UNIVERSIDADE DA BEIRA INTERIOR
Engenharia

Design and Fabrication of a small SI Two-Stroke Engine

Murphy 1.0

(versão final após defesa)

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Resumo

Motores de combustão interna (ICEs, Internal combustion engines) desempenham um papel importante na sociedade atual e vão continuar a fazê-lo pelo futuro próximo. Mesmo com todos os desenvolvimentos e progressos que têm sido feitos em motores elétricos, ainda falta percorrer um longo caminho até que estes consigam substituir o papel dos ICEs, se é que o farão. É então desta forma importante continuar o desenvolvimento de ICEs, caso contrário, poderá não se descobrir vantagens adicionais que os ICEs ainda tenham para oferecer.

Esta dissertação foi feita com intuito de avaliar a possibilidade de fazer o design e fabricar um cilindro e uma cabeça de cilindro para um motor a dois tempos na universidade. Este trabalho vai então consistir num projecto conceptual sobre o design e a fabricação de um cilindro e uma cabeça para um motor a dois tempos controlado pelo pistão (Piston-ported) e acendido a faísca. Como os motores normalmente têm algum tipo de nome/referência, o protótipo desenhado e fabricado foi chamado de “Murphy 1.0”.

Primeiramente, foi feito um estudo teórico sobre motores de combustão interna, mais especificamente sobre motores a dois tempos acendidos por faísca, e com a informação reunida o protótipo foi desenhado em CATIA V5. Para fabricar o “Murphy 1.0” foi estabelecido um processo de fabrico que consegue ser reproduzido na universidade para obter peças de fundição com precisão e detalhe considerável. A maioria dos componentes práticos necessários para a fabricação e para o funcionamento do motor também foram fabricados na universidade e vão ser apresentados ao longo deste documento.

Foram enfrentados vários problemas durante a componente experimental desta tese e os maiores irão ser apresentados ao longo deste documento. Um exemplo destes problemas foi o facto de que o motor foi idealizado e desenhado para funcionar com um carburador e um pequeno sistema de igniçao por descarga capacitiva (CDI, capacitator discharge ignition) mas devido a problemas que ocorreram, ambos tiveram que ser substituídos. Foi então feita a cablagem e a instalação de uma centralina (Microsquirt V2) com os sensores/atuadores necessários para permitir o controlo da ignição e injeção do combustível eletronicamente.

Infelizmente, não foi possível conseguir que o “Murphy 1.0” ligasse. As possíveis razões para tal vão ser apresentadas e explicadas nos capítulos finais mas a razão que estava a impedir com certeza o funcionamento do protótipo era a falta de compressão no cilindro. Esta baixa compressão era proveniente de fugas na junta da cabeça com o cilindro. Devido ao prazo de entrega definido para esta dissertação, não foi possível encontrar uma solução para este problema a tempo. As tentativas feitas para tal, vão ser explicadas também no fim deste documento.

Sem primeiro resolver o problema relativo à compressão, a metodologia seguida para dimensionar as janelas (ports) não pode ser avaliada adequadamente. Foi desta forma inconclusivo se o motor iria funcionar caso o problema da compressão não persistisse. Ao longo deste documento, todas as escolhas não só relativamente ao design mas também ao processo de fabrico usado, serão explicadas para permitir uma replicação do trabalho atual ou uma adaptação.

Palavras-chave

Design; Fabricação; Motor dois tempos; Motores de combustão interna; Ignição por faísca; Centralina; Fundição.

Abstract

Internal combustion engines (ICEs) perform an important role in today's society and will keep doing so in the foreseeable future. Even with all the technological development regarding electric motors, there is still a long way to go before they can substitute ICEs, if they ever do. It is therefore important that the development of ICEs continues, otherwise, the potential further advantages that ICEs can offer will not be realized.

This dissertation was done to assess the possibility of designing and fabricating a two-stroke engine cylinder and cylinder head in the university. This work will then consist of a conceptual project regarding the design and fabrication of a piston-ported two-stroke SI engine cylinder and cylinder head. As engines typically have some sort of name/reference, the prototype designed was named "Murphy 1.0".

Firstly, a theoretical study about internal combustion engines, more specifically two-strokes SI engines, was done and then with the information gathered, a prototype was drawn in CATIA V5. To fabricate the "Murphy 1.0" prototype it was established a fabrication process that can be achieved at the university to obtain cast-pieces with considerable accuracy and detail. All the practical components necessary for the fabrication and for the start up of the engine were also made at the university and will be presented throughout this document.

Several adversities were faced in the experimental component and the major ones will be presented in this document. For instance, the design was made idealizing the usage of a carburetor and a CDI ignition system for small engines but due to experimental problems, both had to be changed. It was then done the wiring harness to an ECU (Microsquirt V2) and afterward, the required sensors/actuators were installed to both the ECU and to the engine to permit the electronical control of the ignition and fuel injection.

Unfortunately, it was not possible to get "Murphy 1.0" to start up. The possible reasons for such will be presented and explained in the final chapters but the reason that was most likely impeding the prototype from working was the lack of compression in the cylinder. This lack of compression was prevenient from a leak in the cylinder head gasket. Due to the deadline set for this dissertation, it was not possible to find a solution to fix this in time. The attempts made to solve this problem in the available time will also be explained at the end of this document.

Without first solving the problem regarding compression, the methodology followed to design the ports cannot be properly assessed. It was therefore not conclusive if the designed engine could or could not work. Throughout this document, all the choices and steps made will be explained, not only regarding the engine design but also the fabrication methods used so that a replication, or an adaption, of this work can be made.

Keywords

Design; Fabrication; Two-Stroke; Internal Combustion Engine; Spark-Ignition; ECU; Metal-casting.

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Nomenclature

A	Cross section area of the cylinder
A_{fin}	Area of one fin
A_{unfin}	Area of cylinder where there are no fins
B	Bore
C	Reboring allowance
C_R	Compression ratio
C_{Rcc}	Crankcase compression ratio
h	Convective heat transfer coefficient
L	Stroke
m_{ar}	Mass of unburned mixture from the previous cycle
m_{as}	Mass of fresh charge supplied per cycle
m_{dref}	Necessary mass to fill the swept volume
m_{ex}	Mass of the exhaust gasses
m_{sref}	Necessary mass to fill the entire cylinder volume
m_{tas}	Mass of fresh charge that is trapped in the cylinder
m_{tr}	Total mass of the charge that is trapped in the cylinder
$\dot{m}_a$	Air mass flowrate
$\dot{m}_f$	Fuel mass flowrate
N	Rotational velocity
N_C	Number of cylinders
N_{fin}	Number of fins
n_R	Number of crank revolutions for each power stroke per cylinder
P	Power
P_{max}	Maximum gas pressure inside the cylinder
$\dot{Q}_{fin}$	Heat dissipation rate from the fins
$\dot{Q}_{total}$	Total heat rate dissipation
$\dot{Q}_{unfin}$	Heat dissipation rate from the unfinned area of the cylinder
sfc	Specific fuel consumption
T	Torque
T_b	Temperature of the engine
T_∞	Temperature of the air flowing through the fins
t	Time
th_c	Cylinder wall thickness
$th_{c,empirical}$	Empirical cylinder wall thickness
th_{ch}	Cyliner head thickness
V_c	Clearance volume
V_{cc}	Volume of the crankcase at bottom dead center
V_d	Swept volume
η_{bth}	Brake thermal efficiency
η_{fin}	Fins efficiency
η_{ith}	Indicated thermal efficiency
η_m	Mechanical efficiency
η_{re}	Relative efficiency
η_{ve}	Volumetric efficiency

ρ_a	Inlet air density
σ_c	Allowable tensile stress

List of Acronyms

ABS	Acrylonitrile Butadiene Styrene
BDC	Bottom Dead Center
BMEP	Brake mean effective pressure
CDI	Capacitive discharge ignition
CE	Charging Efficiency
CI	Compression Ignition
CAD	Computer Aided Design and drafting
CATIA	Computer Aided Three-Dimensional Interactive Application
DE	Delivery Ratio
DCA	Department of Aerospace Sciences
DC	Direct Current
EMF	Electomotive Force
ECU	Electronic Control Unit
EFI	Electronic fuel injection
FMEP	Friction mean effective pressure
IMEP	Indicated mean effective pressure
MAP	Manifold absolute pressure
MEP	Mean effective pressure
MPS	Mean piston speed
PLX-DAQ	Parallax Data Acquisition Tool
PETG	Polyethylene terephthalate glycol
PLA	Polylactic acid
RC	Radio controlled
SE	Scavenge efficiency
SR	Scavenge ratio
SI	Spark Ignition
TDC	Top Dead center
TE	Trapping efficiency
UBI	University of Beira Interior
VE	Volumetric efficiency

Chapter 1

Motivation and Objectives

1.1 Motivation

For this thesis, it was chosen a subject that would allow acquiring a more profound and deep knowledge about propulsion, more specifically engine design and its fabrication. Even though the technological developments in electric motors has been increasing and now being a possible substitute for combustion engines, there is still a long way to go. In several applications, combustion engines are still a better choice. Combustion engines have been used in the aeronautical industry for a very long time now, but piston engines are now mainly used in light aircraft and RC model airplanes. It is, however, within the branch of ultralight aircraft that two-stroke engines stand out above four-strokes. The majority of ultralights produced in the recent past years have been powered by two-stroke engines and good reasons exist as to why. For instance, they require fewer parts and less moving ones too, being, therefore, lighter and mechanically simpler; have a superior power to weight ratio; are simpler and cheaper to maintain and build. It is also of interest to point out that two-stroke engines may be making a comeback, a huge amount of research is currently being made in two-stroke engines by companies like KTM and Honda. Formula 1 Chief Technical Officer (Pat Symonds) has recently stated that the future of F1 may reside in two-stroke engines, and changes may happen as soon as 2025. If such big developments are being made in two-stroke engines and are now even being considered for the future of F1 and MotoGP, it could only further enhance its potential in light aviation.

As two-strokes still play an important part in ultralight aviation and might even be making a comeback, it was of the authors' keen interest to deepen is knowledge in this kind of propulsion system. This thesis will then focus on the conceptual project of a small, simple piston ported two-stroke engine and its construction. It will regard all the steps, as well as the methodology used, to design and fabricate a two-stroke engine cylinder and cylinder head. These were designed for a recovered piston with the corresponding crankcase, crankshaft and connecting rod. The reasons why it was opted only to fabricate the cylinder and cylinder head are the reasons that also justify why the majority of projects regarding engine design rarely go as far as into building a prototype. The design of ICEs is mostly empirical and those who possess that information are usually corporations, whom for obvious reasons won't share them with the public. Consequently, not much empirical information regarding engine design can be freely accessed/-found, which means that any attempt to build a prototype isn't assured of success, given the lack of solid empirical information to base the design at. Hence, having a company build the designed prototype is extremely expensive and could prove ungrateful; and the remaining alternative which is to fabricate it oneself is a long and laborious process with even lower chances of success.

It is, therefore, important as well as interesting to at least try and assess the possibilities of designing and fabricating two-stroke engines in the university. If one proves to be successful, it will permit the creation of empirical data from which conclusions can be drawn and future works can take place with better and better results.

1.2 Objectives

This thesis main objective was to try and create a functional prototype with a simple design that could be fabricated with the resources available at the university. This means that one of the thesis main objectives was to try and replicate a fabrication method that can be used to obtain a functional prototype. As no groundwork has been done at the university as for how to fabricate a two-stroke engine with the resources there available, this project will serve as the groundwork for future projects where the fabrication of an engine is intended through castings. This goes as well to the empirical data regarding the design of a two-stroke engine. So this thesis objectives can be listed as:

- Understand the functioning of two-stroke engines
- Dimension and design a cylinder and cylinder head
- Find a fabrication method that can be used and improve it as much as possible.
- Assemble the fabricated components with the existing ones.
- Check if the prototype is functional.
- Take basic performance parameters of the engine.
- Creation of empirical data in engine design for future works at the university.

1.3 Dissertation outline

This dissertation is divided into 6 main chapters, each dedicated to a different objective but all dependent on one another. It follows the traditional format of a master thesis, being the first chapter the current one which includes the motivation of this project as well as its objectives.

The second chapter is dedicated to the bibliographic review which contains the theoretical content regarding the operation of two-stroke spark-ignition engines and some of the different variations possible in its design. It will also be looked into the main different options for ignition systems, mixture preparation methods as well as the performance parameters of spark-ignition engines.

The third chapter is a description of the methodology followed for the fabrication and the dimensioning of the engine.

The fourth chapter describes all the choices made for the design of the engine such as the cylinder and cylinder head thicknesses and the dimensions of the ports. The process of fabrication and machining are also explained in-depth with all the conclusions that were drawn from the various attempts.

In the fifth chapter, the experimental components necessary for the test runs are presented in detail as well as the troubleshooting done to try and start up the engine.

In the final chapter, are presented the conclusions obtained from the present dissertation and possible future works to be done in this area.

Chapter 2

Bibliographic Review

2.1 Historical context of piston engines

Internal combustion engines date back to 1860 when J. J. E. Lenoir (1822-1900) [1] developed an engine that burned a mixture of coal gas and air at atmospheric pressure. In this engine there was no compression before combustion, the inflammable gas and air were drawn into the cylinder and ignited by an electric spark. Later in that decade, 1867, Nicolaus A. Otto (1832-1891) and Eugen Langen (1833-1895) introduced the atmospheric engine [1]. This engine was more efficient than Lenoir's engine and had half its fuel consumption [2]. In 1876, Otto came up with the four-stroke cycle to overcome the engine's shortcomings of low thermal efficiency and excessive weight [1]. This was the discovery that permitted the internal combustion engine industry to thrive. Soon after, the two-stroke cycle was developed by Dugald Clerk (1854-1913) and James Robson (1833-1913), in England and Karl Benz (1844-1929), in Germany [1]. The two-stroke cycle engine was lighter and with a higher power-to-weight ratio than the four-stroke. The next big development in internal combustion engines came in 1892, brought by Rudolf Diesel (1858-1913) [1]. He purposed a new form of internal combustion engine. In this new type of engine, the combustion occurred with the injection of liquid fuel into air heated only by compression. Allowing this way, the creation of engines with the double of efficiency when compared to other internal combustion engines. Based on what these men invented, several improvements and variations have been made. And even now that over a century has passed there is still room for improvement [1]. Engine manufactures, be them the Diesel type or conventional spark-ignition type, continue to show improvements in their engines power, efficiency, fuel consumption, reduced emissions among many others.

2.1.1 Two-stroke engines history

The first successful two-stroke cycle engine was developed by Dugald Clerk, a Scottish engineer, by the end of the 19th century [3]. However, Clerks' original design was not adaptable to small engines. The crankcase-compression two-stroke engine, as we know, is credited to Joseph Day. He named his original design the "Valveless air compressor", which interestingly had two flap valves, one in the inlet port and one in the piston crown, this because he still hadn't come up with the idea for transfer ports. It was one of his workmen, Frederick Cock, who made the modification which would allow the skirt of the piston to control the inlet port, creating this way the classic piston port engine as we know [4]. Joseph Day ended up naming this new variation the "valveless two-stroke engine". This design used the crankcase compression for the induction process, and the control of the timing and area of the exhaust, transfer and intake ports was made by the piston [3]. It was Joseph Day simplification of the two-stroke engine concept that made possible for the construction of small powerful two-stroke engines.

Early applications of two-stroke engines were in motorcycles [3], and with time they became mainly used when there is a necessity for a lightweight engine. This happened due to the legislative pressure on exhaust emissions that some countries established, making manufacturers swing to four-stroke engines in some applications. Two-strokes are still vastly used in motorcycles, handheld power tools, aircraft/car RC models as well as ultralight aircrafts among others. Two-stroke engines usually have high fuel consumption's but a huge amount of research is being conducted in two-stroke engines by companies like KTM and Honda. Recent technological developments made with direct injection, supercharging, new ignition systems are permitting the creation of new forms of two-stroke engines that are more efficient and Eco-friendly. Formula 1 Chief Technical Officer, Pat Symonds, has recently stated in the Motorsport Industry Association conference that the future of F1 may reside in two-strokes and that he is looking forward to it. He also stated that the opposed piston (two-stroke) is coming back and is already delivering efficiencies superior to 50% in road car form. Basil von Rooyen in 2009/2012 has patented and proved a new type of two-stroke engine, the Crankcase-Independent Two-Stroke (CITS) which eliminates the traditional need to burn the two-stroke oil and dramatically reduce the exhaust emissions. Two-stroke engines have enormous potential and only future research will tell how their future will turn out.

2.2 Two-stroke engine classification

Two-stroke engines are classified, as well as other internal combustion engines, by several different parameters. Some of them are [2], [1]:

1. **Type of ignition**:
   - Spark Ignition (SI): the combustion process is initiated by a spark from a spark plug.
   - Compression Ignition (CI): the combustion process is initiated by the elevated temperature of the gas inside the cylinder (caused by the mechanical compression).
2. **Working Cycle**:
   - Two-stroke cycle: A two-stroke cycle engine completes its cycle in two piston movements, one up and down, during only one crankshaft revolution.
   - Four-stroke cycle: A four-stroke cycle experiences four piston movements to complete its cycle, in which two crankshaft revolutions occur.
3. **Application**: Automobile, locomotive, light aircraft, boats, power tools, among others.
4. **Basic engine design**:
   - Reciprocating: also known as a piston engine, is an engine that has one or more cylinders and uses the reciprocating motion of the pistons to cause the rotary motion of the crankshaft. This type of engine can be subdivided by the arrangement of the cylinders: in-line,V, radial, opposed.
   - Rotary engine: is an engine whose radial cylinders rotate about a fixed crankshaft. Rotary engines also include pistonless engines like the Wankel engine, which uses an eccentric rotor to convert pressure into rotating motion.
5. **Fuel used**: gasoline, diesel, natural gas, ethanol, and others.
6. **Combustion chamber design**: Open chamber and divided chamber. Both of each type have several designs.

7. **Method of mixture preparation**: the mixture preparation can be made through a carburetor, through fuel injection in the intake phase or through fuel injection into the engine cylinder.

8. **Cooling system**: the engine can be air cooled, water cooled or even uncooled.

9. **Port design and location**: The location of the ports in the cylinder, can fall under the following categories: - Cross-scavenged porting, loop-scavenged porting, uniflow-scavenged.

From all these distinct parameters it's possible to distinguish and classify an internal combustion engine. The method of ignition, according to literature, is the principal classifying feature [1], since from it come important characteristics such as the fuel to use, how the mixture is going to be prepared, how the combustion chamber should be designed as well as other classifications even though some fall into subcategories of the ones mentioned above. The second most important classifying feature is the engines operating cycle. Two stroke engines induction usually falls under three different categories. They can be piston ported, which is the simplest and uses the piston to control the ports as it moves up and down, they can use reed valves which is the most commonly used in high-performance two-stroke engines, and they can also use rotary valves.

2.3 Spark-Ignition Engines

Spark-ignition engines are those whose combustion process initiates with recourse to a spark plug, which will ignite the air-fuel mixture with a spark near the end of the compression stroke. The necessary components in this type of engine to ignite the mixture are, an ignition coil as the high-voltage source and a spark plug which delivers the spark in the combustion chamber [5].

2.3.1 Two-stroke Engine Components

In a common two-stroke spark-ignition engine, the major components are listed below and can be seen in figure 2.1:

- **Piston**: Usually made from aluminum alloys, it is located inside the cylinder and is responsible for transferring the force from the expanding gases to the crankshaft. This transfer occurs from the up and down movement of the piston.

- **Cylinder/Liner**: Is a cylindrical vessel typically made of cast iron. It's inside of the liner that the piston moves.

- **Block**: It's where the cylinder/s is placed and in small engines is usually made from aluminum alloys.

- **Crankshaft**: It's a shaft that along with the connecting rod is responsible for converting the reciprocating motion of the piston into a rotational one. The crankshaft has traditionally been made with steel and forged [1].

- **Connecting Rod**: It's a rod that connects the piston to the crankshaft. The connecting rod is usually made of steel or an alloy forging, although in small engines sometimes aluminum is used [1].

- **Combustion Chamber**: Located in the top of the cylinder and is where combustion occurs.
- **Spark Plug**: It's the ignition source and is usually located in the combustion chamber, in a zone out of reach of the piston.
- **Ports**: The intake port, responsible for allowing the new mixture of air-fuel inside the crankcase; the transfer port, responsible for transferring the new mixture of air-fuel from the crankcase into the cylinder; and the exhaust port, whose function is to allow the burned mixture out of the cylinder.

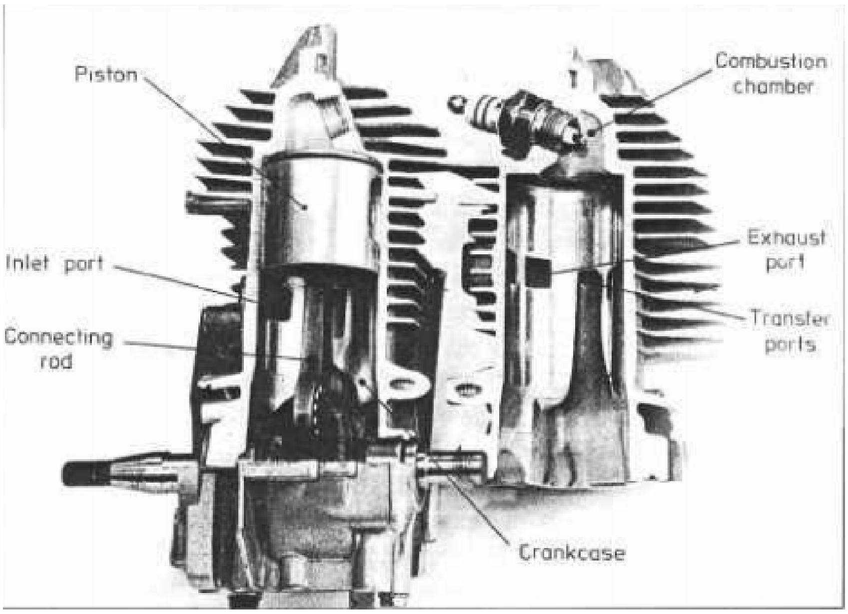

Figure 2.1: Exploded view of a simple two-stroke engine [3]

2.3.2 Two-Stroke Cycle and Timing Diagram

In two-stroke cycle engines we obtain power after only one crankshaft revolution. This crankshaft revolution requires two strokes of the piston, hence its name of two-stroke.

- In the first stroke (compression stroke), the piston goes up in the cylinder compressing the fuel-air mixture trapped there. When the piston is reaching top dead center the mixture is met by a spark plug that will ignite the compressed fuel. As the piston goes up the intake port opens, allowing a new mixture of air-fuel to enter the crankcase.
- In the second stroke (Power stroke), the mixture that was ignited by the spark plug raises the temperature and pressure in the cylinder which forces the piston down [2]. As the piston goes down, the new fuel-air mixture is compressed in the crankcase and when the piston is near the end of its stroke, the exhaust port opens allowing the burnt gases to be expelled. Simultaneously to this, the transfer port opens and the previous compressed charge in the crankcase expands into the cylinder. The process where the burnt gases are expelled from the cylinder as the new mixture is forcing its way into the cylinder is called 'scavenging'.

Another important concept to understand about engines who run on the two-stroke cycle is the timing diagram. In figure 2.2 we can see a typical timing diagram for a piston-ported two-stroke engine, and below an explanation of this diagram.

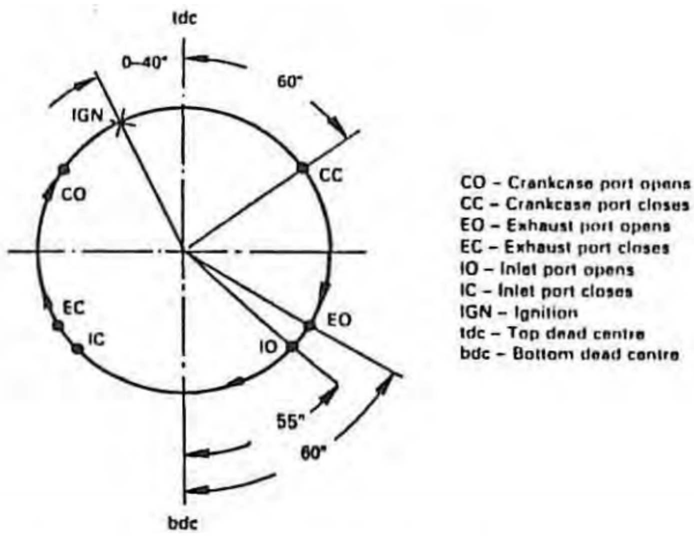

Figure 2.2: Timing Diagram of a typical two-stroke engine [2]

1. The piston uncovers the exhaust port at about 60º before bottom dead center, occurring exhaust blowdown which causes the pressure in the cylinder to approach ambient pressure (EO). This phase described is the end of the power stroke [2].

2. At about 55º to 50º before bottom dead center the inlet port opens (IO), as this happens the compressed charge in the crankcase flows into the cylinder while expelling the burned mixture [2]. This is the scavenging process, which will be explained in Chapter 2.3.3.

3. When the scavenging process ends, both the crankcase and cylinder pressure are near the ambient pressure level [2]. As the piston moves up in the cylinder, the pressure in the crankcase is reduced [2].

4. The inlet port is closed at around 55º after bottom dead center, which marks the end of the scavenging process. The fresh charge is then compressed by the piston in the upward movement [2]

5. The crankcase port is opened at about 60º before top dead center which allows a new mixture into the crankcase. In this stage, the pressure in the crankcase is quite below the ambient pressure [2]

6. Typically, ignition occurs around 10º to 40º before top dead center [2]. The power stroke starts with the ignition of the mixture and ends when the exhaust port is opened (1) [2].

7. Around 60º after top dead center, the crankcase port is closed. It's important to notice that the pressure in the crankcase around top dead center will have risen above the ambient pressure level, which causes some outflow of gas from the crankcase [2].

2.3.3 Scavenging

The process of clearing the burned gases of the cylinder and filling it with a new mixture is called scavenging. It can be defined as the combination of the intake and exhaust processes [1]. There are different categories of scavenging systems, some of them are represented in figure 2.3.

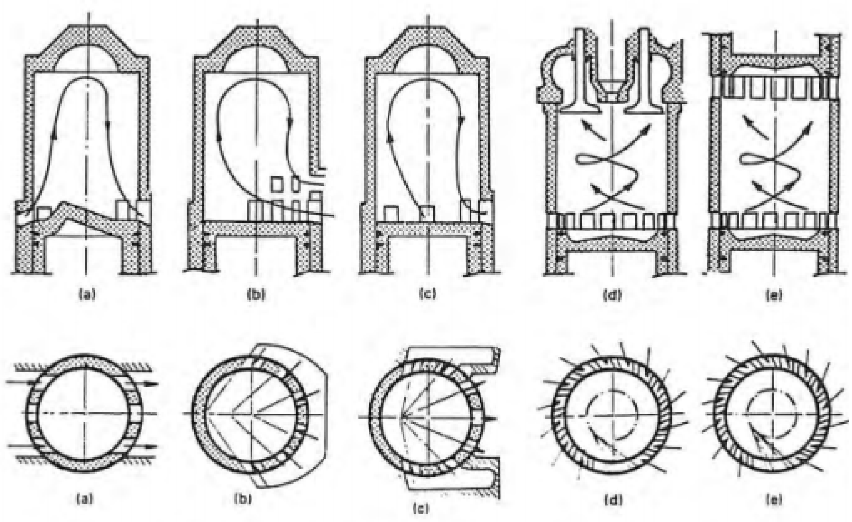

Figure 2.3: Different scavenging arrangements as well as the corresponding port geometry for a two-stroke engine. (a) Cross-scavenging; (b) loop scavenging; (c) Schnurle loop scavenging; (d) uniflow scavenging with poppet exhaust valves; (e) uniflow scavenging with opposed pistons. [2]

The cross scavenge arrangement was used in early crankcase compression two-stroke engines. It has its exhaust and transfer ports in opposite sites of the cylinder wall and is covered/uncovered by the piston. Having the exhaust port and the transfer port on opposite sides forces the burned gases to be pushed out by the crossflow. In this arrangement, the fresh charge could go directly from the transfer port to the exhaust port, so a piston deflector is used to reduce the chance of this unwanted possibility.
Loop scavenging has the benefit of avoiding the use of a piston deflector since the exhaust port is located above the transfer port, reducing this way the possibility of unburnt gases leaving the cylinder directly without using a piston deflector.
The Schnuerle loop scavenging is a system that was designed to improve the efficiency of valveless two-stroke engines. It is a modified version of loop scavenging in which the transfer port is split in two angled ports, one in which side of the exhaust port. This arrangement allows for the piston crown to be of any desired shape and has been almost universally adopted by makers of high output two-strokes [6].
The uniflow scavenge systems are better choices for Diesel engines [2]. In this system, the fresh mixture that enters the cylinder pushes the burnt gases out through the exhaust valve which is in top of the cylinder.
Scavenging is of extreme importance and if not done completely in a two-stroke engine, can lead to bad conditions such as four-stroking. Four-stroking is when combustion occurs every four-strokes or more instead of the original two. This firing is noisy and may even cause damage to the engine. Four-stroking is considered expected in model engines, these small engines initially run as inertially-scavenged four strokes and only after a certain point, when they accelerate enough, they begin to operate as two-strokes [7]. This is an unavoidable condition due to the limitations of their scavenging at low speeds [7].

2.3.4 Port Shapes

There are several different shapes that can be given to the ports of a two-stroke engine, each offers different characteristics to the engine and in the eventuality that the port is not of the appropriate type/size, it can even cause the piston rings to fail. In figure 2.4 we can see several typical port shapes used.

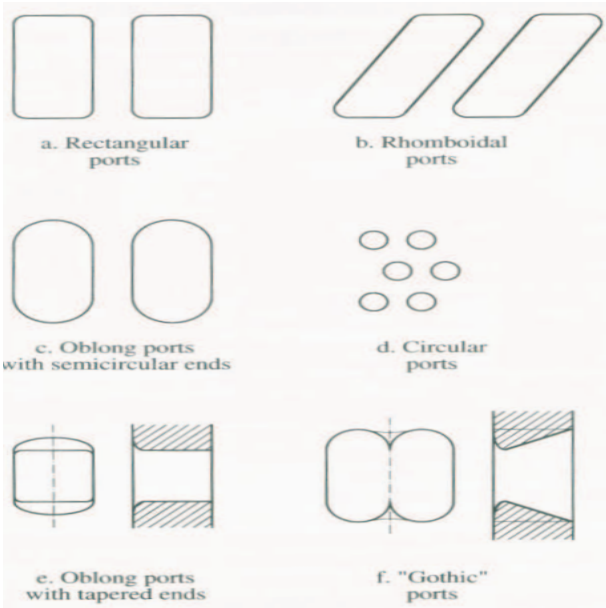

Figure 2.4: Typical port shapes [8]

- Intake Port

  When it comes to the intake port shape, there are several options, some of which can be seen in figure 2.5, but overall the best flow coefficient for any timing-area value is obtained with the widest port [6].

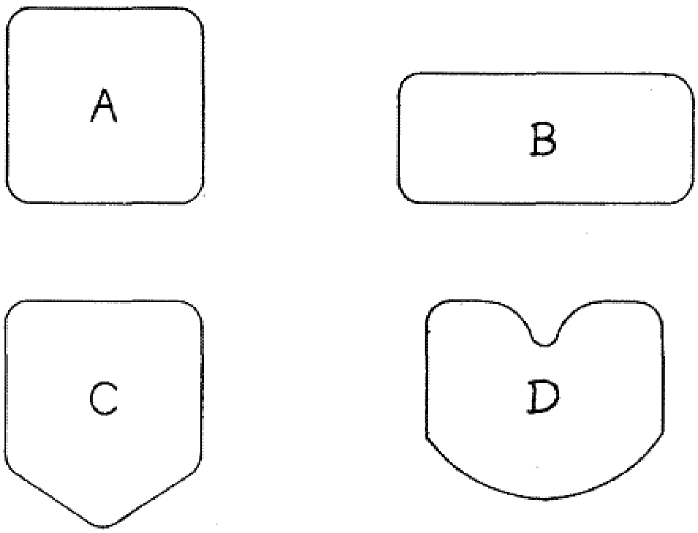

Figure 2.5: Intake port shapes [6]

But widening the port also has its consequences in the intake, a very wide port has the likelihood to cause the rear edge of the piston skirt to snag at the bottom of the port window [6]. This can be addressed by using a window shape with rounded corners instead of squared, which helps prevent rapid wear at the bottom of the piston skirt [6]. Another advantage of a port with rounded corners in the intake instead of a squared one is that it provides a far better flow-coefficient. The ports shown in figure 2.5 all have the same area, but each with different characteristics. The port B has the best flow coefficient, the notched floor on C reduces intake roar and D has a bottom part design that helps prevent piston-skirt snagging with a rib on its top part that acts as a ring-restraint [6].

- Exhaust Port

  As for the exhaust port, simply increasing the width of the port will result in an increase of power but consequently, decrease the ring life drastically [6]. In the eventually of using a too big width, almost instant ring breakage is assured. This because the pistons rings bulge out in the ports as they pass, and while the transfer ports are usually not wide enough for this to happen, the exhaust port tends to be a lot wider and therefore favorable for this to happen. If a squared port with sharp corners and with a total width of around 70 percent of the cylinder bore diameter is used, ring failure is guaranteed to happen in the first revolutions of the crankshaft [6]. Though there are some high-performance racing engines using such widths, there needs to be a very careful process in shaping the port window for such. To improve ring life, it is recommended to have round shapes in the port and have no sharp edges. Typically, an exhaust port shape is rectangular or squared with its corners rounded as we can see in figure 2.6 in port A. But for high-performance engines, the port tends to be widened to an elliptical form as can be seen bellow in port B. In ports with a considerable width, the contours in the elliptical-shaped port help improve ring life by sweeping the ring gently back into its grove, when in a squared port, or a similar type with straight edges the ring may bulge out of its grove and cause its destruction [6].

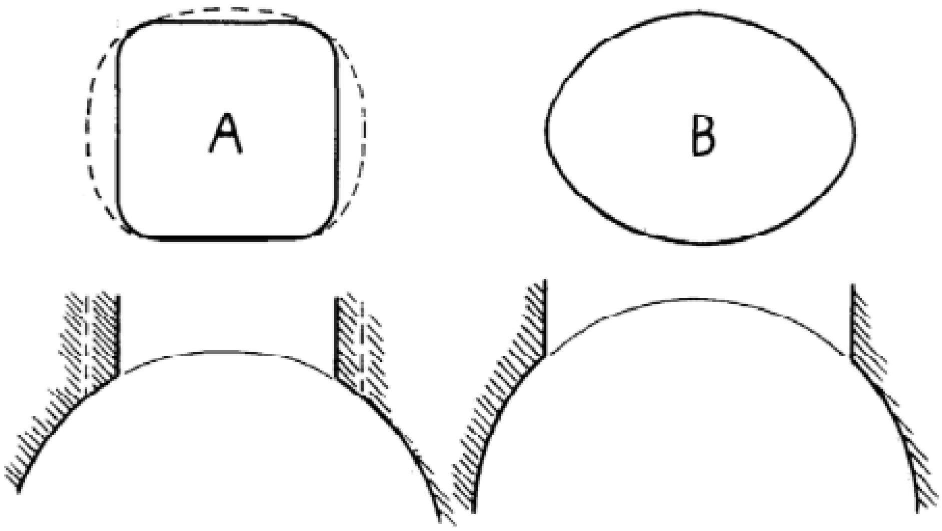

Figure 2.6: Exhaust port widening [6]

- Transfer ports

  As previously said, transfer ports usually aren't wide enough to cause the piston rings to bulge out of their groove and break, but some other important concerns regarding these ports are necessary. The shape of the duct of the transfer ports is one. The best possible shape is a sweeping arc [6], like the one that can be seen in figure 2.7 marked as B. Several engines ducts are shaped with a straight, sharp turn as can be seen in A, this mostly due to manufacturing costs. From the graphic in the figure, it's noticeable that the peak power from one or the other is basically the same, but the scavenging-stream control is better with the sweeping arc which gives a better performance below and above power peak [6].

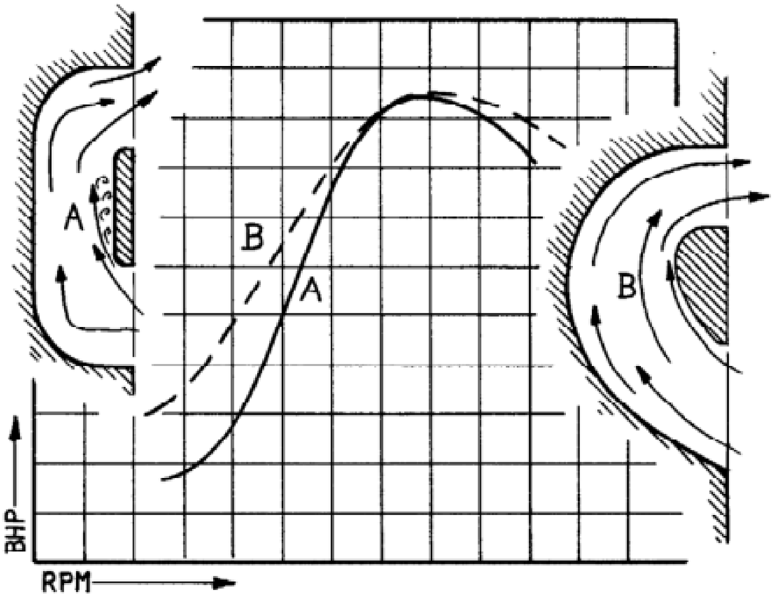

Figure 2.7: Transfer ducts performance comparison [6]

It is also important that the transfer duct should taper down from its entrance to the port window at the top, and it is considered virtually impossible to make an entrance area too large [6]. The transfer port window in the top of the cylinder can also be tilted upward and angled to the rear cylinder wall or simply straight pointing to the center of the cylinder. Each provides a different kind of results in performance. Looking at figure 2.8 as a reference, a slight tilt in the order of 10º like the one in B favors range with a cost in peak power [6].

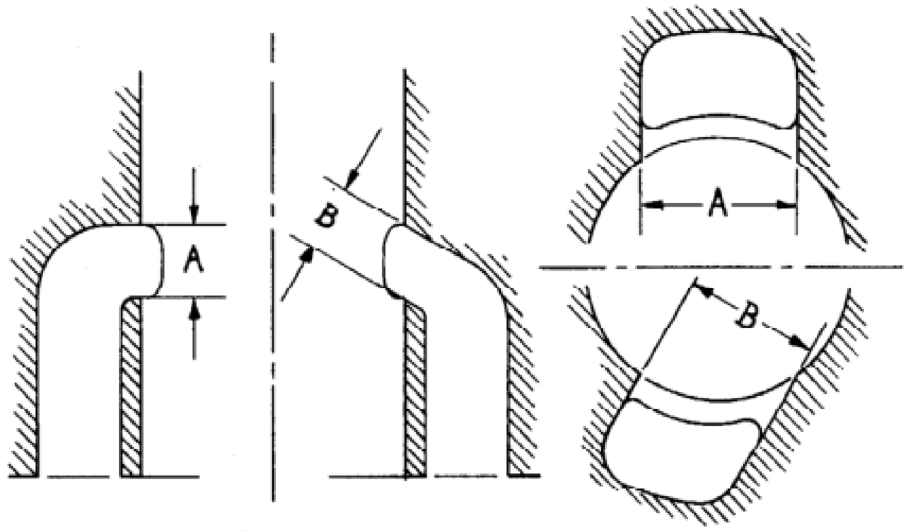

Figure 2.8: Angled port vs straight port[6]

2.3.5 Combustion Chamber

The combustion chamber is located in the cylinder head and as previously said, it's where the combustion takes place. There are different shapes of combustion chambers and different places where the spark plug can be located, both are very important factors that affect how well an engine will run. If these factors aren't properly chosen it can cause knocking, which basically is what happens when the fuel burns in uneven pockets rather than in a uniform burst, reducing the engine's performance or even causing its destruction. Some conditions that can lead to this effect are high compression ratios, a high fuel/air mixture density, a high piston crown among others. To prevent knocking, the end gases of the combustion chamber should be kept as cool as possible and the time needed for the combustion flame to reach end gases, reduced [9]. As such, using a small combustion chamber and locating the spark plug in the center permits the combustion flame to reach the end gases faster since it reduces the flame travel distance to a minimum. As for keeping the end gases cool, the solution comes with the squish band. The squish band reduces the distance between the piston crown and combustion chamber which means that no combustion will occur around the edges of the chamber until the piston has passed TDC, this also permits the large surface area of the piston to act as a heat sink that conducts the heat away from the end gases [9]. Squish bands were originally designed to squish the fuel/air charge from the edges of the chamber into the spark-plug which is what it still does, the gases that are squished quickly meet the spark-plug and carry the combustion flame to all the chamber, and by these means preventing knocking [9].

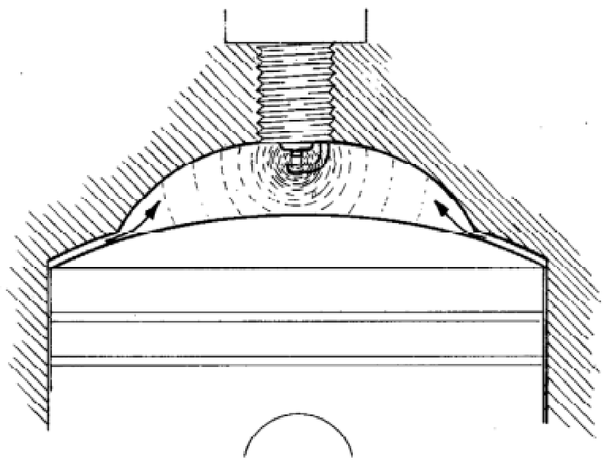

Figure 2.9: Combustion chamber with centrally located spark-plug with the use of a squish band [6]

Over the years, engineers came with several different ideas for combustion chamber shapes. Each has its own set of advantages and disadvantages, the most common type can be seen in figure 2.9, with the centered spark-plug and a centered spherical shape combustion chamber. This configuration allows a shorter flame travel distance in the chamber and a more uniform flame propagation. But not all cylinder heads have the combustion chamber and spark plug centered, and good reasons for such exist. For instance, the area near the exhaust port is always hotter than the one near the inlet port, this due to the hot exhaust gases leaving through the exhaust port every time it opens. To minimize this problem of the cylinder and piston distortion due to the uneven distribution of the temperature, manufactures came with the idea to use a combustion chamber located not in the center but on the side opposite to the exhaust side, as can be seen in figure 2.10. Some combustion chambers are even of conical shape, these combustion chambers reduce the thermal load on the piston because the spark-plug is located further away from the piston. An example of such can be seen in figure 2.11.

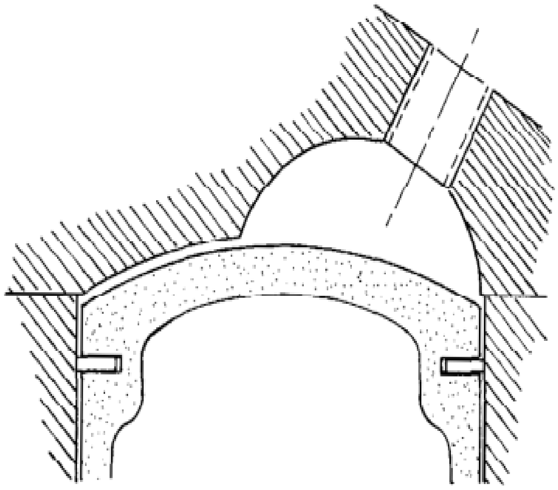

Figure 2.10: Side located combustion chamber[6]

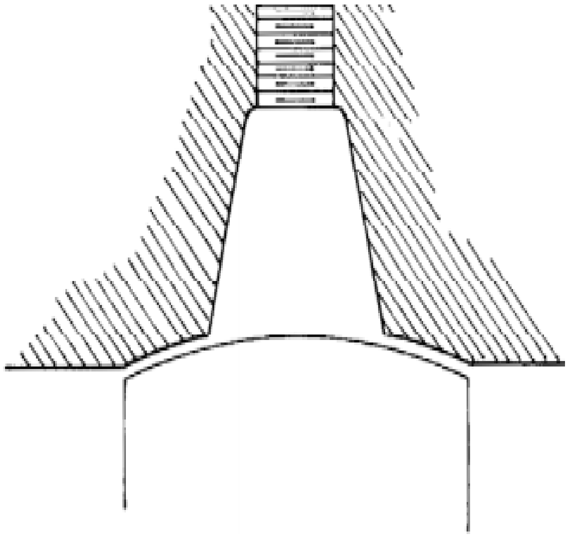

Figure 2.11: Conical combustion chamber[6]

2.3.6 Cylinder Liner

Cylinder liners are one of the most critical parts of an engine's interior. It is inside the liner that the piston will go through its reciprocating motion and therefore suffer an unavoidable fair amount of wear. Engines cylinders can have a liner (sleeved) or not (sleeveless), the reason why most engines are sleeved is due to fabrication costs. The wearing caused by the upward and downward movement of the piston cannot be prevented, and therefore if a liner is not used this means that the cylinder block will itself suffer the wear. This will cause the necessity to replace the block instead of replacing just a liner. Also by using a liner in the designed engine, manufacturers can produce just the liners with superior quality materials (typically cast iron alloyed with chromium) while building the block in inferior quality materials.

Liners are used/needed for several functions such as [10]:

- **The formation of a smooth sliding surface**

  Some of the most important characteristics of a cylinder liner as a sliding surface are as follows [10]:

  1. High galling resistance properties
  2. Less erosion on the liner itself
  3. Less erosion on the piston rings

- **Heat transfer**

  As the liner receives the heat from the combustion and from the reciprocating motion of the piston, it is responsible to help transmit the heat to the coolant in the case of a liquid-cooled engine or to the surrounding air in the case of being air-cooled.

- **Sealing the compressed gas in the cylinder**

  It helps prevent the compressed gases from escaping the cylinder by providing an airtightness between the piston rings and the walls of the liner.

2.3.7 Otto Cycle

One important concept when we talk about internal combustion engines is the Otto cycle. The Otto cycle can be represented as a pressure-volume diagram that models the change that the fuel-air mixture undergoes in pressure and volume [11]. In figure 2.12 the actual P-V diagram of a two-stroke engine can be seen and below is presented a brief explanation of its process.

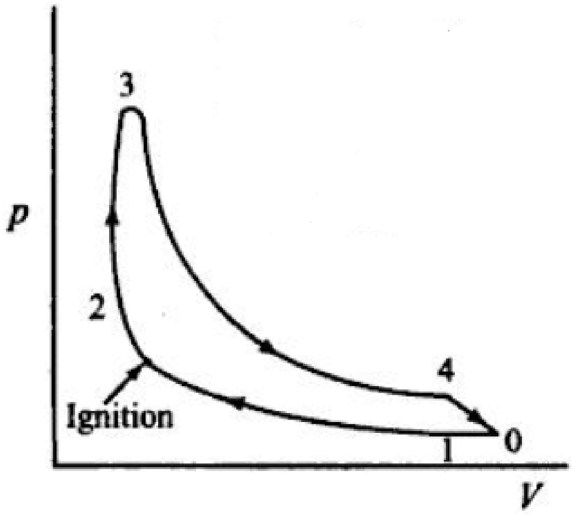

Figure 2.12: Actual P-V diagram of a two-stroke SI engine [12]

- Process 1 to 2: In this phase, the piston will be drawn up opening the intake port, for a new mixture to enter the crankcase, and compressing the fuel-air mixture that had previously entered the chamber [11]. This process is known as the compression stroke.

- Process 2 to 3: In this phase, heat is added at a constant volume. It's when the spark plug ignites the fuel in the combustion chamber, occurring this way the combustion [11]

- Process 3 to 4: The thermal energy generated in the combustion will push the piston down increasing the volume of the chamber. This process is known as the power stroke since it's from it that the thermal energy is turned into motion power [11]

- Process 4 to 0 and 0 to 1: These two processes occur simultaneously, being them the exhaust of the hot gases from the combustion chamber and the entering of a new air-fuel mixture in the cylinder.

2.4 Ignition Systems

An ignition system is made of several components whose function is to ignite the air-fuel mixture inside the combustion chamber. Ignition systems in compression-ignition engines rely on the compression of the air-fuel mixture to ignite while spark-ignition engines rely on the spark generated by the spark-plug. There are a few different types of spark-ignition systems, the most commonly known are the magneto-type system, the battery and coil systems and the electronic ignition system [13].

2.4.1 Magneto Ignition System

Magneto ignition systems are composed by a high-tension magneto, a condenser, one or more spark plugs, and a distributor.

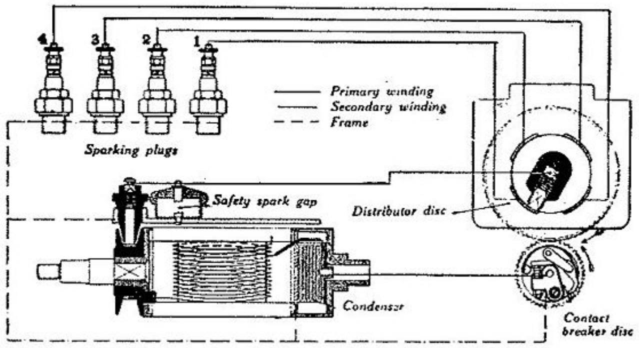

Figure 2.13: Magneto Ignition System [13]

This system represents the simplest ignition method, it is cheap and requires little space, therefore it is often used in small engines like the ones used in lawnmowers. The magneto with high tension generates an electrical current that is connected to the spark-plug, generating this way the spark needed for combustion. As the rotational movement of the magneto generates an electrical current, there is no need for a battery, the system produces its own current.

2.4.2 Battery Ignition System

Battery ignition systems are composed by a battery, ignition switch, ballast resistor, ignition coil, contact breaker, capacitor, distributor, spark plug.

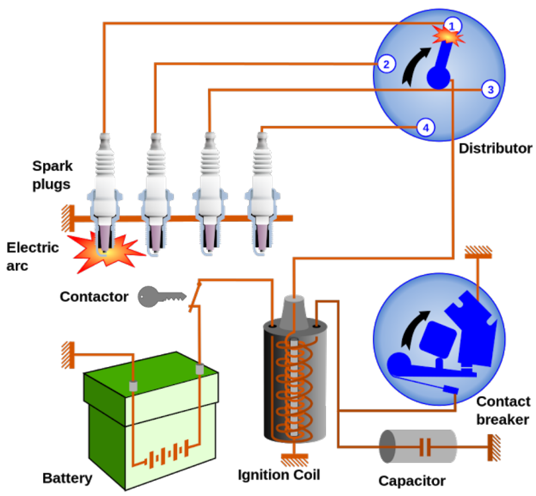

Figure 2.14: Battery Ignition System [14]

This type of system became popular when modern electrical systems appeared, for these allowed a method of charging the battery, which made it a much more reliable ignition system. A brief explanation of how this system works is as follows: As the ignition switch is turned, current flows from the battery through the primary winding as well as the ballast resistor and contact breaker. The ballast resistor function here is to prevent the temperature of the ignition coil to increase as well as to help keep the current down to a safe value. The current that is flowing generates a magnetic field around the ignition coil, whose function is to step up the low voltage to high voltage. The contact breaker function is to make and break the primary circuit, so when it opens the current ceases flowing through the contact breaker and starts flowing through the condenser. The condenser charging, causes the primary current to fall and the magnetic field collapses. This change induces a current in the primary winding which flows in the same direction as the primary current, charging this way the condenser with a much higher voltage than the battery voltage. The high voltage that is generated in the secondary winding is passed into the distributor. The distributor will distribute ignition surges to the spark plug/s in the correct sequence/time.

2.4.3 Eletronic Ignition System

Electronic ignition systems have advantages over the other systems mentioned above. They have no moving parts which avoid mechanisms wear downs. The ignition system is handled by solid-state electronics, this increases its reliability and reduces maintenance requirements. The process to cause ignition in the electric system relies on fewer factors than a mechanical one, resulting this way in a better timing for the engine.
The electronic ignition system is composed by a battery, which is the source of power necessary to the ignition, an ignition switch, which is what turns On and Off the system, an electronic control unit (ECU), whose function is to monitor and control the timing and intensity of the spark automatically, an armature, which is the replacement for the contact breaker as used in battery ignition system, its function is to send the voltage signals to the electronic module in order to make and break the circuit. There is also an ignition coil whose job, as in battery ignition systems, is to produce high voltage for the spark plug, an ignition distributor, that as its name says, is used to deliver/distribute the current to the spark-plug/s, and for last there is a spark plug, which is what produces the spark to cause combustion.

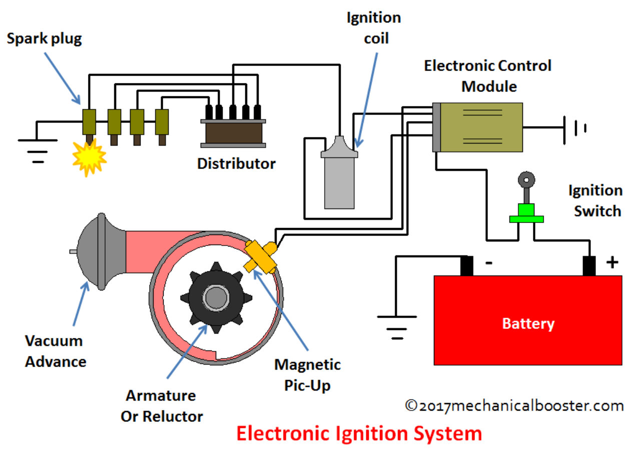

Figure 2.15: Eletronic Ignition System [15]

In figure 2.15 we can see all the components mentioned above connected. When the ignition switch is turned ON, the current starts flowing from the battery through the ignition switch to the coils primary winding which causes the armature to send voltage signals to the ignition module. When the tooth of the rotating armature aligns with the magnetic pic up, the electronic module receives a voltage signal from the pickup coil, what causes the current flow to stop from the primary coil. When the tooth moves from alignment, the ignition module turns ON the current flow again. As in the other systems, a magnetic field is generated around the ignition coil due to the process of breaking and making the signal, causing an EMF (Electromotive force) in the secondary winding which will increase the low voltage greatly. The high voltage generated here is then sent to the distributor, whose function has said before is to distribute the current through the spark plugs in the correct timing. With these currents, the spark plugs will be able to generate the spark needed to cause combustion [15].

2.4.4 Spark-plug and Ignition Advance

The spark-plug, as it was previously said, is responsible for igniting the fuel/air mixture in the combustion chamber. Considering heat range, spark-plugs can be one of two types, "hot" or "cold". A spark-plug is said to be "hot" if it has a long insulator nose and a large area exposed to the combustion gases. In this type of plug, the firing end heats up quickly. The "cold" type has a short insulator nose with a low surface area exposed to the combustion gases, and oppositely to the "hot" type, its firing end doesn't heat up quickly. The insulator tips temperature increases as the engine speed increases, it is therefore recommended for the tip to be at least at 350ºC to prevent misfiring and to be kept lower then 950ºC to prevent preignition [1]. To obtain satisfactory performance results, the central electrode temperature should be between 350ºC to 700ºC [2]. In figure 2.16 we can see a conventional spark-plug design.

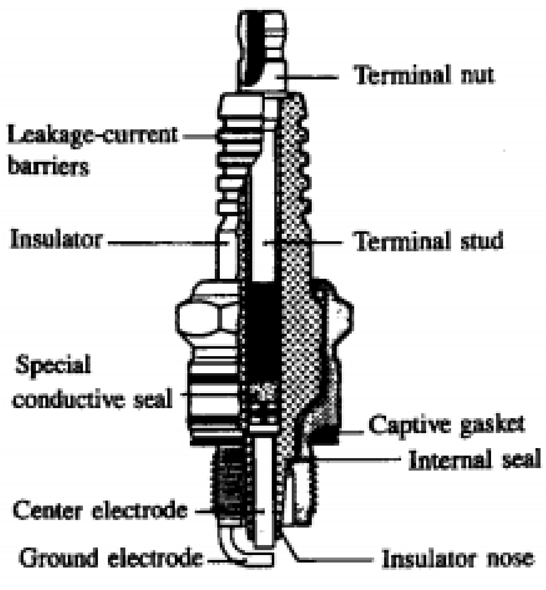

Figure 2.16: Conventional spark-plug [1]

The exact moment when the spark ignites the fuel-air mixture is of crucial importance to the performance of an engine. Ignition advance refers to the timing in degrees of the crankshaft when ignition occurs, usually before TDC. This ignition advance is usually in the order of 10º to 30º before TDC [16]. The reason why it is necessary to advance the ignition timing of an engine is that the burn of the fuel-air mixture does not occur instantly. Even though it only takes a few milliseconds before the flame reaches all the mixture, this time still needs to be accounted for.

This is done by advancing the ignition. The ignition advance is also influenced by the speed at which the engine is working, the geometry of the combustion chamber, the octane ratio of the fuel used, among others. If the engine is working at higher speeds, it will have a lower amount of time for the mixture to burn completely and therefore more ignition advance is needed, the same goes for large combustion chambers, that need more time for the mixture to ignite through all the chamber. The ignition advance used should be one that allows the complete burn of the mixture while preventing engine knocking.

2.5 Fuel

The fuel used in an engine should be chosen accordingly to the engine application and necessity. When we talk about spark-ignition engines the most common fuel is petrol, hence the name of gasoline engines. They differ from the compression-ignition engines whose typical fuel is diesel. Spark-ignition engines also run on fuels like methanol, autogas, nitromethane among several others, but for the application of a small two-stroke engine, the simpler option relies on gasoline. To characterize a gasoline fuel resistance to pre-ignition, the octane rating is used. If a lower octane fuel is used in an engine that should be running on a higher-octane graded fuel, it can cause knocking. Several chemicals are added to the gasoline to improve its chemical stability, some of the chemicals used are ethanol, ethyl tert-butyl ether, and methyl tert-butyl ether. It's important to pay attention to this because certain additives can be dangerous for some engines, such as the case of small engines in which a gasoline fuel without ethanol should be used since ethanol in small engines can cause the engine to fail. Another important parameter of a fuel is its heating value, which will be explained in the chapter Performance Parameters. In small two-stroke engines, there isn't an oil pan or oil pump to lubricate the engine internal components like in four-strokes. This keeps the engine simpler but creates the need to pre-mix the oil with the fuel. The ratio of oil-fuel varies from engine to engine, but typical ratios used in small engines are of 1:32 to 1:50, respectively.

2.5.1 Mixture Preparation

For combustion to occur in an engine, a fuel and an oxidant are needed. The fuel may vary from engine to engine, but the oxidant typically used is atmospheric oxygen. This constitutes the air-fuel mixture and is explained in the chapter Performance Parameters. A certain ratio of air-fuel must be added to the engine for it to function properly. This ratio of air-fuel can either be prepared by a carburetor or by a fuel injector. Both have the same function but have different working principles.

2.5.1.1 Carburetor

Carburetors were the first mechanical component created to provide the air-fuel mixture to an internal combustion engine and were first developed by Karl Benz, founder of Mercedes. They operate on the Bernoulli principle, which in a simple manner says that the faster the air moves, the lower the static pressure is going to be and the higher the dynamic pressure will be. In figure 2.17 we can see the essential components of a carburetor.

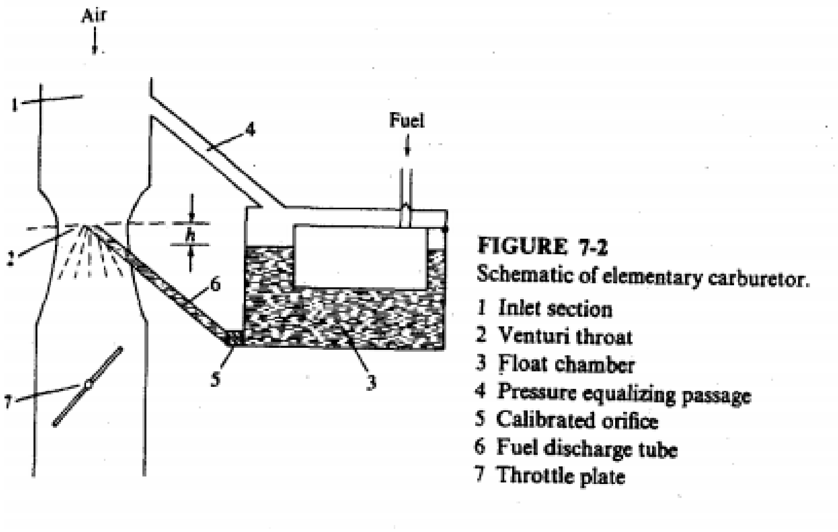

Figure 2.17: Carburetor Schematics [1]

In carburetors, the air-flow passes through a converging-diverging nozzle, which is called a venturi [1]. The pressure difference between the throat of the nozzle and the inlet section is used to measure the quantity of fuel flow that is going to be added to the airflow. The fuel is added to the air stream through a fuel discharge tube and is convected by the air stream as it passes through the throttle plate into the intake manifold [1]. The evaporation of the fuel is initiated inside the carburetor and is continued through the manifold [1]. Typically, carburetors are tuned to provide a stoichiometrically mixture at standard atmospheric conditions, so when this system suffers an atmospheric change it is not prepared to adapt, which makes a great downside for this system in terms of aviation. Carburetors have been increasingly replaced by fuel injection systems, which can offer better power and performance as well as the capability to adapt to changes in atmospheric pressure. But in some cases, they still are a reliable choice. Carburetors are a lot less expensive and have the bonus of simplicity, which makes them far easier to maintain.

2.5.1.2 Fuel Injection

Fuel injection systems were developed to replace traditional carburetors. Initially, fuel injection was created to extract the maximum power output from an engine [2]. This system can operate normally regardless of orientation, contrary to carburetors who cannot operate upside down or in cases of microgravity, such as it happens in aircrafts. As for its downsides, is a system that is far more difficult to implement and far more expensive than the typical carburetor. Fuel injection systems can be divided into two main types, they can either be mechanically controlled or electronically. The fuel injection can also be direct, ported or throttle body. In direct fuel injection, the fuel is sprayed directly into the combustion chamber, while in ported fuel injection the fuel is sprayed in the intake manifold [17]. Throttle body fuel injection is a system where instead of having fuel injectors in each cylinder, the fuel is injected into the throttle body, mixing here the air with the fuel which then travels to the cylinders through the intake valve.

- Mechanical Fuel Injection

  Mechanical fuel injection systems concept was originally introduced around 1900 for use in diesel engines. In World War II these systems were vastly used in high-output aircrafts since they provided better fuel economy and had the ability to work under conditions that carburetors couldn't. But these systems also had disadvantages such as the need for precisely machined components, which made them expensive and had very limited adjustments to obtain the optimal amount of fuel into the engine through a variety of different conditions. Since the appearance of the electronic fuel injection system, the mechanical system is going ever more into disuse.

- Electronic Fuel Injection

  Electronic fuel injection systems came to surpass the failures of its precedent. It has several advantages over it such as the fact that it has a higher durability, is more easily controlled, requires less space, allows a smoother function of the engine, cold-start is facilitated among other advantages. These type of systems can also be classified according to their number of injectors, being single-point or multi-point. The single-point fuel injection system is cheaper than the multi-point fuel injection but has about a 10 percent lower power output when compared to the multi-point system [2]. The multipoint system is a more efficient system but is also more expensive and complex. The choice of which to use depends mainly on its application.

  In electronic fuel injection engines, fuel is drawn from the gas tank by a fuel pump. The fuel after being drawn from the gas tank travels in the fuel lines where it will be filtered before being dispersed to the fuel rail [17]. The fuel pressure is controlled by a regulator, which is responsible to maintain the correct pressure in relation to the intake pressure. What determines how much fuel needs to be released into the cylinders is the engine control unit. This unit is fed with information from the several sensors in the engine, and it's with this information that the unit knows how much fuel needs to be added. The injector valves are opened by solenoids, which are electromagnets, when the control unit signals them to. How much time the valve stays open, influences how much fuel is injected.

2.6 Cooling System

In internal combustion engines, only about 25 to 35 percent of the chemical energy in the fuel is turned into mechanical energy [12]. Of all the heat generated in the engine, around 35 percent is removed by the cooling system [12]. It is extremely important to remove heat from the engine to avoid thermal failure, but it is also desirable to operate the engine as hot as possible to maximize its thermal efficiency [16]. This means that the function of a cooling system is to keep the engine from getting too hot and at the same time keep it from getting too cool. If a cylinder head overheats, it may lead to an overheated spark plug which can cause preignition, and preignition itself can further increase the cylinder head temperature to the point of engine failure or total loss of power [12]. According to literature, there are two main characteristics that define an efficient cooling system:

1. It should have the capability to remove about 30 per cent of the heat generated in the combustion chamber while maintaining the ideal temperature for the engine under all its operating conditions [12].

2. When the engine is hot, it should be capable to remove heat at a faster rate and when the engine is cold, for example when starting, it should keep the cooling to a minimum so that the engines working parts can reach their operating temperatures as fast as possible [12].

2.6.1 Types of cooling systems

There are also two types of systems generally used for cooling in internal combustion engines, they can either be air cooled or liquid cooled, an example of each can be seen in figures 2.18 and 2.19.

- Liquid-cooled system

  In these systems, a liquid, is used to circulate through the engine block in a jacket. The heat is transferred to the liquid by the cylinder walls as well from other parts by means of convection and conduction [12]. As the liquid heats up in its course through the jackets, he is then cooled by an air-cooled radiator system.

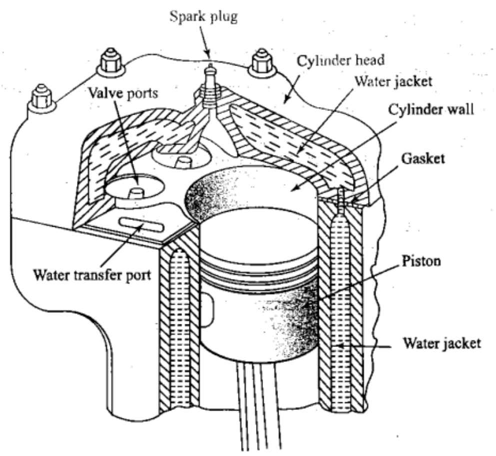

Figure 2.18: Liquid-cooled engine [12]

- Air-cooled system

  Air-cooled engines depend on the flow of air passing through their exterior to cool the engine and preventing it this way from overheating. Engines with this system usually have their exterior covered with fins, which increases the surface area where the airflow passes.

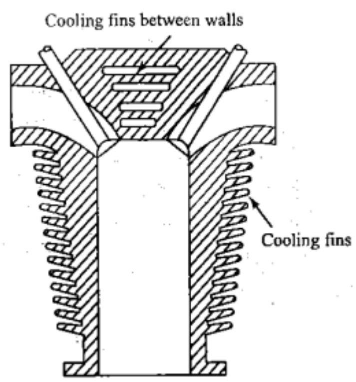

Figure 2.19: Air-cooled engine [12]

Most small engines are air-cooled because this type of system causes a low increase in weight, are simpler and cheaper [16]. When compared to each other, each system has its advantages and disadvantages. Some of the advantages and limitations of each type of cooling system are presented below.

- Advantages of liquid-cooling system:
  - They are more efficient in cooling the engine.
  - Allows for an even cooling of all the engine.
  - The size of the engine does not present a problem in terms of designing the cooling system.
  - An engine with this system produces less noise.
- Limitations of liquid-cooling systems:
  - The cost of this type of system is high.
  - It requires high maintenance.
  - In the eventuality of the cooling system to malfunction, the engine may suffer serious damage.
- Advantages of air-cooling system:
  - There is no danger of coolant leakage.
  - The weight of an air-cooled engine is lesser than of a liquid-cooled one.
  - It does not require external components, like a radiator nor pipes, which makes the system simpler.
- Limitations of air-cooling system:
  - This type of system can only be applied to small and medium size engines.
  - The cooling provided by this system is not uniform.
  - An air-cooled engine works at a higher temperature when compared to a liquid-cooled engine.

As can be seen, the choice of which system to use depends on what is needed for the engine.

2.7 Performance Parameters

Several parameters such as efficiencies, ratios, and others are used to determine an engine performance. According to literature, engine performance is more precisely defined by: the maximum power which is available at each speed within the engines' useful operating range [1]; the range of both speed and power at which the engines operation is satisfactory [1].
Five of the most important efficiencies when we talk about engine performance parameters are: Indicated thermal efficiency (1); Brake thermal efficiency (2); Mechanical efficiency (3); Volumetric efficiency (4); Relative efficiency (5). Other important parameters that are considered basic engine performance parameters are:

- Power and Torque
- Compression ratio
- Crankcase compression ratio
- Swept volume

- Indicated, Brake and Friction Power
- Mean Effective Pressure
- Mean Piston Speed
- Specific Fuel Consumption
- Specific Power Output
- Air-fuel Ratio
- Heating value of the Fuel

2.7.1 Basic Parameters

2.7.1.1 Power and Torque

Power can be defined as the measure of how much work is done per unit of time and is dependent on Torque and velocity. An engine produces power by providing a rotating motion into a shaft, which can apply a certain amount of torque on a load at a certain velocity [18]. Torque is usually measured with a dynamometer and can be defined as a force around a certain point, applied at a radius from that same point, hence torque in engines can be manipulated by using gears.
Torque measures an engine's ability to do work and power measures the rate at which work is done [1]. Power can be calculated with the torque and the rotational speed of the engine with the following formula:

$$P = 2 \times \pi \times \frac{N}{60} \times T \quad (2.1)$$

Where T is torque and N is the rotational velocity in revolutions per minute.

2.7.1.2 Compression Ratio

Compression ratio is the ratio of maximum volume in the cylinder, which happens when the piston is at bottom dead center (BDC), to the minimum volume in the cylinder, which happens when the piston is at top dead center (TDC). This minimum volume is the volume of the combustion chamber. So, compression ratio is given to us by the following formula:

$$C_R = \frac{\text{Maximum cylinder volume}}{\text{Minimum cylinder volume}} = \frac{V_d + V_c}{V_c} \quad (2.2)$$

The higher the compression ratio, the more energy can be extracted from the engine. However certain complications can appear due to high compression ratios in some engines, such as engine knocking if low octane fuel is used. In figure 2.20 a visual representation of the geometry of a cylinder, piston and crankshaft can be seen.

2.7.1.3 Crankcase Compression Ratio

Crankcase compression ratio characterizes how strong the pumping of the fresh charge into the cylinder is. It can be calculated by the following formula:

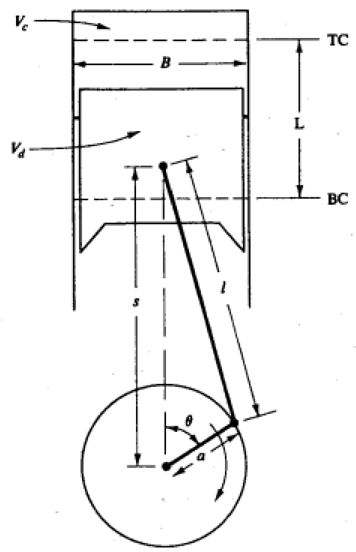

Figure 2.20: Geometry of cylinder, piston, connecting rod, and crankshaft where B=bore, L=stroke, l=connecting road length, a=crank radius, θ=crank angle [1]

$$C_{Rcc} = \frac{V_{cc} + V_d}{V_{cc}} \quad (2.3)$$

Where Vcc is the volume of the crankcase at BDC. Even though the crankcase compression ratio defines how strong the pumping is, its value is greatly influenced by the engine's geometry, such as the size of the bore, the length of the stroke, the size of the connecting rod and flywheel [3].

2.7.1.4 Swept Volume

Swept volume, V_d, is represented in figure 2.20 and is the volume of the cylinder that the piston crown sweeps from TDC to BDC. The swept volume is given by the following formula:

$$V_d = N_C \times \frac{\pi}{4} \times B^2 \times L \quad (2.4)$$

In which B is the bore, L is the stroke and N_C is the number of cylinders.

2.7.1.5 Mean Effective Pressure

The mean effective pressure is a quantity relating to the operation of a reciprocating engine as is a valuable measure of an engine's ability to do work that is independent of engine displacement [1].

$$mep(MPa) = \frac{P(kW) \times n_R}{V_d(dm^3) \times N(rev/s)} \quad (2.5)$$

It is calculated by dividing the work per cycle by the cylinder volume that is displaced per cycle [1]. It is an important parameter for the initial design calculations of an engine. mep can also be defined by the place of its measurement and through its method of calculation. Some important $meps$ regarding the subject of this dissertation are:

- **Indicated mean effective pressure** ($imep$): is basically an average of the pressure that is produced in the combustion chamber during its complete engine cycle.
- **Brake mean effective pressure** ($bmep$): is the mean effective pressure attained from the brake torque.
- **Friction mean effective pressure** ($fmep$): is the mean effective pressure that is lost due to friction.

2.7.1.6 Mean piston speed

The mean piston speed is the average speed of the piston and can be calculated by:

$$mps = 2 \times L \times \frac{N(rev/min)}{60} \quad (2.6)$$

2.7.1.7 Specific fuel consumption

Specific fuel consumption is a parameter used to measure the efficiency of an engine in using the fuel supplied to produce power [1]:

$$\text{sfc}(g/kW \cdot h) = \frac{\dot{m}_f(g/h)}{P(kW)} \quad (2.7)$$

The lower the value the better, this means that less fuel is being consumed. It is calculated by dividing the mass flow of fuel per unit time by the power that is obtained.

2.7.1.8 Specific power output

Specific power output can be defined as the power output per unit piston area, this parameter permits the comparison of different sized engines. It is calculated by dividing the brake power by the piston area.

$$\text{Specific power output } (W/m^2) = \frac{\text{Brake Power}(W)}{\text{Piston Area}(m^2)} \quad (2.8)$$

2.7.1.9 Air-fuel ratio

Air-fuel ratio as the name implies, is the air mass flow rate, $\dot{m}_a$, divided by the fuel mass flow rate, $\dot{m}_f$. It's a crucial parameter in internal combustion engines, for it determines whether the mixture is combustible or not as well as how much unwanted pollutants are produced in the reaction. The best air/fuel ratio for a spark-ignition engine is usually the one that gives the required power output while consuming the lowest amount of fuel possible, compatible with a reliable operation [1].

$$\text{Air/Fuel ratio} = \frac{\dot{m}_a}{\dot{m}_f} \quad (2.9)$$

2.7.1.10 Heating value of the fuel

It is a property of the fuel itself, it's the quantity of heat produced by the combustion of itself. The combustion process produces water vapors and technics exist that permit the recovery of this heat by condensing it [19]. As such, in a fuel with a high heating value, the water in the combustion process is completely condensed and the heat from the water vapors is recovered [19]. Oppositely, a fuel with a low heating value will contain water vapor resulting from its combustion process and the heat from the water vapors is not recovered. In Appendix J are some examples of heating values for the most common fuels.

2.7.2 Efficiencies

To understand these efficiencies, one must first also understand what indicated, brake and friction power are. Indicated power can be defined as the power that is produced inside the cylinder of an engine, not considering losses, and brake power measures the power that is developed at the output of the crankshaft. As indicated power doesn't consider the friction losses, it will always be greater than the brake power by the amount equal to the friction power. Being that friction power is the power that, as its name implies, is loss to friction in the engine parts.

$$\text{Indicated Power} = \frac{imep \times L \times A \times N}{60 \times n_R} \quad [W] \quad (2.10)$$

Where $imep$ is the indicated mean effective pressure in Pa, L is the stroke in m, A is the cross sectional area of the cylinder in m^2, N is the rotational velocity in revolutions per minute and n_R is the number of crank revolutions for each power stroke per cylinder (being 1 for two-stroke cycle engines and 2 for four-strokes).

$$\text{Brake Power} = \frac{bmep \times L \times A \times N}{60 \times n_R} \quad [W] \quad (2.11)$$

Where $bmep$ is the brake mean effective pressure in Pa.

$$\text{Friction Power} = \text{Indicated Power} - \text{Brake Power} \quad [W] \quad (2.12)$$

2.7.2.1 Indicated Thermal Efficiency

Indicated thermal efficiency is calculated through the ratio of Indicated Power with Fuel Energy Rate.

$$\eta_{ith} = \frac{\text{Indicated Power}[W]}{\text{Fuel Energy Rate}[W]} \quad (2.13)$$

This efficiency is useful for with it we can get an idea of the power generated by the engine, it's important to note that this efficiency considers that there are zero losses of heat in the engine. Being fuel energy rate the mass flow rate of fuel multiplied by the heating value of the fuel. It is the actual quantity of energy that is stored in the fuel.

2.7.2.2 Brake Thermal Efficiency

Brake thermal efficiency is calculated through the ratio of brake power with fuel energy rate.

$$\eta_{bth} = \frac{\text{Brake Power}[W]}{\text{Fuel Energy Rate}[W]} \quad (2.14)$$

Brake Thermal efficiency is used to determine how well an engine converts the energy released from the fuel into mechanical energy.

2.7.2.3 Mechanical Efficiency

$$\eta_{m} = \frac{\text{Brake Power}[W]}{\text{Indicated Power}[W]} \quad (2.15)$$

Mechanical efficiency is the ratio of the brake power, which is the useful one, with the indicated power, which is the power produced in the cylinder through the combustion process. Therefore, this efficiency gives us the effectiveness of the engine to convert its input energy into output energy. Mechanical efficiency can also be obtained by dividing Break mean effective pressure by the Indicated mean effective pressure, both terms will be explained later in this chapter.

2.7.2.4 Volumetric Efficiency

$$\eta_{ve} = \frac{m_{as}}{\rho_a \times V_d} \quad (2.16)$$

The volumetric efficiency, according to literature, is the parameter used to measure the effectiveness of an engine's induction process [1]. It is the mass of fresh charge supplied into the cylinder per cycle divided by the theoretical value (inlet air density multiplied by the displaced cylinder volume). Volumetric efficiency, however, is only used in four-stroke cycle engines because this type of engine has a distinct induction process [1]. As for how to measure the induction process effectiveness in two-strokes engines, other parameters are used which will be explained in Chapter 2.7.3.

2.7.2.5 Relative Efficiency

Relative efficiency, also known as efficiency ratio, is the real thermal cycle efficiency divided by the ideal cycle efficiency. In the theoretical cycle the fluids properties are considered constant regardless of temperature, and in reality (actual cycle) both the properties of air and of the burned gases vary with temperature.

$$\eta_{re} = \frac{\text{Actual Cycle Efficiency}}{\text{Ideal Cycle Efficiency}} \quad (2.17)$$

2.7.3 Filling efficiencies of two-stroke engines

Unlike four-strokes engines were presenting the volumetric efficiency as the measure on how effectively the induction process is occurring, two-stroke engines require several more parameters [3].

2.7.3.1 Scavenge Ratio and Delivery Ratio

The scavenge ratio, SR, can be defined as the mass of fresh charge that is supplied during the scavenge period, m_{as}, divided by m_{sref}, which is the necessary mass to fill the entire cylinder volume under constant atmospheric conditions [3]:

$$m_{sref} = \rho_{at} \times (V_d + V_c) \quad (2.18)$$

$$SR = \frac{m_{as}}{m_{sref}} \quad (2.19)$$

Where ρ_{at} is the air density, V_d is the swept volume and V_c is the clearance volume.
The delivery ratio, DR, of an engine is obtained by dividing the mass of fresh charge supplied during the scavenging process by the mass needed to fill the swept volume in the current atmospheric conditions, m_{dref} [3]:

$$m_{dref} = \rho \times V_d \quad (2.20)$$

$$DR = \frac{m_{as}}{m_{dref}} \quad (2.21)$$

2.7.3.2 Scavenging Efficiency

Scavenging efficiency, SE, can be defined as the mass of fresh charge that has been trapped, m_{tas}, divided by the total mass of the charge that has been trapped by the time that the exhaust closes, m_{tr} [3]. The trapped charge is constituted by m_{tas}, the exhaust gas, m_{ex} and by the remaining unburned mixture from the previous cycle, m_{ar} [3]:

$$m_{tr} = m_{tas} + m_{ex} + m_{ar} \quad (2.22)$$

$$SE = \frac{m_{tas}}{m_{tr}} = \frac{m_{tas}}{m_{tas} + m_{ex} + m_{ar}} \quad (2.23)$$

The best scenario possible for the scavenging process is when the scavenging efficiency equals the scavenging ratio.

2.7.3.3 Trapping Efficiency

Trapping efficiency, TE, is the ratio between the mass of fresh charge that has been trapped, m_{tas}, and the mass of fresh charge that has been supplied, m_{as} [3]:

$$TE = \frac{m_{tas}}{m_{as}} \quad (2.24)$$

2.7.3.4 Charging Efficiency

Charging efficiency, CE, is defined as the ratio between the mass of fresh charge air that has been trapped, m_{tas}, by the necessary mass to fill the entire cylinder, m_{sref}:

$$CE = \frac{m_{tas}}{m_{sref}} \quad (2.25)$$

The charging efficiency can also be calculated by multiplying the trapping efficiency by the scavenging ratio.

$$CE = \frac{m_{tas}}{m_{as}} \times \frac{m_{as}}{m_{sref}} = TE \times SR \quad (2.26)$$

Chapter 3

Methodology

In this chapter, will be presented the procedure followed to dimension the cylinder and cylinder head as well as the procedure followed to fabricate the components.

3.1 Conception (3D Modeling)

The 3D modeling of the engine was done in CATIA V5, which is a vastly used design software for the design of 3D CAD products. As the piston, crankcase, crankshaft and connecting rod were all already existing components, they were all measured as accurately as possible with recourse to a caliper, a ruler and a protractor and then replicated into a 3D model in CATIA V5. After the dimensioning of the cylinder and cylinder head, which will be explained later in this chapter, they were also designed in CATIA V5. As it is of extreme importance to adapt the design to the fabrication process, several alterations were made from the initial design to the final one, all these alterations will be explained in the Chapter Practical Case. The design was made with reference to the usual shapes given to small two-stroke engines, and it was made as simple as possible since this was the first time it was attempted to cast an engine cylinder at the university and so, no experience was possessed in the process to fabricate one.

3.2 Fabrication Method

As engine cylinders and cylinder heads are typically cast in one piece, it was initially considered to do a sand casting which is a common foundry process used for small and simple casting projects. But a sand casting has limitations when it comes to making hollow spaces, like the cylinder bore and the ducts for the admission/exhaust/transfers. As such, it was then opted to do an investment casting with a process known as “lost PLA/ABS”. This process requires a 3D print of the object to cast, in PLA, PETG or ABS, which is sunk in plaster and after a 24h wait, so it hardens, is taken into an oven where it will go through a burnout schedule which will melt/burn the plastic. Leaving this way, a negative mold for the pouring of the molten metal. The 3D print before being sunk in plaster needs to be properly gated so that there is a way for the plastic to leave the mold when it goes into the oven and later for the molten metal to be able to enter in the mold. It was chosen to use a casting technique instead of machining a block of metal due to the liberty in the design that the casting permits. For example, as was explained in the previous chapter, the transfer ducts in two-stroke engines should be as close as possible to a sweeping arc. This would be extremely difficult, if not impossible, to achieve if it were decided to machine a block of metal with the resources available.

The metal chosen for both the cylinder and cylinder head was aluminium which was obtained from old pistons and from an old engine block. Aluminium presents itself as an easier material to work with when compared for example to steel, which is another common type of material used in engines. Aluminium offers a better surface finishing when compared to other cast alloys, as well as it has lower melting point than iron or steel. Some properties that also justify the choice of this material for the building of the cylinder and cylinder head are that it has a low density, a high electrical and thermal conductivity and a high resistance to corrosion.

As this lost PLA/ABS process only gives one try per print, some testing with plastic prints was done first in order to find a burnout schedule that worked. The pieces used for testing were made from different materials, some were in ABS and others in PLA. The plaster sand mix was also rehearsed to find a mixture that was vicious enough to enter in all the small spaces inside the mold while presenting the least amount of cracks and defects as possible after the burnout. When a working schedule was obtained, it was decided to try and build the cylinder head first and later the cylinder, this because the cylinder head is smaller and far less complicated to fabricate than the cylinder block. Until obtaining a useable piece, different gating systems were tested, alterations in the design were made to facilitate the foundry process and different technics were used to try and improve the quality of the aluminum. When the pieces were obtained they were properly machined and a cylinder liner was built and inserted in the cylinder.

In figure 3.1 is a flowchart of the process done to obtain the cylinder and cylinder head pieces. In chapter 4 all this procedure will be explained in detail while presenting the attempts made with the results attained.

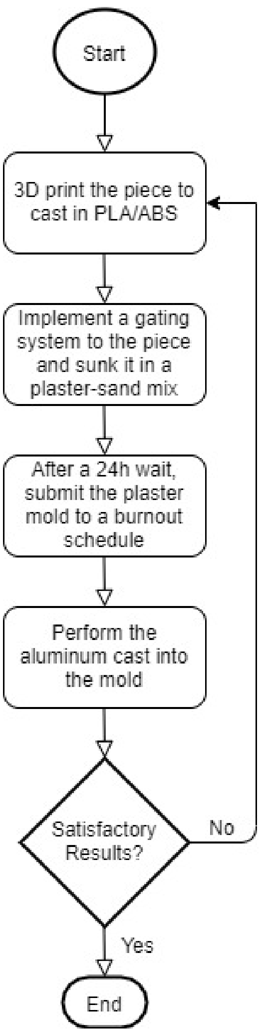

Figure 3.1: Flowchart of the fabrication process

3.3 Cylinder and Cylinder Head dimensioning

Regarding the process to dimension and design the cylinder and cylinder head, little to no information exists in the internet/books on how to properly design and dimension either. Engine design data mostly relies on empiricism, and it may seem obvious, but no engine manufacturer is going to share their secrets for someone else to replicate their work. Several computer programs exist that assist in engine design allowing simulations on what to expect from the inserted design, but none was found that was free or had an affordable price. According to such, *Bhandari, V.B. – Design of Machine Elements* came very handily since it provided a set of formulas that even though aren't the most precise nor the most accurate, do serve as good guidelines for the sizing of the cylinder and cylinder head.

As this is a conceptual project and for the reasons previously mentioned, some assumptions regarding BMEP, mechanical efficiency, the gas pressure inside the cylinder and others had to be made in order to obtain values to work on. In figure 3.2 is a flowchart of how the dimensioning was done. The assumed values will be presented in chapter 4.

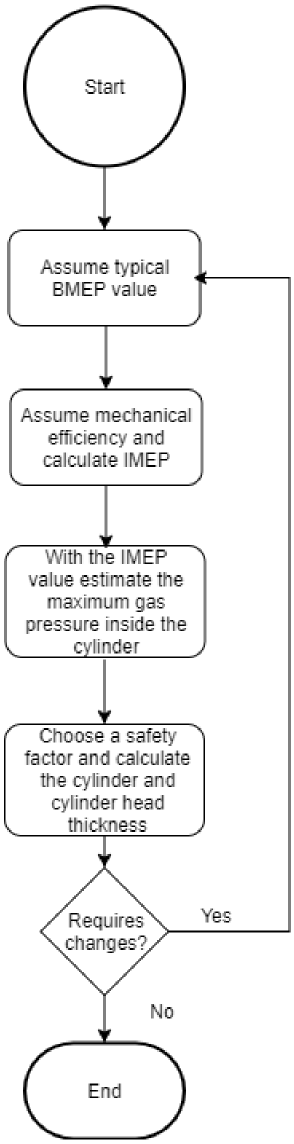

Figure 3.2: Flowchart of the dimensioning process

3.3.1 Ports size and timing

As this was the first time it was attempted to design a two-stroke engine and not much empirical data regarding such was found, it was opted to design the engine to be of the piston-ported type. To design the ports, several factors had to be taken into consideration. The duration of which the ports stay open, their height in the cylinder walls and their area were all factors that needed to be determined so that they could be drawn in the 3D modeling software. So, to start the process of figuring what port size to use as well as the port opening timing, it is of extreme convenience to use typical values used in two-stroke engines. These guidelines are provided by *Gordon Jennings*, in his book *"Tuners Handbook"*. The author provided values for intake, transfer and exhaust port time-areas which can be observed below:

- For piston-controlled intake ports, 0.00014 to 0.00016 $sec - cm^2/cm^3$
- For transfer ports, 0.00008 to 0.00010 $sec - cm^2/cm^3$
- For exhaust ports, 0.00014 to 0.00015 $sec - cm^2/cm^3$

The expression $sec - cm^2/cm^3$ is obtained by dividing the cylinder volume in the corresponding unit, cm^3, into the mean port area, in cm^2, multiplied by the total time which the corresponding port stays open in seconds [6].

There are several different port shapes that can be adopted. For our case, an elliptical shaped layout was used for all the ports, just like the one shown in figure 2.4 c). This layout type with the rounded corners has a better flow coefficient than a simple squared type port with the same area and helps prevent ring snagging [6]. As for the dimensioning of the ports, *Tuners Handbook*, as well as *Gordon P. Blair, Design and Simulation of Two-Stroke Engines*, provided guidelines as to what values to use in the port's width and the radii of its corners. All these values will be presented in chapter Practical Case.

To obtain the values of specific time-area for the designed ports and compare them with the labeled ones, a set of assumptions and a process of trial and error began:

- To work on any time-area problem, it is first needed to convert the engine's timing, in degrees, into time for a given engine speed [6]. For such, we use the following formula:

  $$t = \frac{\theta}{N(rev/min) \times 6} \quad (3.1)$$

  Where θ is the port-open period in degrees and t is the time open duration in seconds. Initially a working speed from which maximum power would be obtained from the designed engine had to be assumed along with a port-duration period. This gave us t.

- Now all that remains to obtain the value of specific time-area and compare it with the labeled ones, is to divide the ports mean area by the cylinder volume and multiply it by t.

This process was repeated several times for each of the ports, alternating the assumed variables such as the engine working speed, the port open-period, and the ports area until our values were inside the range provided by *Tuners Handbook*. In figure 3.3 we can see a flowchart of the process just described.

Once the designed ports specific time-area value was inside the range of the guidelines, all that remained was to figure the exact height at which the port was going to be placed in the cylinder. For such, with recourse to a pencil, a ruler, a compass and a protractor it was drawn a circle that represents the path followed by the crankpin with the angles where the ports begin to open and close marked, something like what can be seen in figure 3.4. From the center of the circle, a vertical line is drawn that represents the cylinder axis and from each and any given point from the circle perimeter, we can draw a line with the exact length of the connecting rod connected to the vertical line and we obtain the height at which the piston is located. One important detail that mustn't be overlooked, is that the time-area values are based on a "mean" port area which is taken when the piston is halfway towards the fully open position [6]. This is best illustrated through an example, so let's take the figure 3.4 as a reference for it is also used in *Tuners Handbook* to illustrate this. In the case of an exhaust port that opens 90 degrees before bottom dead center, when the crank-pin is at 45 degrees from bottom dead center the piston will uncover around 70% of the total port window area in most engines [6]. This is for the exhaust, the intake port when its halfway through its fully open position will uncover about 65% and the transfer ports in halfway through its fully open position will uncover roughly 75% of its window [6]. For the design of the engine ports it was then assumed that the uncovered area of them, when halfway through their fully open position, would fall near these average values provided by *Tuners Handbook*.

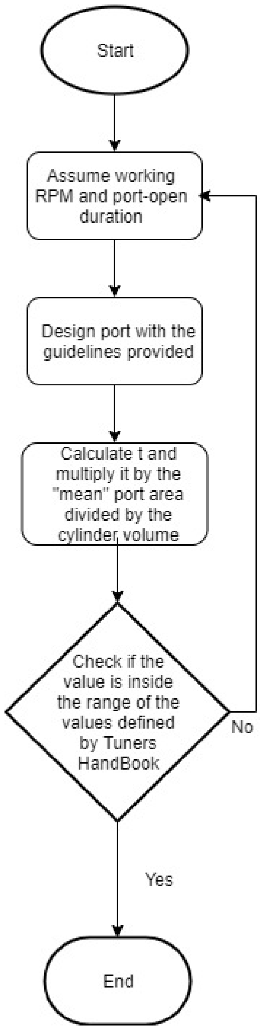

Figure 3.3: Flowchart of the dimensioning of the ports

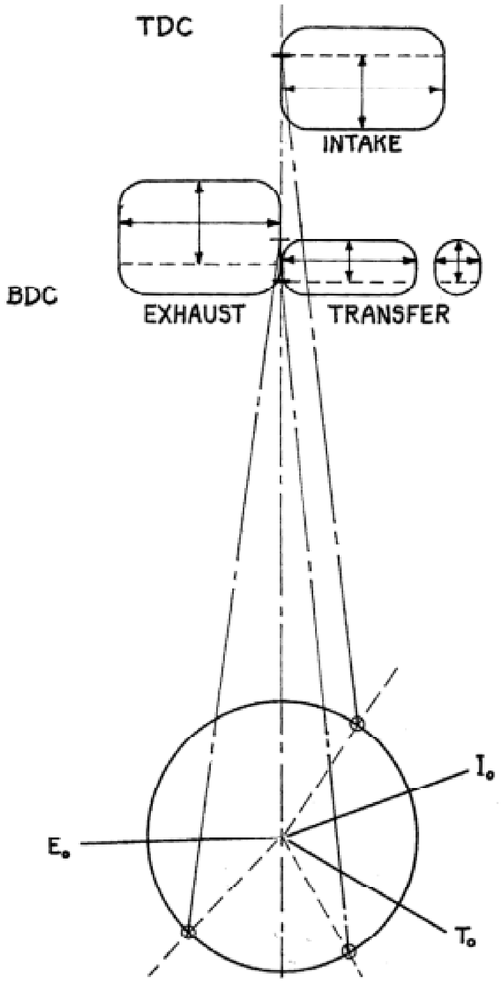

Figure 3.4: Ports height in the cylinder [6]

3.3.2 Combustion Chamber (Squish Band)

To design the combustion chamber firstly it was decided whether it should have a squish band or not. The implementation of a squish band in this design was an ambitious idea, for it complicates the fabrication process. Complicates in the sense that it would be harder to obtain the cylinder head casting with no imperfections due to the level of detail needed. But as the empirical dimensioning process for a simple squish band was very simplified in *Tuners Handbook* and the rewards in engine performance may be considerable, it was a possibility that couldn't be overlooked. The guidelines in this book say to use a squish band with about 50% of the cylinder bore area and regarding the clearance between the squish band and the piston, in engines between 50 to 80 cc, it is recommended to use a clearance from 0.6 to 0.8 mm [9].

The next step was to decide the volume of the combustion chamber as well as its shape and the location of the spark plug. Several different options can be taken and some of the most commonly used were presented in chapter 2. After making these decisions the combustion chamber was designed in the 3D modeling software and below in figure 3.5 is a flowchart of the process just described.

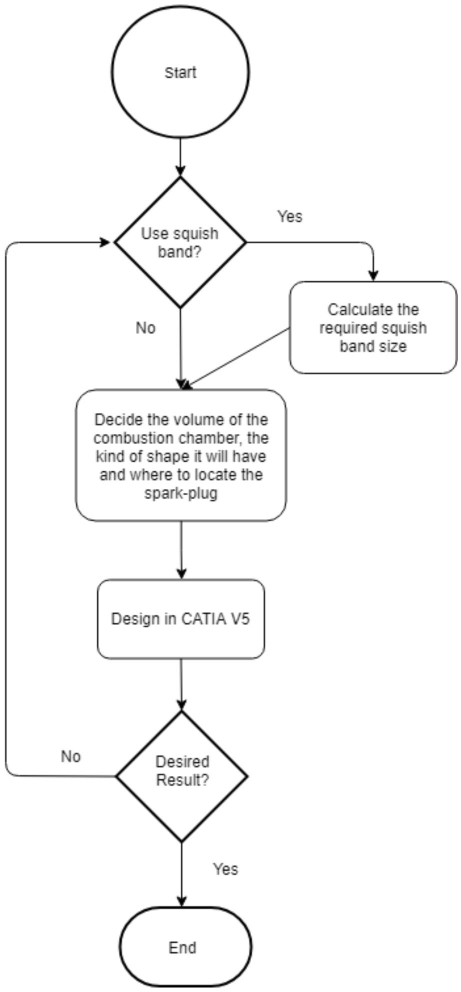

Figure 3.5: Flowchart of the process used to design the combustion chamber

3.4 Cooling system

For the cooling system of the engine, it was opted for the air-cooled alternative. This type of system is the most common choice found in small engines due to its perks of being considerably lighter, far cheaper and simpler than the liquid-cooled alternative. The usage of this type of system also doesn't need any other external components, such as radiators or pipes, which increases simplicity and makes the propulsion system occupy less space.

Given that the designed engine wasn't being made to run for long periods of time, only to perform some tests in short periods of time, the choice had to fall towards simplicity. The air-fins were then designed following simple calculations which permitted to estimate their capacity to dissipate heat and by checking how other similar small two-stroke engines air-fins were designed and dimensioned. The formulas used to assess the designed fins capacity to dissipate heat were found in *Yunus A. Cengel – Introduction to Thermodynamics and Heat Transfer* and these only provide a rough estimate on how much heat the fins will dissipate, this because several assumptions and simplifications had to be made to obtain the heat dissipation values. For instance, it was assumed an equal distribution of the temperature around the engine and assumed that the fins were rectangular and had the same constant convection heat transfer coefficient, h. The results and the formulas used will be presented in the next chapter.

Chapter 4

Practical Case

In this chapter, the results from the dimensioning of the components will be presented, as well as their design. It will also be explained in depth the process of fabrication followed to obtain the components and how they were machined.

4.1 Existing Components

As was previously stated, these existing components were all measured as accurately as possible and then replicated into 3D models which can be seen in Appendix L. These 3D models are simplified versions of the existing components since there was not a necessity to replicate every small detail existent in each piece.

4.1.1 Piston

The piston can be seen in figure 4.1, it is a symmetrical piston made of aluminum with a concave crown with a diameter of 42.5 mm and a diameter below the ring belt of approximately 42.7 mm. This difference in diameters is intentional and is explained by the fact that the temperatures reached by the top of the piston are different from the ones reached by the skirt, which causes different levels of expansion. This piston also has two transfer cuts, each 180º from the other, to allow the entrance for the transfer ducts in the crankcase to be unblocked when the piston is at BDC.

Two-stroke pistons usually have one or two rings, instead of three like four-stroke pistons. This because two-strokes do not require the third ring (Oil ring) [20], since the lubrification of the engine is provided by mixing the oil with the fuel and there is no oil reservoir. Both one ring or two ring configurations have their advantages and disadvantages but, in our case, the piston has two rings which provides a better engine performance durability, due to the capability of the second ring to seal the compression in the eventuality that the first ring seal gets compromised. It also permits a better heat transfer due to the existence of two rings instead of one.

Figure 4.1: Piston used

4.1.2 Connecting Rod and Crankshaft

The connecting rod's basic function is to transfer the forces from the piston pin to the crankpin [21]. In figure 4.2 we can see the used connecting rod, it consists of a simple design with a small end and a big end, being connected to the piston pin and crankpin respectively. It is made of steel and its length from the small end to the big end is 64 mm. As it was previously stated in this thesis, the crankshaft is responsible, alongside with the connecting rod, to convert the reciprocating motion of the piston into a rotary one [21]. The crankshaft is composed by the crankpin, crank webs and shaft and can be seen in figure 4.2 as well as the connecting rod. As the crankpin is located 16 mm from the center of its rotation axis, this represents a stroke of 32 mm.

Figure 4.2: Crankshaft and Connecting rod

4.1.3 Crankcase

Regarding the crankcase, it is a simple design with a circular shape made of aluminum. It has a volume of approximately 100 cubic centimeters when the piston is at bottom dead center and a crankcase compression ratio of around 1.45. It also possesses two entrances for transfer ports, located 180º from each other and each on the side where the piston has the air transfer cuts. The crankcase and these entrance shapes can be seen in figure 4.3. The entrances sidelines measure approximately 7.5 mm and the top part has 24 mm from side to side with a small curvature. In figure 4.4 we can see the crankcase assembled with the crankshaft as well as the connecting rod and piston.

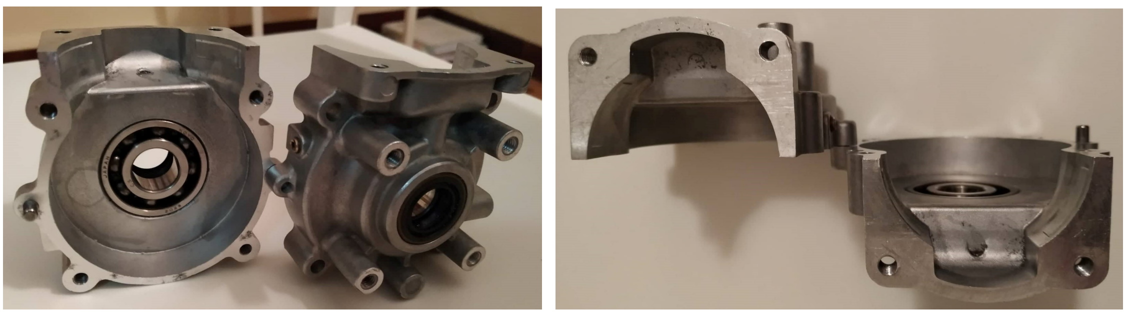

Figure 4.3: Crankcase

Figure 4.4: Existing components assembled

4.2 Results from the dimensioning

To start the dimensioning process, it was first necessary some values to work on, therefore some assumptions had to be made. The $bmep$ assumed was of 550 kPa, which is an empirical value expected for small two-stroke engines used in model airplanes [1],[16]. From this value and assuming a mechanical efficiency of 70% it was possible to calculate the $imep$. With the $imep$ value and following the guidelines provided in *V.B. Bhandari - Design of machine elements*, it was assumed that the maximum gas pressure inside the cylinder was 10 times the $imep$ value. As the material chosen for the fabrication of the engine cylinder and cylinder head was aluminum, it was assumed an yield strength of 250 MPa and a safety factor of 3. All these values were necessary for the process that will be explained below.

4.2.1 Cylinder and Cylinder-head

To find the necessary cylinder length, the distance between the top part of the crankcase up to the piston crown when the piston is at BDC was measured and added the stroke value. This resulted in a cylinder length of 54 mm. As for the cylinder base, it was made to fit the crankcase shape and the cylinder head base was made accordingly with the same design.

The formulas used to dimension the cylinder and cylinder head thickness were provided by *V.B. Bhandari - Design of machine elements* and are as follows:

$$th_c = \frac{P_{max} \times B}{2\sigma_c} + C \quad (4.1)$$

Where th_c is the cylinder wall thickness in mm, P_{max} is the maximum gas pressure inside the cylinder in MPa, B is the inner diameter of the cylinder or cylinder bore in mm, σ_c is the allowable tensile stress for the cylinder material in MPa and C is the reboring allowance in mm. The reboring allowance value used was of 1 mm as it is recommended for engines of this size [21].

This formula suggested a thickness of approximately 3 mm and to verify this value, another formula is presented in the same book which is derived from empirical relationships:

$$th_{c,empirical} = 0.045 \times B + 1.6(mm) \quad (4.2)$$

The value obtained from this empirical formula is approximately 3.5 mm, as can be seen, it is extremely close to the one obtained from equation 4.1.
To calculate the cylinder head thickness the following formula was used:

$$th_{ch} = B \times \sqrt{\frac{K \times P_{max}}{\sigma_c}} \quad (4.3)$$

In this equation, th_{ch} represents the thickness of the cylinder head in mm and K is a constant with a value of 0.162. From this formula, it was suggested a cylinder head thickness of approximately 5.2 mm.

These formulas aren't the most precise nor do they take into consideration several factors, therefore a higher value for both thicknesses was used. For the cylinder wall thickness, the minimum value is 7 mm, except in the location of the transfer ducts, and for the combustion chamber where the forces will be greater, a minimum thickness of 7 mm.

4.2.2 Combustion Chamber (Squish Band)

The combustion chamber was designed following the flowchart in figure 3.5. Following this flowchart, it was opted for the implementation of a squish band, which for this engine means a squish band with an area of approximately 7.1 cm^2. Next in line in the flowchart, was to decide the volume of the combustion chamber. Since the piston used has a diameter of 42.5 mm and a stroke of 32 mm, the displacement volume is approximately 45.4 cm^3. As such, it was aimed for a total engine capacity of roughly 50 cc, designing this way a combustion chamber with a volume of approximately 6 cm^3. This represents an engine with a total engine volume of around 51 cm^3 with a compression ratio of approximately 8.6. But as this is a two-stroke engine that has ports in the cylinder walls, a more realistic value for the compression ratio is obtained by only considering the displacement value to be measured from the point where both the transfers and exhaust port are closed. Under these conditions, the compression ratio is approximately 5.9.

Regarding the combustion chamber design, several different designs can be used but it was opted for a "spherical" design with a centrally located spark plug and with a squish band. Similar to the one seen in figure 2.9, mainly for permitting a shorter flame travel distance and for its simplicity.

4.2.3 Ports

The ports were dimensioned following the process described in the flowchart in figure 3.3, then as described in the text below that chart, the height of the ports in the cylinder walls was found and later designed into the cylinder in the 3D modeling software. The working rotational velocity assumed, from which maximum power would be attained, was of 6000 revolutions per minute, since this is a modest value that most likely would prevail true for this engine.

The intake port was designed with a 2.75 cm width and with its corner radius with 0.6 cm. Several tries were made, alternating the port duration, the width and the corner radius before arriving at these values. To dimension the port, the guidelines provided in *Gordon P. Blair - Design and Simulation of Two-Stroke Engines*, came very handily and advised using a width of around 65% of the cylinder bore for the intake port. The port that was conceived has a total port area of 3 cm^2 and it has a port open-duration of 140º. With these values, the specific time-area value for the intake is 0.000147 $sec - cm^2/cm^3$ which as can be observed in chapter 3.3.1, falls in the middle of the guidelines provided by the *Tuners Handbook*.

The exhaust port for high-performance engines usually has a width of 62%-70% of the cylinder bore [6], so for our case, it was dimensioned using a width of around 60% of the cylinder bore. This is a safer course of action because if found necessary, the port can be enlarged but if the port is too wide it can cause the destruction of the piston rings. The port has a total width of 2.45 cm and a corner radius of 0.55 cm. This results in a total area of approximately 2.44 cm^2 and the open-duration period for this port is 160º. Resulting in a specific time-area of 0.0001484 $sec - cm^2/cm^3$ which falls inside the range of the values provided by *Tuners Handbook*.

Regarding the transfers, there are two, since the crankcase used has two entrances for transfer ports. This means our engine will be of the Schnuerle Porting type. Both transfer ducts were designed to make the flow enter angled towards the intake side and slightly tilted upward. The entrance area for the duct has the same size as the entrance presented in the already existing crankcase. Both ducts are symmetrical and taper down from the entrance to the ports themselves. The ports have a total width of 1.55 cm and a corner radius of 0.35 cm. Combined, they present an area of approximately 1.96 cm^2 which with an open-duration period of 120º, gives us a specific time-area value of 0.000953 $sec - cm^2/cm^3$. This value falls in the middle of the guidelines provided by *Tuners Handbook*.

With the port-duration periods defined, it is now possible to know the engines blowdown and draw its timing diagram, the later can be seen in Appendix C figure C.1. Blowdown can be defined as the amount of time, most commonly in degrees, between the openings of the exhaust and transfer ports and for the designed engine has a value of 20º, which falls accordingly to the recommended value for the assumed working velocity [22]. To arrive at this value, it is only needed to subtract the degrees at which the exhaust port opens (80º before TDC) to the degrees at which the transfer ports open (60º before TDC).

Literature recommends a minimum distance between the exhaust port and the transfer port. The minimum value recommended to engines of 125 cc and up is 8.9 mm. If there is a failure to maintain at least this separation it will result in a partial charge short-circuiting [6]. Even if this value only holds true for 125 cc engines and up, it will serve as a guideline in designing the distance between the transfer and exhaust of our engine. As the designed engine has 50 cc, and consequently a much smaller bore size than a 125 cc, it was decided to try and a implement a distance of approximately 8 mm between the transfer ports and the exhaust port to reduce the chances of partial short-circuits happening. In figure 4.5 we can see a visual representation of this.

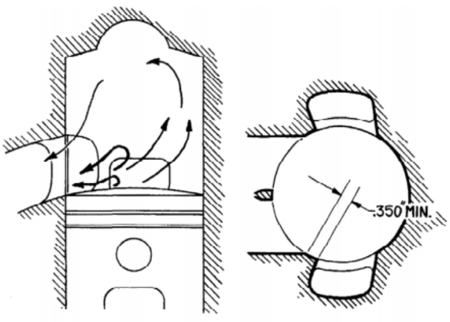

Figure 4.5: Minimum distance recommended between the exhaust and transfer ports [6]

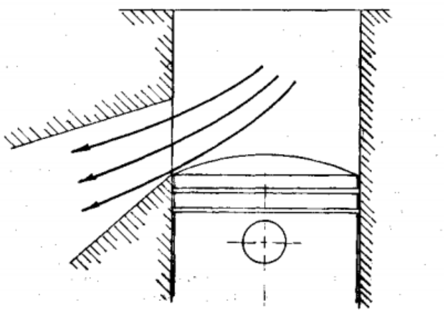

Figure 4.6: Visual representation of how the exhaust flange should be designed [9]

It is also recommended for the exhaust flange to be lower than the piston at BDC [9], as can be seen in figure 4.6. In the eventuality of the port not being designed in this way, it can cause high speed gas flow disruption. As such the engine exhaust flange was designed holding this into account.

4.2.4 Air-fins

Firstly, to determine the heat dissipation from the engine it was necessary to calculate the area of the cylinder surface that had no fins, A_{unfin} in m^2, and the area that each fin had, A_{fin} in m^2. It's important to point out that these calculations were made for only one surface of the cylinder block, and later multiplied by the remaining, nearly symmetrical, number of surfaces. So, to calculate the heat dissipation rate from the fins, the following formulas were used:

$$\dot{Q}_{total} = \dot{Q}_{unfin} + \dot{Q}_{fins} \quad [W] \quad (4.4)$$

$$\dot{Q}_{unfin} = h \times A_{unfin} \times (T_b - T_\infty) \quad [W] \quad (4.5)$$

$$\dot{Q}_{fins} = \eta_{fin} \times h \times A_{fin} \times (T_b - T_\infty) \times N_{fin} \quad [W] \quad (4.6)$$

Where T_b is the engine's temperature, T_∞ is the temperature of the air that is flowing through the fins, N_{fin} is the number of fins and η_{fin} is the fins efficiency.

The value assumed for the convective heat transfer coefficient, h, was of 200 $W/m^2 * C$, which is inside the range of values provided for typically forced convection of gases [23]. The fins efficiency value was obtained from the plot presented in Appendix K and has a value of approximately 85 %. With these values and with the assumptions made, the calculations from these formulas indicate that one cylinder wall would be able to dissipate approximately 2000 Watts when T_b = 250°C and T_∞ = 20°C. Assuming that all the cylinder walls dissipate the same heat through their fins, plus the finned surface of the cylinder head, the total heat dissipation rate from the engine at the presented temperatures, would be above 8000 W. This value is enough to assure that even if the engine eventually got to the temperature of 250°C, the air fins would be dissipating enough heat to not only prevent it from going any higher but also to cool it.

4.3 Draw of the components - *Murphy 1.0*

With the values obtained from the dimensioning process, the 3D modeling of the components was done in CATIA V5 and below in figures 4.7, 4.8, 4.9, 4.10, 4.11, 4.12, 4.13 and 4.14 we can see the idealized design for the cylinder and in figure 4.15, 4.16, 4.17, 4.18 and 4.19, the idealized design for the cylinder head. As will be explained later in chapter 4.4, the designs had to be altered due to the fabrication process, in Appendix B are the 2D draws of the idealized design for both pieces as well as the final designs with the alterations needed for the fabrication process.

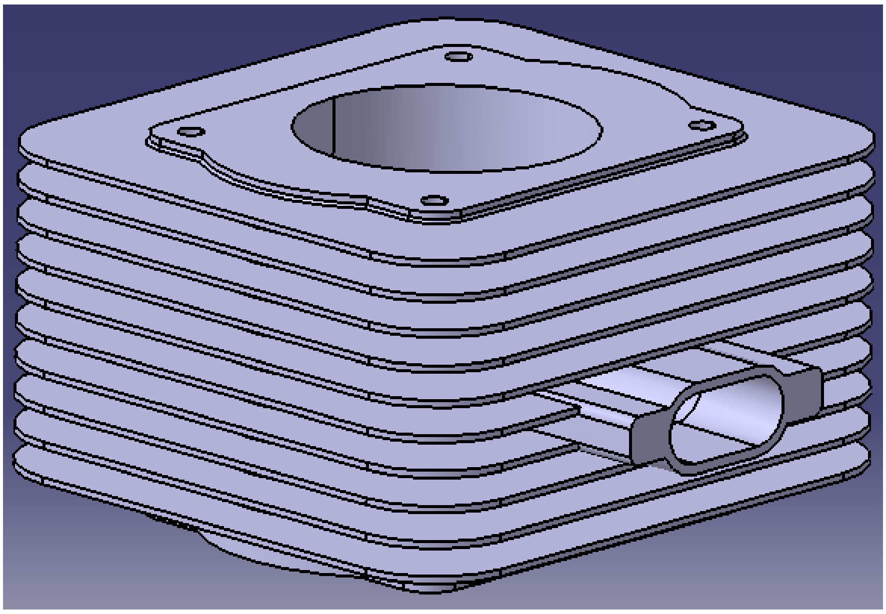

Figure 4.7: Isometric view of the cylinder

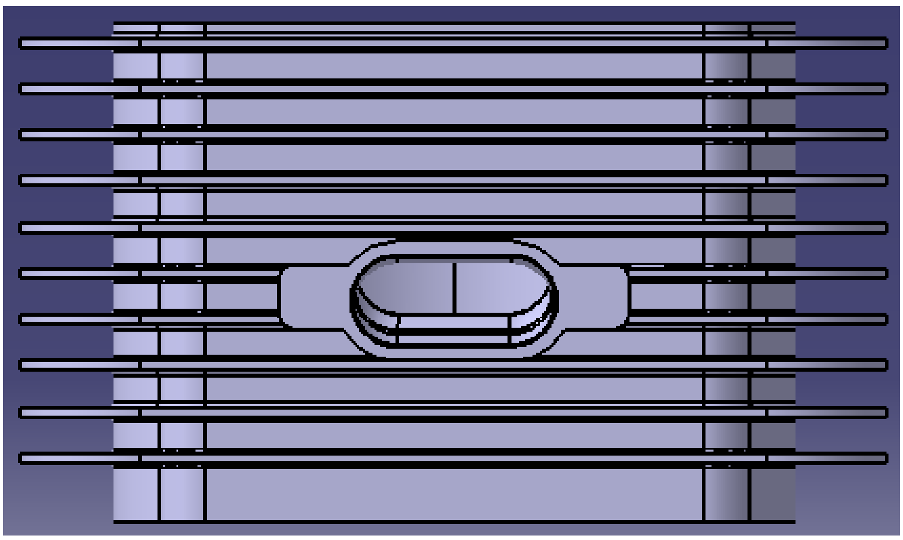

Figure 4.8: Exhaust side view

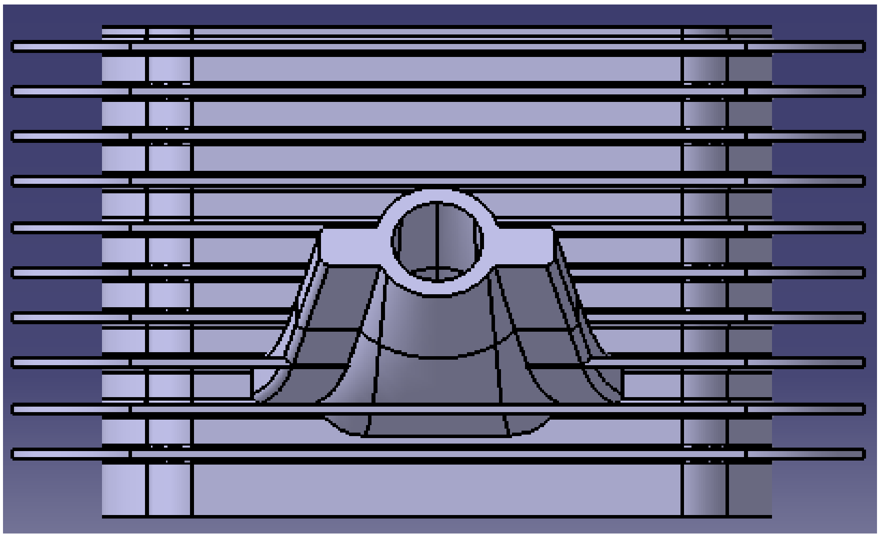

Figure 4.9: Intake side view

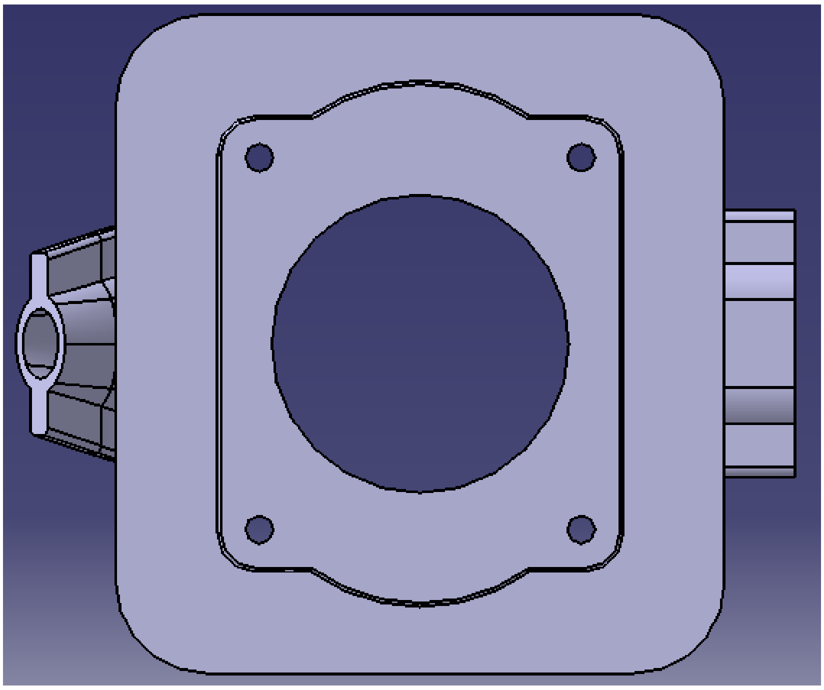

Figure 4.10: Top view

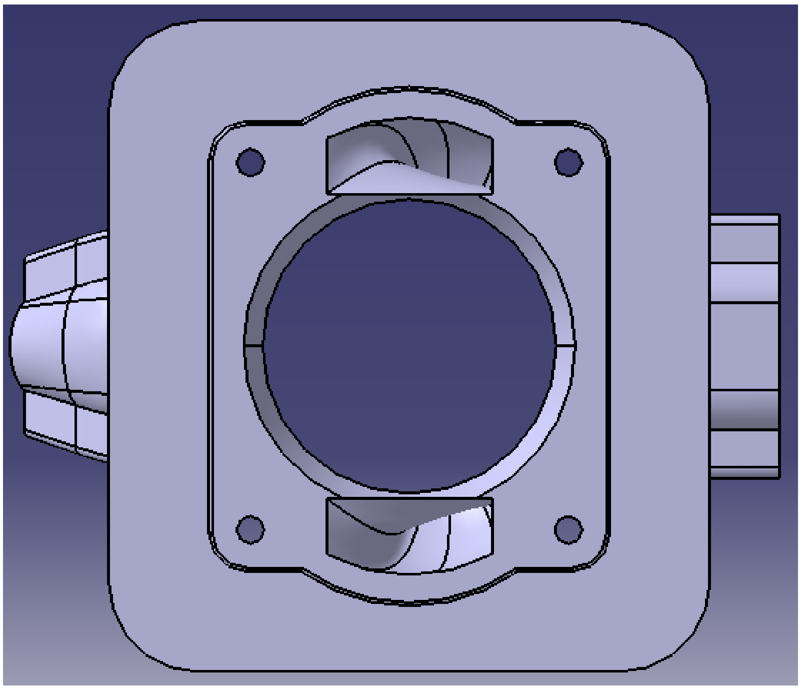

Figure 4.11: Bottom view

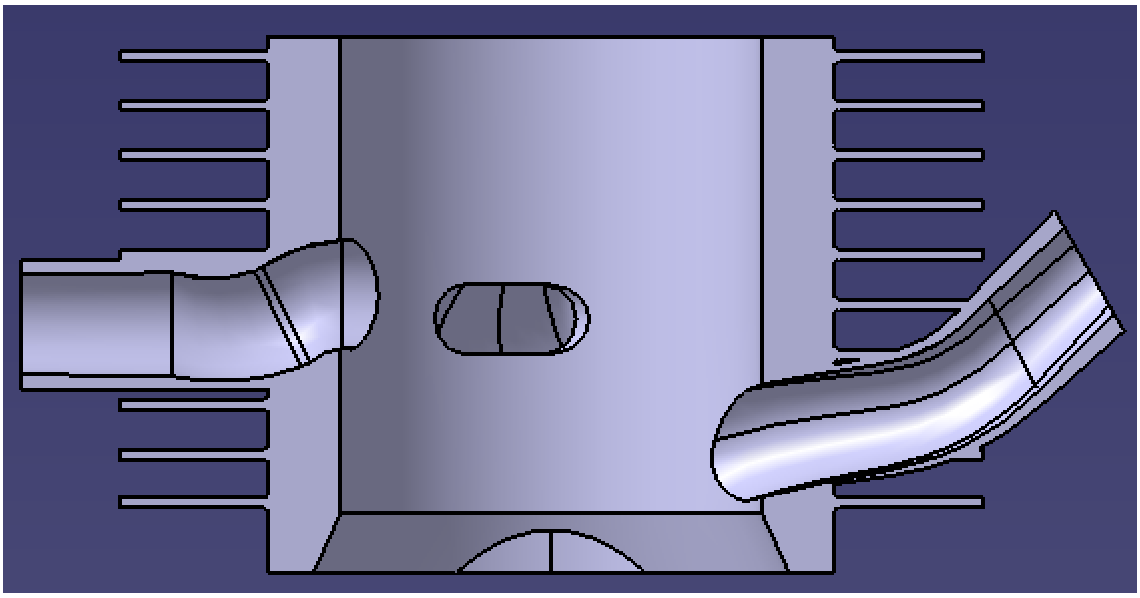

Figure 4.12: Transfer port inside view

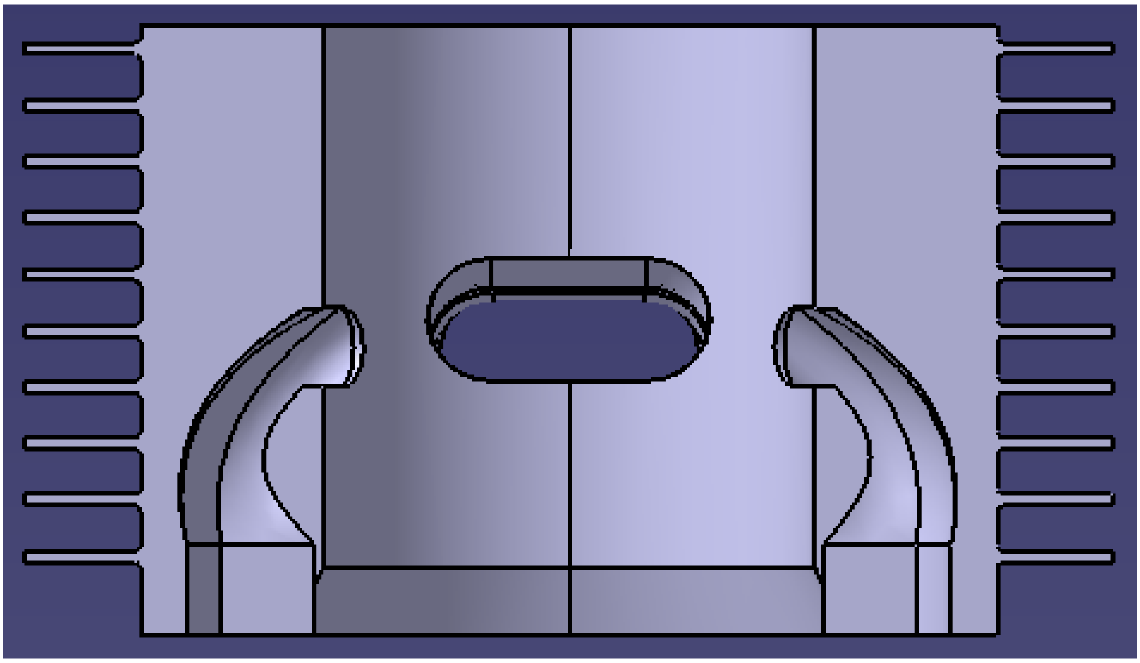

Figure 4.13: Exhaust port inside view

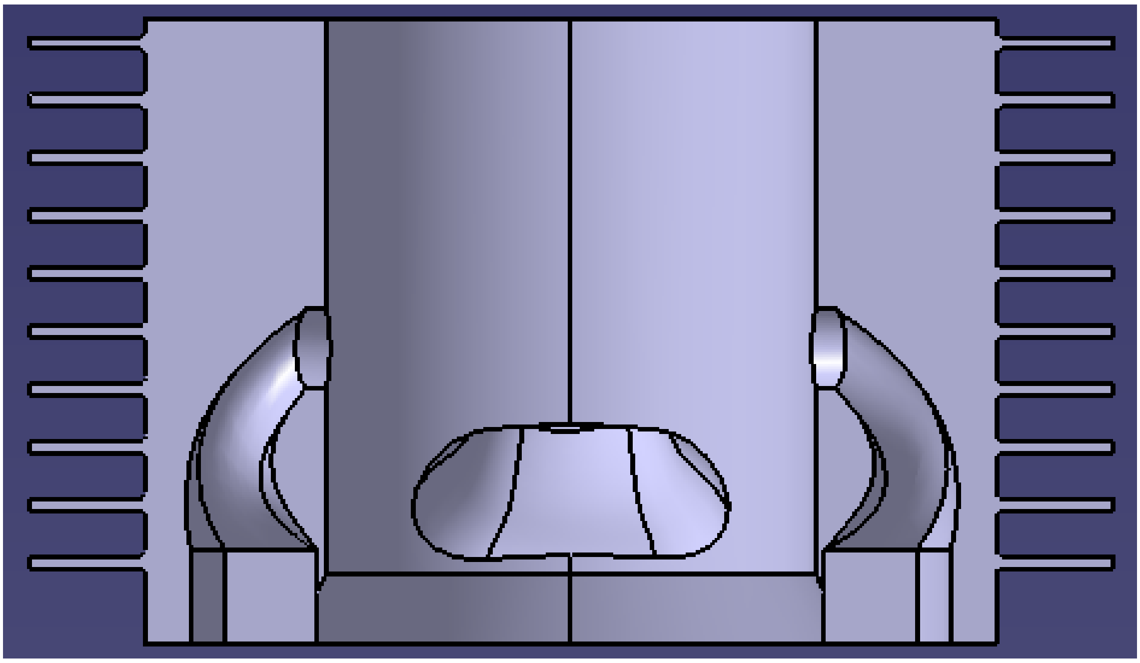

Figure 4.14: Intake port inside view

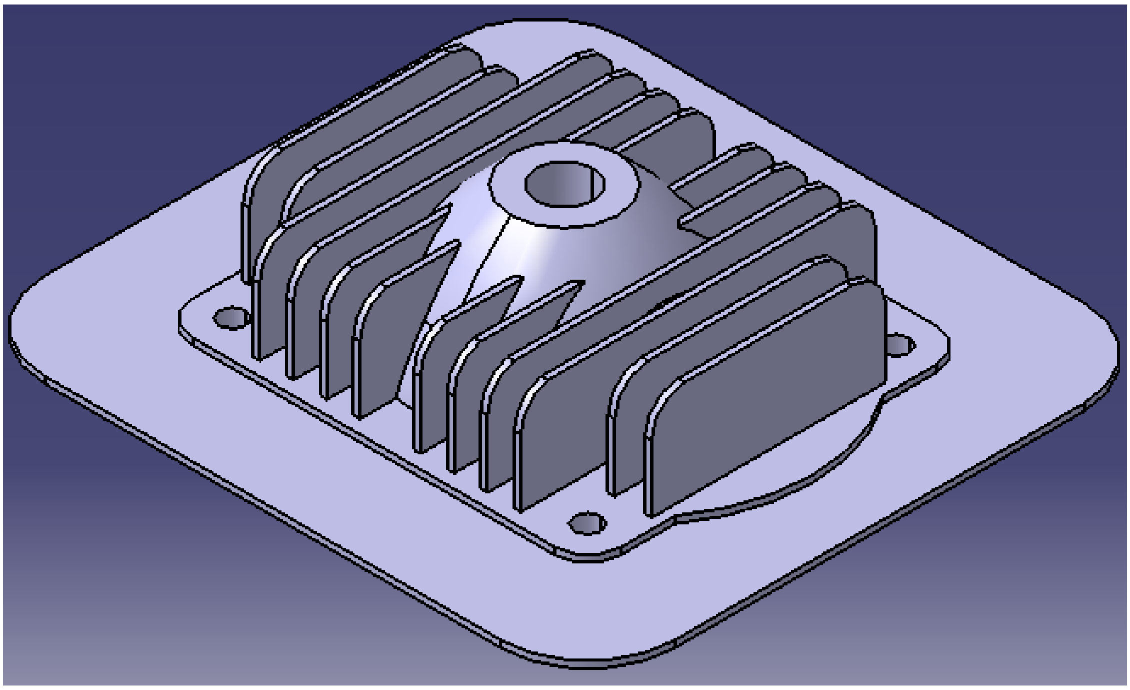

Figure 4.15: Isometric view of the cylinder head

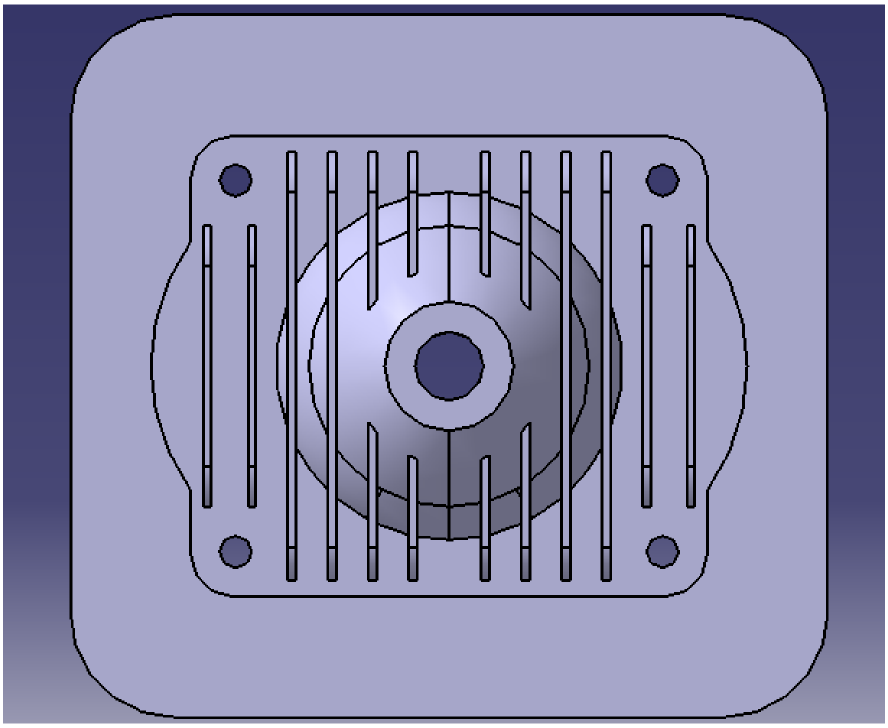

Figure 4.16: Top view cylinder head

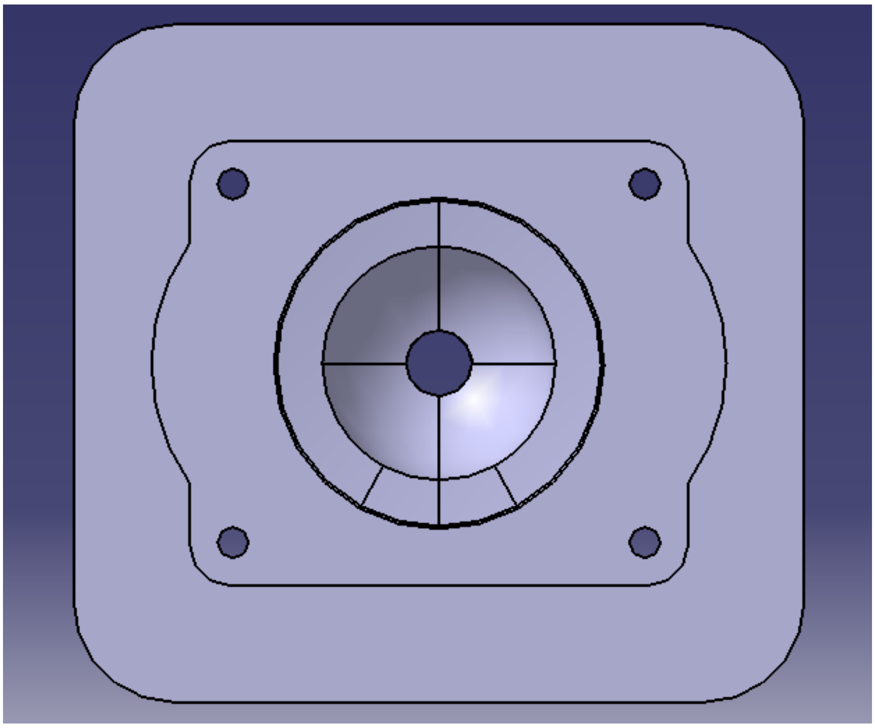

Figure 4.17: Bottom view cylinder head

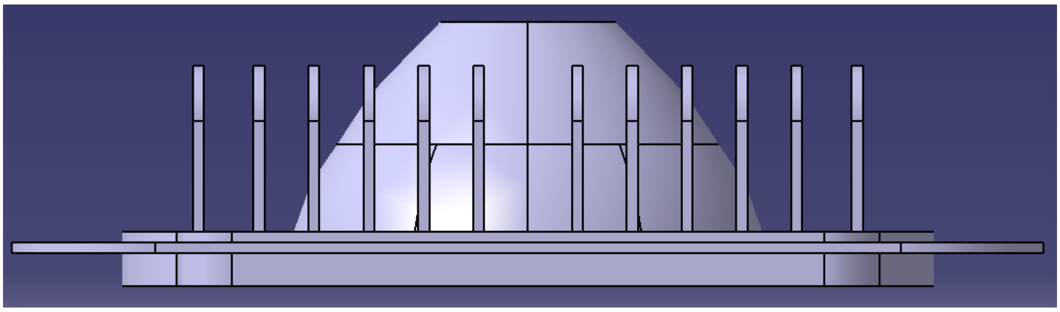

Figure 4.18: Front view cylinder head

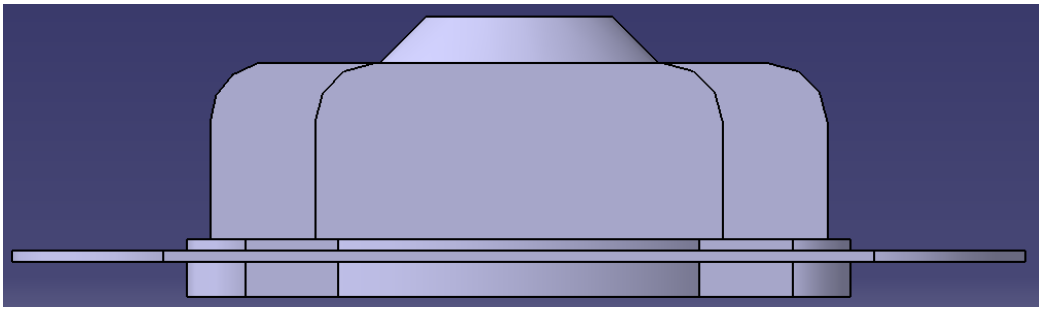

Figure 4.19: Side view cylinder head

4.4 Fabrication of the components

4.4.1 Furnace

To begin the fabrication of our components, first was built a small furnace with refractory cement and sand inside a small metal bucket with a side hole for the torch to be inserted. It was also built a small cover for the furnace so that it would be able to conserve more heat. The cement-sand ratio used for the furnace was 3-4 respectively, which provided good results for the furnace itself since it did not crack through the entire process, only showed signs of deterioration due to the multiple uses. The cover for the furnace was built using a ratio of 4-4 which with some use made the part crack and lose its use. To replace it, a plaster-sand cover was made instead, this because the cement-sand mix needs to be moist cured, which involves watering it during a 7 day period minimum before it's fully cured. The plaster mix offered a better, quicker solution because instead of needing 7 days to cure, it is ready to use within a day. The plaster-sand-water ratio used for the cover was of 10-10-7 respectively. Below in figure 4.20, we can see the furnace with the original cover and in figure 4.21 with the new plaster cover.

Figure 4.20: Furnace built with the original cover

Figure 4.21: Furnace with the plaster cover

While building the furnace, it was also started the construction of a propane burner to use in the furnace. Initially, the burner didn't produce a flame hot enough with only the propane gas, it needed pressurized air boosting the flame. As such, an adapter was added so that the pressurized air would mix with the gas before exiting the torch. In figure 4.22 we can see the propane burner that was built and used for all the foundries.

The crucible for the furnace was made by welding a steel pipe to a base of steel with two side handles also welded to the pipe, as can be seen in figure 4.23. This simple arrangement was enough since the furnace would only be heated up to about 900ºC, which is higher than the melting point of the aluminum and far below the melting point of steel.

Figure 4.22: Propane burner built

Figure 4.23: Crucible

4.4.2 Detailed explanation of the fabrication process

With the furnace operational to melt the aluminum, it began the process to test plaster-sand mixes as well as burnout schedules to find a mixture and a schedule that provided a useable mold. The process to find an optimal ratio of plaster-sand-water, a working schedule and a gating system were all done simultaneously in each attempt. This because every attempt needs a 3D piece properly gated with a riser and a feeder, which is sunk in a plaster-sand-water mix that needs to be fluid enough to enter all the small cavities of the 3D piece but can't be too watery, otherwise it will cause the mold to have extensive cracks or even break during the burnout. So all these factors interact with each other simultaneously and were tested and perfected through each attempt.

Through several attempts and a process of trial and error, the plaster-sand-water ratio found to provide satisfying results was **2-1-1.5** respectively. The plaster used for the molds was "Fine Stucco Plaster" from the brand *Sival*, one attempt was made using "Finishing Plaster" and resulted in a cracked open mold after the burnout process as will be seen later in this chapter. It was found that the plaster-sand-water mix has to be done in a certain order to get a fluid mix. First, the quantity of water to use has to be placed in a recipient, like a large bucket, where the mixture is going to be made, then the plaster is added slowly and it should be let to sit for 30seconds-60seconds before mixing it thoroughly with the water. Afterward, the sand has to be added slowly while stirring it until it is fully mixed. This process has to be done very quickly because there is only about 5 to 10 minutes before the mix starts to harden. The sand also plays an important role, different sands will provide different results. It was found that the thinner the sand was, the more fluid the plaster-sand mix was before hardening. As for the gating system, several variations were tried and what was found to provide better results was to add an excess of material, to remove after the cast, at the bottom part of the casting and to make the riser as big as possible. The excess of material at the bottom helps to improve the quality of the piece because the impurities left inside the mold, such as small grains of sand or bits of plaster, will accumulate at the bottom part of the piece when the aluminum is poured. As for making the riser as big as possible, it's to try and guarantee that the riser solidifies after the casting, if this doesn't happen, shrinkage and porosity can occur. All variations were made using styrofoam to create the riser/feeder as well as any excess dimmed necessary. The design for both pieces was also adjusted to the foundry process, as will be shown throughout chapter 4.4.3, by increasing the thickness of the fins, the length of the cylinder as well as other characteristics that had to be adjusted. In every single foundry process initiated, it was tried to improve the quality of the aluminum as much as possible. What was found to be extremely helpful in this manner was the use of sodium carbonate. Attaching a small quantity of sodium carbonate to a steel bar and holding it down the crucible when the aluminum was already molten helped to degas it, which reduced its porosity significantly. Regarding the burnout schedule, it was done some basic research online as for schedules used in this "lost PLA/ABS" process and from there some alterations were made to adapt it to our case. Initially, it was thought that the 3D model for the cylinder and cylinder head could only be done in ABS, so the initial schedule was done considering this the material to melt. This schedule also worked for PLA as well since PLA has a lower melting and combustion point than ABS. Later throughout the process, the opportunity to use PLA pieces for the cylinder and cylinder head instead of ABS surged and this meant that the burnout schedule could be altered to a lower ceiling temperature, which would help prevent possible cracks in the mold. Both burnout schedules can be seen in Appendix A, figures A.1 and A.2.

Different recipients were also tried to hold the plaster with the mold in it until it dried. The size of the recipient was conditioned by the size of the oven where it would go through the burnout, and besides this, the recipient also had to be easily removed after the plaster dried up so the mold wouldn't end up damaged while doing so. The first recipient attempt was done using two A4 sheets of acetate glued together in a cylindrical shape with a bottom base made of styrofoam. As soon as the plaster dried up, the acetate sheets were easily removed but the styrofoam wouldn't come off as easily as one might expect. But the greatest downside of this recipient was its stability, as these were only sheets glued to one another, they didn't provide a very stable recipient. So, it was thought to use plastic water carboys of 7L, these are easily obtainable, easily removed from the plaster once it dries up and they fit with a significant tolerance inside the oven used. This type of recipient was used in the majority of the attempts and the cylinder head itself was built using this recipient. Its major downside was the fact that it had little width, which means that the riser/feeder would be near the edge of the mold. This can cause the mold to crack and open up during an aluminum pour. For the cylinder head, the plastic carboy was a reliable choice but for the cylinder itself, as it was a bigger piece, the plastic carboy was right near the edge of acceptable, so after a failed attempt it was decided to try and improve the recipient for the mold. The solution found was to use a small plastic bucket with the bottom part removed and using the cover as its base. This plastic bucket almost had the maximum dimensions allowed by the oven entrance, which means that with this recipient the mold would be as big as possible giving more margin for centering the piece, keeping the riser/feeder away from the edges of the mold and making the riser bulky.

In chapter 4.4.3, the relevant attempts made will be presented, in chronological order, with the respective results and with the conclusions drawn from each one until the end results were attained.

4.4.3 Foundry attempts

The first pieces that were subjected to the process which was previously described can be seen below in figure 4.24 and 4.25, the two small identical pieces were made of PLA and the bigger one was made of ABS. These were the first test pieces to what was a long process of trial and error to find a plaster-sand-water ratio, a burnout schedule and a gating system that worked.

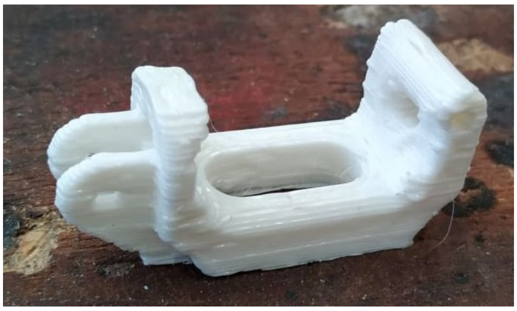

Figure 4.24: Test pieces in PLA

Figure 4.25: Test piece in ABS

It was only after obtaining these results that it was known with some level of confidence that this was a process that with some refinement, could be used to obtain more complicated pieces such as the cylinder and cylinder head. The burnout schedule used was the one previously mentioned above where ABS is the material considered to melt. Below in figure 4.26 are the pictures of the molds of the three pieces after going through the burnout schedule and in figure 4.27 the results from the foundry process.

Figure 4.26: Molds of the test pieces

Figure 4.27: Results from the test pieces

The pieces didn't come out exactly has the 3D models that were used for the mold, but for the first attempt whose main goal was to see if the process defined could work at all, it came very close to. The parts didn't form exactly like the models due to the fact that the plaster-sand-water mixture was not fluid enough to enter the small cavities of the pieces. The reason why the mixture was not fluid enough was because of the proportions of plaster-sand-water experimented at that time and due to the order used to perform the mix. The plaster and sand were mixed together and only then the water was added, this resulted in a mixture full of agglomerates which were very hard to get rid of in the small amount of time it took for the mixture to start hardening.

Considering everything from these results, it was decided to pass on and try to build the cylinder head because it was easier to print and most likely easier to cast than the cylinder. A new ratio for the plaster mix was tried to make it more fluid while still maintaining the same mixture procedure. The burnout schedule used was the same from the previous attempt, even though this 3D print of the cylinder head was made of PLA, because it proved to work while providing satisfying results for the PLA pieces previously tested. Also, a simple gating system with the same ideology as the one used in the small ABS piece was tested, in figure 4.28 we can see the 3D print used, in figure 4.29 the plaster mold, which was done in a different recipient, before going into the oven and in figure 4.30 the results obtained after the foundry.

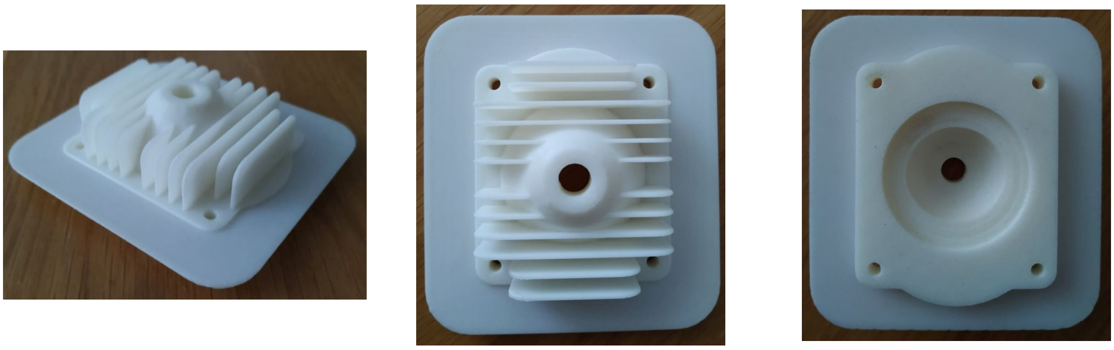

Figure 4.28: 3D print of the first cylinder head attempt

Figure 4.29: Mold of the first cylinder head attempt

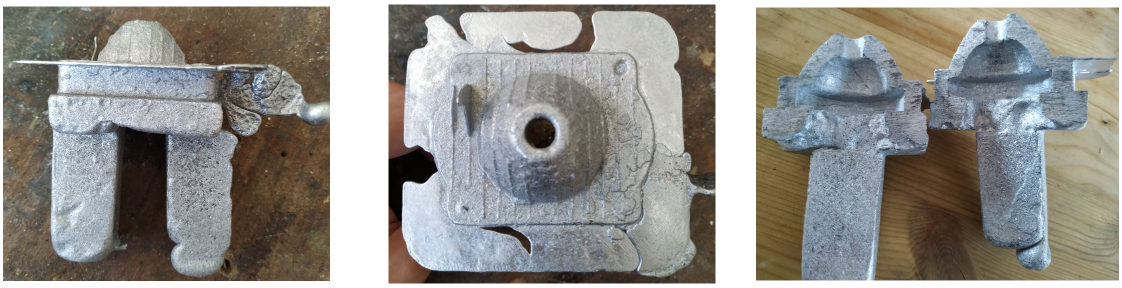

Figure 4.30: Results from the first cylinder head attempt

The results obtained showed a piece resembling the print used but not nearly good enough. The fins of the top of the cylinder head didn't form at all and the side fin barely formed. We can also see a great excess of impurities in the cylinder head top surface which in the mold was located in the bottom part. It was from here that it was concluded that adding an excess of material on the side of the piece that was at the bottom of the mold would help to prevent impurities from getting in somewhere critical. At the time it wasn't possible to conclude if either the fins weren't formed due to their low thickness or due to the gating system. So another attempt was made with the same recipient for the mold, the same burnout schedule, an identical design only adding an excess of material in the spark-plug entrance location and changing the gating system. The plaster-sand-ratio water attempted at this time was **2-1-1,5** which provided excellent results and was the one used from here forward in all the attempts. It was also at this point that the correct mixing order was discovered, which was already explained at the beginning of this subchapter. In figure 4.31 we can see the 3D printed PLA piece and in figure 4.32 the results obtained from the foundry.

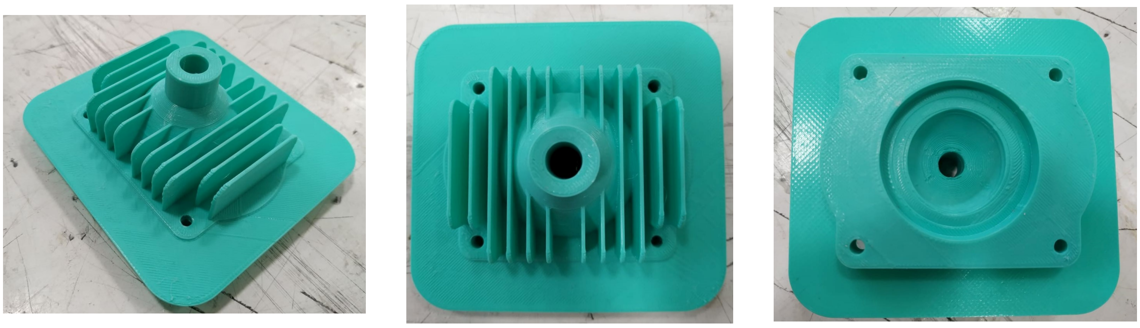

Figure 4.31: 3D print of the second cylinder head attempt

Figure 4.32: Results from the second cylinder head attempt

As can be seen in the figure 4.32, the gating system broke off. This happened during the process of sinking the 3D piece in the plaster mix but thankfully did not ruin the mold and it was possible to test it and collect the results. From here it was possible to conclude that the problem for the fins not forming came from the fact that they did not possess enough thickness. So another attempt was made doubling the thickness of the fins and increasing their length to try a new gating system that helped assure that the fins formed. Even though the burnout schedule was working and providing consistent results, it was decided to modify it for PLA, since every piece for the tests was being done in this material and for the reasons previously mentioned. In figure 4.33 we can see the 3D piece used, in figure 4.34 we can see how it was gated and in figure 4.35 we can see the results from the foundry process.

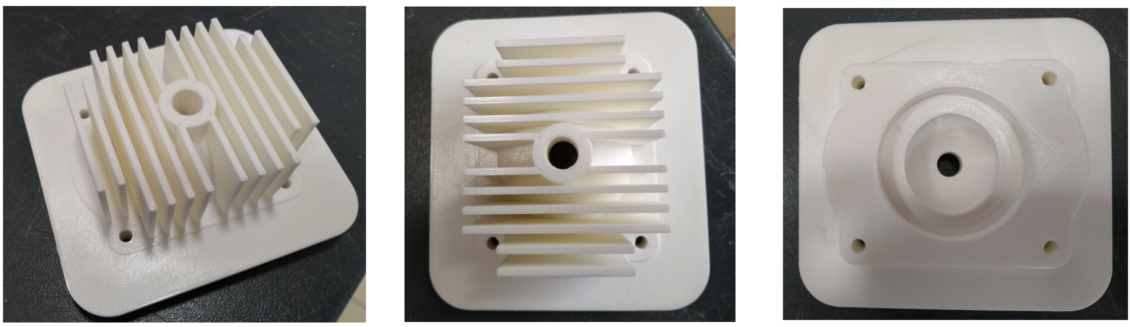

Figure 4.33: 3D print of the third cylinder head attempt

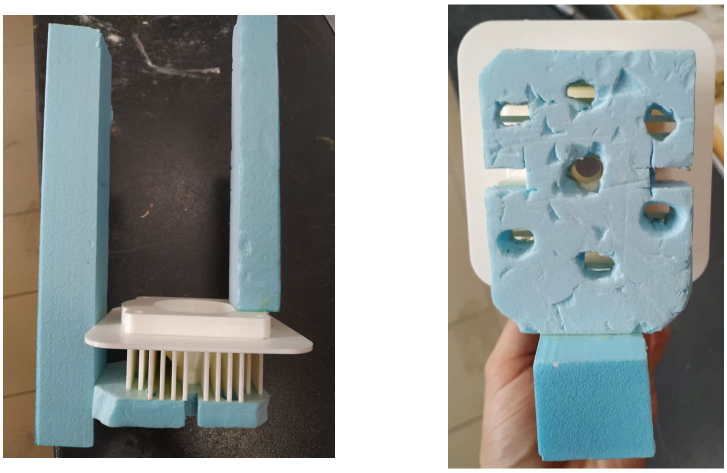

Figure 4.34: Gating system used for this attempt

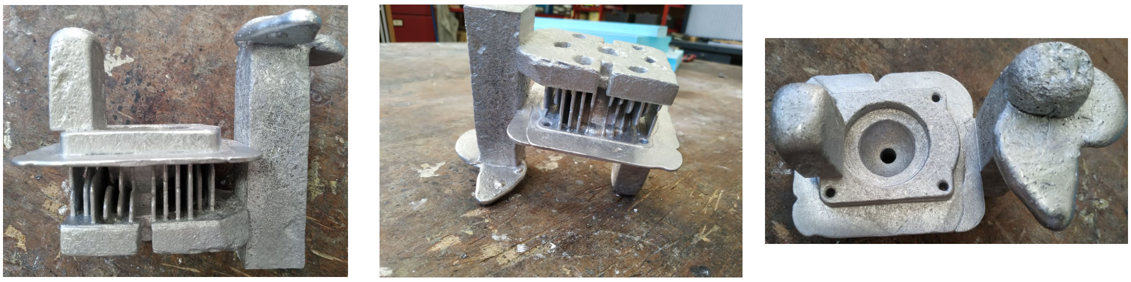

Figure 4.35: Restuls from the third cylinder head attempt

Even though the fins on the top part of the cylinder head didn't fully form, it can be considered a success. After some machining, which was done afterward and whose results will be shown later in this chapter, this piece could be used as a prototype for testing. In this attempt, the feeder was glued together in two different spots of the whole piece as can be seen in the pictures. This was done to try and increase the stability of the piece while sinking it in plaster and to try and assure that the fin formed. This proved to be a bad decision because if the aluminum enters from two different points, it will have different temperatures when it meets one another and consequently solidify at different times causing what can be seen in the fin as two different layers.

Despite already having a possible working prototype for the cylinder head it was decided to try and do one more attempt to get it fully right. So for this new attempt, everything was done as in the previous one only altering the gating system feeder that was no longer attached to two points and by increasing the thickness of the fins significantly. In figure 4.36 we can see the 3D print with the alterations made and in figure 4.37 we can see the results obtained from the foundry process after being machined to remove its excesses.

The results from this attempt can be called a success, the only problem presented by this piece was the fact that the side fin didn't fully form which is a minor concern that can be overlooked. The piece obtained presented better results than the previous attempt and therefore was the one used for testing while the other was saved as a backup. Nonetheless, the backup cylinder head was also machined to remove its excesses, and the results can be seen below in figure 4.38. The process of machining will be explained later in chapter 4.4.4.

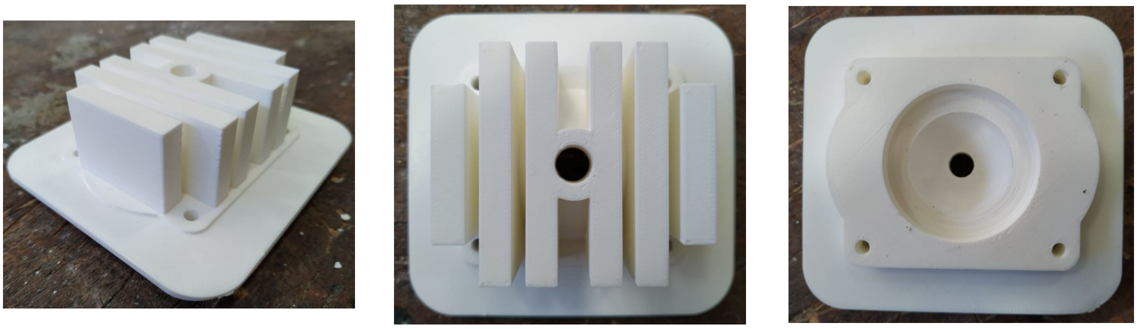

Figure 4.36: 3D print of the fourth cylinder head attempt

Figure 4.37: Results from the fourth cylinder head attempt

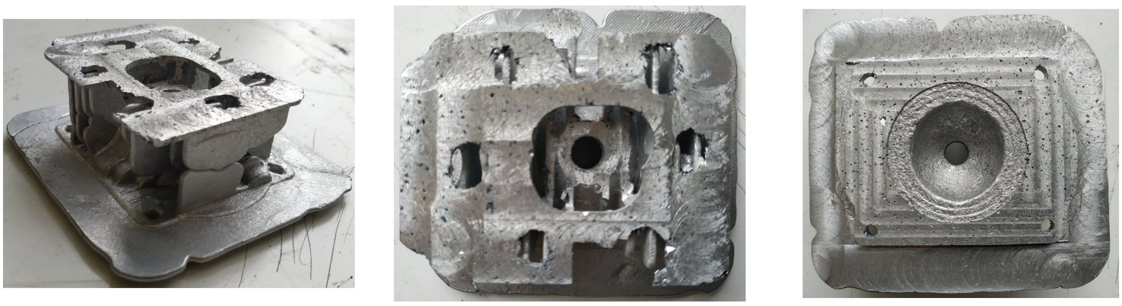

Figure 4.38: Third attempt cylinder head after being machined

With the prototype of the cylinder head ready, it was time to try and build the cylinder. As a 3D print of the cylinder in ABS had already been obtained at a previous date, it did not possess the necessary thickness in its fins that was found to be necessary for the foundry process. To try and repair this, the space between the fins was filled with glue to increase the thickness of the fins that would be formed in the mold. The burnout schedule used for this attempt was the one in figure A.1, since this was an ABS piece and as for the gating system, it was used the exact same ideology as for the last cylinder heads since it provided satisfactory results. The piece was placed inside the recipient upside down, which still was a 7L water carboy, with the entrance for the transfer ports upwards so that the plaster mix would be able to enter through all the cavities. In figure 4.39 we can see the 3D print of the cylinder used with its fins already glued.

Unfortunately, for this attempt it was tried a different kind of plaster that supposedly would help provide better results in terms of cracks and fluidity of the plaster-sand-water mix. This resulted in a broken mold after the burnout with a texture that was not solid at all. In figure 4.40 we can see the results after the burnout.

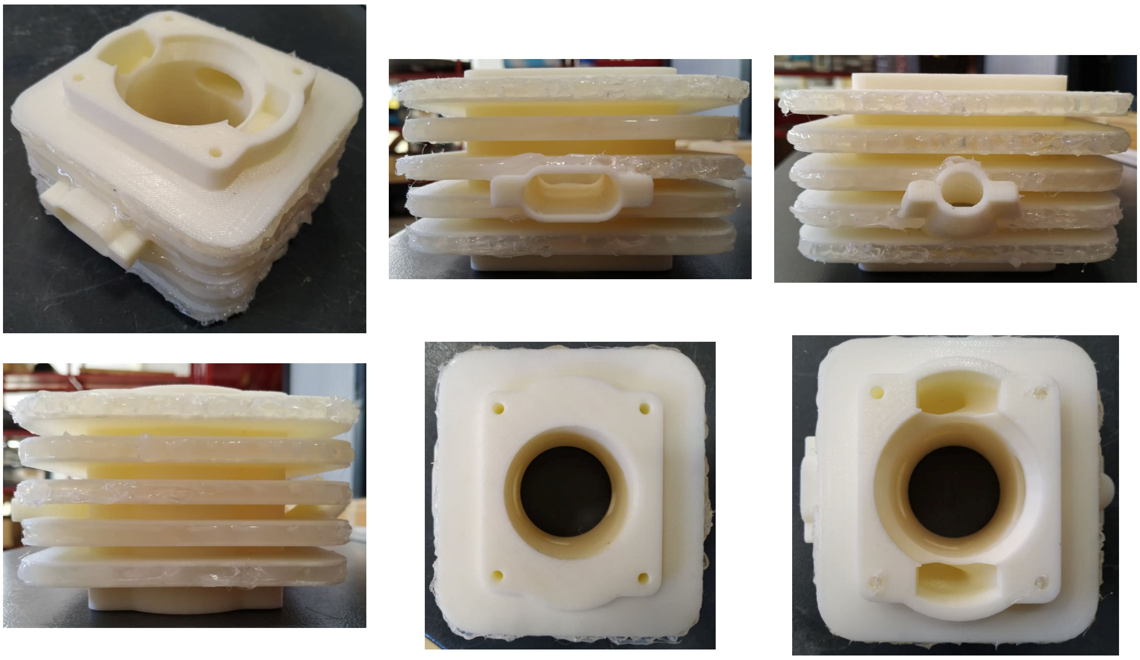

Figure 4.39: 3D print of the first cylinder attempt

Figure 4.40: Mold of the first cylinder attempt after the burnout schedule

So to make a new attempt, another 3D print of the cylinder was done but this time in PLA, so the schedule used was the one in figure A.2. This piece had the thickness of the fins already increased, the gating system was done exactly the same way as the previous one, the plaster used was the one that was used so far and the recipient was changed to a bigger one for the reasons previously explained in subchapter 4.4. In figure 4.41 is the 3D printed cylinder that was used, in figure 4.42 is the 3D piece with the gating system already set and in figure 4.43, the results obtained after machining the piece.

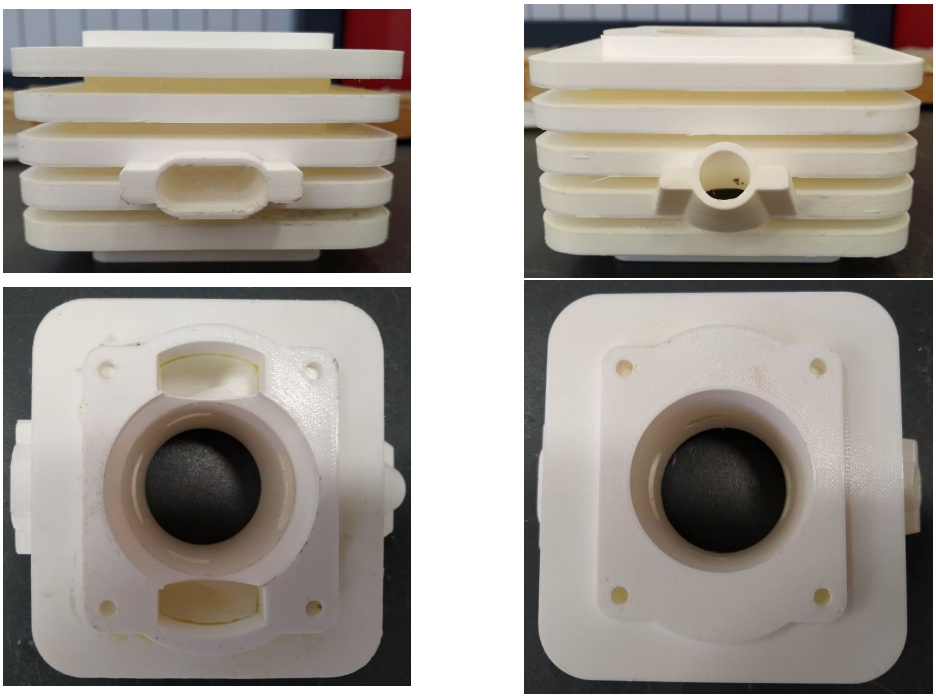

Figure 4.41: 3D print of the second cylinder attempt

Figure 4.42: Gating system used for the second cylinder attempt

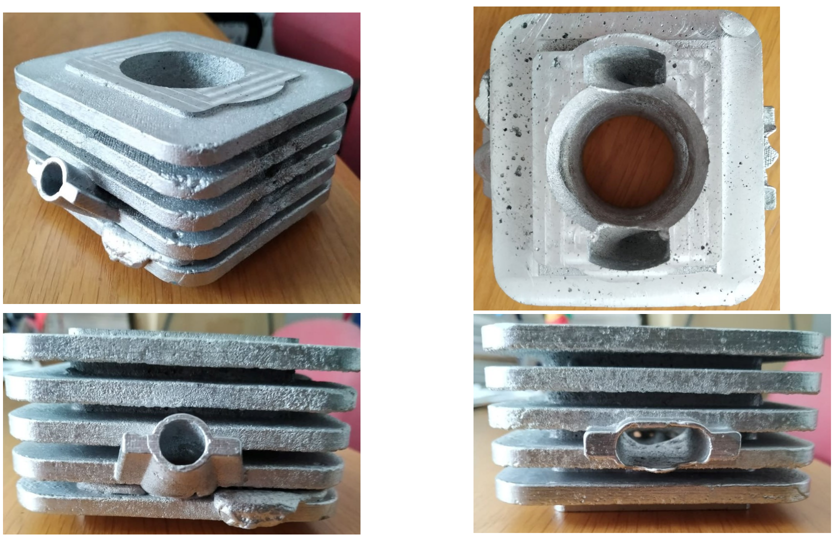

Figure 4.43: Results from the second cylinder attempt after being machined

The result wasn't perfect but it certainly was workable, this piece presented some problems like for example the fact that the exhaust and intake didn't fully form. This was most likely due to its low thickness and could be easily fixed using J.B. Weld, a more concerning issue was the shrinkage of the bore size. The 3D print had a bore of 42.5 mm which was the desired value not taking into consideration the space needed to insert a liner, and it shrunk to about 41 mm. The ports windows also didn't form exactly like the ones in the 3D model, but this was not very concerning because the ports would be made with the appropriate size in the cylinder liner.

Despite this being a piece that could be worked with to a functional prototype it was decided to make one more attempt. This mainly because the piece obtained was not designed taking into consideration the shrinkage of the aluminum nor the amount of machining it would be required to fit the liner into the cylinder. The new attempt was made increasing the bore size to 47 mm, to compensate for the possible shrinkage and to reduce the machining needed to fit the liner afterward. It was also increased the thickness of the cylinder externally, so the increased bore diameter wouldn't comprise the minimum thickness value determined in the dimensioning process. The thickness of the exhaust and intake was also increased to see if they would form and everything else of the procedure was done exactly the same way as of the previous attempt. Below, in figure 4.44 we can see the 3D printed cylinder used for this final attempt and in figure 4.45 the results from the foundry.

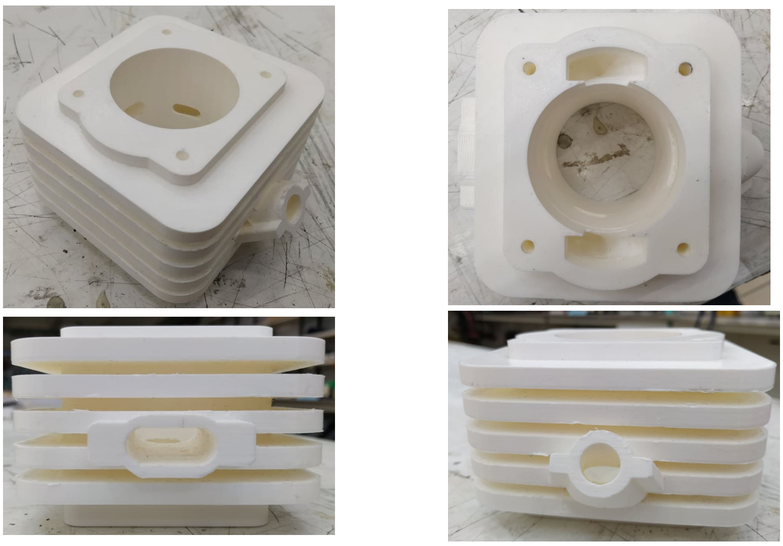

Figure 4.44: 3D print of the third cylinder attempt

Figure 4.45: Results from the third cylinder attempt after the foundry

As can be seen from the images, the intake entrance appeared with the same problem as the previous attempt only this time the hole was a bit bigger and the bottom part of the cylinder didn't fully form. This could be due to the aluminum not being hot enough when it was poured, making it impossible to reach everywhere in the mold before starting to turn solid. The bore size obtained in the aluminum piece was of 46 mm, which was 1 mm smaller than the value presented in the 3D print. Nonetheless, this attempt can be considered a huge success, the quality of the aluminum obtained was better than the one obtained in the previous attempts and the problems that appeared could be fixed simply by filling the hole in the intake and the incompletely formed bottom part of the cylinder with J.B. Weld. Holding everything into account, it was opted to use this piece instead of the one previously obtained due to the quality of the aluminum and due to the bore size being larger in this last attempt so less machining would be needed around the bore to insert the liner.

4.4.4 Machining and final adjustments

In figure 4.46 we can see the cylinder piece after being machined in the lathe and in the milling cutter machine and filled with J.B. Weld in the areas where the aluminum didn't reach.

Figure 4.46: Final cylinder after the J.B. Weld was applied and after the machining

With the cylinder block ready, it was started the fabrication of the cylinder liner. The liner was made of a seamless tube of steel which is not the recommended material at all. But as at the time no better alternative was available and since only a small amount of tests would be done to the engine, it was thought it should work as intended for the required duration. The liner was machined in the lathe to the proper dimensions and later taken to the milling cuter machine where the ports were made as accurately as possible. The steel liner can be seen in figure 4.47.

Figure 4.47: Steel liner built

To insert the liner into the cylinder, the liner was placed in a freezer while the cylinder was heated to about 200ºC in an electric oven, this caused the cylinder to expand and the liner to 'shrink', so it would be possible, with the help of a press, to place the liner inside the cylinder which would be sealed under pressure when the temperature of the components reached room temperature. The results obtained can be seen in figure 4.48.

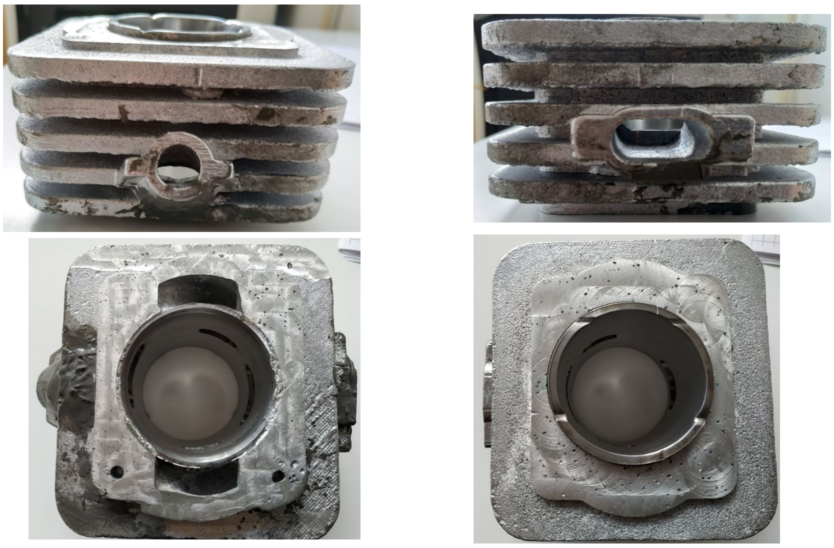

Figure 4.48: Cylinder after the liner was inserted

The process to press the liner into the cylinder was more complicated than expected which resulted in the liner being misaligned with the cylinder, this can be seen through the ports. This misalignment was small but yet considerable. To try and reduce this misalignment, the dremel was used to unblock the ports windows as much as possible. The inside diameter of the liner also needed some machining in the lathe for the piston to be able to slide through with the right tolerance. The liner had already been machined until its inner diameter was the same as the piston, but due to the deformations it suffered from the process of inserting it into the block, it needed to be rectified. It is crucial for the functioning of the engine that the piston has the correct tolerance relatively to the liner walls (piston-cylinder clearance). As no thermal study was made on this engine design, it was not possible to know exactly how much each component would expand when the engine was working. So, according to several internet forums regarding piston-cylinder clearance, a rule of thumb was used, which recommends 0.03 mm of clearance for each 10 mm of the cylinder diameter. This represents a clearance of approximately 0.13 mm for our engine, as such the piece was taken to the lathe and with sandpaper, it was polished until the piston slid through with a clearance near as possible the recommended one. The holes for the cylinder to be attached to the crankcase and cylinder head were made in the milling cutter machine, the results after all this machining can be seen in figure 4.49 and this is the cylinder piece ready to be tested.

The cylinder head to use after being removed from its excesses was previously shown in figure 4.37, theses excesses were removed in the milling cutter machine and the holes to attach it to the cylinder were also made in the milling cutter. The combustion chamber and squish band were machined in the lathe so they would have a better surface finish. The final cylinder-head piece ready to test can be seen below in figure 4.50. The excess in the fins of both the cylinder and cylinder head was initially thought to be removed in the milling cutter, but this could prove to be more harmful than helpful. The milling cutter could easily break during this and cause irreparable damage to both pieces. Therefore it was opted to leave the fins as they were since they should be able to dissipate enough heat for the small amount of time the engine would be run.

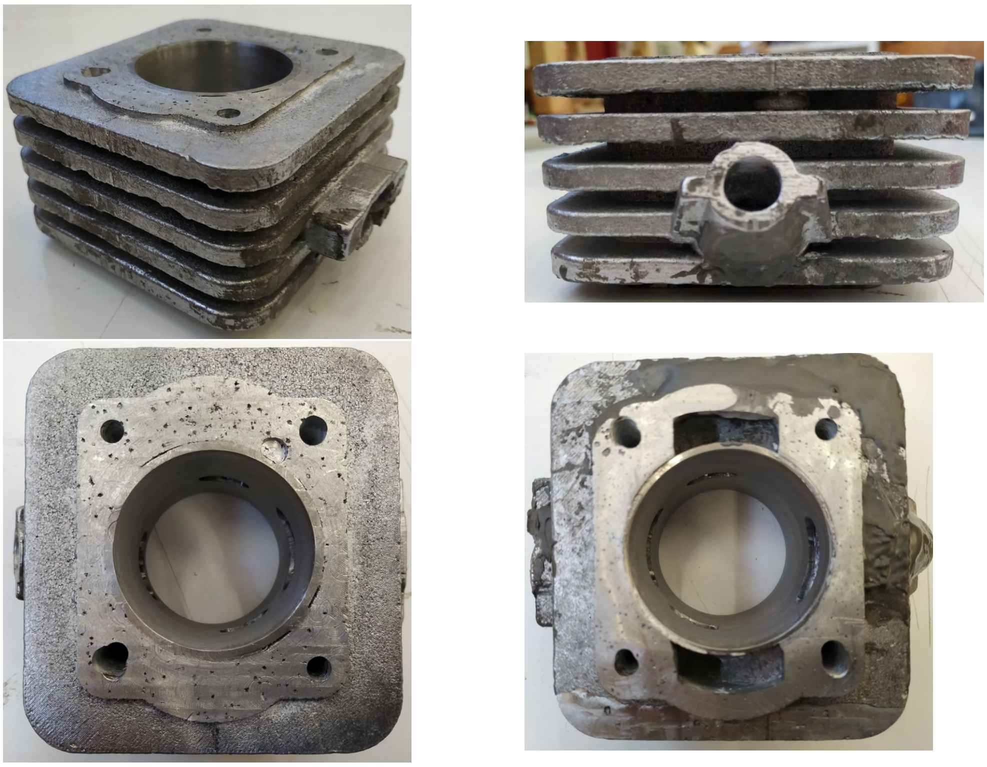

Figure 4.49: Cylinder with the steel liner, final result

Figure 4.50: Cylinder head, final result

With both the cylinder and cylinder head ready to be tested, to assemble the engine it was still needed to fabricate a head gasket and a cylinder to crankcase gasket. The head gasket was made out of copper sheet and was cut with recourse to a water jet. The purpose of the head gasket is to seal the cylinder so it can ensure the maximum compression possible while preventing leakage. Cooper is commonly used in head gaskets for it provides an excellent combustion chamber seal, it is highly conductive, its strength-malleability combination is great for it allows the gasket to conform to the surface where it is being held without losing much of its strength [24], among several other advantages. The copper gasket was annealed with recourse to a blowtorch to heat the copper gasket through all of its area. This was done because when copper is annealed it becomes soft and pliable, which gives a great seal once it is compressed [25]. As for the cylinder to crankcase gasket, it was made of “paper joint” and it was cut in the DCAs laser machine. Its main purpose is to prevent leakages from the crankcase/cylinder joint. In figures 4.51 and 4.52 are both gaskets built and used.

Figure 4.51: Copper head gasket

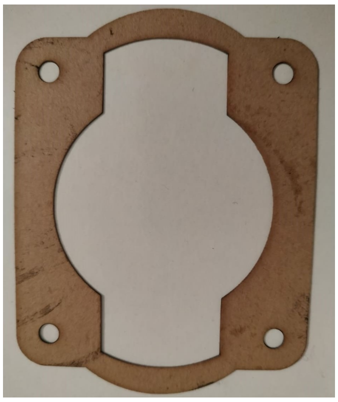

Figure 4.52: Cylinder to crankcase paper gasket

4.5 *Murphy 1.0*, engine data

Below in table 4.1 are presented some of the basic specifications of the designed engine.

Table 4.1: ***Murphy 1.0*** basic engine data

Stroke	3.2	[cm]
Connecting Rod	6.4	[cm]
CrankRadius	1.6	[cm]
Bore	4.25	[cm]
Cross sectional area of the cylinder	14.2	[cm^2]
Swept volume	45.4	[cm^3]
Cleareance volume	≈6	[cm^3]
Engine volume	≈51.4	[cm^3]
Total Compression Ratio	≈8.6	[-]
Compression ratio w/ all the ports closed	≈5.9	[-]
Crankcase Compression ratio	≈1.45	[-]
Piston-Liner Cleareance	≈0.13	[mm]
Piston-Squish Band Clearance	≈0.6	[mm]
Cylinder block Weight, After-machined	0.6	[Kg]
Cylinder head Weight, After-machined	0.15	[Kg]
Assembled engine Weight, (Block,head,piston,crankcase,crankshaft,etc...)	2	[Kg]

Chapter 5

Results and Discussion

5.1 Assembly, preliminary tests and required alterations

With all the components fabricated and ready to assemble, all that remained to try and start up the engine was to mount it in a bench test, install the selected ignition system, mixture preparation method and select a propeller. All of these steps will be explained throughout this chapter in chronological order.

5.1.1 Engine assembly

To assemble the cylinder, cylinder head and crankcase it was initially used four M5 threaded rods. These rods passed inside the through holes made in the cylinder and cylinder head and entered in the threaded holes of the crankcase. To squeeze the assembled parts, nuts were placed in each of the rods and then tightened as much as possible. Due to problems that will later be explained in this document, these rods had to be substituted by superior quality screws to increase the pressure made in the cylinder and cylinder head.

5.1.2 Load cell

If the engine worked, it would be necessary to gather results from its performance. For this project, it was then chosen and prepared to use a load cell via Arduino type of system to measure the engines' torque in case it worked. Arduino is an open-source platform vastly used to build electronic projects. An Arduino consists of a physical programmable circuit board, of which several different types exist, and of an IDE software that runs on the computer, which is where the user writes the code to upload into the circuit board. The load cell consists of a solid aluminum bar that allows the conversion of a force into an electrical output that can be read. Different types of load cells exist, but for this project, it was used a bending beam load cell with a maximum capacity of 10 kg of force. This cell works by bending when a force is applied to it and the resistances in it installed create a measurable differential from this bending. The load cell used contains four wires that connect to a micro amplifier, which in this work was the HX711 board, to amplify the electrical signal provided by the load cell resistances to the Arduino board. In figure 5.1 the connections made between the HX711 and the Arduino board can be seen, which for this case was an Arduino Nano.

Regarding the code prepared to use in the Arduino software, there are two since the load cell has to be calibrated first. To do such the code that can be seen in Appendix D is used, this code after uploaded into Arduino has to be run while the load cell isn't being subject to any weight. Afterward, a known weight has to be placed in the load cell to check what values it prints in the Arduino IDE, and from there the calibration factor has to be adjusted until the printed value is equal to the already known weight. When the calibration factor is found, it has to be copied

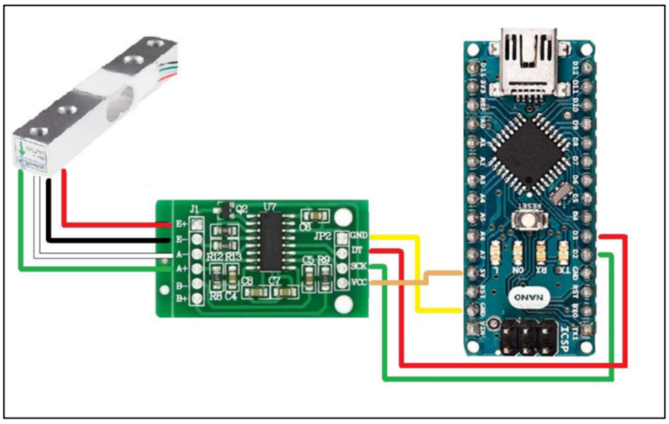

Figure 5.1: Connections made between the Arduino Nano, load cell and HX711 amplifier. Taken from [26]

and inserted in the code from Appendix E to read the values provided by the load cell.

5.1.3 Test stand

To try and start the engine, it was necessary the use of a test stand. As it was not possessed a bench test that could be used for the designed engine, it was decided to adapt one that was built during previous projects. The original test stand was built for a Honda GX31 which is also a small-sized engine and in Appendix F figure F.1 is a visual representation of the test stand in CATIA v5. This type of test stand is known as an absorber, which normally uses a brake to apply load to the engine at constant velocity while a load cell measures the load transmitted to the brake [27], but for this case acting as the brake is the propeller. This test stand was designed holding into consideration Newtons' 3rd Law, which says that any action will have an equal and opposite reaction [26]. The rotation of the crankshaft will cause a momentum and the propeller, despite rotating in the same direction, will cause an opposite and equal momentum with the crankcase, which represents an example of Newtons' 3rd Law [28]. This test stand was built to be able to measure and quantify this reaction in the form of a force through the use of a load cell. In figure 5.2, is a representation of how this test stand works. Being "A" the point at which the engine support platform is fixed to the work table, this support platform uses two ball joints and a metal plate to permit the insertion of the load cell at the desired height on the other end. As such, point "B" represents the load cell which is fixated to the test stand through the means of two screws to measure the resulting force. The load cell aside from being fixated to the test stand also has to be fixated to the work table at the correct height for the engine to remain vertical.

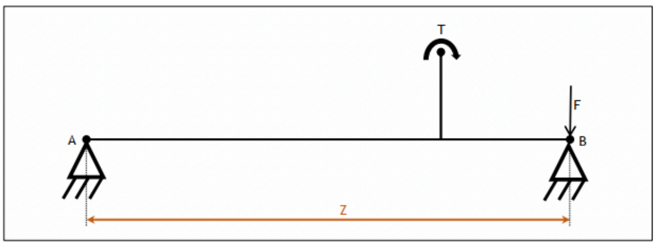

Figure 5.2: Representative diagram of the test stand, taken from [28]

As it was previously said, the load cell measures the force that results from the working engine in kg. So to find the torque, the force read by the load cell has to be converted to Newton by multiplying it by gravity (9.81 m/s^2) and afterward, this force is multiplied by the distance from points A to B.

The original test stand could not be used for our engine as it was already said, so it was modified by removing the small segment that was on top of the U shaped structure to clear the space to weld a steel beam with a square transverse section with 6 cm. This segment is where the engine was going to be attached to, so four holes were made for the threaded rods that would connect the engine to the stand and a bigger hole in the center for the crankshaft to pass through. On the opposite side of where the engine was attached, bigger holes were made to facilitate the tightening of the nuts on the threaded rods. To install the hall sensor and to locate it firmly at the desired position, a rectangular piece of wood was cut to fit tightly inside this segment so the sensor had a steady support. The test stand assembled to the engine can be seen in Appendix F as well as detailed views of the test stand and of how the hall sensor was fixated.

5.1.4 Propeller and adapter

As it was already explained, to ensure that the engine doesn't run load-free and given the aeronautical interest of this thesis, it was decided to use a propeller to apply a load factor. The load applied to the engine will, therefore, be approximately constant throughout all the tests due to the fact that they will be done in a static environment and the propeller's efficiency will be near zero [26]. As this is a prototype and even though an estimation was made regarding how much power the designed engine would produce, this is only a rough approximation that could error by a great margin. So, estimating the ideal dimensions of a propeller and buying one specifically for this purpose could prove to be pointless. As such, it was used the software "Propselector" to estimate what the ideal propeller dimensions would be for the designed engine and then from the several available propellers at the DCA one was chosen whose dimensions would be near the one that was idealized. The propeller chosen from the available ones was one with 18x10"in" made of wood. Though this is a relatively smaller propeller than the one that was idealized through the "PropSelector" software, 23x10"in", it should work for the conditions that the engine would be put through and provide results from which conclusions can be drawn.

To fixate the propeller to the crankshaft, it was necessary to build an adapter since the threaded part of the shaft meant for the propeller was relatively small. So, a small aluminum piece was designed and made in the lathe which would fit in the small conical section present in the crankshaft just before the threaded section begins. This adapter was locked in place by a nut with a flat washer in the threaded shaft. The adapter built can be seen below in figure 5.3 and its dimensions can be seen in Appendix G figure G.1.

Figure 5.3: Adapter built for the propeller

This piece already possessed three holes around its center which were used to fixate the adapter to the propeller with three screws and nuts. It was also fixed a plastic cone that was fabricated in previous projects to facilitate the engine start by using a specially made adapter in the drill. The cone was placed in the center of the propeller axis and was attached with screws to a small aluminum disc that was itself screwed to the propeller center. All these components can be seen in Appendix F.

5.1.5 Ignition system and Mixture Preparation

- Ignition System

  For the ignition system, it was bought a Pegasus DC-CDI ignition that can be seen in figure 5.4. It is a small system that uses the "Hall Effect" to send a signal to the microprocessor, which is fed by a battery, to cause the spark to happen in the spark-plug. This was an appealing system for its simplicity, it comes already fully prepared to use, without the need to configure or tune anything, only needing the hall sensor that comes with it to be installed firmly in the correct place. It is composed of a small unit with a cable to connect to a power supply (battery), a cable to connect to the "hall sensor" and a high voltage line that is connected to a cavity cap where the spark-plug is attached. The spark-plug model that was installed was an NGK – CMR6H, whose heat range is near the middle of the scale, being neither "hot" nor "cold".

  To the installation of this system, it was necessary to attach a magnet to the crankshaft for the "Hall Sensor" to send the signal to the processor and cause the spark at the desired time. It was decided to use a 25° ignition advance, which means that the magnet was placed at approximately 25° before top dead center. To do this, it was used a CDI test kit that was bought alongside the CDI system. This test kit permits to do a CDI ignition test of the ignition system to check if there are any malfunctions as well as to perform a Hall sensor test to check if the sensor after assembled would detect the magnet without the need to initiate the engine. This kit also provides a degree wheel to assist in setting the ignition timing. In figure 5.5 we can see the test kit that was used.

Figure 5.4: Pegasus DC-CDI ignition system [29]

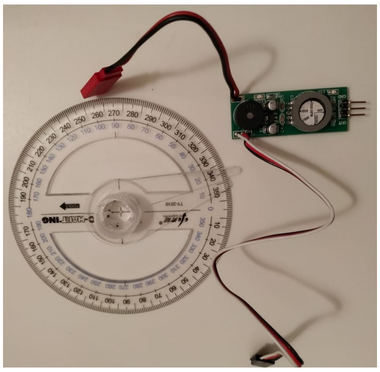

Figure 5.5: Test Kit

- Mixture Preparation

  The fuel chosen was regular unleaded gasoline, which is the most common type of fuel used in SI engines and could be easily obtained for the tests of this engine. To prepare the air-gasoline mixture it was initially opted to use a small carburetor for 50cc engines. It was chosen to use a carburetor instead of a fuel injector mainly by being far simpler to install and easier to get to work. As this is the first attempt of building a functional prototype of a two-stroke engine, increasing the complexity of the mixture preparation method could prove to be pointless for there were no assurances that the designed model would work. So, for preliminary tests of this engine, the carburetor alternative was a safer option. In figure 5.6 we can see the carburetor used.

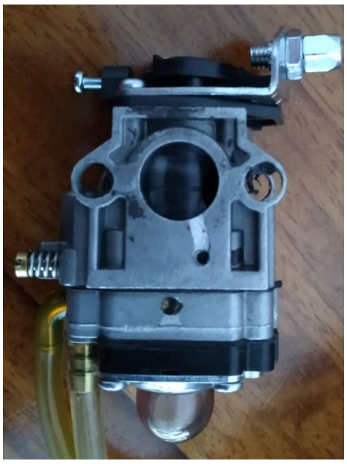

Figure 5.6: 50 cc Carburetor used

5.1.6 Alterations required to the ignition system and mixture preparation

5.1.6.1 Ignition module replacement

Unfortunately, the ignition module was destroyed due to a short-circuit right before trying to start up the engine for the first time. As it was not possessed a similar ignition system to use and the time it would take for a new one to be bought and received would compromise the deadline of this dissertation, it was decided to assemble and install an ECU, more specifically a MicroSquirt V2. This unit was in possession of Professor Francisco Brójo and was bought in a previous date to be used in an on-going Ph.D. dissertation. This ECU was brand new, thus it was necessary to do the wiring harnesses and to install the required sensors/actuators to it. As future works with this unit might need more or different inputs/outputs than the ones that were going to be needed for this project the wiring harness was done to every input/output of the ECU and was done as the fabricant specifies in the user manual. The wiring harness system was done with recourse to a welding iron, wires, and several other small utilities. For an easier identification of the several wires installed, it was assigned a color to each input/output, excluding the ones that had to be in the same color due to the lack of options. In Appendix H, figure H.1 is a schematic of how the wiring harness was done and in figure 5.7 the ECU that was used with the wiring harness already done.

Figure 5.7: MicroSquirt V2

The use of this unit allows the user to have direct control over several characteristics in most engines. It is the electronics brain responsible for all the circulation of information from both the sensors inputs to the actuators outputs [26]. With this unit, it is possible to program and define several engine operating parameters, if the correct sensors and actuators are installed, through the use of the TunerStudio software. The ECU connects to a computer through an RS232-to-USB which then allows the user to tune the engine parameters, some examples of what can be done and seen with this program are:

- Correction maps for the correction of the air-fuel ratio
- Correction maps for the ignition advance angle
- Engine RPM
- Engine and admission temperature
- Throttle Position

Several sensors can be used with this unit but for this project, not all the ECU inputs/outputs were required so only the necessary ones to run the engine were installed. As the mixture preparation method initially chosen to employ was a carburetor for 50 cc engines, the only required sensor input was the “Crank Tach In” which connected to a hall sensor (Model Haltech S3) that will detect the previously installed magnet in the crankshaft. With this input, it is possible to know the engine velocity and to control the ignition advance with the use of the TunerStudio software. For the ignition to happen it was necessary to install an ignition coil and an external ignition module that would connect to the ECU “Spark A” output, which receives the signal from the “Crank Tach In” input and causes the spark to happen in the spark-plug. The ignition coil used was an Intermotor 12304, produced by Peugeot-Citroen and its wiring harness was made according to its technical sheet, as for the ignition module it is a model 1 227 022 008 made by Bosch. Both can be seen in Appendix F figure F.9 already attached to the test stand.

5.1.6.2 TunerStudio basic configurations for ignition control

After everything was connected and assembled, it was necessary to do some basic configuration on the TunerStudio in order for the ECU to function as intended. As there was only the need to control the ignition of the engine, the only settings that needed to be adjusted were in “Basic/Load Setting$\rightarrow$ Engine and sequential settings” and in “Ignition Settings$\rightarrow$ Ignition Options / Wheel Decoder”. The finding of the correct configurations was a process of trial and error and the final configurations tried can be seen in Appendix I in figure I.1.

5.1.6.3 Troubleshoot to start the engine - Carburetor and Electronic ignition

With everything configured, it was tried to start the engine using a 40-1 fuel/oil mix. As at this point it was only being tested if the engine was working, the test bench wasn't assembled to the work table with the load cell. Instead, it was vigorously fixed to the table with clamps to ensure it did not move. Then the ECU, which was already installed with the sensors to the engine, was connected to a power source (Battery) and to the computer. It is important to underline the safety required to perform these tests, the propeller rotates at thousands of RPM and so in every attempt, it was decided that two people should always be present. While one with the help of a drill assembled with the special adapter would push it against the plastic cone installed on the propeller, the other would be adjusting the throttle position from behind the engine and outside the propeller's rotational plane. As this is a two-stroke engine with no transmission system installed, it could probably be rotated both clockwise and anti-clockwise. But since the propeller used was made to rotate anti-clockwise, it was opted for this direction. In Appendix F figure F.14 we can see the engine before being fixated to the work table to be tested with the carburetor installed.

After several attempts to initiate the engine while adjusting the "*Idle Screw*" in the carburetor to different positions, there was no ignition happening. This because the carburetor had some kind of malfunction and the fuel was not being drawn from it into the engine crankcase. Since no other similar carburetor was in the possession of the DCA, and because time was of the essence, it was opted to try and install a Throttle body fuel injection system that was used in previous projects instead.

5.1.6.4 Substitution of the carburetor system for fuel injection

The installation of a fuel injection system increases the complexity of this system greatly and can even be considered an "over-kill" due to this being a prototype, but it was a quicker option than to wait for another carburetor to be bought and received. As the wiring harness for the ECU had already been done in full, what was necessary to install the fuel injection system was to install a fuel injector, a fuel pump, two temperature sensors one for the admitted air and one for the engine, a MAP sensor, a throttle position sensor, and an accelerator body. All these components were from an EFI conversion kit for small engines from Ecotron which can be checked online at [30].

5.1.6.5 TunerStudio configurations for fuel injection

With all the sensors/actuators required for the fuel injection system installed, it was necessary to go to TunerStudio as before and change its configurations in order for the ECU to function as intended. This was done first by going to "Basic/Load Setting→ Engine and sequential settings", where the injector size had to be defined as well as the control algorithm. The control algorithm chosen was the Alpha-N which calculates the amount of fuel required through the throttle position sensor. It was also tried to use the Speed Density algorithm which controls the amount of fuel injected through the MAP sensor, but for some unknown reason this sensor was not working properly and therefore this option was discarded. It was also necessary to go to "Basic/Load Settings→ General Settings" and select the Alpha-N control algorithm for the fuel and ignition load. The last configuration required for the fuel injection to work as intended lies in “Fuel Settings→ Fuel VE Table 1”. This configuration page is a table that is totally customizable for how much volumetric efficiency is desired in each operating point of the engine. The TunerStudio software can generate a table by inserting engine characteristics such as idle speed, engine redline, peak power, peak torque, and engine displacement. As the engine was a prototype and none of these characteristics were known, the values inserted had to be assumed and several different tables were tried. In figures I.2 and I.3 are examples of the configurations used to try and start up the engine.

5.1.7 Troubleshooting to start the engine - Fuel Injection and Electronic ignition

As was previously said, several different configurations were tried in the TunerStudio software to get the engine running and even though progress was made with the fuel injection and ignition was happening, the engine would not run on its own. The reason why it would not start up was not known with certainty, it was initially thought that the problem might be related to lack of compression. So, a compression test was done with a compression gauge and it was found that the engine had a compression of approximately 25 psi. This value is far below what is expected of an engine of this size and this could be the reason or one of several, that was preventing the engine from working. To find the cause of this low compression, the engine was checked for possible leaks and it was found that the engine was leaking from the head gasket and from the pores of the cylinder head. To repair the problem regarding the leaks from the pores, J.B. Weld was added to cover them and this seemed to work in preventing these leaks. As for the leaks from the head gasket, it was decided to replace the threaded rod that was being used to assemble the crankcase, cylinder and cylinder head by four M5 “hardened steel” screws that would permit a bigger squeeze. These were all tightened with a torque of 10 Nm to ensure a uniform squeeze of the copper gasket and then another compression test was made to check for improvements. The results from this second compression test were even worse than the first, presenting a compression of around 24 psi. The compression not only remained far below what it should be, but it even got worse. The compression was still low because the leak from the head gasket remained and the explanation thought to why it got even worse was due to the fact that the cylinder liner might have been wearing off from the strokes of the piston. The liner was checked with the piston inside and the tolerance between them both was slightly larger than the one that was implemented. As the liner was made from steel instead of cast-iron, its inner walls were getting worn out and gradually increasing the piston-liner clearance. This consequently, removes the capacity for the rings to seal the chamber and causes loss of compression. At the time of its fabrication, it was not possessed such material and therefore improvisations had to be made. It was not thought at the time, that the start-up of the engine would prove so laborious and take so many attempts.

Even though the odds of getting the engine working decreased at each time one attempted to do so, it was thought to install an exhaust from an old 50 cc chainsaw to try and improve the chances of getting the engine to start up. Exhausts on two-stroke engines play a particularly important role, they can improve an engine’s performance significantly. They assist the engine: by helping pull extra charge from the crankcase into the cylinder; by preventing the charge that is entering through the transfers to leave directly through the exhaust port; and by boosting the compression [31]. The prototype should be able to work without one, but as the installation of one could only help, it was decided to install it and test the engine.

To install the exhaust in the prototype, an adapter had to be made to connect them both. This adapter was made out of a rectangular piece of aluminum that was taken to the milling cutter where the engine exhaust shape, which was almost of the exact same size as the entrance for the exhaust from the chainsaw, was made. Afterward, two holes were made for screws to attach this adapter to the engine block and another one, to attach the exhaust to it. Both the adapter built and the exhaust used can be seen in Appendix F. Another compression test was done after the install and it did not make any visible difference, the compression remained the same and the engine still didn't start.

5.1.8 Alterations to the cylinder liner

In a last-ditch effort to try and start up the engine, it was decided to remove the steel liner and replace it with a new one in a more suitable material (cast-iron). This was done to try and resolve the problem regarding the low compression and the misalignment from the steel liner that was blocking the ports partially. To remove the existing liner from the engine block, it was used a hand saw to cut it vertically and afterward, with a hammer and a similar sized tube, punched out. The new liner was machined and inserted into the engine block by the same method as the previous one but this time with more careful planning to insert it correctly and avoid misalignments. It was also decided that the inner diameter would only be adjusted after the insertion into the engine block to prevent the deformations that come with the insertion method. In figure 5.8, the cast-iron liner can be seen after it had its exterior machined to the dimensions of the engine block cylinder and with the ports holes already made. Before inserting the liner into the engine block, the port windows on the block were further enlarged with recourse to a dremel. This was done to prevent the windows from being blocked by the liner in the event of it being misaligned as the previous one. In figure 5.9 the engine block can be seen with the new liner already inserted and ready to be tested. This time the liner was almost perfectly fitted inside the engine block, it still came out a bit misaligned from the correct position but it was a much smaller deviation than the previous attempt and thanks to the precaution of enlarging the engine block ports, only the exhaust had a small blockage in its port.

Figure 5.8: Cast-iron liner built

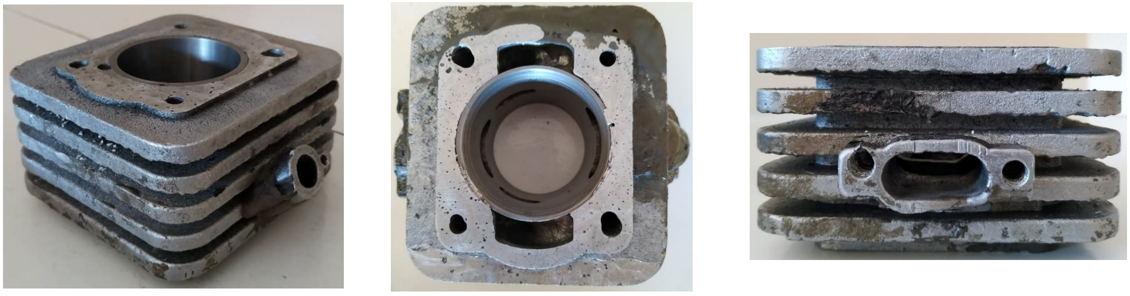

Figure 5.9: Cylinder with the cast-iron liner, final result

5.2 Final attempts to start up the engine

With the new liner inserted into the block, the engine components were all assembled and another compression test was done. The compression remained unaltered, 25 psi, which indicates that the low compression was being caused mainly by the leak in the cylinder head gasket. Nonetheless, it was tried to start up the engine and even though ignition was happening and pushing up the engines RPMs, it still did not sustain itself to work. Normally, for a two-stroke engine around the same size, the compression should be somewhere between 80-120 psi and 25 psi is not even close to that.

To try and repair the main cause of leakage, that was found to be the head gasket, it was tried to further increase the squeeze in the M5 screws to 15 Nm. Despite this making two of the screw threads in the crankcase to yield, another compression test was done and the compression had gone up to 30 psi. In light of this improvement obtained by just slightly increasing the tightness of the screws, all the four holes where the screws were placed were enlarged to fit M6 screws and the thread in the crankcase was remade. These larger screws permitted to perform a tightening of 15 Nm to all of them without causing the screw threads to yield. Interestingly another compression test was done afterward and the compression went back to the initial value of approximately 25 psi. It was even tried to furthermore increase the tightening of the screws to 20 Nm, but this made the thread in the crankcase to yield once more and the compression remained unaltered.

It was then not possible to get the engine to start up, and these were the last attempts made in the time available.

5.3 Results and discussion

The results from the fabrication process established were satisfactory. Holding into account that this was the first prototype attempt and several further improvements can be made in the fabrication process, it produced good results in replicating the 3D model of the engine cylinder/cylinder head. As it was shown and explained in chapter 4, alterations had to be made to the design to adapt it to the fabrication process. None of the cast pieces came out exactly like the designed model but thanks to the alterations made in the design, it was possible to obtain pieces with considerable accuracy. The causes for the problems/imperfections in the pieces are mostly known and can be further reduced, or even erased, to obtain pieces with a higher level of precision. These can be achieved with some refinement to the fabrication process and to the design of the piece to cast. Examples of how the fabrication process can be further improved are as follows:

- **Electric oven**: A bigger oven would permit the use of bigger molds which would not only permit the cast of bigger pieces but it would also give more margin to center the piece and avoid cracking issues.
- **Furnace/Crucible**: The fabrication of a bigger crucible and a bigger furnace with an in-built thermostat would permit a more precise temperature control of the molten metal and of the timing to pour it.
- **Design and gating system**: Further adapt the design for the fabrication method as well as improve the gating system.
- **3D Printer**: Free access to a 3D printer for more liberty in experimenting models and on improving the gating system with 3D print parts instead of styrofoam.
- **Plaster**: Experiment different plasters to obtain an even better mix, being the most widely used and recommended for this kind of procedure "Plaster of Paris".
- **Exothermic toppings**: The use of an exothermic topping for the metal pour would help increase the quality of the cast-piece overall.

As for the engine itself, even after everything attempted it still didn't start. This could be due to several reasons, some that can be impeding the engine from working are as follows:

- **Lack of compression**
- **Wrong ignition configuration**
- **Wrong fuel injection configuration**
- **Failed port design**
- **Misalignment of the cylinder liner**
- **Wrong fuel-oil ratio**

The one reason that is for certainly impeding the engine from working is its lack of compression. With a compression of just 25 psi, it's basically impossible to get the engine to start up and work. The only place found to be leaking was the cylinder head gasket and in the time available, it was thought of ways to seal this leak but sadly the attempts made were unsuccessful. What was most certainly also helping this occur, was the extended vibration of the engine components when tried to start it up. The test stand used despite being vigorously attached to the worktable still produced high-level vibrations across the workbench as it was already pointed out in previous works using this test stand [28]. Without first solving the problem regarding the leakage, other reasons that can also be impeding the engine from working can't be completely excluded nor included.

Regarding the liner, even though it still was slightly misaligned, approximately by 1 mm, the port windows were all unblocked and it had no leaks. It was tested if the liner was not sealing the chamber, and thereby cause loss of compression, with a compressed air gun. The spark-plug was removed, the piston brought to TDC and afterward, it was shot air through the spark-plug hole and air wasn't exiting through the ports, only from the gasket.

What else could be preventing the engine from working, would be a wrong ignition and fuel injection configuration. As for ignition configuration, it is not very likely that it was wrongly configured. This because even if the ignition angle wasn't at the exact desired angle, it was very close to and this should only alter engine performance, not prevent it from working. The fuel injection configuration could also not be injecting the correct amounts of air/fuel and therefore preventing it from working. Or it could also be a bad oil/fuel ratio, but again, this wasn't likely the case since when was tried to start up the engine, there was ignition happening and the engine itself was pushing up the RPMs higher than the drill used could go (from approximately 1900 to 2100-2500). The fact that ignition was happening and pushing the RPMs, indicates that the ignition and fuel injection configurations, even if not optimal, were in a state that should not keep the engine from working.

Ignition happening and pushing the RPMs can also indicate that the designed ports can/might work. As constant ignition was happening with a slight increase in RPMs, it means that the fuel was entering the crankcase through the intake port, going through the transfers into the cylinder and exiting as exhaust gases through the exhaust. Therefore the process followed to dimension the ports could not be validated nor discarded. To assess its validity one must first eliminate the other already known existing problems that are impeding the engine from working such as its lack of compression.

As it was not possible to get the engine to work, there were no performance parameters to be collected. Hence, power and torque values could not be attained to be compared with the expected power from the conceptual project calculations. These expected values would most likely be wrong by a considerable margin for most of the RPM range calculated. This mainly because it was assumed a mechanical efficiency of 70%, and the real value would most probably prove to be much lower. This value was also assumed constant through all the engine operating range and in reality, this doesn't happen. In Appendix K are presented some expected power values for different velocities. It is also important to point out that the arrangements for the measurements of engine performance parameters were made. This was done at a previous date when the cast pieces were still being machined to prepare for the eventuality that the engine worked. So, the load cell system was prepared as described in Chapter 5.1 but it was never assembled to the test-stand, given that there was never a necessity for it.

Chapter 6

Final Considerations and Future Works

6.1 Final considerations

The main objectives for this dissertation involved the design, fabrication and performance tests of a small single cylinder two-stroke engine. Therefore, this includes the assembly of the same with the installation of an ignition system and a mixture preparation method to test if the prototype was functional and if so, the installation of a data logging system to measure its performance parameters.

The engine design mostly relied on the engine ports, which are critical for the functioning of the engine. As was explained previously in this document, little to no information regarding engine design was found, and this is easily explained by the fact that no manufacturer is going to give away the method through which their engines are designed. Engine design mostly relies on empiricism and as this was the first two-stroke prototype built in the DCA, no previous empirical data was possessed. The ports were then designed following the empirical guidelines found in the books previously presented and made in the cylinder liner as meticulously as possible.

The design process was followed by the fabrication, which was an extremely experimental procedure and involved a long and laborious process of trial and error until the fabrication method had been refined enough to present satisfactory results. As said before, everything from the fabrication process was done at the university, the fabrication method was learned and refined to a point that can be considered a reliable option in future projects that require the cast of pieces with some degree of complexity. The machining of the components and adapters was also done at the university with the help of Prof. Dr. Francisco Brójo, mainly recurring to the milling cutter, the lathe, and the Dremel. The process of machining done was not the most accurate nor appropriate but produced acceptable results and therefore, can also be considered a reliable option in future works with similar objectives.

The designed engine was idealized to run via carburetor and with a CDI ignition system, as was presented before in this document, but due to the several adversities that came with the experimental procedure, it was necessary a change in both. This change was to an ECU as it was already explained, and required the necessity to better understand how electronic control systems work. It was necessary to install sensors/actuators and to learn how to control them through the TunerStudio software. This system was far more complex than the initial idea, but it was a necessity to try and start up the engine that permitted a considerable understanding of how to electronically control an engine.

Even with everything attempted it was not achieved the start-up of the engine. This, as was previously explained, was almost certainly due to the leaks from the cylinder head gasket which were causing an extreme loss of compression. The recovered piston-crankcase set was most likely from a two-stroke engine with the cylinder and cylinder head in one piece. As the designed engine was in two pieces, to facilitate the machining process, it needed a bigger squeeze than the crankcase screw thread could handle to seal off this leak. To solve this, as there is probably only one more chance to increase the crankcase screw thread to a bigger size, a new cylinder head could be made with an improved design to guarantee that the piece won't yield to the augmented tightening with the bigger screws. It could also be attempted to use a copper sheet gasket with a smaller thickness to reduce the force needed to squish it between the cylinder and cylinder head.

With this compression problem, it was not possible to conclude if the designed ports could actually work and hereby present a process from which future two-stroke engine variations can be made at the university. It is, however, thought that given the fact that the engine had constant ignition happening and a slight increase in velocities, the ports were working as intended and the lack of compression was the factor that was preventing it from sustaining itself to work. Nonetheless, only after solving the compression problem and then attempt to initiate the engine will it be possible to take any conclusive information regarding if the design worked or not.

To sum everything up, despite the unsuccess in obtaining a working prototype and the inconclusive data regarding if the ports were working as intended or not, it was achieved a deeper knowledge about two-stroke engines; a prototype (Murphy 1.0) was designed, fabricated and machined; it was found a method to design the engine ports that unfortunately was not conclusive; it was established a fabrication process that can provide satisfactory results at the university; and in spite of not being planned, the knowledge regarding ECU's was also considerably improved.

6.2 Future works

Several future works can derivate from this dissertation, but as a first priority in the authors' opinion, one must assess if the guidelines followed to design the ports actually present a viable method. To do such, it is first needed to solve the compression problem prevenient from the cylinder head gasket. If this problem is solved, one can attempt to start up the engine and verify if the designed prototype is functional. If it works, it will be possessed a functional prototype from which empirical data regarding engine design can start to be gathered. So, below are presented some future works that can be implemented to this dissertation and works that can be done related to it.

Implemented to this dissertation

- Solve the compression problem with the "Murphy 1.0" and test the engine.
  - Design a new cylinder head with increased thickness to permit a bigger squeeze in the screws without making it yield, correct its air fins direction, which were only noticed to be in the opposite direction after the fabrication of the components, and use a thinner cylinder head gasket to try and solve the compression problem. Or, design/cast the cylinder and cylinder head as one piece.
  - Improve the fabrication and machining process to obtain higher quality pieces.

- If the engine works, collect the performance parameters so empirical data is gained regarding engine design. If not, verify what's impeding it and if it can be solved.

Related works:

- Assess different methods to design a two-stroke engine
- Study of how squish bands influence engine performance
- Design and fabrication of an expansion chamber
- Design and fabrication of other small prototypes.

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Appendixes

Appendix A - Burnout schedules

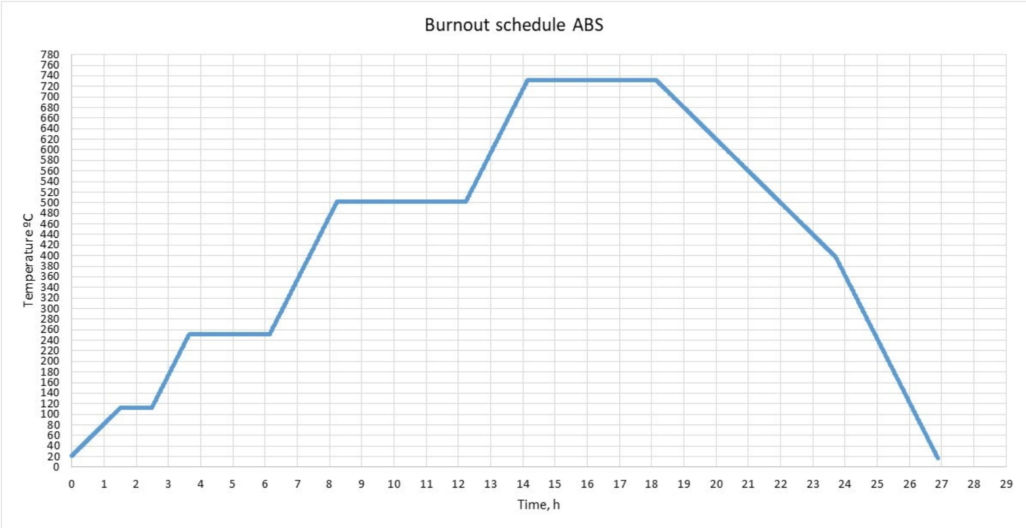


Figure A.1: ABS burnout schedule

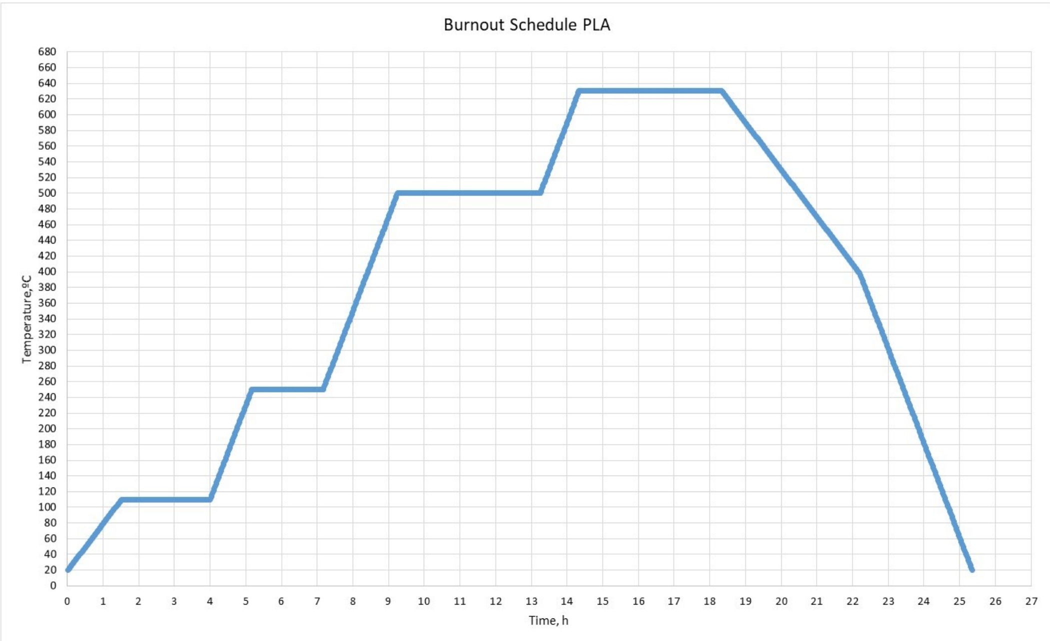


Figure A.2: PLA burnout schedule

Appendix B - 2D Draws of the cylinder and cylinder head, Murphy 1.0

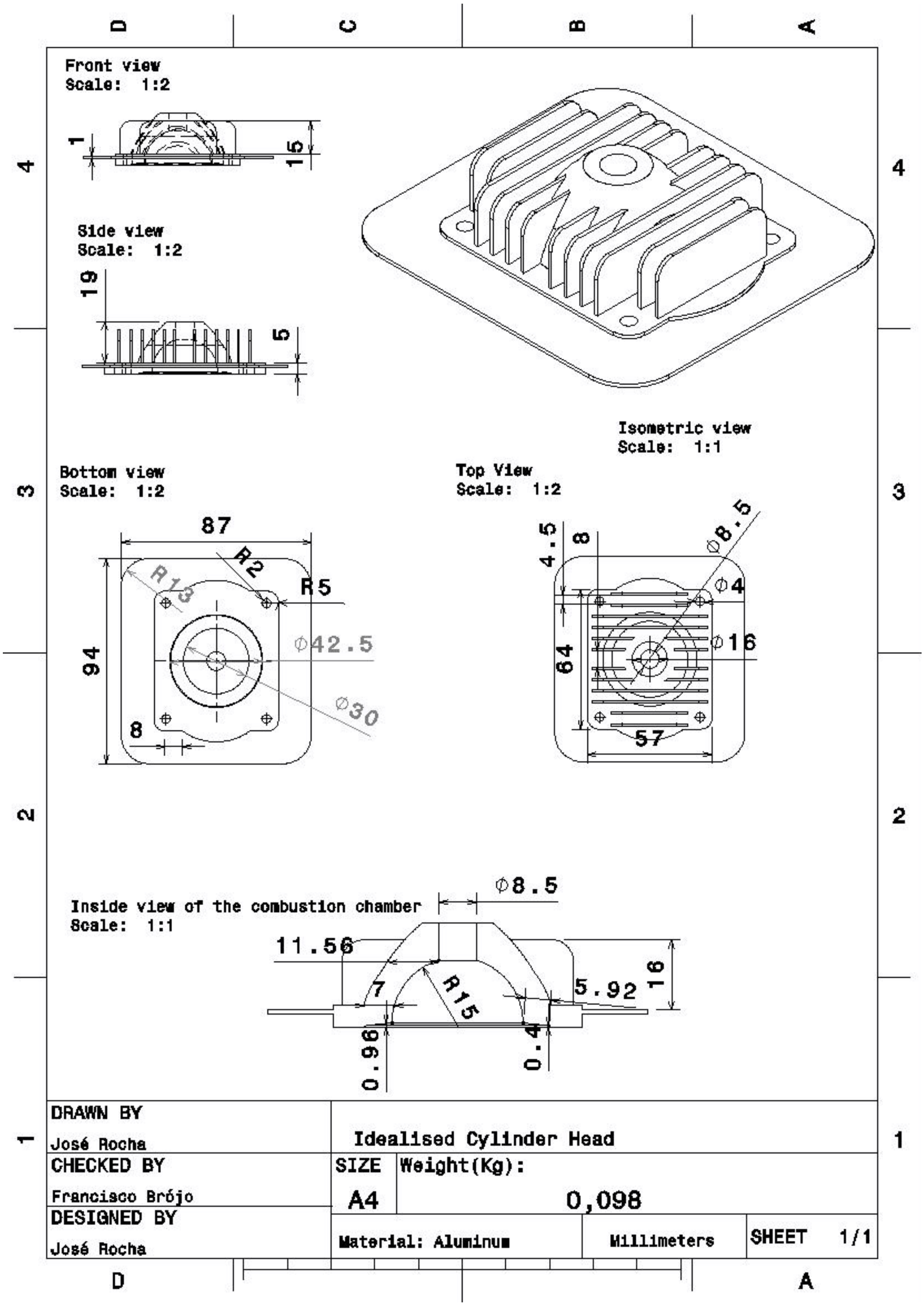


Figure B.1: "Idealised cylinder head"

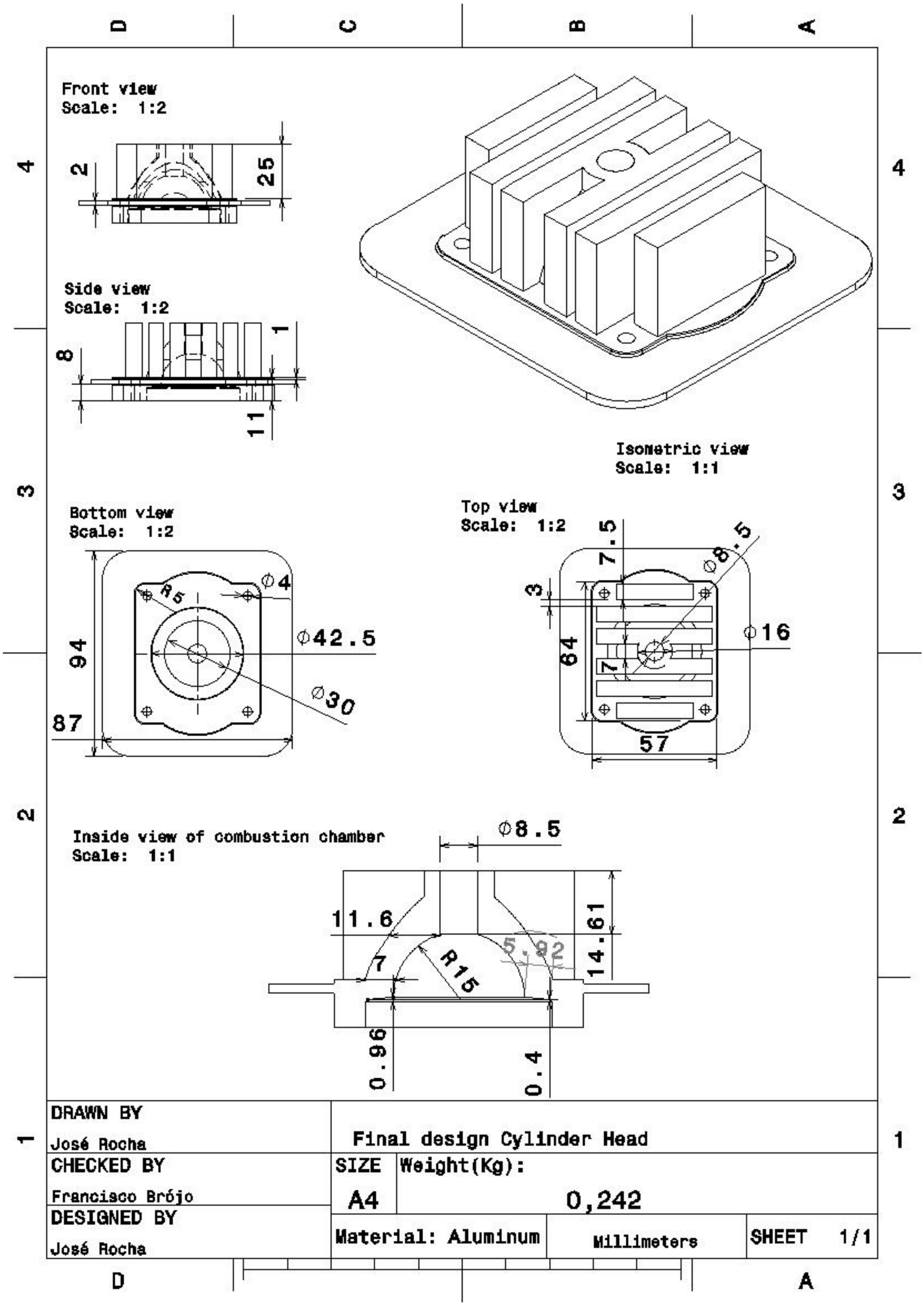


Figure B.2: "Final design cylinder head"

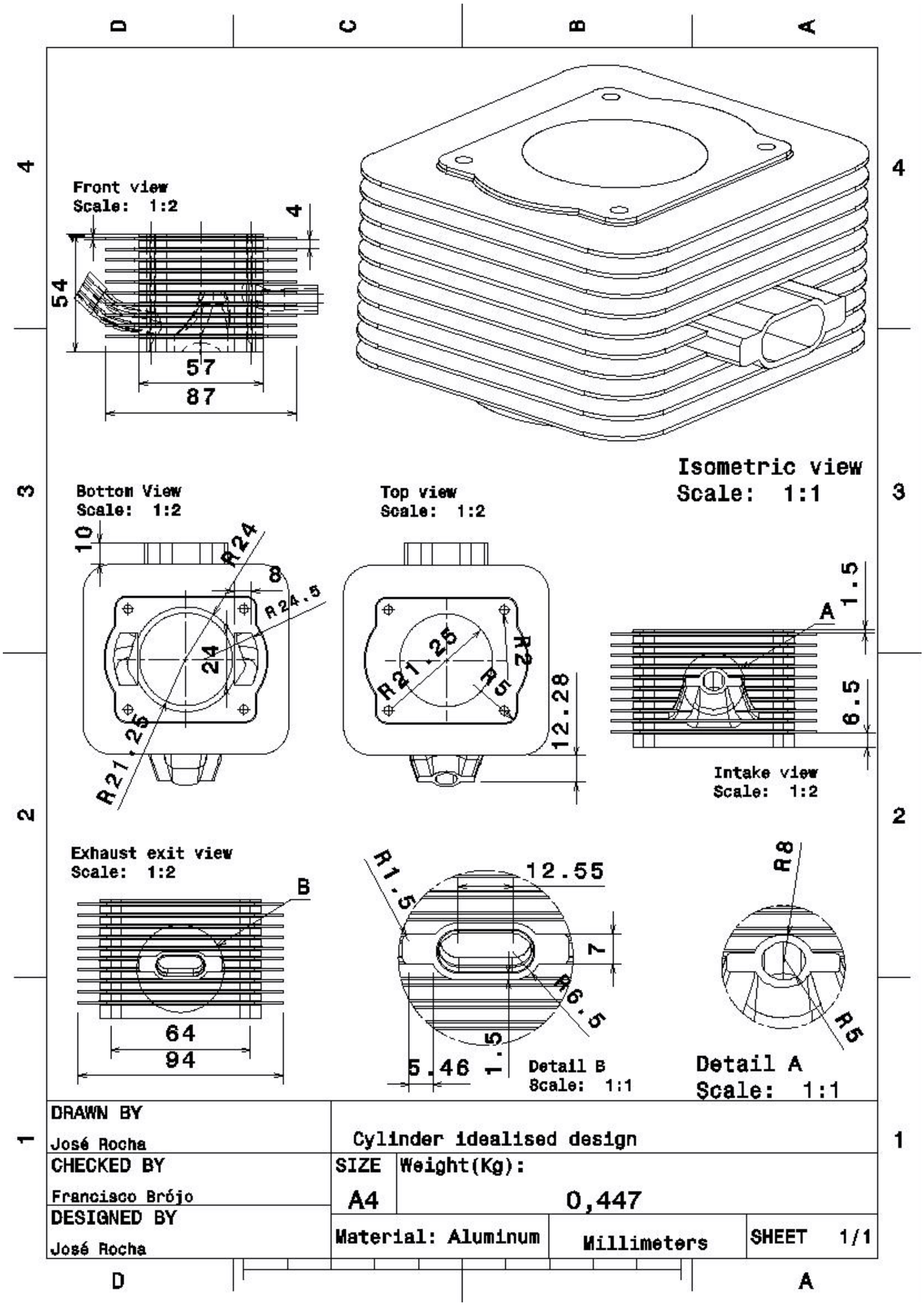


Figure B.3: "Idealised Cylinder design"

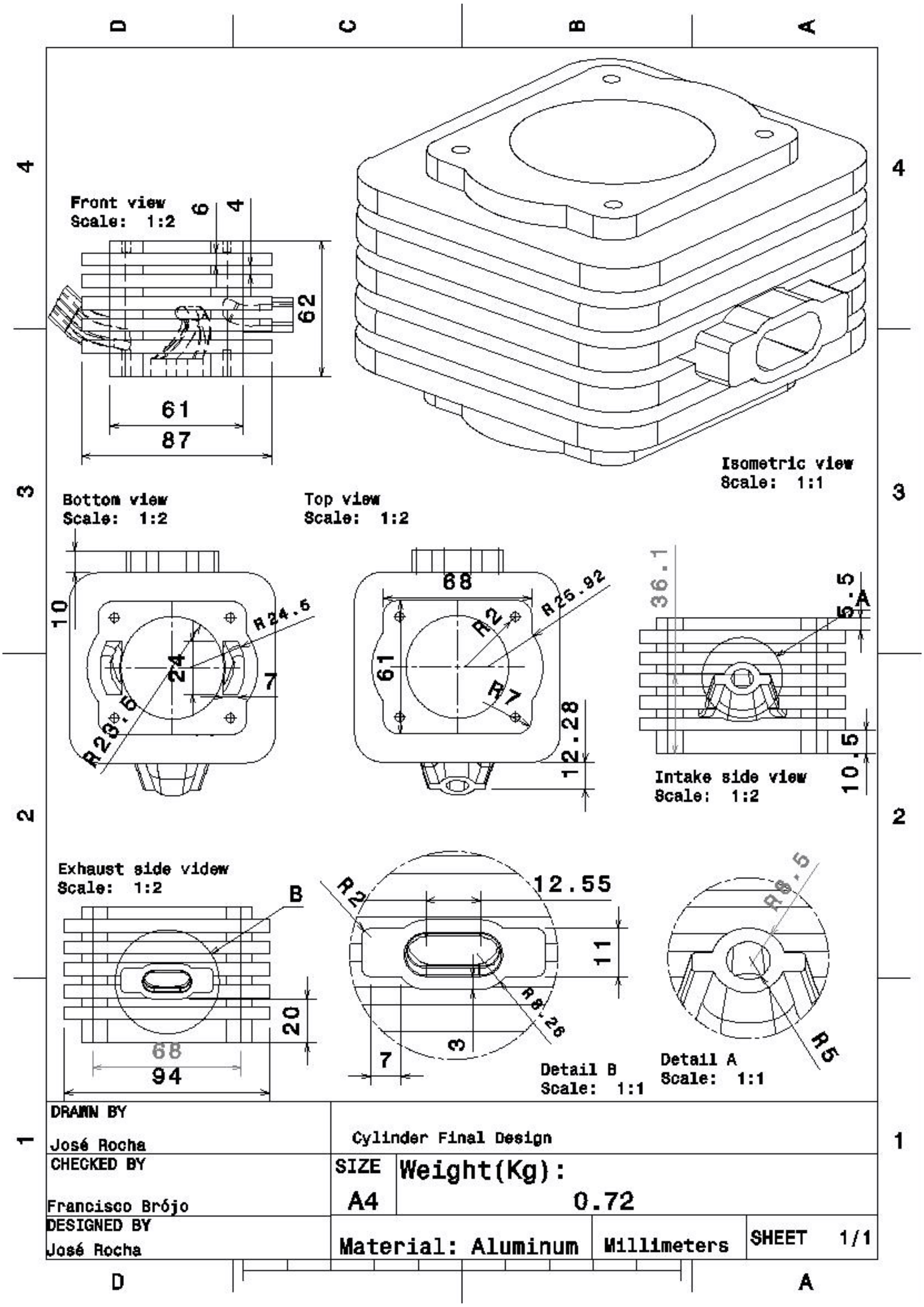


Figure B.4: "Final Cylinder design"

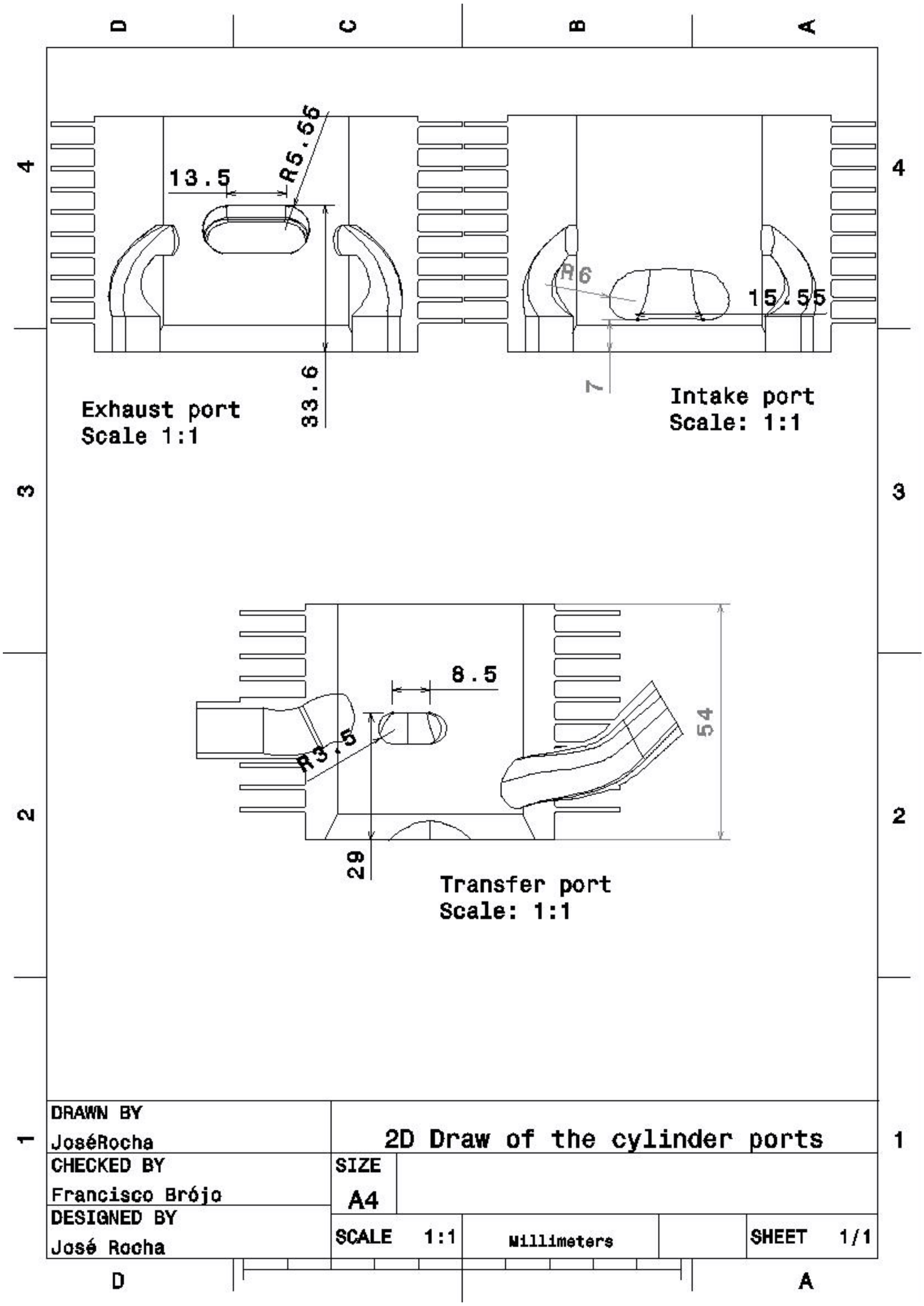


Figure B.5: "Ports dimensions"

Appendix C - Timing diagram

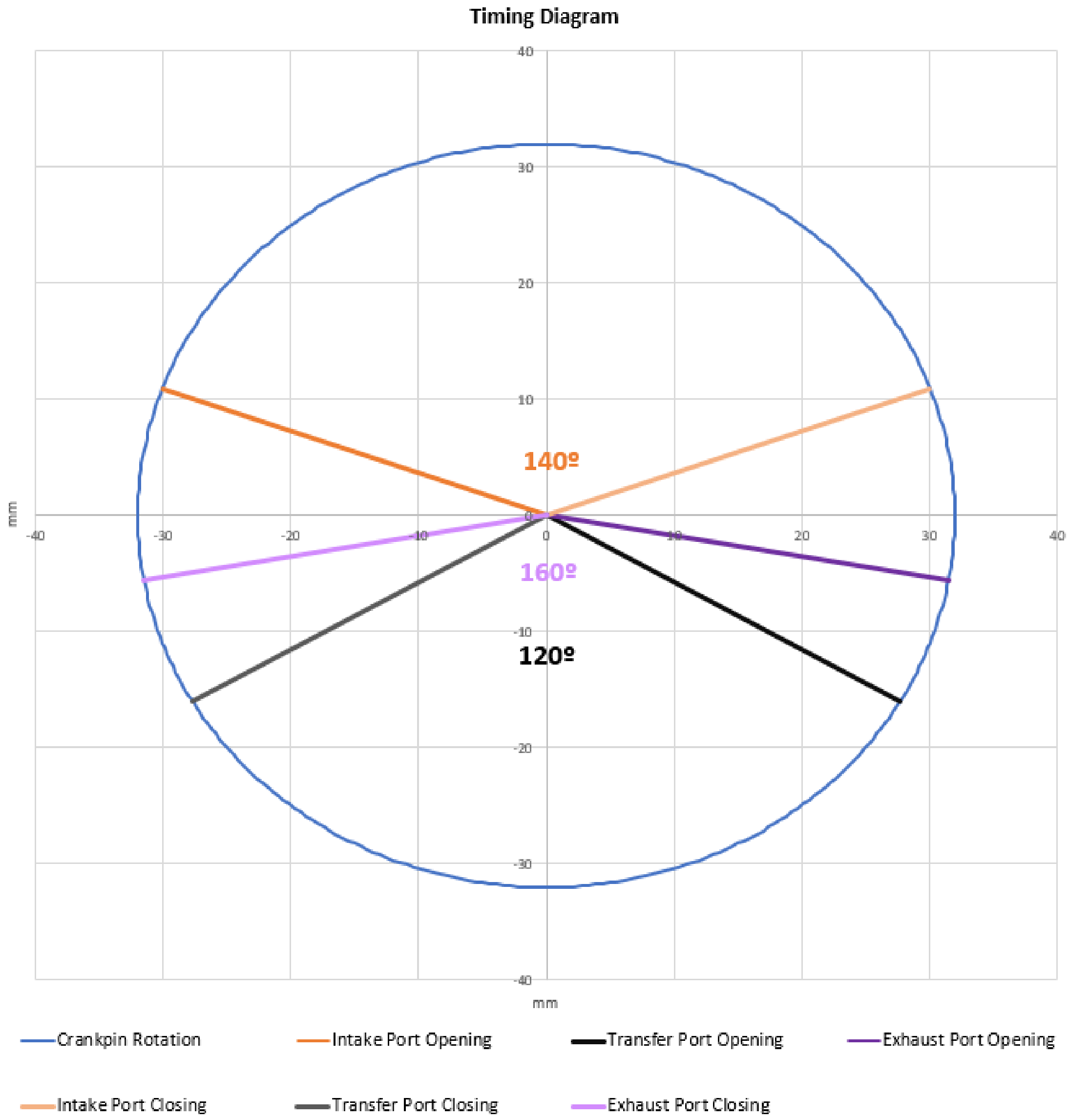


Figure C.1: Timing diagram of the engine designed

Appendix D - Arduino code used for calibrating the load cell

```
#include "HX711.h"

// HX711 circuit wiring
const int LOADCELL_DOUT_PIN = 2;
const int LOADCELL_SCK_PIN = 3;

HX711 scale;
float calibration_factor = 323000;
//=============================================================================
// SETUP
//=============================================================================
void setup() {
Serial.begin(9600);
scale.begin(LOADCELL_DOUT_PIN, LOADCELL_SCK_PIN);
Serial.println("HX711 Calibration");
Serial.println("Remove all weight from scale");
Serial.println("After readings begin, place known weight on scale");
Serial.println("Press a,s,d,f to increase calibration factor by 10,100,1000,10000 respectively");
Serial.println("Press z,x,c,v to decrease calibration factor by 10,100,1000,10000 respectively");
Serial.println("Press t for tare");
scale.set_scale();
scale.tare(); //Reset the scale to 0
long zero_factor = scale.read_average(); //Get a baseline reading
Serial.print("Zero factor: "); //This can be used to remove the need to tare the scale.
Serial.println(867944);
}
//=============================================================================
// LOOP
//=============================================================================
void loop() {
scale.set_scale(calibration_factor); //Adjust to this calibration factor
Serial.print("Reading: ");
Serial.print(scale.get_units(), 3);
Serial.print(" kg");
Serial.print(" calibration_factor: ");
Serial.print(calibration_factor);
Serial.println();
if(Serial.available())
{
char temp = Serial.read();
if(temp == '+' || temp == 'a')
calibration_factor += 10;
else if(temp == '-' || temp == 'z')
calibration_factor -= 10;
else if(temp == 's')
calibration_factor += 100;
else if(temp == 'x')
calibration_factor -= 100;
else if(temp == 'd')
calibration_factor += 1000;
else if(temp == 'c')
calibration_factor -= 1000;
else if(temp == 'f')
calibration_factor += 10000;
else if(temp == 'v')
calibration_factor -= 10000;
else if(temp == 't')
scale.tare(); //Reset the scale to zero
}
delay(1000);
}
//=============================================================================
```

Appendix E - Arduino code used to read the load cell values

```
#include "HX711.h"
const int LOADCELL_DOUT_PIN = 2;
const int LOADCELL_SCK_PIN = 3;
HX711 scale;
float calibration_factor = 323000; //Calibration Factor obtained from the calibration code
//=============================================================================
// SETUP
//=============================================================================
void setup() {
Serial.begin(9600);
scale.begin(LOADCELL_DOUT_PIN, LOADCELL_SCK_PIN);
Serial.println("Press T to tare");
scale.set_scale(calibration_factor);
scale.tare(); //Reset the scale to 0
Serial.println("CLEARDATA"); //clears up any data left from previous projects
Serial.println("LABEL,Time,Weight kg");
}
//=============================================================================
// LOOP
//=============================================================================
void loop() {
Serial.print("DATA,TIME,");
//Serial.print("Weight: ");
Serial.println(scale.get_units(), 3); //Up to 3 decimal points
//Serial.println(" kg");
//Serial.print("Time");
//Serial.print("Weight");
if (Serial.available())
{
  char temp = Serial.read();
if (temp == 't' || temp == 'T')
scale.tare(); //Reset the scale to zero
}
delay(250);
}
//=============================================================================
```

Appendix F - Practical components

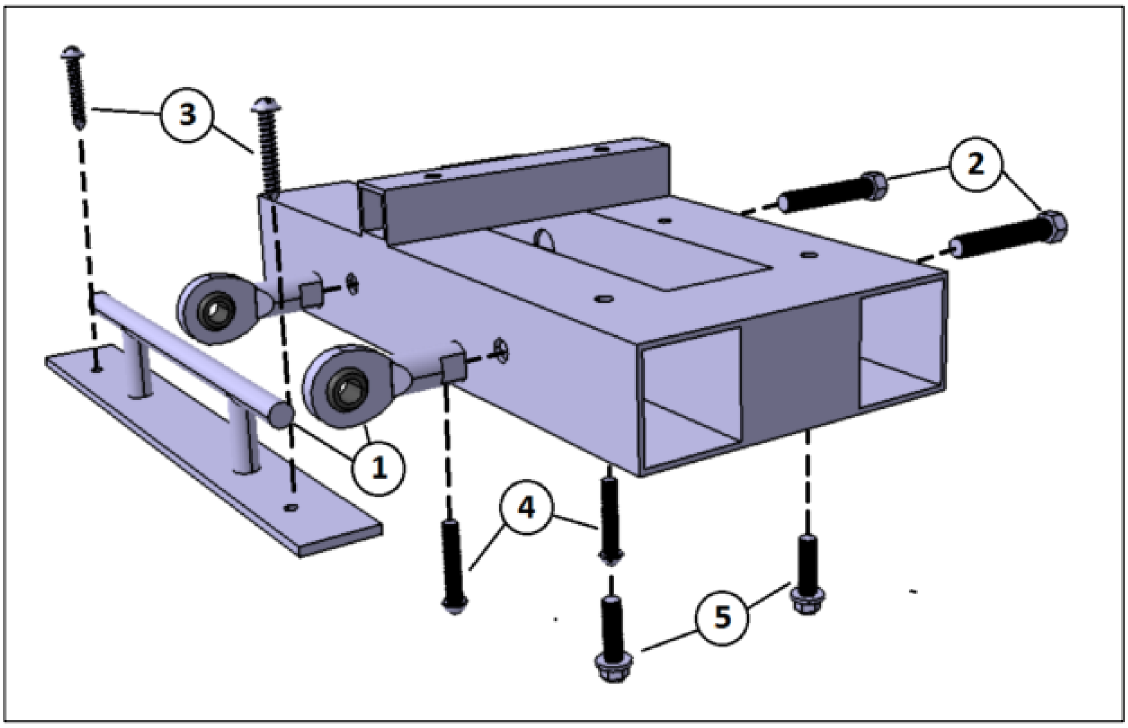
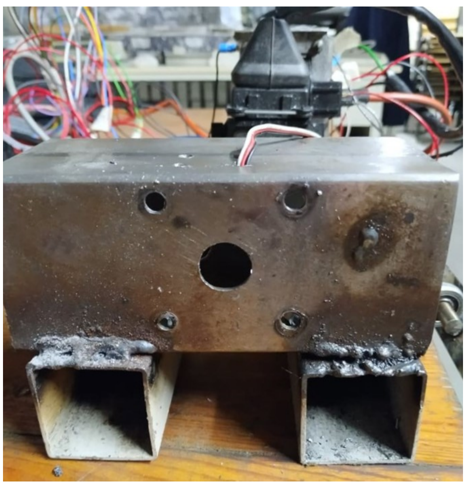


Figure F.1: Representative CATIA V5 design of the support platform built in previous projects. Taken from [26]



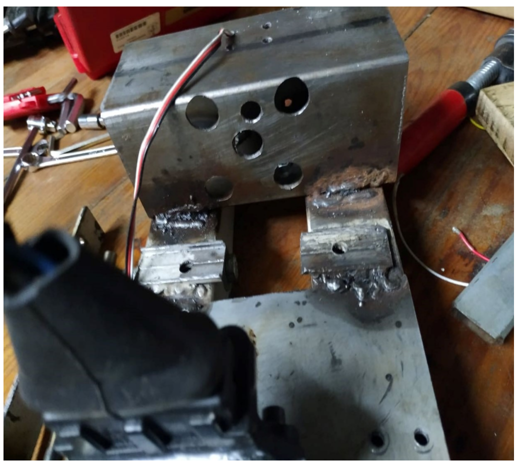


Figure F.2: Detailed view of the modification made to the test stand

Figure F.3: Detailed view of the engines backside

Figure F.4: Engine assembled to the test stand

Figure F.5: Plastic cone

Figure F.6: Adapter for the drill

Figure F.7: Aluminum disc used to connect the cone to the propeller

Figure F.8: Propeller assembled with the adaptor and the cone

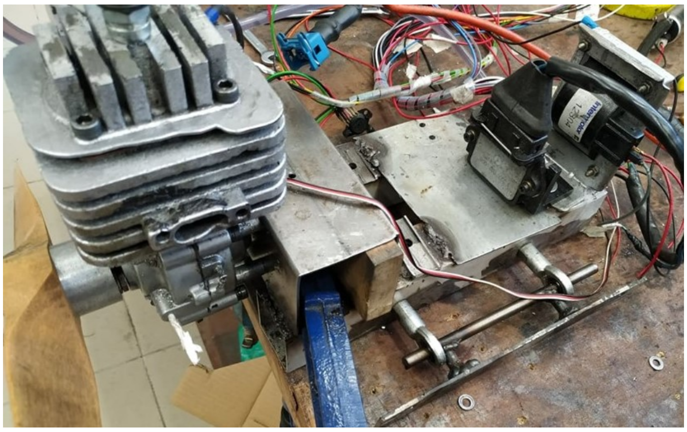


Figure F.9: Ignition module and ignition coil assembled to the test stand

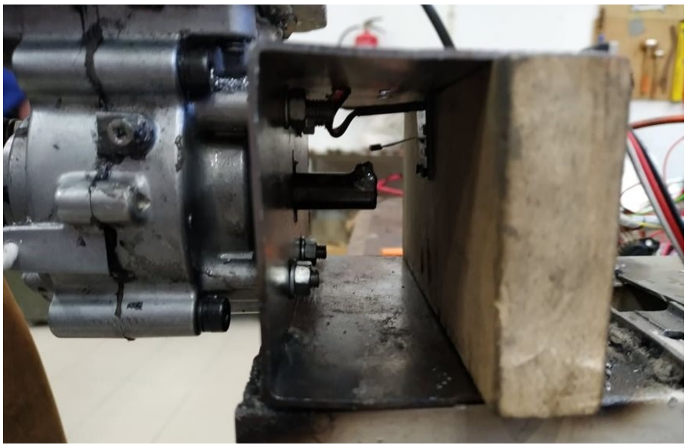


Figure F.10: Detailed view of the hall sensor

Figure F.11: Adapter built to install the exhaust

Figure F.12: Exhaust installed

Figure F.13: Adapter built for the intake

Figure F.14: Engine with the carburetor installed

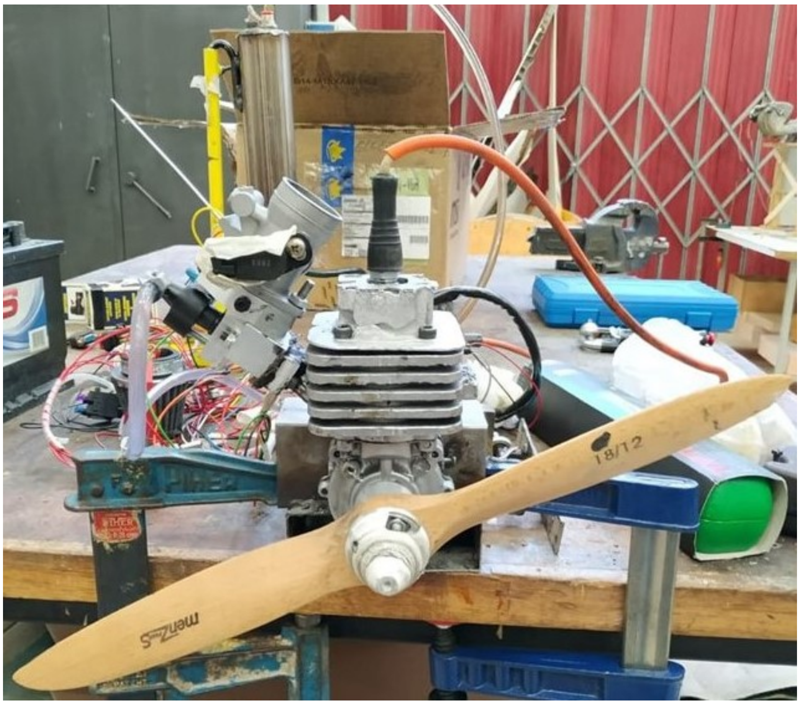


Figure F.15: Engine with the fuel injection system installed, final assembly

Appendix G - 2D Draw of the Adapter built for the propeller

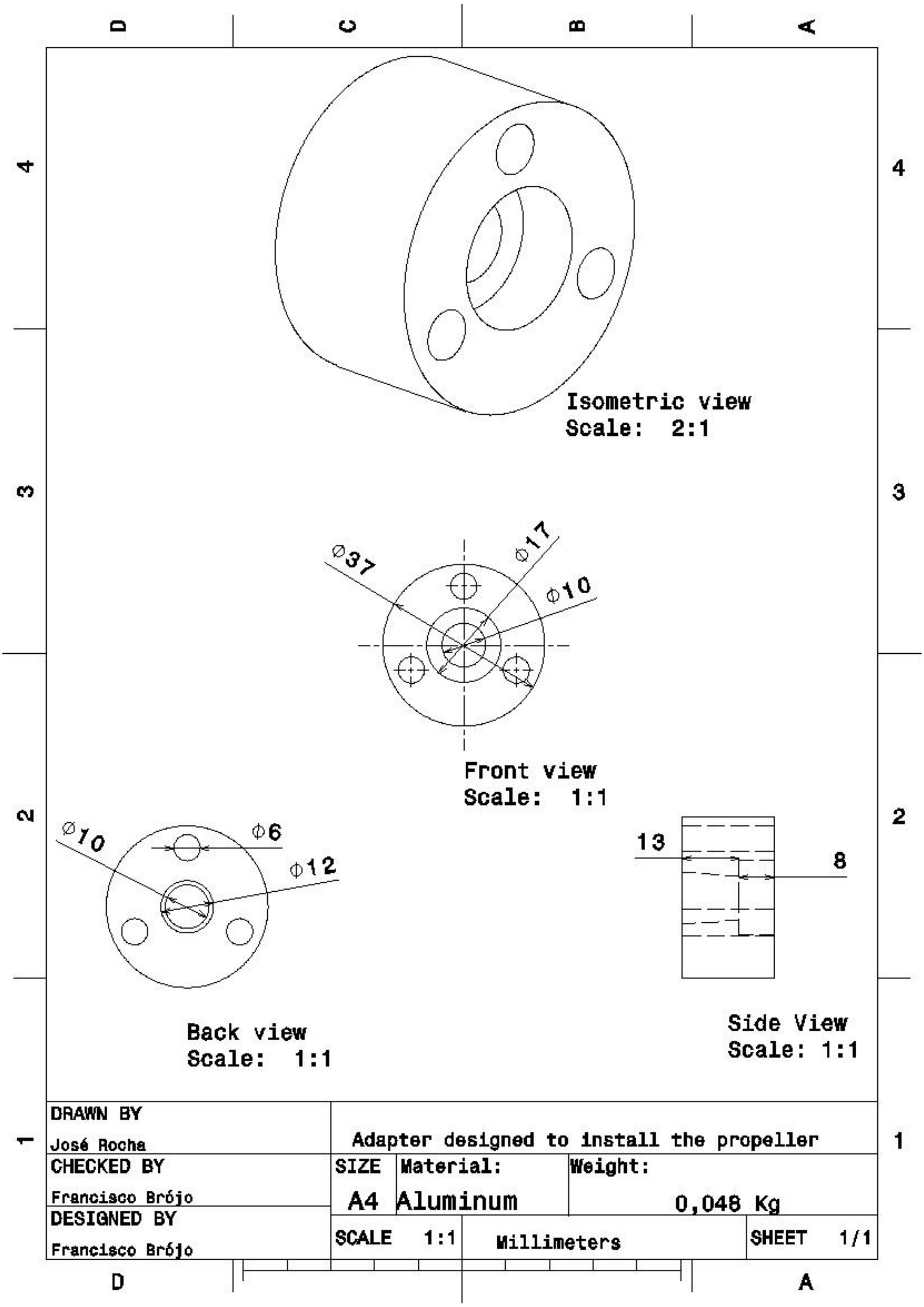


Figure G.1: Adapter designed to install the propeller

Appendix H - Wiring harness diagram

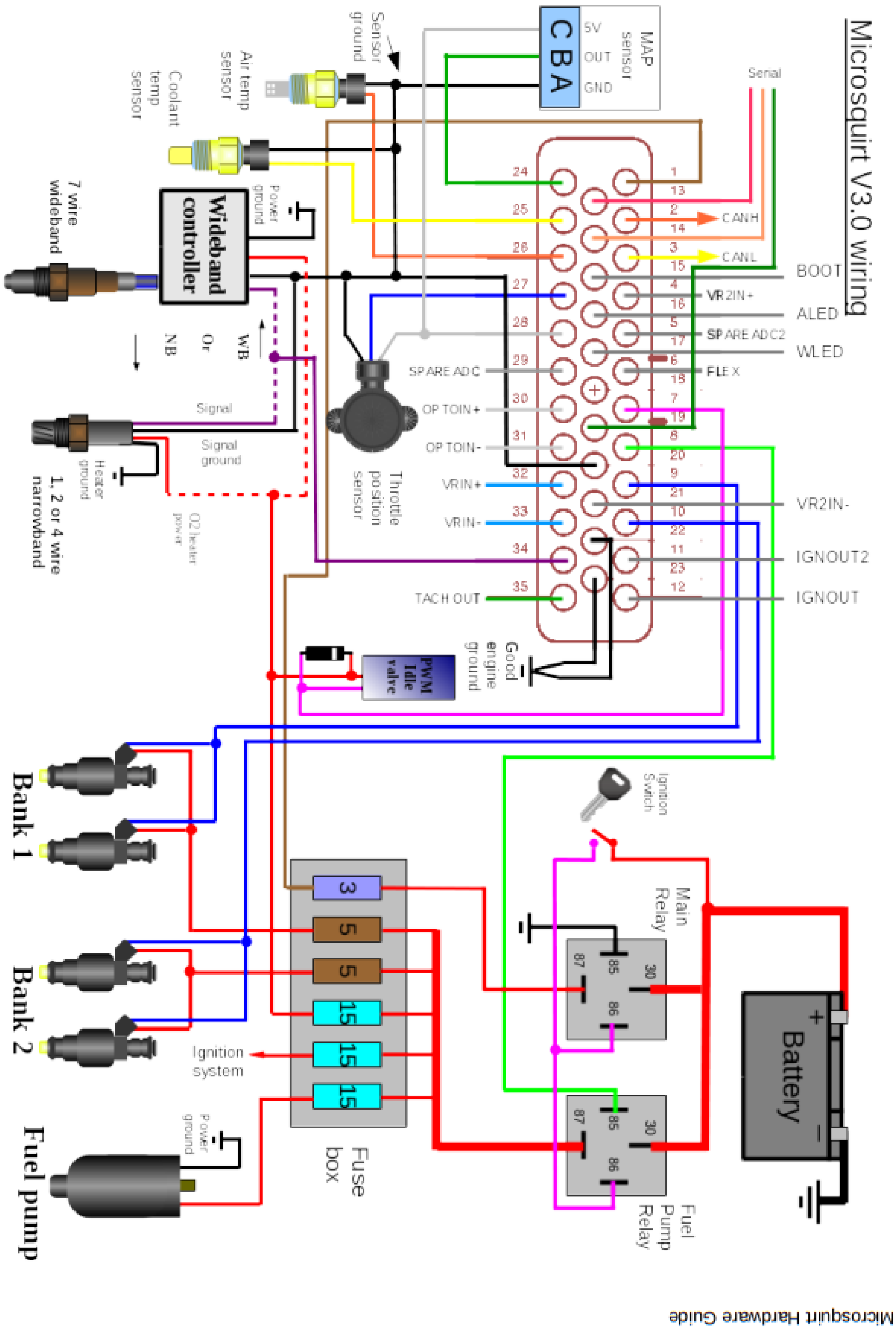


Figure H.1: Wiring harness diagram, taken from [32]

Appendix I - TunerStudio Configurations

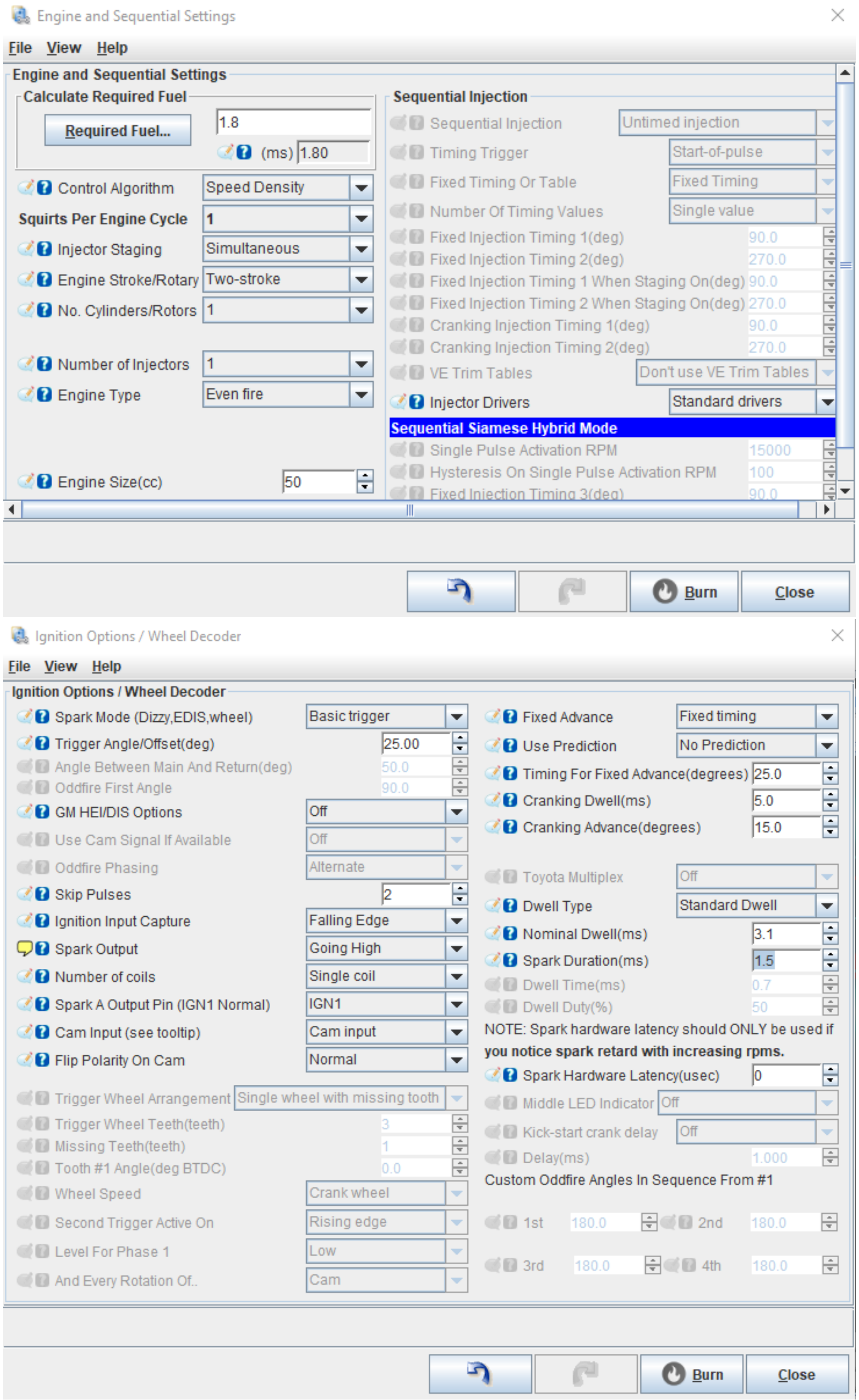


Figure I.1: TunerStudio basic ignition configuration

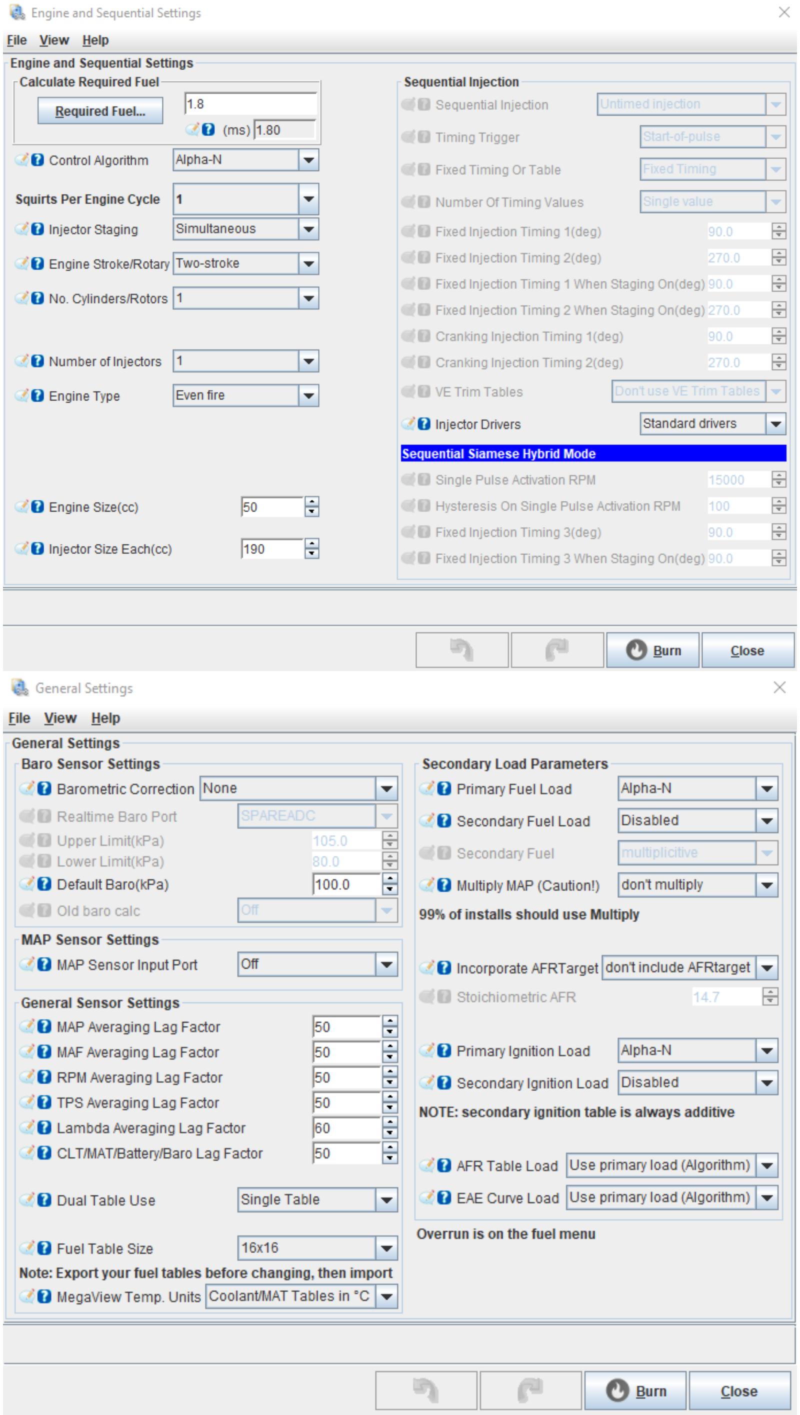


Figure I.2: TunerStudio basic configuration for fuel injection

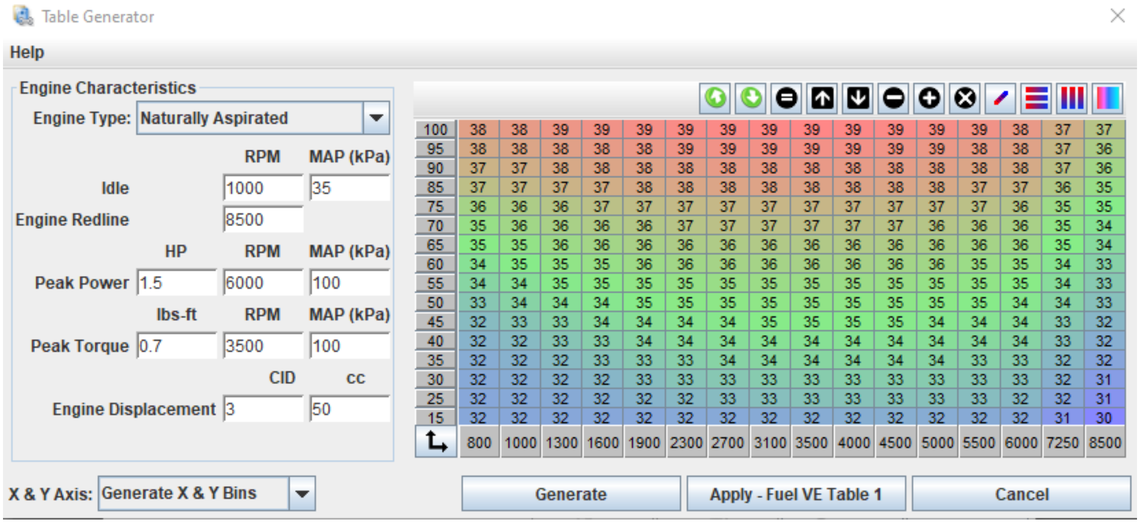


Figure I.3: Table VE generated with recourse to TunerStudio MS

Appendix J - Heating values for different fuels

Fuel	Density		Higher Heating Value (HHV) (Gross Calorific Value - GCV)					Lower Heating Value (LHV) (Net Calorific Value - NCV)				
	@0°C/32°F, 1 bar											
Gaseous fuels	[kg/m^3]	[g/ft^3]	[kWh/kg]	[MJ/kg]	[Btu/lb]	[MJ/m^3]	[Btu/ft^3]	[kWh/kg]	[MJ/kg]	[Btu/lb]	[MJ/m^3]	[Btu/ft^3]
Acetylene	1.097	31.1	13.9	49.9	21453	54.7	1468					
Ammonia				22.5	9690							
Hydrogen	0.090	2.55	39.4	141.7	60920	12.7	341	33.3	120.0	51591	10.8	290
Methane	0.716	20.3	15.4	55.5	23874	39.8	1069	13.9	50.0	21496	35.8	964
Natural gas (US market)*	0.777	22.0	14.5	52.2	22446	40.6	1090	13.1	47.1	20262	36.6	983
Town gas						18.0	483					
	@15°C/60°F, 1 bar											
Liquid fuels	[kg/l]	[g/gal]	[kWh/kg]	[MJ/kg]	[Btu/lb]	[MJ/l]	[Btu/gal]	[kWh/kg]	[MJ/kg]	[Btu/lb]	[MJ/l]	[Btu/gal]
Acetone	0.787	2.979	8.83	31.8	13671	25.0	89792	8.22	29.6	12726	23.3	83580
Butane	0.601	3.065	13.64	49.1	21109	29.5	105875	12.58	45.3	19475	27.2	97681
Butanol	0.810		10.36	37.3	16036	30.2	108359	9.56	34.4	14789	27.9	99934
Diesel fuel*	0.846	3.202	12.67	45.6	19604	38.6	138412	11.83	42.6	18315	36.0	129306
Dimethyl ether (DME)	0.665	2.518	8.81	31.7	13629	21.1	75655	8.03	28.9	12425	19.2	68973
Ethane	0.572	2.165	14.42	51.9	22313	29.7	106513	13.28	47.8	20550	27.3	98098
Ethanol (100%)	0.789	2.987	8.25	29.7	12769	23.4	84076	7.42	26.7	11479	21.1	75583
Diethyl ether (ether)	0.716	2.710	11.94	43.0	18487	30.8	110464					
Gasoline (petrol)*	0.737	2.790	12.89	46.4	19948	34.2	122694	12.06	43.4	18659	32.0	114761
Gas oil (heating oil)*	0.84	3.180	11.95	43.0	18495	36.1	129654	11.89	42.8	18401	36.0	128991
Glycerin	1.263	4.781	5.28	19.0	8169	24.0	86098					
Heavy fuel oil*	0.98	3.710	11.61	41.8	17971	41.0	146974	10.83	39.0	16767	38.2	137129
Kerosene*	0.821	3.108	12.83	46.2	19862	37.9	126663	11.94	43.0	18487	35.3	126663
Light fuel oil*	0.96	3.634	12.22	44.0	18917	42.2	151552	11.28	40.6	17455	39.0	139841
LNG*	0.428	1.621	15.33	55.2	23732	23.6	84810	13.50	48.6	20894	20.8	74670
LPG*	0.537	2.033	13.69	49.3	21195	26.5	94986	12.64	45.5	19561	24.4	87664
Marine gas oil*	0.855	3.237	12.75	45.9	19733	39.2	140804	11.89	42.8	18401	36.6	131295
Methanol	0.791	2.994	6.39	23.0	9888	18.2	65274	5.54	19.9	8568	15.8	56562
Methyl ester (biodiesel)	0.888	3.361	11.17	40.2	17283	35.7	128062	10.42	37.5	16122	33.3	119460
MTBE	0.743	2.811	10.56	38.0	16337	28.2	101244	9.75	35.1	15090	26.1	93517
Oils vegetable (biodiesel)*	0.92	3.483	11.25	40.5	17412	37.3	133684	10.50	37.8	16251	34.8	124772
Paraffin (wax)*	0.90	3.407	12.78	46.0	19776	41.4	148538	11.53	41.5	17842	37.4	134007
Pentane	0.63	2.385	13.50	48.6	20894	30.6	109854	12.60	45.4	19497	28.6	102507
Petroleum naphtha*	0.725	2.745	13.36	48.1	20679	34.9	125145	12.47	44.9	19303	32.6	116819
Propane	0.498	1.885	13.99	50.4	21647	25.1	89963	12.88	46.4	19927	23.1	82816
Residual oil*	0.991	3.752				41.8	150072	10.97	39.5	16982	39.2	140470

Table J.1: Heating values for common fuels [19]

Appendix K - Engine parameters and usefull information

Rpm	Indicated Power (W)	Indicated Horse Power (Hp)
0	0	0
1000	594.47	0.8
2000	1188.94	1.59
3000	1783.41	2.39
4000	2377.89	3.19
5000	2972.36	3.99
6000	3566.83	4.78
7000	4161.3	5.58
8000	4755.77	6.38

Table K.1: Expected Indicated Power

Rpm	Brake Power (W)	Brake Horse Power (Hp)
0	0	0
1000	416.13	0.56
2000	832.26	1.12
3000	1248.39	1.67
4000	1664.52	2.23
5000	2080.65	2.79
6000	2496.78	3.35
7000	2912.91	3.91
8000	3329.04	4.46

Table K.2: Expected Brake Power

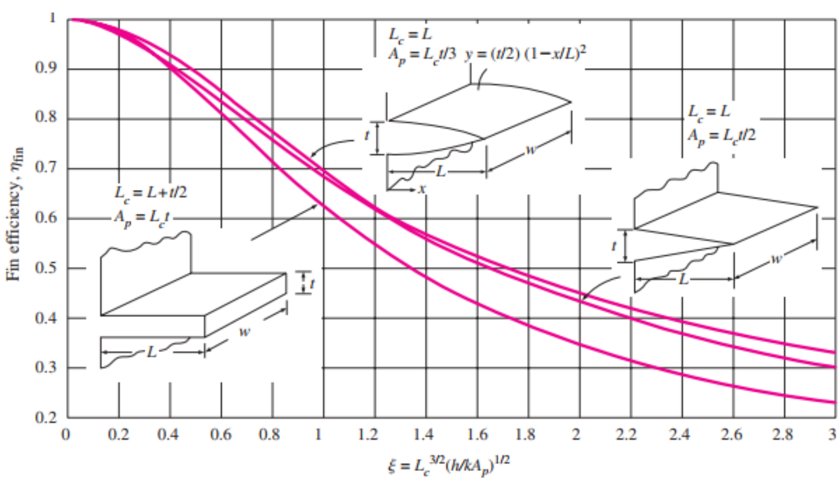


Figure K.1: The efficiency of straight fins of rectangular, triangular and parabolic profiles. Taken from [23]

D	75	100	150	200	250	300	350	400	450	500
C	1.5	2.4	4.0	6.3	8.0	9.5	11.0	12.5	12.5	12.5

(*Note: D* and *C* are in mm)

Table K.3: Reboring allowance for IC engine cylinder, taken from [21]

Appendix L - 3D models of the existing and designed components assembled

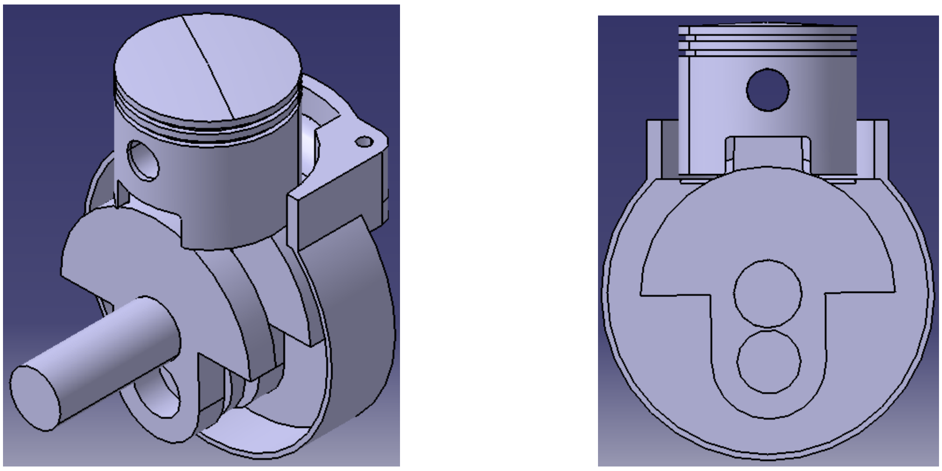


Figure L.1: Existing components assembled, 3D model

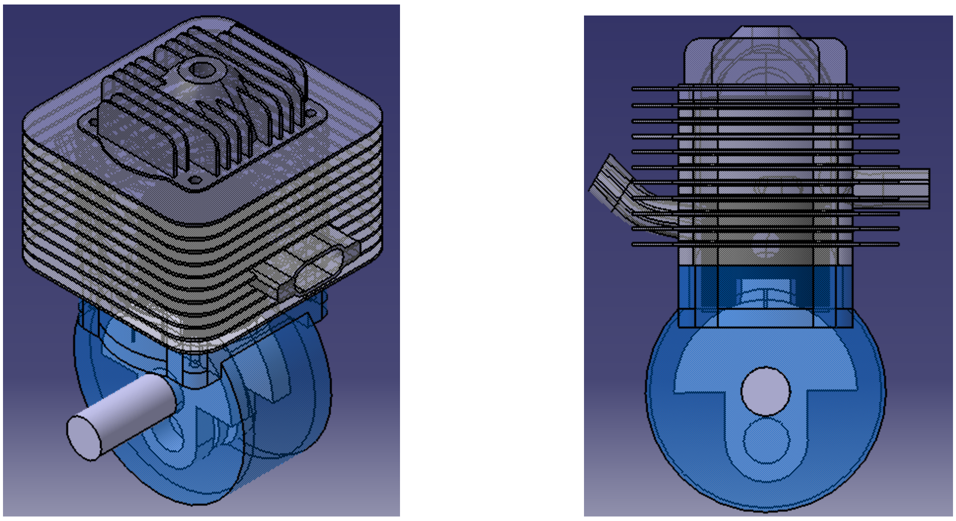


Figure L.2: All the components assembled, 3D model