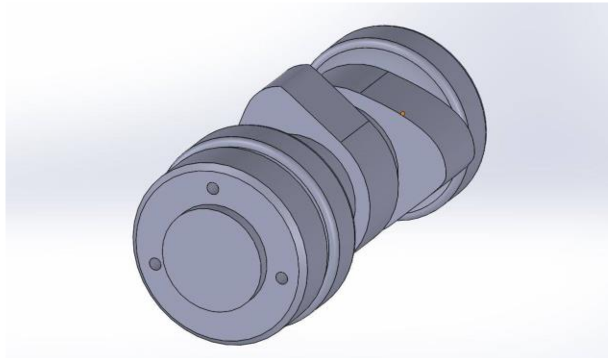


5. Valves

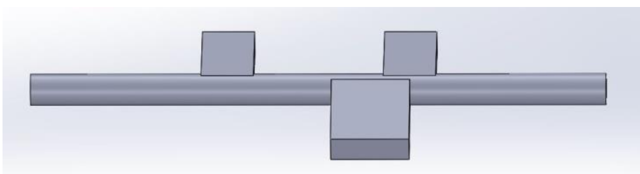


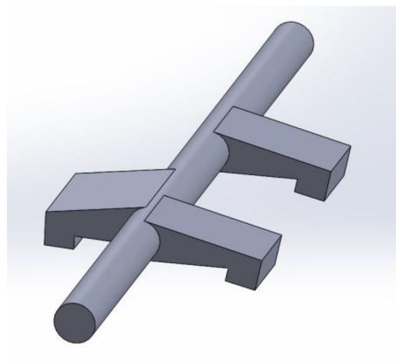
6. Camshaft



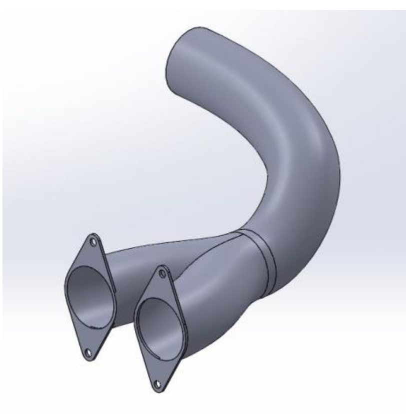


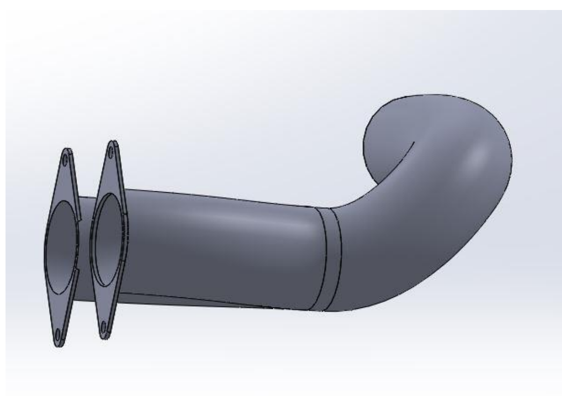
7. Rocker Arms



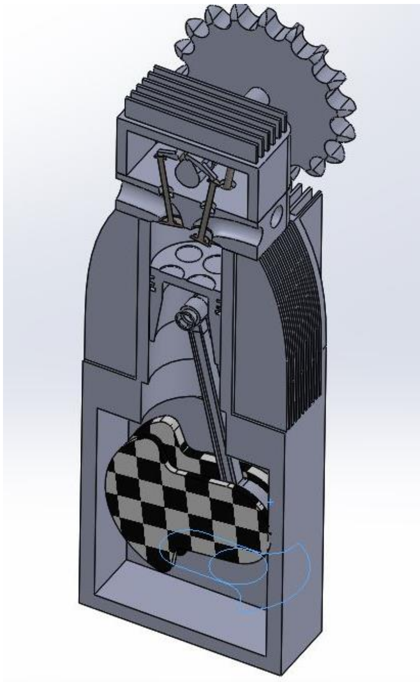


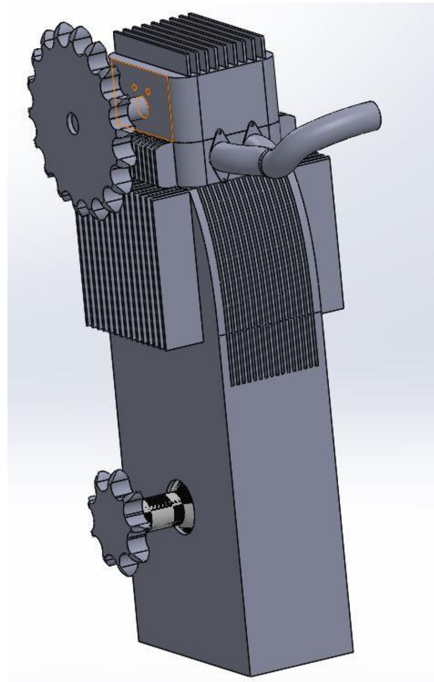
8. Intake Manifold



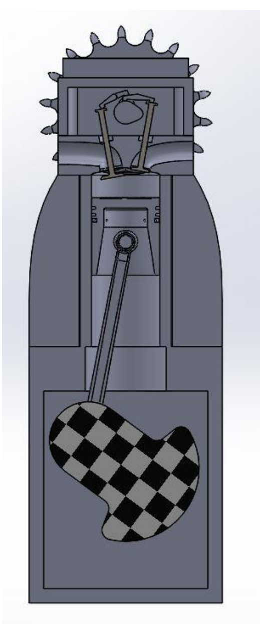


9. Assembly

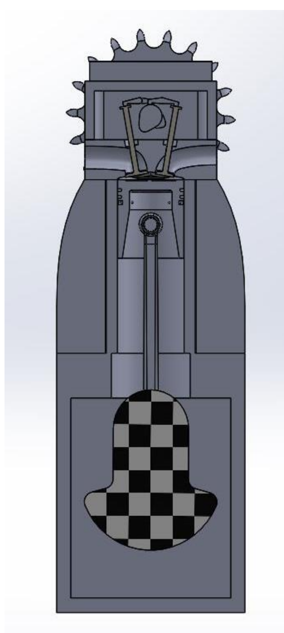




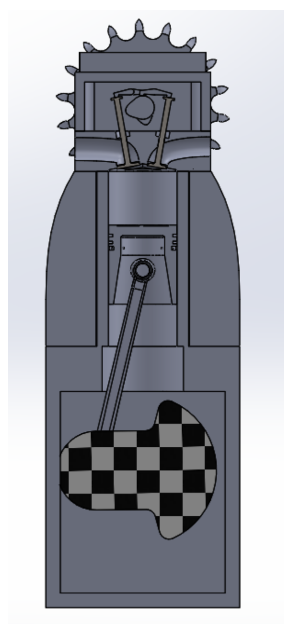
10. Strokes



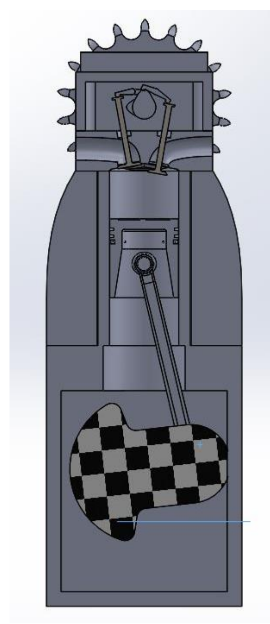
INTAKE



COMPRESSION



POWER



EXHAUST