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Supercharging of a 4-stroke Junkers engine

(Versão final após defesa)

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Dissertação para obtenção do Grau de Mestre em
Engenharia Aeronáutica
(mestrado integrado)

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novembro de 2023

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Dedication

Dedicated to my family.

Acknowledgements

First of all, I would like to thank all my family for their support during the hard times this journey has provided. A special thank you to my father José, my mother Sónia and my brothers Filipe and João.

Big appreciation to Professor Dr. Francisco Brójo, for the motivation he conveyed to me and the help he provided during the adversities encountered throughout this thesis. Always supported my ideas and guided me towards greater knowledge in the area.

A big thank you to engineer João Antunes who with his experience and knowledge of engines and manufacturing processes, helped me make all the necessary components and provided essential assistance during the tests. Also, to engineer Pedro Oliveira who always showed interest in helping me.

I also want to mention all my friends who accompanied me during these years. All the stories we have been through will be remembered for a long time. A special mention to Silveira, Pinho, Samu and Nuno, who were always there since the beginning, to A., a special person to me and to PES.

Lastly, I would like to appreciate the financial help from Grupo AFA that allowed me to finish this chapter of my life as well as the support from Câmara Municipal da Calheta.

Resumo

No campo da aviação, a sobrealimentação permite que os motores a pistão desenvolvam a potência máxima quando operam a grandes altitudes ou aumentem a potência de uma aeronave durante a decolagem. Um motor naturalmente aspirado, sem um compressor, perderá potência a grandes altitudes devido à densidade reduzida do ar que entra no sistema de admissão. Este projeto tem como objetivo sobrealimentar um motor de pistões opostos (OPE) desenvolvido por Jorge Gregório utilizando o turbocompressor RHB31 VZ21 para melhorar o seu desempenho. O UBI/UDI – OPE-BGX286 é um OPE monocilíndrico de ignição por faísca a 4 tempos com uma configuração de cambota dupla semelhante ao Junkers Jumo 205 operando numa posição horizontal. Um novo sistema de escape e um sistema de admissão contendo um capacitador de ar foram projetados e fabricados para permitir a sobrealimentação do motor. Um dinamômetro de correntes de Eddy e vários sensores conectados à unidade de controlo do motor foram utilizados para medir os parâmetros do motor durante os testes de aspiração natural e de sobrealimentação. O desempenho do motor foi quantificado pelos valores de binário, potência e consumo específico de combustível. Ao longo dos testes foram encontrados vários problemas que impossibilitaram o funcionamento correto do motor, sendo explicadas as modificações feitas para os retificar. São apresentadas então possíveis razões para o insucesso do estudo.

Palavras-chave

Sobrealimentação; Projeto; Fabricação; 4-tempos; Ignição por Faísca; Motor de Combustão Interna; Junkers

Abstract

In the field of aviation, supercharging allows piston engines to develop maximum power when operating at high altitudes or boost an aircraft's power during take-off. A naturally aspirated engine, without a supercharger, will lose power at high altitudes because of the reduced density of the air entering the intake system. This project aims to supercharge an opposed piston engine (OPE) developed by Jorge Gregório using the RHB31 VZ21 turbocharger to improve its performance. The UBI/UDI – OPE-BGX286 is a single cylinder 4-stroke spark ignition OPE with a twin crankshaft configuration similar to the Junkers Jumo 205 operating in a horizontal position. A new exhaust system and an intake system containing an air capacitor were designed and fabricated to allow the supercharging of the engine. An Eddy current dynamometer and various sensors connected to the engine control unit were utilised to measure the engine parameters during the natural aspiration and supercharging tests. The engine performance was quantified by the values of torque, power and specific fuel consumption. Several problems were found throughout the tests that made it impossible for the engine to function correctly and the modifications made to rectify them are explained. Possible reasons for the failure of the study are then presented.

Keywords

Supercharging; Design; Fabrication; 4-Stroke; Spark Ignition; Internal Combustion Engine; Junkers

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List of Acronyms

AEO	Advance to Exhaust Opening
AFR	Air/Fuel Ratio
AIO	Advance to Intake Opening
BDC	Bottom Dead Centre
BSFC	Brake Specific Fuel Consumption
CAD	Computer-Aided Design
CNC	Computer Numerical Control
CI	Compression Ignition
DOHC	Double Over Head Camshaft
ECD	Exhaust Closing Delay
ECU	Engine Control Unit
IA	Ignition Advance
IAT	Intake Air Temperature
ICD	Intake Closing Delay
ICE	Internal Combustion Engine
ISFC	Indicated Specific Fuel Consumption
LHV	Lower Heating Value
MAP	Manifold Air Pressure
MBT	Maximum Brake Torque
MIG	Metal Inert Gas
MS2	MegaSquirt 2
NA	Naturally Aspirated
OHC	Over Head Camshaft
OHV	Over Head Valves
OPE	Opposed Piston Engine
PVC	Polyvinyl Chloride
PTO	Power Take-Off
RPM	Revolutions Per Minute
SC	Supercharged
SFC	Specific Fuel Consumption
SI	Spark Ignition
TDC	Top Dead Centre
TPS	Throttle Position Sensor

UBI	Universidade da Beira Interior
WOT	Wide Open Throttle
WWII	Word War II

Nomenclature

a	Crank Radius	[m]
A	Cross-sectional Area	[m²]
AFR	Air/Fuel Ratio	
AFR_stoich	Stoichiometric Air/Fuel Ratio	
b	Distance from axis	[m]
B	Bore	[m]
$bmep$	Brake Mean Effective Pressure	[Pa]
$bsfc$	Brake Specific Fuel Consumption	[g/kW.h]
DT	Duty Cycle	
F	Net Load	[N]
fc	Fuel Consumption	[kg/h]
$isfc$	Indicated Specific Fuel Consumption	[g/kW.h]
l	Connecting Rod Length	[cm]
LHV	Lower Heating Value	[MJ/kg]
m_a	Mass of Air	[g]
$\dot{m}_a$	Mass Flow Rate of Air	[g/s]
m_f	Mass of Fuel	[g]
$\dot{m}_f$	Mass Flow Rate of Fuel	[g/s]
$\dot{m}_{inj}$	Mass Flow Rate of Injector	[cc/min]
n	Revolutions per Cycle	
N	Engine Speed	[rpm]
N_c	Number of Cylinders	
p	Pressure	[Pa]
P	Power	[W]
P_b	Brake Power	[W]
P_i	Indicated Power	[W]
P_s	Specific Power Output	[W/l]
PW	Pulse Width	
r_c	Compression Ratio	
S	Stroke Length	[m]
sfc	Specific Fuel Consumption	[g/kW.h]
T	Torque	[N.m]

$\bar{U}_p$	Mean Piston Speed	[m/s]
v	Flow Velocity	[m/s]
V	Volume	[cm^3]
V_{BDC}	Cylinder Volume	[cm^3]
V_c	Clearance Volume	[cm^3]
V_d	Displacement	[cm^3]
V_{TDC}	Clearence Volume	[cm^3]
W	Work	[J]
W_b	Brake Work	[J]
W_f	Parasite Work	[J]
W_g	Gross Work	[J]
W_i	Indicated Work	[J]
W_n	Net Work	[J]
W_p	Pumping Work	[J]
η	Overall Performance	
η_m	Mechanical Efficiency	
η_V	Volumetric Efficiency	
θ	Crank Angle	[^{o}]
λ	Air/Fuel Equivalence Ratio	
ρ	Density of Fluid	[g/m^3]
ρ_a	Intake Air Density	[g/m^3]
ω	Engine Speed	[rad/s]

Chapter 1 – Introduction

1.1 Motivation

For several decades, a piston engine coupled with propellers provided the necessary power for aircraft. Aero piston engines were the primary powerplant used in aeronautics until the 1930's, however, they continue to power small aircraft and helicopters, as nearly 80% are powered by piston engines. [1]

Aero piston engines provide a good amount of power relative to their weight, which is essential for smaller, lightweight aircraft where engine weight can significantly impact performance. They are generally more cost-effective to manufacture and maintain than other type of engines such as jets. Small aircraft typically operate shorter distances and at lower airspeeds compared to larger commercial jets, making piston engines a reasonable choice. These engines are more fuel-efficient for lower-speed, short-distance flights and are designed to use fuels readily available at most airports. All of this makes the aero piston engine a more attractive choice for smaller aircraft operators which often have budget constraints and limited maintenance resources.

Some of the most successful aero piston engines are the Junkers engines. They are opposed piston engines characterized by having a pair of pistons operating in opposite directions in a single cylinder, sharing the same combustion chamber. These engines established many standards (records) that currently exist regarding power-to-weight ratio, dynamic balance refinement, fuel tolerance, compactness, thermal efficiency, and manufacturing simplicity, making the concept viable for certain applications such as aviation. [2]

By supercharging an aero piston engine, the engine will develop maximum power when operating at high altitudes or boost an aircraft's power during takeoff. At high altitudes, an engine without a supercharger will lose power because of the reduced density of the air entering the induction system of the engine. [1] A supercharged (SC) engine will have the same performance as a naturally aspirated (NA) engine with a much larger displacement and consequently greater volume and mass. [3] This makes it an attractive option for small aircraft as the weight is an essential factor.

1.2 Objectives

The main objective of this dissertation is to modify a 4-stroke spark ignition Junkers engine by adding a turbocharger to improve performance. To achieve the main objective, some specific points were considered in favour of better work organization, results and understanding of engine operation. These points can be listed as:

- Understand the operation of 4-stroke Junkers engines;
- Design changes to add a turbocharger;
- Build components necessary for the change;
- Carry out preliminary tests on the engine and rectify possible problems;
- Test the engine to quantify performance;
- Write the dissertation to present the discussion;
- Write a scientific article summarizing the description of the state of art, methodology and discussion of results.

1.3 Dissertation Structure

This dissertation is divided into five main chapters.

The current chapter is constituted by the motivation behind this project as well as its objectives.

The second chapter is dedicated to the state of art. In this chapter, the fundamentals of internal combustion engines are covered. First, a general approach to internal combustion engines is taken into account. Afterwards, a more specific approach is made to the 4-stroke engine operation, components, supercharging and performance characteristics. At the end, a brief review of Junkers engines is made.

In chapter three, the methodology behind this project is described. It is presented some options that were considered for designing the necessary components to supercharge the engine as well as the fabrication methods and materials used in the manufacturing process of the components.

Chapter four consists of the practical case. All the components used in this project are mentioned and the newly designed ones are presented with their specifications in terms of dimensions and how they were fabricated. The experimental procedure is described and some tests and problems that were encountered are shown, as well as all the changes made to the engine to overcome those setbacks.

The final chapter presents the main conclusions of the project, the possible reasons for its failure and some future works that can be done to improve the quality of the study.

Chapter 2 – State of Art

2.1 Reciprocating Internal Combustion Engines

The reciprocating internal combustion engine (ICE), also called piston engine, is a thermal engine that converts chemical energy contained in the fuel into mechanical energy. In internal combustion engines, as distinct from external combustion engines, this energy is released by burning or oxidizing the fuel inside the engine. [1] This causes the conversion of the chemical energy into thermal energy, raising the temperature and pressure of the gases within the engine. The high-pressure gases expand against a moving displacer (piston), converting the expansion into mechanical energy by a reciprocating motion (repetitive up-and-down or back-and-forth linear motion) that is, afterwards, turn into a rotational motion by a mechanical component connected to the piston (crankshaft).

The occurrence of a periodically changing working chamber as a result of the motion of the piston is characteristic of the manner of operation of this type of engine. As driven machines, they absorb mechanical energy to increase the energy of the conveyed fluid. On the other hand, as drive machines, mechanical energy is released in the form of useful work at the piston or the crank mechanism. They are thus part of the category of fluid energy machines. [1]

The fuel-air mixture before combustion and the burned products after combustion are the actual working fluids. The work transfers which provide the desired power output occur directly between these working fluids and the mechanical components of the engine. [2]

The engine is constituted by cylinders, inside which slide pistons are connected to a crankshaft by the connecting rods. The steady rotation of the crank produces a cyclical piston motion. If we rotate the crankshaft, the pistons go up and down in the several cylinders. The piston subjected to high pressure turns the crankshaft. So that the engine doesn't stop when a piston is compressing air in the cylinder, or so that it doesn't have an irregular operation, one end of the crankshaft is equipped with a flywheel, that conserves the kinetic energy. [3]

All the ICE's have the same basic geometry represented in Figure 2.1:

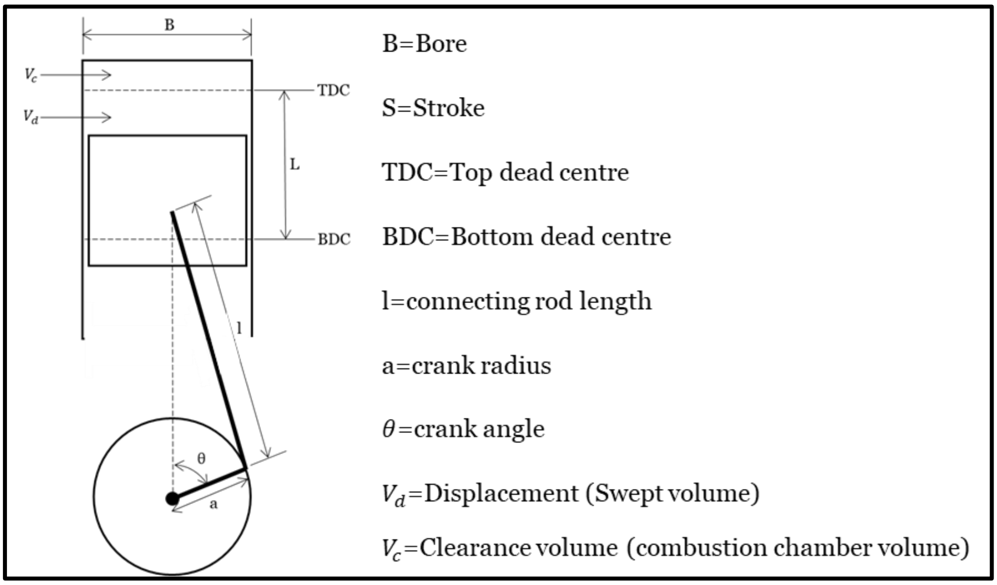


Figure 2.1: ICE piston and cylinder geometry

The bore of the cylinder is its diameter. The TDC and BDC are, respectively, the highest and lowest points the piston can reach inside the cylinder and this distance is called stroke. When the piston is at TDC, the volume inside the cylinder is minimum (clearance volume) and when it is at BDC, the volume is maximum (cylinder volume). The clearance volume is also called combustion chamber because it is in this volume that combustion begins. Displacement corresponds to the volume swept by the piston as it descends from TDC to BDC. [3]

The huge number of different applications, designs and characteristics allow us to classify the ICE's in several parameters:

- Application
  - Automobile, aircraft, locomotive, marine, etc.
- Basic engine design (Figure 2.2)
  - Single cylinder/piston;
  - In-line – cylinders are positioned in a straight line, one behind another;
  - V-block – two banks of cylinders with an angle between them from 15º to 120º;
  - Boxer – two banks of cylinders opposite to each other;
  - W-block – same as V-block engine but with three banks of cylinders;
  - Radial – the pistons are positioned in a circular plane;
  - Opposed piston – two pistons in each cylinder with the combustion chamber between them;

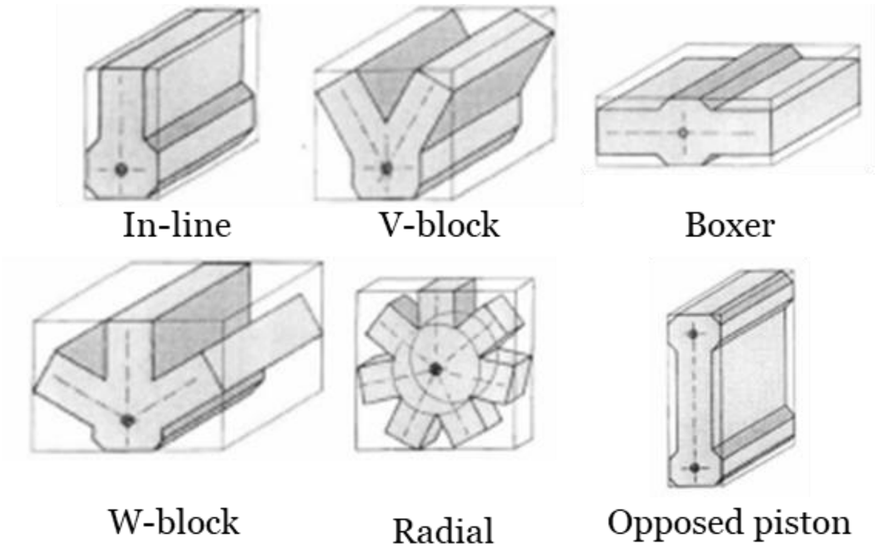
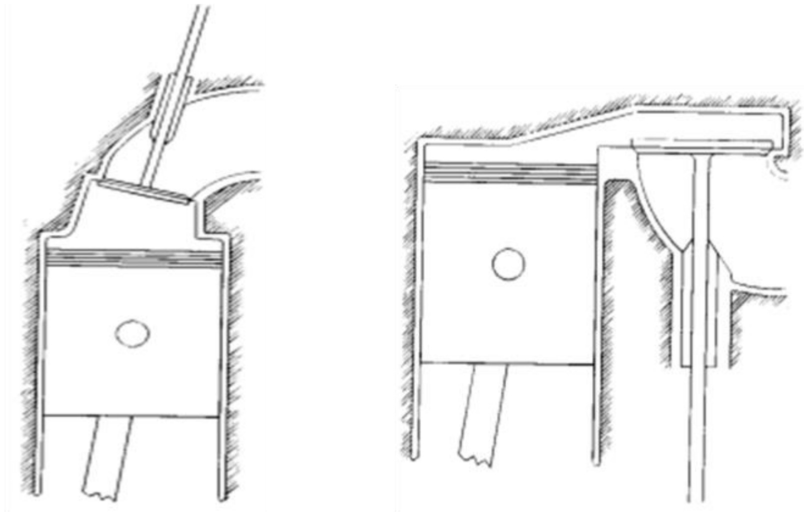


Figure 2.2: Cylinder arrangements in ICE's [5]

- Working cycle
  - 4-stroke – four piston movements over two engine revolutions for each cycle;
  - 2-stroke – two piston movements over one revolution for each cycle;
- Charging system
  - Naturally aspirated – the fresh charge is drawn into the cylinder by the working piston;
  - Mechanically supercharged – the compressor is driven directly by the engine;
  - Turbo-supercharged/turbocharged– a turbine powered by the engine exhaust drives the compressor;
- Fluid inlet-outlet control (Figure 2.3)
  - Valves in head (I-head)
    - Overhead valve (OHV) - valves are built in the cylinder head and the camshaft is near the cylinder;
    - Overhead camshaft (OHC) or Double Overhead Camshaft (DOHC) - valves and camshaft are built in the cylinder head;
  - Valves in block (L-head);
  - Rotary valves;
  - Cross-scavenged porting (inlet and exhaust ports on opposite sides of the cylinder at one end);
  - Loop-scavenged porting (inlet and exhaust ports on the same side of the cylinder at one end);
  - Through/uniflow scavenged (inlet and exhaust ports or valves at different ends of the cylinder);



Valves in head (I-head) Valves in block (L-head)

Figure 2.3: Valves arrangement in ICE's [6]

- Fuel used
    - Liquid fuels (Gasoline, Diesel, Kerosene, etc.);
    - Gaseous fuels (Natural gas, Biogas);
    - Solid fuels (Pulverized coal);
- Air-fuel mixing methods
    - Carburettor;
    - Intake port or intake manifold injection;
        - Single-point;
        - Multi-point;
    - Direct injection into the working chamber;
- Method of ignition
    - Spark ignition (SI) - An electrical spark ignites the mixture in the cylinder;
    - Compression ignition (CI) - The fuel injected ignites spontaneously in the air heated by compression in the cylinder;
- Cooling systems
    - Air cooling;
    - Water cooling;

2.2 4-Stroke Engine Cycle

One of the most important methods of classification of ICE's is the principle of operation. Most of them, both SI or CI engines, operate on either a 4-stroke or a 2-stroke cycle.

In a 4-stroke ICE, the piston executes four distinct strokes within the cylinder for every two revolutions of the crankshaft to complete the sequence of events which produces one power stroke.

Since the cycle is completed only once every two revolutions, the valve gear (and fuel injection equipment) has to be driven by mechanisms operating at half engine speed. Some of the power from the expansion stroke is stored in a flywheel, to provide the energy for the other three strokes. [4]

Both SI and CI 4-stroke engines use this cycle (Figure 2.*4*):

1. Intake Stroke
   a. Starts with the piston at TDC and ends with the piston at BDC, increasing the volume in the cylinder, which in turn creates a vacuum. The resulting pressure differential through the intake system from atmospheric pressure on the outside to the vacuum on the inside draws fresh charge into the cylinder. To increase the mass inducted, the intake valve opens shortly before the stroke starts and closes after it ends. The exhaust valve is closed at this stage. [5]
2. Compression Stroke
   a. When the piston reaches BDC, the intake valve closes and the piston travels back to TDC with all valves closed. The charge inside the cylinder is compressed to a small fraction of its initial volume, raising the temperature and pressure in the cylinder. [2] Combustion is then initiated, beginning near the end of the compression stroke and continuing through the first part of the expansion. This combustion process results in a high-pressure, high-temperature gas mixture (exhaust products). [6]
3. Power Stroke
   a. Starts with the piston at TDC and ends at BDC as the exhaust products expand and push the piston down, with all valves closed, and force the crank to rotate. At the end of the power stroke, the exhaust valve opens and exhaust blowdown occurs. Pressure and temperature in the cylinder are still high relative to the surroundings at this point, and a pressure differential is created through the exhaust system which is open to atmospheric pressure. This pressure differential causes much of the hot exhaust gas to be pushed out of the cylinder and through the exhaust system. [5]
4. Exhaust Stroke
   a. The exhaust valve remains open and the piston travels from BDC to TDC, expelling the remaining gases out of the cylinder into the exhaust system. As

the piston approaches TDC the intake valve starts to open, so that it is fully open by TDC when the new intake stroke starts the next cycle. Just after TDC the exhaust valve closes, and the cycle starts again. This period when both the intake valve and exhaust valve are open is called valve overlap. [5]

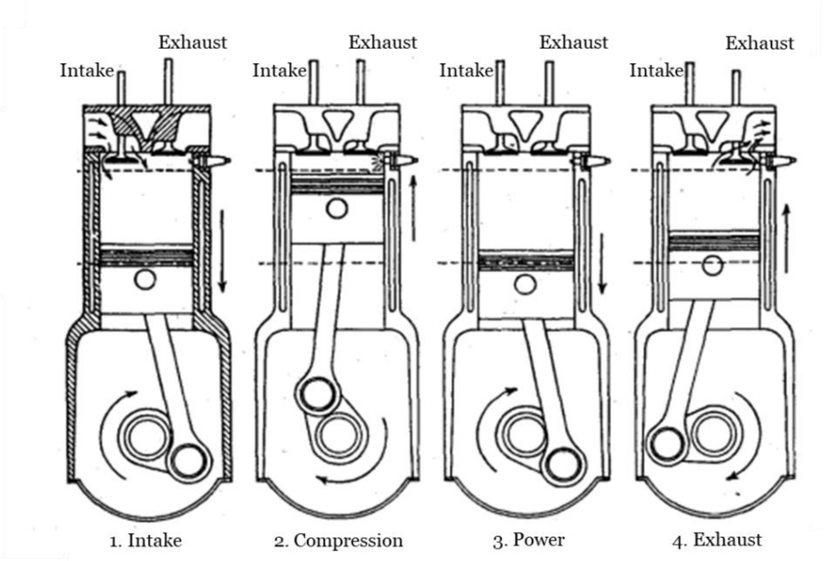


Figure 2.4: 4-stroke engine cycle [4]

2.2.1 SI Engine

In SI engines the air and fuel are usually mixed in the intake system before entering the engine cylinder using a carburettor or a fuel-injection system. Several different types of injection systems are plotted against crank angle through the entire 4-stroke cycle. Crank angle is a useful independent variable because engine processes occupy almost constant crank angle intervals over a wide range of engine speeds. [2]

The valves don´t open or close coincidently with the TDC or BDC. During the intake stroke, the intake valve opens before TDC and remains open for some time after BDC to maintain high mixture flows at high engine speeds (and hence high power outputs). [2]

The intake closing delay (ICD) causes the inertia of the gas column to allow the mixture to enter during the first part of the piston's ascent. The high air velocities at the cylinder inlet create inertia, causing the mixture to continue to enter the cylinder even if its pressure is higher. If the intake manifold is at atmospheric pressure, the filling pressure can be much higher, that is, it's possible to admit a mass of air greater than what would fit in the cylinder at atmospheric pressure. [3]

The advance to intake opening (AIO) serves so that the inertia of the exhaust gases still exiting at high speed through the respective valve, causes the aspiration of the intake air, washing the cylinder from the exhaust gases. [3]

After the intake valve closes (both valves closed), the mixture is compressed to above atmospheric pressure and temperature as the cylinder volume is reduced by the upward movement of the piston.

Between 10 and 40 crank angle degrees before TDC, an electrical discharge across the spark plug starts the combustion process. [2] The ignition advance (IA) serves for the combustion of the mixture to take place in such a way that the maximum pressure of the cycle occurs after TDC, to crease the useful pressure component. [3] A turbulent flame develops from the spark discharge, propagates across the mixture in the cylinder, and extinguishes at the combustion chamber wall. As the fuel-air mixture burns, the cylinder pressure rises above the level due to compression alone. [2] It is also important that combustion is completed as quickly as possible.

There is an optimum spark timing which, for a given mass of fuel and air inside the cylinder, gives maximum torque. More advanced (earlier) timing or retarded (later) timing than this optimum gives lower output. Called maximum brake torque (MBT) timing, this optimum timing is an empirical compromise between starting combustion too early in the compression stroke and completing combustion too late in the power stroke. [2]

About two-thirds of the way through the power stroke, long before the piston reaches BDC, the opening of the exhaust valve takes place. The cylinder pressure is greater than the exhaust manifold pressure and a blowdown process occurs. The burned gases flow through the valve into the exhaust port and manifold until the cylinder pressure and exhaust pressure equilibrate. The piston then displaces the burned gases from the cylinder into the manifold during the exhaust stroke. [2]

The advance to exhaust opening (AEO) has the function of allowing a large part of the burnt gases to escape the cylinder as soon as possible, so that the blowdown process does not last too far into the exhaust stroke. Thus, the piston doesn't encounter much resistance during the upward stroke of the exhaust and not many pumping losses incur. [3]

The valve closes after TDC (exhaust closing delay (ECD)), to allow the combustion chamber to be completely flushed. When closing the exhaust valve is important to have a negative pressure wave at this location, to remove as much burnt gas as possible, so that they don't escape into the intake manifold and to facilitate the entry of the fresh charge. [3]

The distribution diagram of a 4-stroke engine is shown in Figure 2.5.

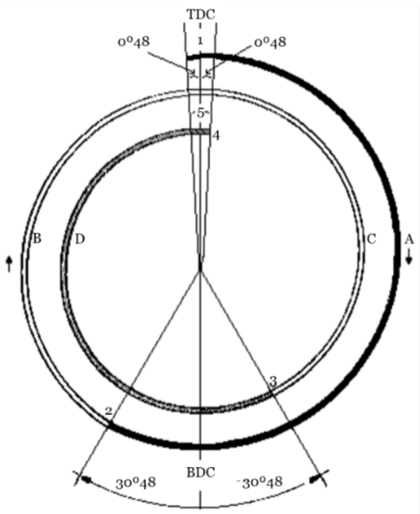


Figure 2.5: Distribution diagram of a 4-stroke engine [8]

2.2.2 Otto Cycle

The actual thermodynamic and chemical processes in internal combustion engines are too complex for complete theoretical analysis. Under these circumstances, it is useful to imagine a process which resembles the real process in question but is simple enough to lend itself to easy quantitative treatment. [8]

For SI engines the equivalent air cycle is usually taken as the constant volume air cycle or Otto cycle whose p-V diagram is shown in Figure 2.6. At the start of the cycle, the cylinder contains a mass of air at the atmospheric pressure p_0 and volume indicated by state 1 after an isobaric intake (6-1). The piston is then moved upward, and the gas is compressed reversibly and adiabatically (isentropic process) to state 2. Next, heat is added at constant volume to increase the pressure to 3. Isentropic expansion takes place to the original volume at state 4, and the medium is then cooled at constant volume and returns to its original pressure at state 5 (exhaust blowdown). Finally, an isobaric exhaust occurs while the piston goes from BDC to TDC (5-6). [9]

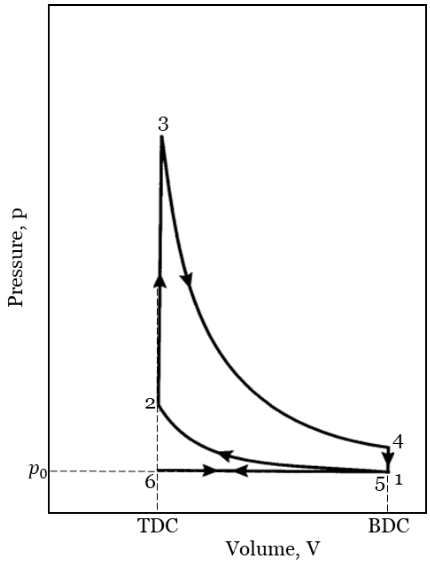


Figure 2.6: p-V diagram of an Otto cycle [6]

2.3 Engine Components

The ICE is constituted by different components with specific roles for the proper functioning of the engine (Figure 2.7). These components can be made of different materials considering the forces and range of temperatures to which it's subjected as well as their role in the functioning of the engine.

The engine cylinders are where the pistons reciprocate back and forth and are contained in the engine block. The walls of the cylinder have highly polished hard surfaces. [5] The block has traditionally been made of grey cast iron because of its good wear resistance and low cost, despite some smaller SI engine blocks being made of aluminium to reduce the weight, and passages for the cooling water are cast into it. [2]

The crankcase is often integral to the engine block and surround crankshaft. The crankshaft is a rotating shaft, supported by main bearings, through which engine work output is supplied to external systems and has traditionally been a steel forging, even though some are made of cast iron. [5] The connecting rod big-end bearings attach to the crank pin on each throw. Both main and connecting rod bearings use steel backed precision inserts with bronze, babbitt, or aluminium as the bearing materials. The crankcase is sealed at the bottom with a pressed-steel or cast aluminium oil pan which acts as an oil reservoir for the lubricating system. [2]

Pistons are cylindrical-shaped masses that reciprocate back and forth in the cylinder and are made of aluminium in small engines or cast iron in larger slower-speed engines. The piston both seals the cylinder and transmits the combustion-generated gas pressure to the crank pin via the connecting rod. The connecting rod, usually a steel or alloy forging (though sometimes aluminium in small engines) is fastened to the piston using a steel piston pin (wrist pin) that is usually hollow to reduce its weight. [2]

The oscillating motion of the connecting rod exerts an oscillating force on the cylinder walls via the piston skirt (the region below the piston rings). The piston is fitted with rings (piston rings) which ride in grooves cut into the piston head to seal against gas leakage and control oil flow. [2] The upper rings are compression rings made of highly polished hard chrome steel. The purpose of these is to form a seal between the piston and cylinder walls and to restrict the high-pressure gases in the combustion chamber from leaking past the piston into the crankcase (blowby). [5] The lower rings assist in lubricating the cylinder walls and scraping the surplus oil from the cylinder wall and returning it to the crankcase. [2]

The cylinder head seals off the cylinders and is made of cast iron or aluminium. It must be strong and rigid to distribute the gas forces acting on the head as uniformly as possible through the engine block. [2] The cylinder head contains the spark plug (an electrical device used to initiate combustion in an SI engine by creating a high-voltage discharge across an electrode gap) or fuel injector (a pressurized nozzle that sprays fuel into the incoming air on SI engines or the cylinder on CI engines), and in overhead valve engines, parts of the valve mechanism. [5]

Valves are made from forged alloy steel and are used to allow flow into and out of the cylinder at the proper time in the cycle. [5] The valve type normally used in 4-stroke engines is the poppet valve, which is spring loaded, closed and pushed open by camshaft action. [7] Most modern SI engines have overhead valve locations, leading to a compact combustion chamber with minimum heat losses and flame travel time and improving breathing capacity. The valve stem moves in a valve guide, which can be an integral part of the cylinder head (or engine block) or may be a separate unit pressed into the head (or block). A valve spring, attached to the valve stem, holds the valve closed. [2]

A camshaft is a rotating shaft used to open or close valves at the proper time in the engine cycle, either directly or through mechanical or hydraulic lifters (push rods, rocker arms, tappets) sliding in the block. [5] Camshafts are made of cast iron or forged steel with one cam per valve and are gear, belt, or chain driven from the crankshaft. [7] In 4-stroke cycle engines, camshafts turn at half the crankshaft speed. [2]

An intake manifold (a piping system which delivers incoming air to the cylinders), usually made of cast metal, plastic, or composite material, and an exhaust manifold (a piping system which carries exhaust gases away from the engine cylinders), generally of cast iron, complete the SI engine assembly. Another engine component specific to SI engines is the carburettor, a venturi flow device which meters the proper amount of fuel into the air flow by means of a pressure differential, which can be replaced by a fuel injection system. [5]

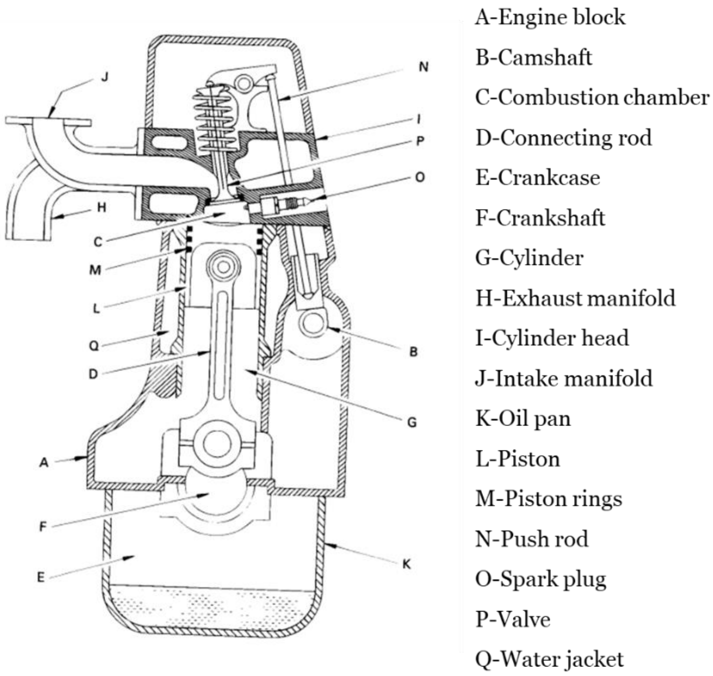


Figure 2.7: Cross-section of a 4-stroke SI engine [6]

2.4 Supercharging

The power that can be extracted from an engine is limited by the amount of fuel that can be burned efficiently within the combustion chamber. This is limited by the amount of air being introduced into the cylinder each cycle. [8] One of the ways to increase the performance of an engine is to increase the amount of air allowed in each cycle. This can be achieved by increasing the inlet air pressure (thus increasing its density), in a process called supercharging. [3] This increases the fuel that can be burned and hence raises the potential power output.

The origin of supercharging was due to deficiencies felt in aircraft engines when flying at altitude and not in an attempt to improve their performance. [3] In the field of aviation, supercharging allows piston engines to develop maximum power when operating at high altitudes or boost an aircraft's power during takeoff. At high altitudes, an engine without a supercharger will lose power due to the drop in pressure and reduced density of the air entering the induction system. Superchargers have compressors mounted in the intake system used to raise the pressure and density of the incoming air. [1] A supercharged engine will have the same performance as a naturally aspirated one with a much larger displacement and consequently greater volume and mass. [3] The thermodynamic effect of supercharging can be seen from a p-V diagram in Figure 2.8.

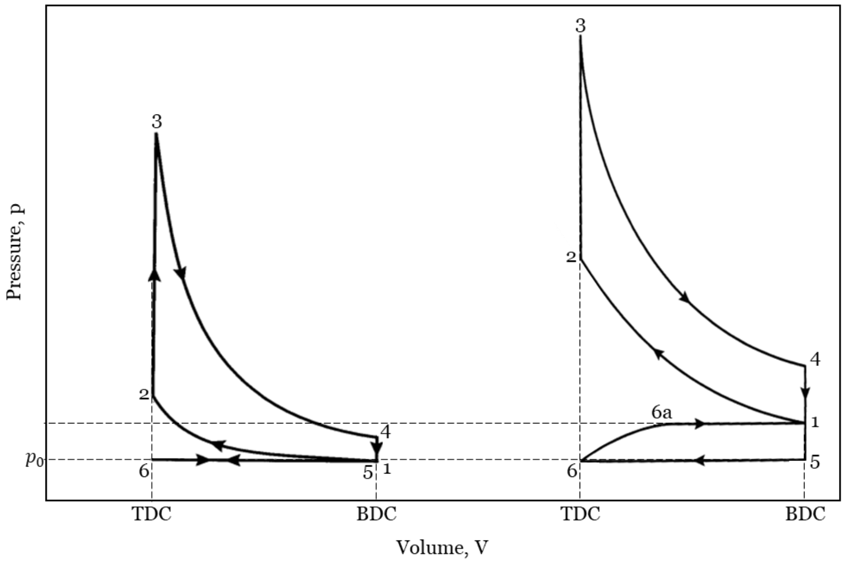


Figure 2.8: p-V diagram of a NA engine (left) and a SC engine (right) [6]

Superchargers and turbochargers are two types of forced induction employed in ICE's to add large amounts of power to their engine. The key difference between a turbocharger and a supercharger is the power supply. Superchargers are mechanically driven directly off the engine through belts, gears, shafts, or chains connected to the engine's crankshaft. [1] On the other hand, the engine and the turbocharger are linked thermodynamically and not mechanically. [5]

Turbocharging is a particular form of supercharging in which a compressor is driven by an exhaust gas turbine. [6] This turbine is mounted in the exhaust flow of the engine, which, in turn, spins the compressor element situated between the air intake and the engine. [1] The compressor and the turbine are placed on the same shaft which rotates at very high speeds. Generally, the exhaust gases enter the turbine radially and exit axially, the reverse happening in the compressor. So that the turbocharger does not reach high speeds, producing very high pressures, many are fitted with a wastegate valve that diverts part of the exhaust gases from passing through the turbine. It responds to the intake pressure, opening a bypass to the exhaust gases. [3]

In theory, a turbocharger is more efficient as it uses the "wasted" energy in the exhaust stream for its power source. [1] However, the turbine in the exhaust causes a more restricted flow and a certain amount of back pressure in the exhaust system, resulting in a slightly higher pressure at the engine exhaust port and tends to provide less boost until the engine is running at higher RPMs. A disadvantage of turbochargers is turbo lag, which occurs with a sudden throttle change. It takes several engine revolutions to change the exhaust flow rate and to speed up the rotor of the turbine. [7]

With the turbocharged SI engines, the higher charge pressure results in higher ultimate compression temperatures. This increases the risk of autoignition and of knocking, and may result in severe cylinder head and piston damage. The mixture temperature during compression must be kept below the self-ignition temperature of the fuel. For this reason, measures such as low compression ratio, retarded ignition timing or charge air cooling (intercooler), are used to offset the effect of the temperature rise in the compressor. [9]

2.5 Engine Performance Characteristics

2.5.1 Geometrical Properties

For an engine with bore B, crank radius a, stroke length S, turning at an engine speed of N or ω:

$$S = 2a \text{ [m]} \tag{1}$$

$$N = \frac{Number\ of\ cranckshaft\ revolutions}{Time} \text{ [rpm]} \tag{2}$$

$$\omega = \frac{2\pi N}{3600} \text{ [rad/s]} \tag{3}$$

An important characteristic speed is the mean piston speed $\bar{U}_p$:

$$\bar{U}_p = \frac{2SN}{60} \text{ [m/s]} \tag{4}$$

Mean piston speed allows us to judge whether the engine is intrinsically slow or fast and will normally be in the range of 5 to 15 m/s due to gas flow resistance into and out of the engine or stresses caused by the inertia of the moving parts. [2] This speed determines the velocity at which the gases enter the engine, regardless of whether the engine is large or small. [3]

Bore sizes of engines range from 50 to 0.5 cm. The ratio of bore to stroke, B/S, for small engines is usually from 0.8 to 1.2. An engine with $B = S$ is often called a square engine. If the stroke length is longer than the bore diameter, the engine is under square, and if the stroke length is less than the bore diameter, the engine is over square. [5]

The piston displacement or swept volume V_d for an engine cylinder is the distance travelled by the piston during one piston stroke from BDC to TDC.

$$V_d = V_{BDC} - V_{TDC}\ [\text{cm}^3] \tag{5}$$

For an engine with N_c cylinders:

$$V_d = N_c \left(\frac{\pi}{4}\right) B^2 S\ [\text{cm}^3] \tag{6}$$

Minimum cylinder volume occurs when the piston is at TDC and is called combustion chamber or clearance volume V_c.

$$V_c = V_{TDC}\ [\text{cm}^3] \tag{7}$$

When the piston is at BDC the cylinder volume is maximum and is called simply cylinder volume V.

$$V = V_{BDC} = V_c + V_d\ [\text{cm3}] \tag{8}$$

The compression ratio r_c is defined as:

$$r_c = \frac{maximum\ cylinder\ volume}{minimum\ cylinder\ volume} = \frac{V}{V_c} = \frac{V_d + V_c}{V_c} \tag{9}$$

Typical values of r_c for SI engines are between 8 and 12, limited by the knock and by autoignition, while for CI engines is between 12 and 24. [2] Supercharged engines usually have lower compression ratios than naturally aspirated engines. [5]

2.5.2 Torque and Power

The specification of an engine is usually given by the maximum torque and power values. Both are functions of engine speed.

Torque T is a good indicator of an engine's ability to do work and is measured by dynamometers. It is obtained by reading off a net load F at a known distance b from the axis of rotation. [7]

$$T = Fb\,[\text{N.m}] \tag{10}$$

Torque is related to work by:

$$2\pi T = W_b = \frac{(bmep)V_d \times 10^{-6}}{n} \text{ [J]} \tag{11}$$

The number of revolutions per cycle n is 2 for a 4-stroke engine and 1 for a 2-stroke engine. Therefore, for a 4-stroke engine:

$$T = \frac{(bmep)V_d \times 10^{-6}}{4\pi} \text{ [N.m]} \tag{12}$$

The usable power delivered by the engine to the load is called brake power and is the rate at which work is done.

$$P_b = \frac{2\pi NT}{60} \text{ [W]} \tag{13}$$

The specific power output P_s can be defined by the brake power per litre of displacement V_d.

$$P_s = \frac{P_b}{V_d \times 10^{-3}} \text{ [W/l]} \tag{14}$$

2.5.3 Air/Fuel Ratio

The energy input to an engine comes from the combustion of fuel. Air is used to supply the oxygen needed for this chemical reaction and for combustion to occur, the proper relative amounts of air (oxygen) and fuel must be present. Air/fuel ratio AFR is the parameter used to describe the mixture ratio. [5]

$$AFR = \frac{m_a}{m_f} = \frac{\dot{m}_a}{\dot{m}_f} \tag{15}$$

where m_a is mass of air, $\dot{m}_a$ is mass flow rate of air, m_f is mass of fuel, $\dot{m}_f$ is mass flow rate of fuel.

When the ratio is such that all the fuel is burned using all of the available air, the mixture is said to be stoichiometric (AFR_{stoich}). The evaluation of the mixture is made by the air/fuel equivalence ratio λ. [3]

$$\lambda = \frac{AFR}{AFR_{stoich}} \tag{16}$$

An engine to which more fuel is supplied than required by stoichiometry ($\lambda = 1$) is said to burn a mixture with an air deficiency ($\lambda < 1$) and is referred to as a rich mixture. The opposite happens when it burns a mixture with excess air ($\lambda > 1$) and is referred to as a lean mixture. [1]

2.5.4 Fuel Consumption

The fuel consumption is measured as a flow rate ($\dot{m}_f$) and is the fuel mass flow per unit time, but a more useful parameter is the specific fuel consumption sfc which is the fuel flow rate per unit power output. It measures how efficiently an engine is using the fuel supplied to produce work: [2]

$$sfc = \frac{\dot{m}_f \times 3{,}6 \times 10^6}{P} \text{ [g/kW.h]} \tag{17}$$

Brake power gives brake specific fuel consumption $bsfc$ and indicated power gives indicated specific fuel consumption $isfc$.

$$bsfc = \frac{\dot{m}_f \times 3{,}6 \times 10^6}{P_b} \text{ [g/kW.h]} \tag{18}$$

$$isfc = \frac{\dot{m}_f \times 3{,}6 \times 10^6}{P_i} \text{ [g/kW.h]} \tag{19}$$

2.5.5 Work

Work is the output of any engine, and in a reciprocating ICE, this work is generated by the gases in the combustion chamber of the cylinder. [5] The cylinder pressure and corresponding cylinder volume throughout the engine cycle can be plotted on a p-V diagram shown in Figure 2.9. [2]

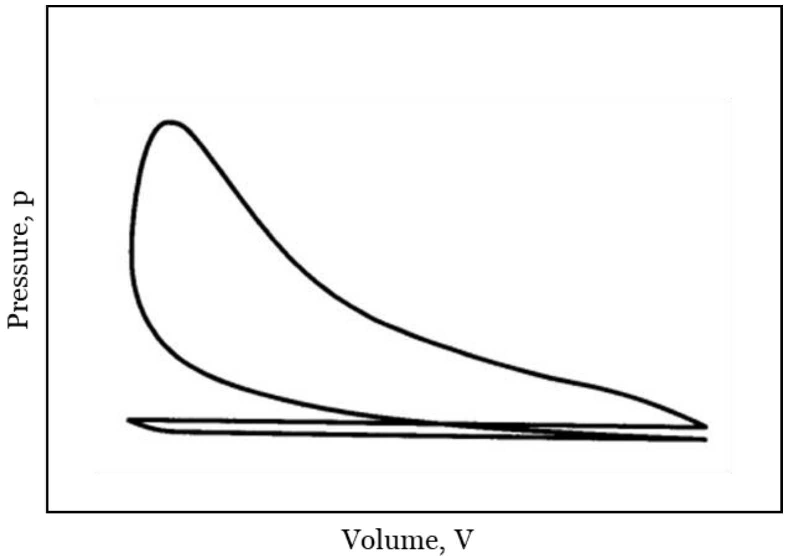


Figure 2.9: p-V diagram of a 4-stroke SI engine cycle [6]

$$W = \int p \; dV \text{ [J]} \tag{20}$$

The indicated work W_i is equivalent to the specific work acting on the piston excluding mechanical friction and parasitic loads of the engine (W_f). Actual work available at the crankshaft is called brake work W_b.

$$W_b = W_i - W_f \text{ [J]} \tag{21}$$

Gross work W_g - Work delivered to the piston over the compression and power strokes only.

Net work W_n - Work delivered to the piston over the entire 4-stroke cycle.

The work transfer between the piston and the cylinder gases during the intake and exhaust strokes is called pumping work W_p and is negative for NA engines or positive for SC engines. [2]

$$W_n = W_g + W_p \text{ [J]} \tag{22}$$

The ratio of brake work at the crankshaft to the indicated work in the cylinder defines the mechanical efficiency of an engine and will be on the order of 75% to 95% for modern engines operating at wide open throttle (WOT). [5]

$$\eta_m = \frac{W_b}{W_i} \tag{23}$$

2.6.6 Volumetric Efficiency and Overall Performance

One of the most important processes that governs how much power and performance can be obtained from an engine is getting the maximum amount of air into the cylinder during each cycle since more air means more fuel can be burned and more energy can be converted to output power. Ideally, a mass of air equal to the density of atmospheric air times the displacement volume of the cylinder should be ingested for each cycle. [5] However, the intake system restricts the amount of air that an engine of a given displacement can induct and less than this ideal amount of air enters the cylinder. [2]

The parameter used to measure the effectiveness of a 4-stroke cycle engine's intake process is the volumetric efficiency η_v.

$$\eta_v = \frac{m_a \times 10^6}{\rho_a V_d} \tag{24}$$

$$\eta_v = \frac{2\dot{m}_a \times 10^6}{60 \times \rho_a V_d N} \tag{25}$$

where ρ_a is the intake air density.

For NA engines, η_v is always smaller than 1 while SC engines have operating states in which η_v is larger than 1, as it is possible to supply an engine with more air than would fit in its swept volume under intake conditions. [1]

The ratio of the work produced per cycle to the amount of fuel energy supplied per cycle that can be released in the combustion process is commonly used to relate the desired engine output (power) to the necessary input (fuel flow). It is a measure of the engine's overall performance η. [5]

$$\eta = \frac{Power\ output}{Fuel\ input} = \frac{P}{\dot{m}_f LHV} \times 10^{-9} \tag{26}$$

$$\eta = \frac{1}{3600 \times (sfc) LHV} \tag{27}$$

where LHV is the lower heating value of fuel, which for the commercial hydrocarbon fuels used in engines are in the range 42 to 44 MJ/kg. [2] Gasoline has a LHV of 43.4 MJ/kg. [6]

2.6 Junkers Engine

OPE's are characterized by having two pistons operating in the same cylinder and moving in opposite directions, eliminating the need for cylinder heads. They began to be used commercially around 1900 for numerous land, marine, and aviation purposes, despite the first appearance in public use of the 2-stroke, gas-fuelled, OPE engineered by Wittig (Figure 2.10) in Germany around 1878. [7]

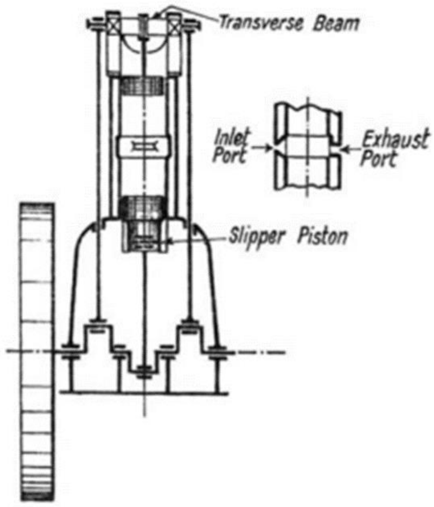


Figure 2.10: Wittig gas OPE arrangement [2]

The OPE has set many of the existing standards for power-to-weight ratio, dynamic refinement, fuel tolerance, package space, fuel efficiency, and manufacturing simplicity, remaining viable for certain applications that require outstanding power and package density, simplicity, and reliability, such as aviation. [7]

Most OPEs operated on a 2-stroke cycle and almost all have been CI diesel engines, as they were intended to achieve high thermal efficiency as well as high power density. [7]

Similar to 4-stroke CI engines, 2-stroke CI engines are constructed with a cylinder, piston, crankshaft, connecting rod, and crankcase. Common to both are the compression and power strokes. The 2-stroke cycle (Figure 2.11) eliminates the separate intake and exhaust strokes, making each cycle with one power stroke completed in one crankshaft revolution. In their places, the fresh charge and the burnt gases enter and leave the cylinder through openings made in the cylinder walls (ports) which are closed and opened by the piston in its reciprocating movement. [3]

A fresh charge enters the crankcase when the piston uncovers the intake port while moving upwards, while it compresses the charge in the cylinder head above the piston. It has no external mixture preparation system and only air is taken in and compressed instead of an air-fuel mixture (as in SI engines) and the fuel is injected at the end of the compression stroke by a fuel injector fitted instead of a spark plug. Air is compressed to a sufficiently high pressure and temperature for combustion to occur spontaneously when fuel is injected (auto-ignition), that is why it has high compression ratio values, between 12 and 25. The resulting explosion of gases is what drives the piston downwards, uncovering the exhaust port through which the exhaust gases flow out of the cylinder. As the piston moves down even further the transfer port becomes uncovered and the compressed air in the crankcase rushes into the cylinder head above the piston, helping to displace any remaining exhaust gases (scavenging). The piston now starts back on its upward stroke. [8]

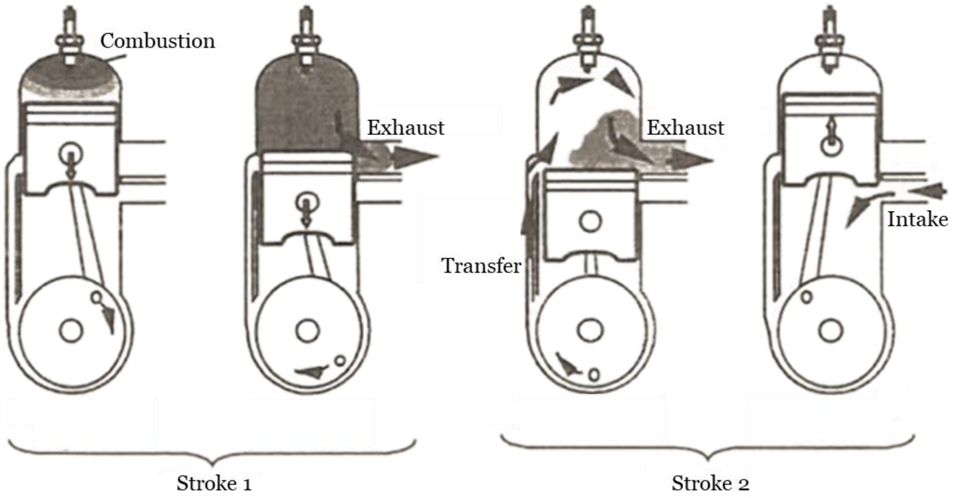


Figure 2.11: 2-stroke engine cycle [3]

One of the downsides to the OPE was that some prominent applications were not entirely durable or required relatively high maintenance due to several reasons such as the engines being undersized for their applications, being so advanced to its time in terms of performance versus any other diesel, having short development periods while their technology was pioneering and the added high public visibility and potential liability because of their applications (major military vehicles, railway locomotives and aircraft). The weakest link of the OPE was, and probably remains, the oil consumption issue, which is a fundamental characteristic of piston-ported liners and sleeve valve engines. [7]

The high risk and liability for aeronautical power units is largely why there are very few successful commercial piston engine applications in this field. While there have been many aeronautical diesel engines through the years, few have been OPE's. [7] The Junkers Jumo 205 (Figure 2.12) is one of the most famous aeronautical OPE.

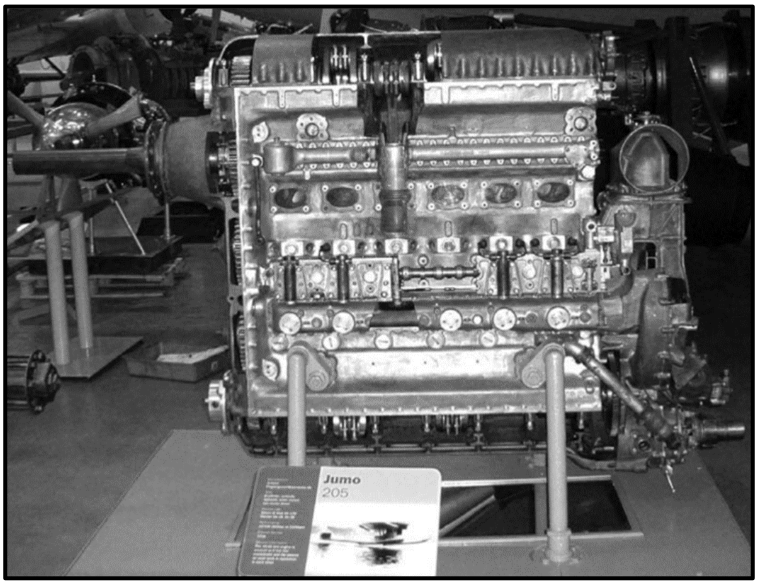


Figure 2.12: Junkers Jumo 205 [2]

The Junkers Jumo 205 was a renowned engine in pre WWII civil aviation. It was also used militarily before and during WWII and was the only diesel engine used in regular aircraft service in significant quantities worldwide. It set many long distance records and historically remains the most efficient piston aero engine in aviation. The power ratings ranged from 373 kW at 2200 rpm, to 735 kW at 2800 rpm for the turbocharged versions. [7]

Original factory information on Jumo engines is scarce as the Junkers works at Dessau were destroyed after WWII and the knowledge base transferred to the Soviet Union. [7]

This family of engines was based on a vertical liquid-cooled light-alloy cylinder arrangement with an upper "exhaust" crankshaft linked to the exhaust pistons and the lower "air" crankshaft controlling the scavenge pistons. [7] The exhaust piston controls the exit of the combustion gases and the scavenge piston controls the admission of fresh air by uncovering the proper ports in the cylinder wall. [9] A set of five spur gears linked the exhaust and air crankshafts at the front of the engine. The centre of these gears drove two camshafts that control individual fuel injection pumps for each cylinder. The two intake manifolds, which were cast integral with the cylinder block, had their entries on the rear face of the crankcase. Two exhaust manifolds were bolted one on each side of the crankcase, connecting to the exhaust. [7]

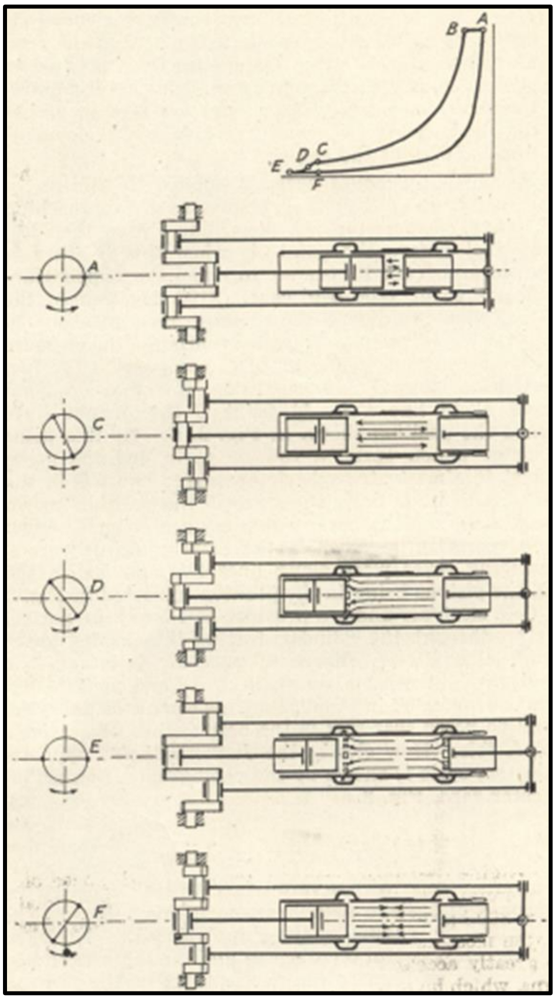


Figure 2.13: Junkers engine cycle [14]

The operating principle seen in Figure 2.13 is similar to that of a 2-stroke CI conventional engine. Firstly, the pistons approach each other, compressing air alone. Just before the state A, the fuel injection begins. At state B the fuel injection is stopped and expansion of the gases commences. At state C the exhaust piston opens a row of ports and the gases exhaust to almost atmospheric pressure. At state D the scavenge piston uncovers a second row of ports through which the cold clean air under pressure enters the cylinder, forcing the exhaust gases out ahead of it (scavenging). A charge of pure air is left in the cylinder when the ports are again closed and compression begins, completing the cycle. [10]

All of the fundamental steps leading to the Junkers aviation engine are characterized by the transition to the two-shaft arrangement, to airless injection, and to spiral scavenging. With the two-shaft arrangement, the long connecting rods of the single-crank 3-throw Wittig system are eliminated, allowing high-speed operation and reducing the weight of the moving parts and the size of the engine to a minimum. [9]

An active spiral motion is imparted to the scavenging air flow by the diagonal position of the transfer ports. This whirling motion persists in the straight cylindrical clearance space throughout the compression and power strokes. Into this rotating disk of air, there is injected fuel as well atomized and distributed as possible at the end of the compression stroke. The injection is accomplished without air using hydraulic pumps which force the fuel directly into the combustion chamber through open nozzles. [9]

Chapter 3 – Components Design

3.1 Conception (3D Modelling)

The 3D modelling of all the components and their interaction with each other were done in SolidWorks, a solid modelling computer-aided design (3D CAD) software published by Dassault Systèmes. In order to make the most of existing components in the engine, the intake collector and the carburettor body, converted to function only as a throttle, from a previous work were maintained. These two components, as well as the exhaust ports, were replicated into a 3D model in SolidWorks, considering all the distances measured as accurately as possible using a digital calliper, a set-square and a ruler. This initial 3D model helped design and assemble all the further necessary components for this work which will be explained later in this chapter. The components were designed in the simplest possible way to facilitate the fabrication process without, however, compromising their performance. All the results from the design process will be explained in Chapter 4 – Practical Case.

3.2 Exhaust Manifold Design

The idea behind the exhaust manifold, as the name implies, is to collect all the exhaust gases from the engine to a common flange where the rest of the exhaust system is connected (NA engine) or the turbocharger is bolted (SC engine).

The manifold contains a port for each exhaust port in the engine. A flange on the manifold fits against a matching surface on the exhaust port area in the engine, separated by gaskets that prevent leakage of gases.

There are two main types of exhaust manifolds (Figure 3.1):

Log manifold - all of the exhaust gases flow into a common plenum where they often collide with each other causing a lot of turbulence before the turbocharger. It has very short unequal length runners which increase back pressure, causing a lot of backflow of exhaust. On the other hand, it is the most economical, compact and reliable type of exhaust manifold. [11]

Tubular manifold - it is made of separate equal length cylindrical tubes. They offer superior flow characteristics and less backflow of exhaust gases back into the engine as each runner is longer and with smoother curves. However, tubular manifolds are often prone to cracking causing them to lose points in reliability. [11]

Figure 3.1: Log manifold (left); Tubular manifold (right) [15]

The exhaust manifold must be designed to ensure the maximum utilization of exhaust gas energy at relatively low engine speeds, since an aero piston engine usually does not exceed 3500 rpm. Thus, the narrower and shorter the runners, the better low-speed performance will be. [12] Short runners will provide a fast responsive turbo while long runners will have better flow. [13]

The collector angle is also important as it defines how the exhaust gases will come together and merge before the turbocharger. The less the angle the better the flow. [11]

It was then decided to proceed with the design of a tubular manifold considering the characteristics of the engine and this process will be explained in Chapter 4 – Practical Case.

3.3 Exhaust System Design

The exhaust pipe is attached to a muffler and it will guide the exhaust gases to the atmosphere from the exhaust manifold or the turbocharger turbine outlet. Usually, it is used a large-bore tube for the exhaust pipe since a small-bore system will create a flow restriction and back pressure at the turbine exit. [12] The fact that the two engine exhausts are now connected into a single manifold, it is not possible to use one of the original exhaust systems, as the quantity of exhaust gases doubles at the junction of the individual tubes of the manifold. The new system must be capable of carrying the gases out without any difficulty and it should be as straight as possible. The continuity equation in fluid mechanics states that the mass flow rate of fluid in a closed system must remain constant:

$$\rho A v = constant \tag{28}$$

where ρ is the density of the fluid, A is the cross-sectional area and v is the flow velocity.

As the two manifold tubes have the same diameter and flow rate, the mass flow rate into the junction must be equal to the mass flow rate leaving the junction. Therefore, the total cross-sectional area of the new pipe must be at least equal to the sum of the cross-sectional areas of the two tubes. The length of the exhaust pipe was purely estimated considering the original exhaust system size. The results will be explained in Chapter 4 – Practical Case.

To test the engine as NA and SC versions using the same manifold, so that we see the real effect of the turbocharger on the engine's performance, it is needed a piece that can adapt the exhaust system for use on both versions. The differences in the tube's diameter of the exhaust manifold and the exhaust pipe, as well as the different flanges that connect the manifold to the turbocharger turbine and the turbine outlet to the rest of the exhaust system, make this piece essential for the easy adaptation of the engine for the NA test. This piece connects the new exhaust manifold to the exhaust pipe for the NA test and it is not needed for the SC test.

3.4 Intake System Design

The intake system connects the turbocharger compressor outlet to the engine's intake ports. As an intake manifold was already made in a previous work, it was not needed to design a new one since the manifold fulfilled all the requirements for the expected tests on the engine. Therefore, it was only necessary to pay attention to the connection from the manifold to the turbocharger compressor outlet.

Turbocharged engines are traditionally multi-cylinder because they can be timed in such a way that when one cylinder is exhausting, thus providing the turbo, another cylinder is in the intake process. In a single cylinder engine, the exhaust and intake strokes are out of phase. This means when the engine is exhausting, which is when the turbocharger is powered, the engine is not intaking and the air has nowhere to go. [17]

Adding a volume between the turbocharger and the engine intake is expected to smooth out the peaky nature of a turbocharger operating under pulsing conditions. [17] One way to create this volume in the intake system is by keeping a constant and sufficiently large radius along the system. However, this method will make the system long. Another way is to create a portion of the intake system with a larger volume, that is, with a larger radius than the rest of the system. This allows the reduction of the length of the intake system and keeps the overall volume big enough. This large volume intake is referred to as the air capacitor (Figure 3.2).

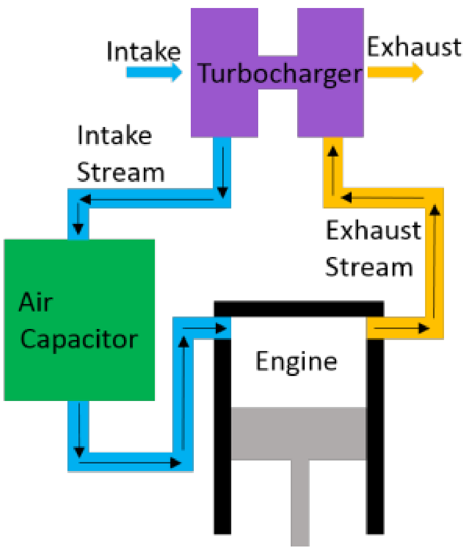


Figure 3.2: Diagram of engine flow with air capacitor system [17]

The shape and size of the air capacitor is important. A volume too large will cause excess turbo lag due to the pressurization time of the capacitor, whereas a volume too small will induce a large pressure drop during the intake stroke, negating the benefits of the turbo. The shape is important to minimize pipe losses and resonances in the system. [17]

The air capacitor must be reasonably sized and cannot take too long to pressurize. The volume has to be as small as possible to minimize cost, incorporate easily into the engine, and minimize turbo lag. But it must be large enough not to experience a significant pressure drop when the engine intakes air. The intake system with the capacitor must have a reasonable volume of between four and five times the engine capacity with a charge time probably lower than two seconds. [17] All results will be explained in Chapter 4 – Practical Case.

3.5 Fabrication Methods and Materials

Exhaust manifolds are usually made from cast iron, and some are made from stainless steel or heavy-gauge steel. [18] Casting is a laborious process as it is necessary to melt iron, which has a high melting temperature, and manufacture an initial mold. Due to the molten metal, it becomes a dangerous process. Manifolds using cast iron have rough surfaces and are difficult to weld. Stainless steel was the best option as the smoother internal surface helps increase gas speed and reduce back pressure. It is versatile, durable, and highly resistant to corrosion and staining. All the individual parts, such as tubes and flanges were welded together to achieve the exhaust manifold design.

Exhaust pipes may be made from stainless steel or zinc-plated steel. [18] To simplify, the exhaust pipe was made from stainless steel as well and connected the turbine outlet flange to the muffler by welding.

It was necessary to fabricate all the turbo flanges as the turbocharger kit only came with the turbine gaskets. These flanges should be in mild steel 6-7 mm thick. [19] To be precise, these were designed in SolidWorks and manufactured using a CNC water jet cutting machine at FabLab.

The intake system is usually made of cast metal, plastic, or composite material. [6] For the air capacitor, it was used a stainless steel plate in different formats to achieve the pretended design and two stainless steel pipes that connected the air capacitor to the rest of the intake system. This material will allow the capacitor to withstand the high air pressures coming from the turbocharger compressor while maintaining a constant shape. A smooth PVC hose connects the air capacitor to a stainless steel pipe linked to the turbocharger turbine inlet flange.

All the welding processes were carried out with a MIG welding machine due to its simplicity, cleanliness and technical efficiency. A metal band saw was used to cut all the straight pipes needed as it has a precise straight cut. These cutting and welding methods were also used in the fabrication of the NA adaptive piece.

All the connections using flanges need gaskets to prevent leakage of gases and are usually made of copper, asbestos-type material, or paper. [15] The turbo provided stainless steel compression style gaskets. These gaskets must not be reused because once compressed they will not seal again. [17] That is why it was made some gaskets with gasket paper for the NA engine tests. The turbo gaskets were outlined on paper to make them similar to the flanges and with a utility knife (box cutter) they were cut into their respective shapes.

In Chapter 4 – Practical Case all these fabrication procedures will be explained in detail and the final product will be shown.

Chapter 4 – Practical Case

4.1 Existing Components

The engine used was developed by Gregório from the junction of two Robin EY15 engine blocks. The UBI/UDI – OPE-BGX286 engine (Figure 4.1) is a single cylinder 4-stroke spark ignition (SI) OPE with a twin crankshaft configuration similar to the Junkers Jumo 205 operating in a horizontal position. [18] It was later modified by Fernandes to have electronic injection and ignition, controlled by a Megasquirt 2 (MS2) engine control unit (ECU) shown in Figure 4.2. [19]

Figure 4.1: UBI/UDI – OPE-BGX286 engine [21]

Figure 4.2: MegaSquirt 2 (MS2)

The intake collector (Figure 4.3), made by Fernandes, remained as well as the throttle body. The collector joins the two intakes into one and already has an intake air temperature sensor (IAT), a connector to the manifold air pressure sensor (MAP) and two inputs for the injectors. [19] For the throttle body (Figure 4.4), an old carburettor was used to which a throttle position sensor (TPS) was applied. Only the carburettor butterfly is used together with the TPS and the rest of it is unused. [19]

Figure 4.3: Intake collector [22]

Figure 4.4: Throttle body [22]

The collector was replicated into a 3D model in SolidWorks, based on the dimensions given by Fernandes. Then, with the help of a digital calliper, a set square and a ruler, the two exhaust ports were also replicated into the same 3D model. This initial 3D model (Figure 4.5) helped design the exhaust manifold and, by adding the throttle body model into it, it helped the calculations of the intake manifold volume.

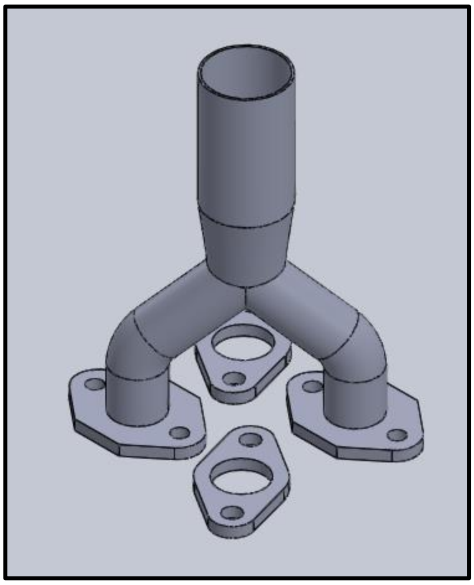


Figure 4.5: Initial 3D model

4.2 Exhaust manifold

The engine has two exhaust ports that need to merge into one before reaching the turbine inlet of the turbocharger. It was then decided to design a tubular style manifold with two runners of equal length so that the exhaust pulse reaches the collector before the turbine at the same time. If they do not arrive at the same time, they may create more turbulence and some restrictions of flow, as well as some unwanted backflow to the other runner.

Using the initial 3D model, a tubular manifold was designed to be as smooth as possible, with the minimum bend possible. The positioning of the turbine inlet flange was the first thing to do. It represents the position of the turbo in relation to the intake collector.

To determine the best position of the turbocharger, as well as the angle of the turbine inlet flange, the fuel injectors were conceptually designed into the intake collector in their positions. This helped to determine the height and the distance of the turbo from the intake collector to be as close as possible, as well as the range of angles it was possible to work with.

For the turbine inlet flange angle, was considered also the angle of the compressor outlet flange in order to make the intake system as smooth as possible. To represent the turbocharger in the 3D model, all the flanges were designed in their original position, as part of the turbo. It was decided that the compressor outlet flange normal should be, at least, in the horizontal direction. This allows the turbine inlet flange normal to be around 45º, a good angle considering the objective to smooth the exhaust manifold.

After designing the manifold (Figure 4.6), each runner has an internal diameter of 22 mm and a total length of 160 mm. Both runners merge into a collector linked to the turbine inlet flange of the turbocharger. The collector is 20 mm in length and 25 mm in diameter, the same as the turbine inlet port so that there are no flow restrictions before the exhaust pulse reaches the turbine.

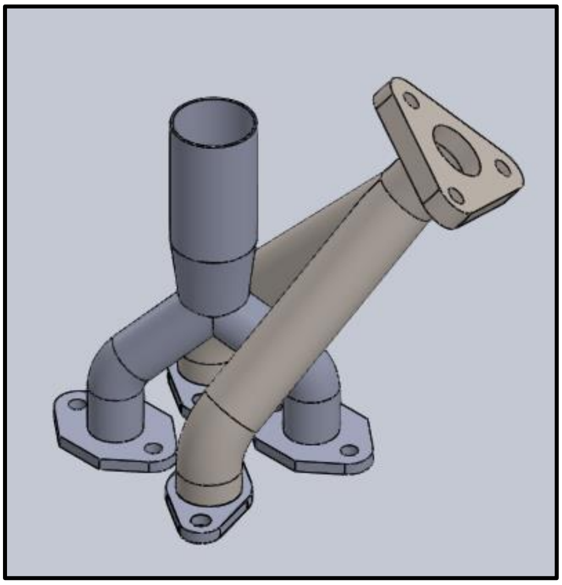


Figure 4.6: Exhaust manifold 3D model

For the fabrication of the exhaust manifold, it was used a 2 mm wall stainless steel tube with 22 mm internal diameter. It was cut two 110 mm pieces for the straight part of the manifold. At one end of each tube a cut was made to create a mitre joint with about 40º. The elbows, as well as the engine exhaust flanges, were repurposed from the original exhaust system to make the most of what was available. The design was already taking into account this possibility. For the collector, 20 mm of the same stainless steel tube was used, but a slit cut was made to widen it and then welded together again so that the diameter reached approximately the 25 mm desired.

The turbine inlet flange, as well as the other turbo flanges, were designed in SolidWorks and then manufactured, using a CNC water jet cutting machine, from a 6 mm thick steel plate. The holes to screw the flanges to the turbo came too small, so it was needed to widen them using a drill with 8.5 mm so that the M8 screws fit.

The pipes were initially cut with the metal band saw to reach the desired length. To do the mitre joint, the cut was made using an angle grinder with a cutting disc because it was impossible to achieve the pretended angle using the metal band saw or a metal cutting machine.

All the individual pieces were then welded together using a MIG welding machine according to the design, achieving the final piece shown in Figure 4.7.

Figure 4.7: Exhaust manifold

4.3 Exhaust System

The turbocharger turbine is 28 mm in diameter. The exhaust pipe must have the same or a greater diameter than this in order not to cause any flow restrictions and its cross-sectional must be at least equal to the sum of the cross-sectional areas of the two exhaust manifold runners. Each runner has a cross-sectional area of approximately 380 mm^2, so the pipe must have at least 760 mm^2. This means that the exhaust pipe must be no less than 31 mm in diameter.

However, when analysing the turbine outlet (Figure 4.8), it was noticed that the pipe could be 37 mm in diameter. So, a pipe with 37 mm width was utilized to make the exhaust pipe. The flange has the same design as the turbine outlet with a 37 mm hole in the turbine direction to attach the exhaust pipe. The reason why the maximum possible width was used is to exhaust the gases after the turbine as quickly as possible. This helps to mitigate the high gas pressures in that zone that could harm the performance of the turbocharger and engine.

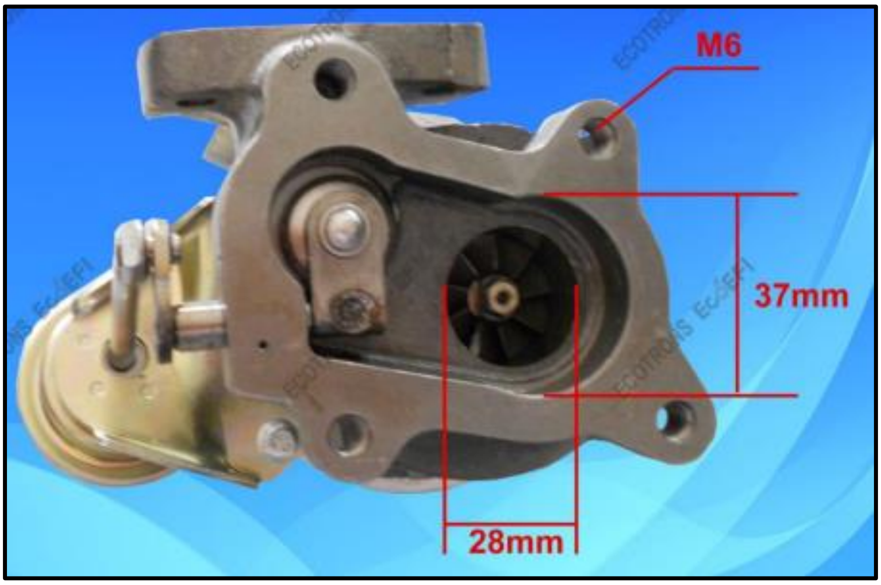


Figure 4.8: Turbine outlet of the RHB31 VZ21 [23]

However, a stainless steel tube with 38 mm width and 50 mm long was used for the exhaust pipe because it was the tube available with the closest diameter to the intended one. The length of the pipe was purely estimated considering the original exhaust system size that had approximately 50 mm long before the muffler.

The exhaust pipe was cut using a metal band saw and was welded to the turbine outlet flange on one end and to the muffler on the other end with a MIG welding machine (Figure 4.9). The muffler was slightly bigger than the originals as the amount of exhaust gases through the muffler was now twice the amount each original muffler was designed for.

Figure 4.9: Exhaust pipe with muffler

The lambda sensor was also placed in the muffler, which allows us to obtain the AFR values. The placement was made through a threaded adapter that was in one of the original mufflers. It was only necessary to cut a 35 mm square hole into the muffler and weld this adapter with a MIG welding machine (Figure 4.10).

Figure 4.10: Placement of the lambda sensor

For the NA test, a piece that directly connects the exhaust manifold to the exhaust pipe was made. It consists of a stainless steel pipe with a 38 mm width, the same as the exhaust pipe, with 25 mm long linked to a flange similar to the exhaust manifold flange (turbine inlet flange) on one end and a flange similar to the exhaust system (turbine outlet flange) on the other end. The length of the pipe is such that it is as short as possible so that its influence on the overall size of the exhaust system is minimal and that it is possible to have space to tighten the bolts on each end of the piece. The pipe was welded to each flange using a MIG welding machine (Figure 4.11).

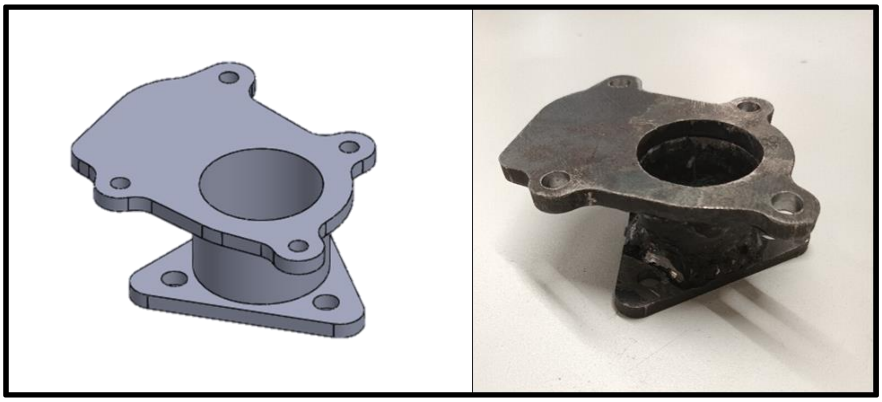


Figure 4.11: Adaptive piece for NA test

4.4 Intake System

An intake collector was already made from a previous work, so, the connection from the turbocharger compressor outlet to the collector was the only concern. The air capacitor model was followed and for this engine with 286 cm^3 displacement, the total volume of the intake system between the turbocharger and the engine should be between 1144 and 1430 cm^3.

The throttle body remained in the same position as in the NA test, which means, it is placed between the turbocharger compressor outlet and the engine. Despite this throttle position can lead to compressor surge as the throttle is closed, it allows better engine control by increasing the throttle responsiveness in comparison if it was placed before the turbocharger compressor inlet. Also, if the throttle was placed before the turbocharger compressor inlet, lubricating oil leakage from the turbocharger into the intake manifold may occur due to the large vacuum formed in the compressor when the throttle is closed. [12]

The total volume of the current intake manifold and the throttle body is around 242 cm^3. This means that the air capacitor and the rest of the intake system should have a total volume of around 1000 cm^3. The air capacitor consists of a cylinder with 10 cm in diameter and 10 cm long with 785 cm^3 in volume (Figure 4.12). To connect the air capacitor to the rest of the intake system, two tubes 35 mm long were attached to each end of the air capacitor. One of the tubes is 26 mm in width, the same diameter as the turbocharger compressor inlet, and the other is 50 mm, the same size as the throttle body inlet.

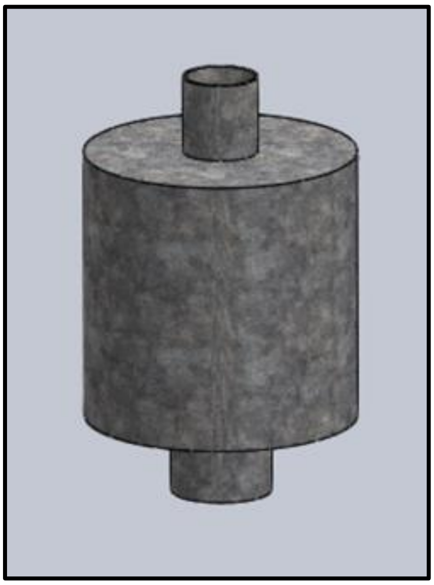


Figure 4.12: Air capacitor 3D model

The cylinder is made from 10cm of a stainless steel pipe with 10cm in diameter. On one steel plate it was drilled a 26mm hole in the middle. On another steel plate, a 50mm hole was made. Both holes were drilled using a vertical milling machine with a Forstner bit with its respective sizes. Each steel plate was welded at each end of the cylinder with a MIG welding machine.

The two tubes were cut using a metal band saw and then MIG welded to the respective end of the cylinder (Figure 4.13). Another tube 26mm in diameter and 35mm long was welded to the compressor outlet flange to allow the connection from the turbocharger to the air capacitor.

Figure 4.13: Air capacitor

The air capacitor is directly connected to the throttle body by the 50mm width tube and a PVC pipe with 26mm width connects the air capacitor to the turbocharger (Figure 4.14). The total volume of the intake system is around 1240 cm3.

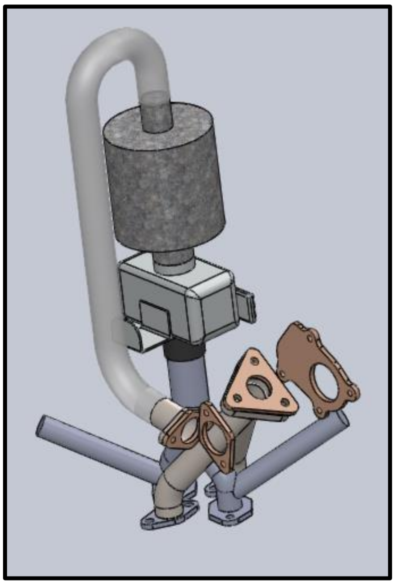


Figure 4.14: 3D model of the SC configuration

4.5 Tests

4.5.1 Experimental procedure

The engine was tested using an Eddy current dynamometer for both NA and SC versions. As the engine is controlled electronically by an ECU, data such RPM, manifold air pressure, intake air and engine temperature, Air/Fuel ratio and a couple of parameters used to calculate the fuel consumption are collected by the ECU and analyzed using the MegaLogViewer and the TunerStudio MS, software compatible with MS2 and used also to program the ECU. All data is obtained through sensors present in specific places of the engine. These sensors and their respective cable management were maintained from previous work, as well as all the ECU programming procedures (Annex F – ECU Programming). [21]

Values of torque and power are given by an Arduino code present in the bench test (Annex A – Test Bench Arduino Code) and saved in an Excel file through the DataStreamer plugin. As for fuel consumption, data such as Pulse Width (PW) and Duty Cycle (DT) provided by the ECU logs, are taken into account and analysed through the MegaLogViwer software. The fuel consumption (fc) calculation process is divided into three parts. First, it is calculated the time interval in which the injector is injecting fuel. Then, it is possible to calculate the real working cycle. Multiplying then by the mass flow rate of the injector ($\dot{m}_{inj}$) we have the amount of fuel delivered. [20]

$$real\ pulse = \frac{PW - \frac{DT}{100}}{PW} \quad (29)$$

$$real\ working\ cycle = real\ pulse \times \frac{DT}{100} \quad (30)$$

$$fc = \frac{real\ working\ cycle \times \dot{m}_{inj} \times fuel\ density \times 60}{1000}\ [\text{Kg/h}] \quad (31)$$

PW and DT values are obtained through an average of the values obtained during each test. The $\dot{m}_{inj}$ has a value of 95.69 cc/min and the gasoline density is around 0.737 kg/l at 15ºC. [19] [6]

The procedure is based on a "braking engine test" technique. After the engine stabilizes at 4500 rpm with WOT conditions, the current on the dynamometer is increased, exerting a greater load on the engine. This causes torque to increase and rpm to decrease until the engine stalls, allowing the engine performance graphs.

The NA test (Figure 4.15) is carried out with the same configuration as Fernandes's dissertation. However, a newly generated VE table was used as the cranking rpms were also different because of the starting engine of the test bench. The cranking rpms was set to 900 and the idle MAP was set to 50 kPa, the pressure the MAP sensor indicated when the TPS was set to 0 % (closed throttle). The VE table values were then increased or decreased to achieve higher values of power and torque. This way, a final VE table is obtained.

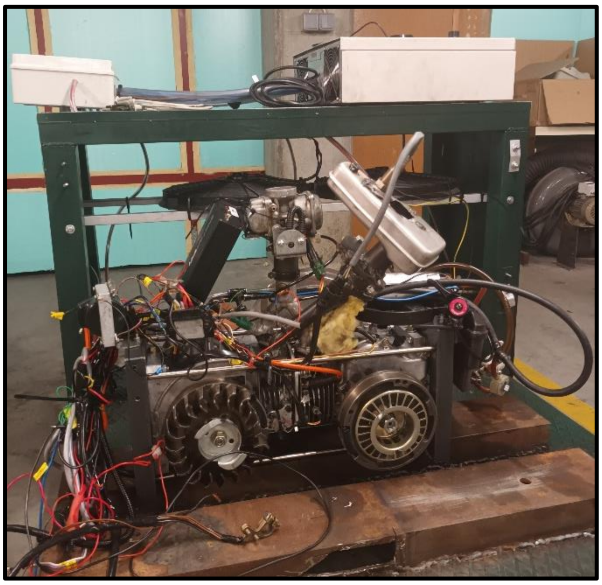


Figure 4.15: NA engine configuration

For the SC test (Figure 4.16), a new ignition table is needed, with the difference of the maximum boost level of 83 kPa, the maximum pressure increase the turbo RHB31 VZ21 is capable of providing. The VE table is modified from the final VE table obtained with the NA test with the modification of the fuel load axis. On the 100 kPa is set the value of 180 and on the 75 kPa is set to 100. Between the new 180 kPa and the 100 kPa, the values are replaced with new ones with equal intervals. The lower fuel load values are also replaced to have equal intervals through all the axis. Then, new values of the VE table are interpolated considering the new axis values. An increase of 10-20 % of the values above 100 kPa of fuel load is made to give a good starting point for the SC test. Depending on the results, these VE values are modified to achieve better engine performance, as was done for the NA test.

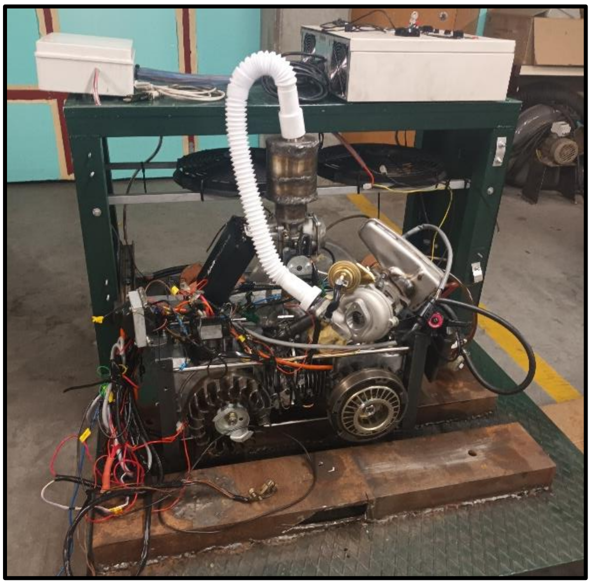


Figure 4.16: SC engine configuration

4.5.2 Engine modifications

The first results with the NA configuration were not satisfactory. The engine was not producing the expected torque and, consequently, the power was too low compared to the previous work of Fernandes. The results are presented in Figure 4.17.

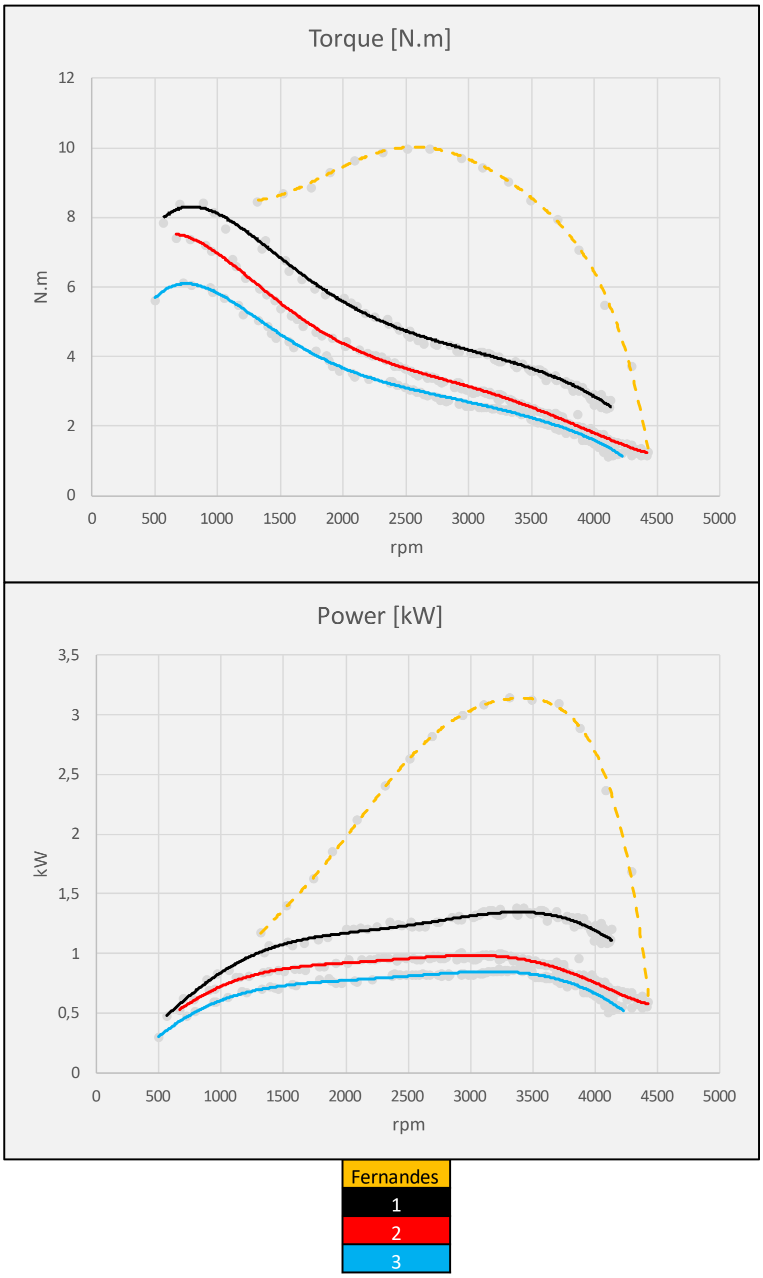


Figure 4.17: First NA tests

Despite the enrichment of the air/fuel mixture through the tests, the results were getting worse. It was noticed that the throttle was never wide open, and its position was more and more closed each test (90%; 85%; 81%). This could justify the worse results achieved each test even though it does not justify the lack of performance of the engine. When a WOT test was to be performed, a variety of new problems were detected.

One of the head gaskets burst during the test and had to be replaced (Figure 4.18). It is possible that the gasket was already in poor condition when the first tests were carried out. A defective head gasket causes cylinder leakage, lowering compression and causing poor engine performance.

Figure 4.18: First damaged head gasket

To prevent further anomalies regarding the head gaskets, these were both replaced with new ones and assembled correctly. These gaskets maintained their original position in each block and the 5.3 mm aluminium spacer was placed between them. This spacer is tightened to one of the blocks with 6 bolts with 20 N.m of torque. [21] Two guides were utilized to guarantee the alignment of the cylinders along a common axis. The two blocks were put together and the stainless steel rods were tightened with 22 N.m of torque, slightly more force than with the 6 previous bolts to prevent unexpected movements and leakage between the blocks.

At the same time, the power take-off (PTO) was slightly loose, allowing movement in the normal direction. This slack did not exist before the tests were carried out and it was feared that the crankshaft on that side was damaged in some way. However, when the pulley was removed, it was observed that the screw that tightened the PTO to the crankshaft was unscrewed and there was no problem with the crankshaft (Figure 4.19).

Figure 4.19: PTO screw problem

As the engine is working, the PTO rotates and unscrews the screw nut. To prevent this from happening again in the future, the screw nut was welded to the crankshaft at some points with a MIG welding machine after being tightened.

After the reassembly of the engine, it was difficult to start it regardless of ECU options. Even though the engine was burning fuel, it did not have enough power to do the compression stroke without the help of the starter. The only way to make it work by itself was by increasing the amount of air entering the intake manifold using a compressed air compressor. The occurrence of this phenomenon may mean the presence of leaks after the MAP sensor, which could affect the composition of the air/fuel mixture entering the cylinder.

There was a small leak in one of the intake ports because of the presence of gasoline in that area. The other, on the contrary, had no traces of fuel leakage. The joint of the intake collector to the intake port flange was sealed with silicon as this appears to be where the problem was, just below the injector port. However, to prevent further problems with gaskets and as they were a bit damaged and old, new intake and exhaust port gaskets were made of gasket paper. After all, the leakage problem on the ports was solved.

Although, the engine still would not start. This means that the engine was losing compression, and the compressed air/fuel mixture was leaking out from somewhere. The junction of the two blocks was perfectly sealed, without any sign of leaks. The problem now appeared to be the spark plug as there were traces of gasoline around that area. The spark plug had too much gap allowing the air/fuel mixture to leak during compression in the cylinder. When the spark plug was removed, the thread made in the engine block to engage the spark plug was almost non-existent. To repair it, a M10 x 1.0 threaded bushing was used (Figure 4.20).

With the engine disassembled, it was noted that the intake and exhaust valves had no clearance. This may also result in loss of pressure in the cylinder during the compression and/or power strokes. The correct tappet clearance for both intake and exhaust valves is 0.12 mm. [21] Therefore excess material from the valves was removed using an electric bench grinder until the clearance was the desired one, having been constantly checked with a feeler gauge.

After all these modifications, the engine was working but still had smoke leaks on the side. One of the head gaskets is not screwed into the engine block, which means that the holes that the screws were supposed to be in are open. The marks on the end of each hole, especially on the middle ones, indicated that there was leakage there. So, 6 threaded steel rods were inserted into those holes to cover them. Only the holes for the guides remained open (Figure 4.20).

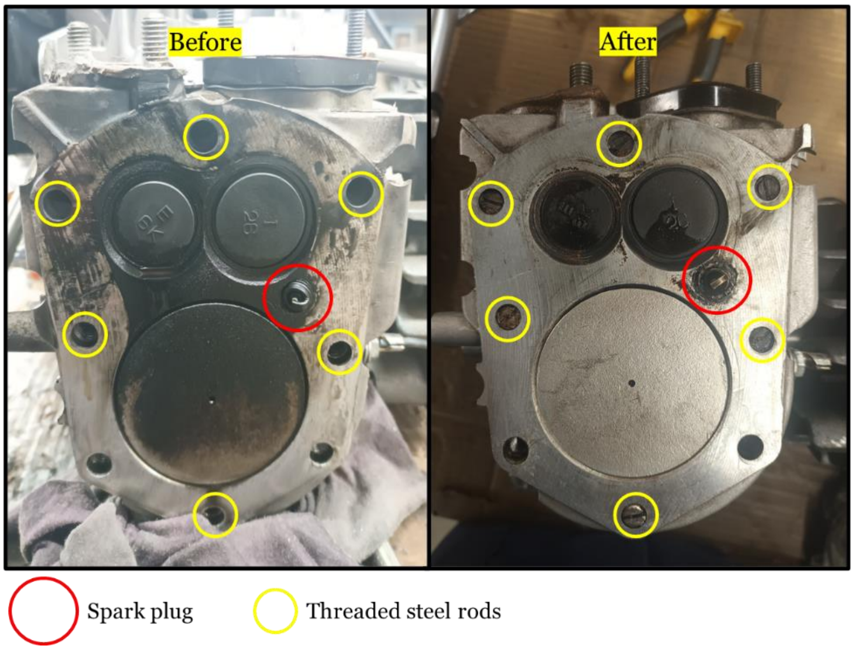


Figure 4.20: Spark plug and engine block holes modification

The engine worked better, but still, it had leaks of smoke on the side, around the head gasket from the spark plug side. New head gaskets (Figure 4.21), one with just the holes for the two guides and the other “normal”, were made out of 1.5 mm thick copper after another head gasket burst near one of the original bolt holes near the spark plug (Figure 4.22). In Annex H – Copper Head Gaskets Fabrication, the fabrication process of the copper gaskets is explained and shown through images.

Figure 4.21: Copper head gaskets

Figure 4.22: Second damaged head gasket

The aluminium spacer that sits between the head gaskets was also investigated. It was noticed that it was slightly bent and not entirely levelled. An ultra fine (1000 grit) sandpaper was used and put on a levelled glass table. The sandpaper was then wet and the aluminium spacer was moved on top of it until all the surface was polished. This meant that the surface was levelled, and the process was repeated on the other side to achieve the same objective on both sides of the spacer. The engine block from the spark plug side was also rectified using a milling cutter machine as it did not seem to be perfectly levelled as well.

The spacer has a cut made, as well as one of the head gaskets, to allow the fitting of the spark plug. As seen in Figure 4.22, it looked like the gasket was not well sealed against the spacer. This was because the threaded bushing used to repair the spark plug fitting was maybe touching the aluminium spacer, allowing the air/fuel mixture and exhaust gases to escape through that area. So, a bigger cut was made to perfectly seal the engine head on the spark plug side.

The test bench base, where the engine is bolted to, was also reinforced with two U profile steel bars above and a regular square steel tube under the base. The engine is now bolted to the U profile steel as it is stronger and could give more stability than when it was bolted to the steel plate on the base.

The PTO, as well as the drive belt pulley (Figure 4.23), were also rectified as they were not perfectly concentric with the gear attached to the crankshaft, causing the timing belt, which connects the engine to the test bench, not to have the same tension through each revolution of the PTO. This may cause the two blocks of the engine to split up on the opposite side of the gears box, allowing the leak of gases through the head gasket.

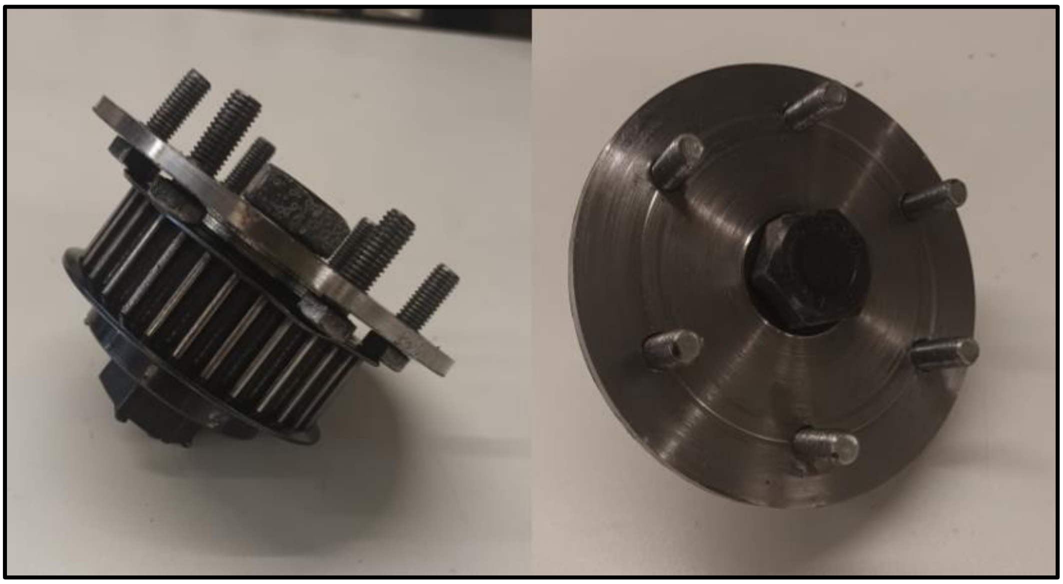


Figure 4.23: Drive belt pulley

After all the modifications made to the engine and the ECU programming, the engine had always the same leak problem through the head gasket of the spark plug side. This way, it was not possible to get results from both NA and SC tests. Possible reasons for this problem are explained in Chapter 5 – Final Considerations.

Chapter 5 – Final Considerations

5.1 Conclusions

The main objective of this dissertation was not achieved due to the impossibility of making the engine work properly, making it impossible to obtain useful results. It ended up being very frustrating after all the modifications made to the engine and ECU settings, and all the work and time spent trying to achieve all the objectives. The engine is a unique prototype, and it took a lot of creativity and research to try to solve all the problems found during the tests.

Starting with the design and manufacturing of the components for the supercharging of the engine, some of the materials used could be different. Stainless steel was used for almost all of the new components, but for some of them, such as the exhaust manifold, copper pipes with thinner walls could be a better choice, especially because of its flexibility and adaptation during the range of working temperatures of the engine. It would mitigate some manufacturing errors that may exist with the current manifold. The fact that the manifold was not a single piece but the junction of two separate pipes and elbows, makes this component susceptible to welding errors, especially on the angles of the elbows and the pipes.

All the attempts to solve the problems were very enriching in terms of practical knowledge about this type of engine and all of them improved the life range of the engine. The new intake and exhaust gaskets substituted old and damaged gaskets that were causing leak problems. The design and fabrication of the new copper head gaskets were also a good improvement as they are more durable and improve heat exchange.

The covering of the engine block holes also seemed to be a good choice as there were no leaks after the rectification. The adjustment of the spark plug was more difficult to solve. The angle of the hole was odd and it was not easy to make the thread and install the threaded bushing. However, after a couple of attempts, the leak problem of the spark plug was perfectly solved.

The lack of results during engine testing due to the persistent leaks can be the result of a wide variety of reasons. First of all, it was the first time this engine was tested on the Eddy current dynamometer test bench in question. Being a constantly improving project made and modified by students it has its flaws, some of them were repaired or, at least, its issues were alleviated during this dissertation.

Another possible reason could be the drive belt pulley and PTO unity. Even after the modification of both components, it was notable that the unity was not concentric with the crankshaft. A slight error in the holes for fastening the gear to the PTO is enough to prevent them from being perfectly concentric. 1-2 mm of level difference of the drive belt pulley during the revolutions of the engine was enough for the timing belt to have tension peaks. In some positions the belt was loose, in others the tension was very high. This could make the engine split up during those tension peaks on the belt.

The changes in the tappet clearance of both intake and exhaust valves could be also the reason for the failure of the engine. With the new clearance, the exhaust valves take longer to open and the pressure in the cylinder can reach higher values than was supposed to. The exhaust gases have no way to go so they are forced through the weakest point of the engine, between the aluminium spacer and the engine block on the spark plug side.

The final possible reason could be the ignition advance used or the ignition offset angle. Despite these values being the same from a previous work on the engine, it seemed that it could be too high, causing higher temperatures and pressures inside the cylinder.

After all, a lot of knowledge was gained regarding 4-stroke engines, especially the OPEs, and the supercharging of an original NA engine.

5.2 Future Works

Making alterations to an already deeply modified engine that had been idle since the last dissertation it was used became a job with a lot of setbacks. However, several future works can be done based on this dissertation. First, the drive belt pulley and PTO unity could be machined to be perfectly concentric, removing one of the problems during the tests. Also, the conception of a numerical modelling and flow analysis of the same study could provide additional insights on the subject. Another idea is the optimization of the exhaust manifold for minimal pressure drop, allowing a higher efficiency of the turbocharger. The change from a single scroll turbocharger to a twin scroll would be an improvement as the pulse of the two runners would not interfere with each other. Also, being a SI engine, a study of the feasibility of using an intercooler would be interesting to prevent knock and improve performance. The final recommendation is the design and manufacture of a new OPE with only one intake and exhaust port/valve, allowing an easier design of intake and exhaust systems.

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Annex A – Test Bench Arduino Code

```
//INCLUDE LIBRARIES
#include <Wire.h>                                            // I2C Library
#include <LiquidCrystal_I2C.h>                               // LCD Library
#include <HX711.h>                                         // HX711 Library

//INITIALIZE INTERFACE PINS
HX711 scale;                                         // Start link with HX711
LiquidCrystal_I2C lcd(0x27, 16, 2);           // (address, #columns, #rows)

//VARIABLES FOR RPM
const int rpmPin = 2;                           // Pin 2 detects hall sensor
int rpm;                                        // RPM value to be displayed
float PrevTime = 0;  // Time of previous magnet detection in microseconds
float Duration = 0;             // Time elapsed between magnet detection in
microseconds

//VARIABLES FOR TORQUE
const int CLK = 4;                            // Pin 4 connects to HX711 CLK
const int DOUT = 5;                          // Pin 5 connects to HX711 DOUT
const int ResetPin = 6;              // Pin 6 connects to reset push button
const int Distance = 500;             // Distance of the arm in millimetres
float Weight;                      // Weight measured by load cell in grams
float Calibration = 218.5;                   // Load cell calibration factor
float Torque;                          // Torque calculated in Newton.metre
bool ResetButton;
float PowerKW;
float PowerHP;


void setup()
{
  pinMode(ResetPin, INPUT_PULLUP);        // Activate internal resistor and
connect to pin 2
  attachInterrupt(digitalPinToInterrupt(rpmPin), RPMISR, RISING);
```

```
  scale.begin(DOUT, CLK);                          // Start link with HX711
  scale.tare();                                       // Tare scale to zero
  scale.set_scale(Calibration);     // Adjust scale to calibration factor

  Serial.begin(9600);

  lcd.init();                                  // Start I2C link with LCD
  lcd.backlight();                               // Turn on LCD backlight
  lcd.setCursor(0, 0);
  lcd.print(" The Dyno Bench "); // Delay routine to stabilize scale.tare
  lcd.setCursor(0, 1);
  lcd.print("   C-MAST UBI   ");
  delay(1500);
}

void loop()
{
  RPM();
  TORQUE();
  POWER();
  LCD();
  SERIALCOM();
}

void RPMISR()
{
  Duration = micros() - PrevTime;    // Calculates time difference between
revs in microsecond
  PrevTime = micros();             // Store time for next rev calculation
}

void RPM()
{
  rpm = 60000000 / Duration;      // rpm = (1/ time millis)*1000*1000*60;
  if (micros() - PrevTime  > 2*1000000)         // Check if motor stopped -
unchanged after 2s
  {
    rpm = 0;
```

```
  }
}

void TORQUE()
{
  ResetButton = digitalRead(ResetPin);       // Read state of push button
  if (ResetButton == LOW)
  {
    scale.tare();                  // Tare scale if push button is pressed
  }
  Weight = scale.get_units();                          // Average 3 readings
  Torque = Weight * Distance * 0.001 * 9.81 * 0.001;             // Converts
grams.mm to N.m
}

void POWER()
{
  PowerKW = rpm*Torque/9550;
  PowerHP = PowerKW*1.341;
}

void LCD()
{
  lcd.clear();
  lcd.setCursor(0, 0);
  lcd.print("SPD: ");
  lcd.print(rpm);
  lcd.print(" rpm");
  lcd.setCursor(0, 1);
  lcd.print("TRQ: ");
  lcd.print(Torque);
  lcd.print(" N.m ");
}

void SERIALCOM()
{
  Serial.print(rpm);
  Serial.print(" ");
```

```
  Serial.print(Torque);
  Serial.print(" ");
  Serial.print(PowerKW);
  Serial.print(" ");
  Serial.println(PowerHP);
}
```

Annex B – Exhaust Manifold

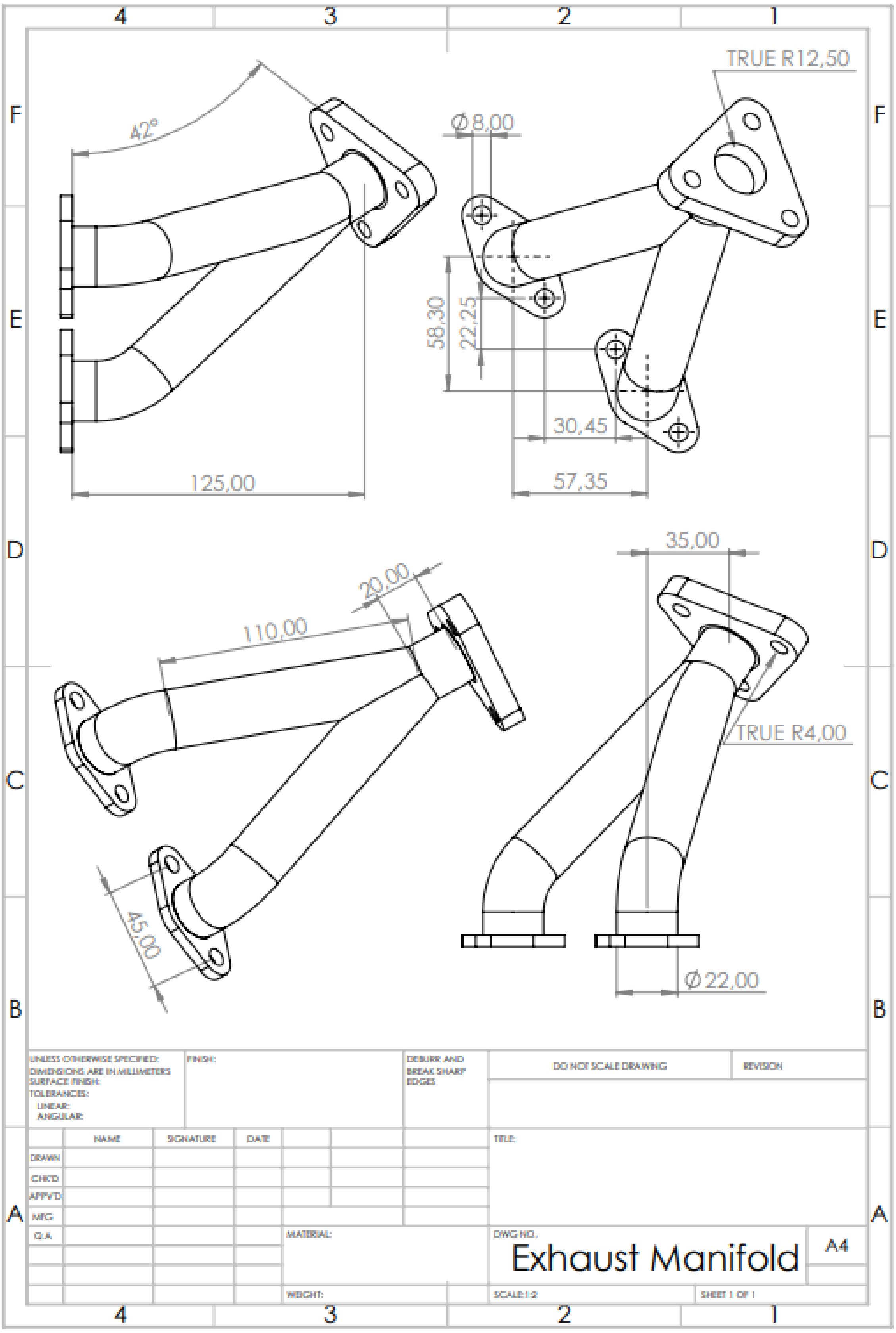

Figure B.1: Exhaust manifold 2D design

Annex C – Air Capacitor

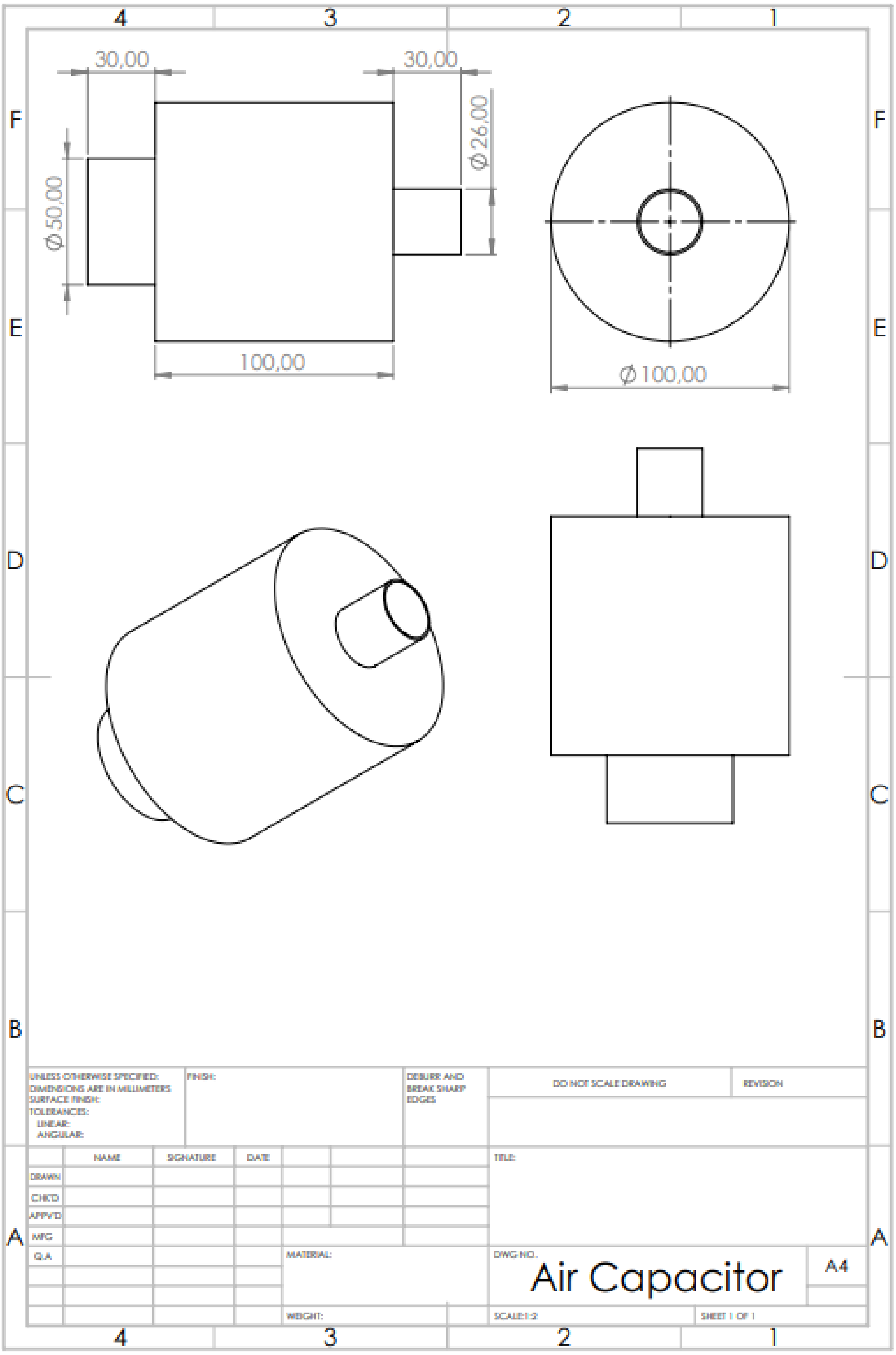

Figure C.1: Air capacitor 2D design

Annex D – RHB31 VZ21 Turbo Flanges

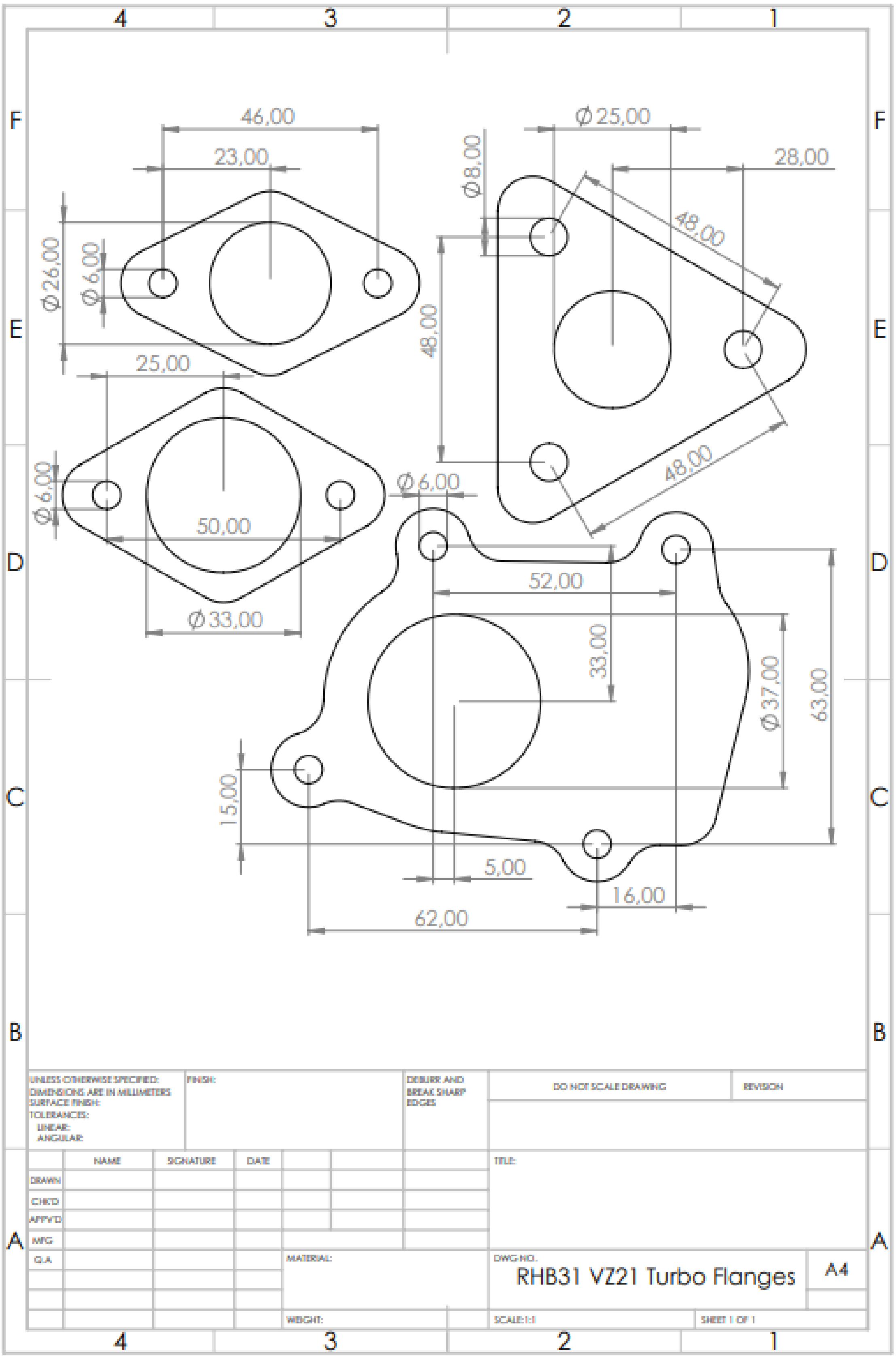

Figure D.1: RHB31 VZ21 turbo flanges 2D design

Figure D.2: RHB31 VZ21 turbo flanges

Figure D.3: RHB31 VZ21 compressor outlet flange

Annex E – RHB31 VZ21 Turbo 3D Model

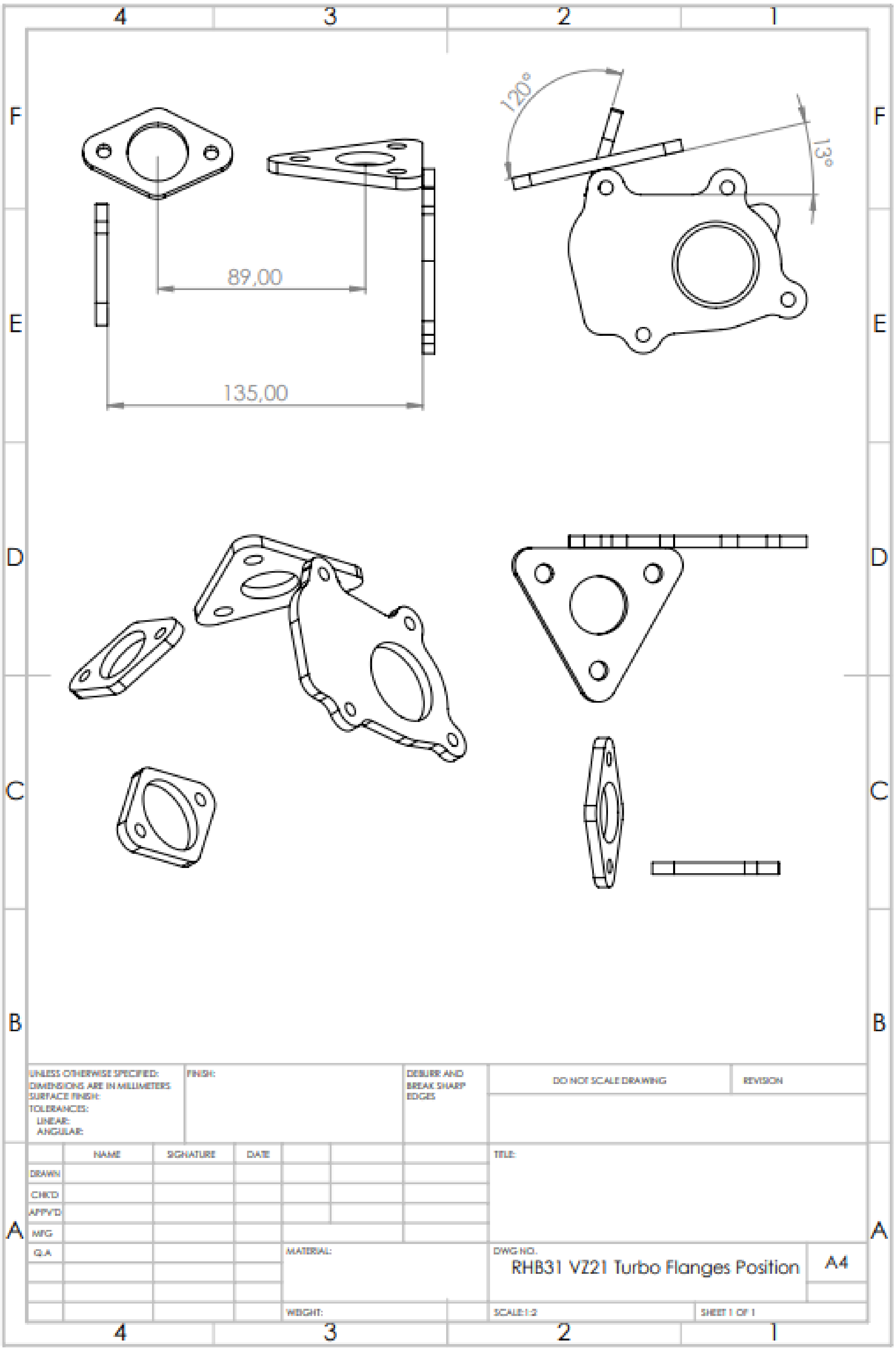

Figure E.1: RHB31 VZ21 Turbo Flanges Position

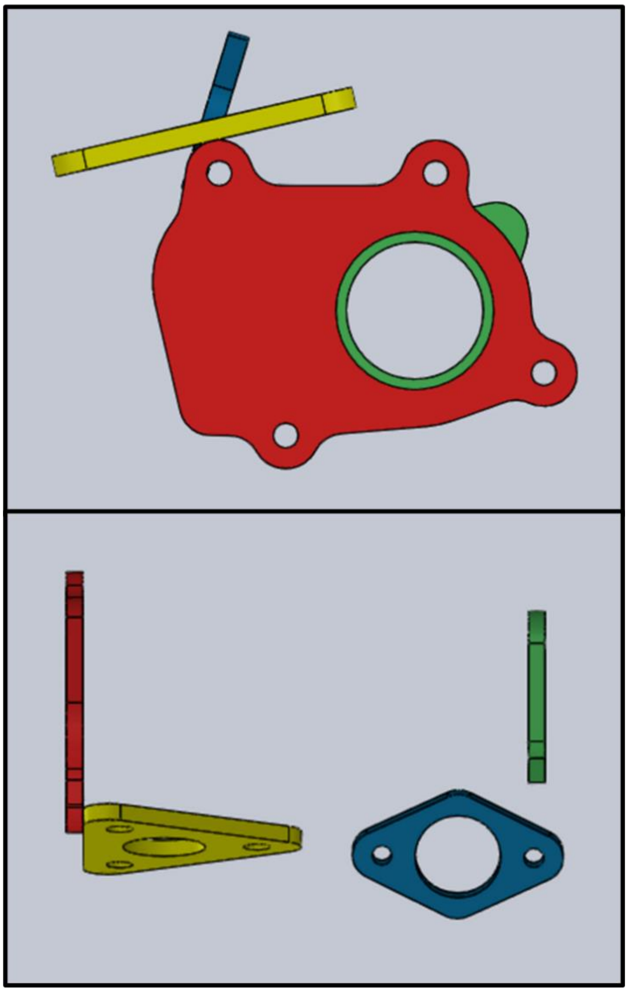

Figure E.2: RHB31 VZ21 Turbo 3D Model

Annex F – ECU Programming

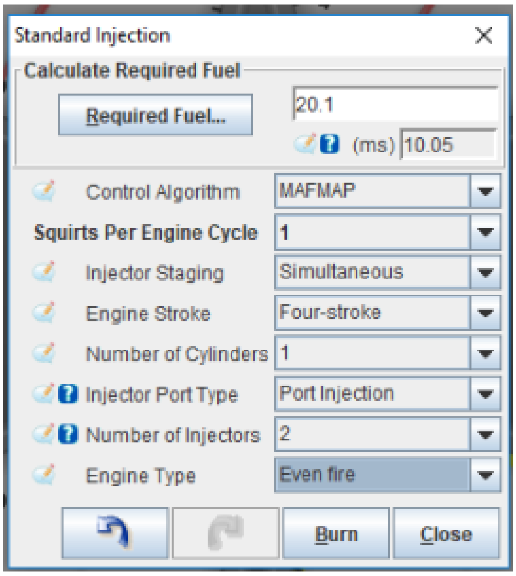

Figure F.1: Engine and sequential settings [21]

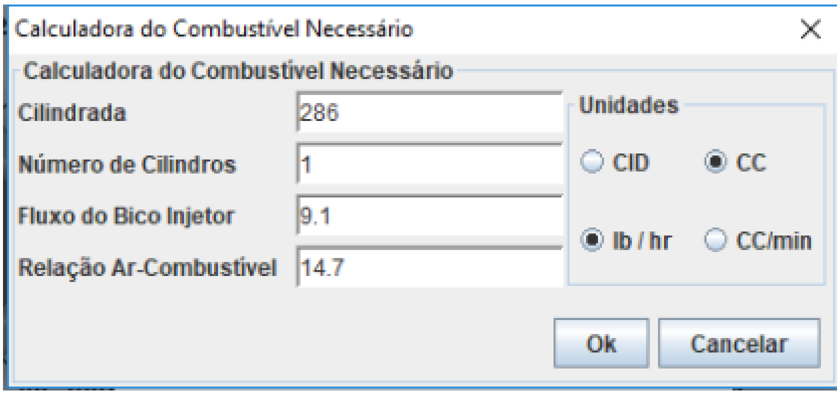

Figure F.2: Required fuel calculator [22]

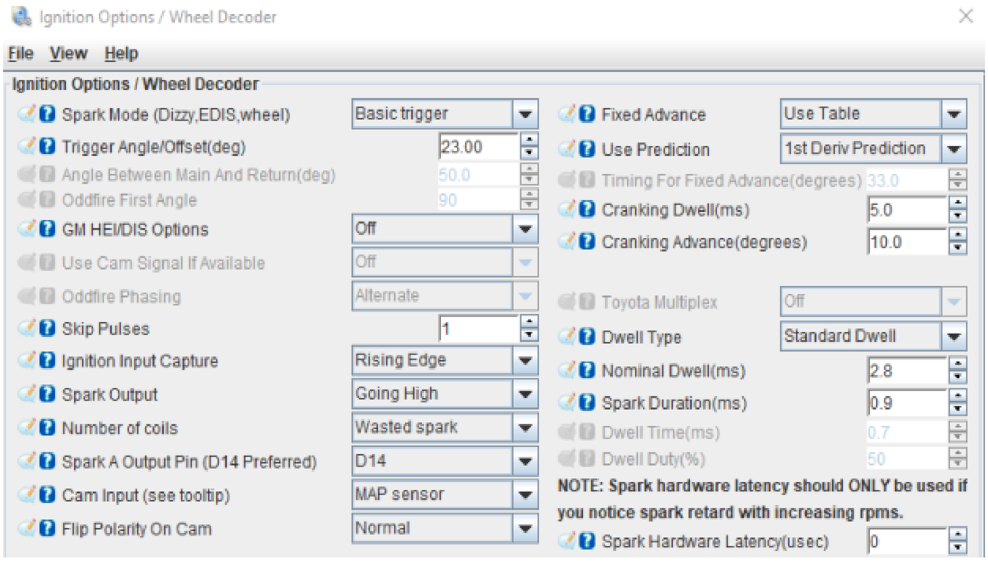

Figure F.3: Ignition options [21]

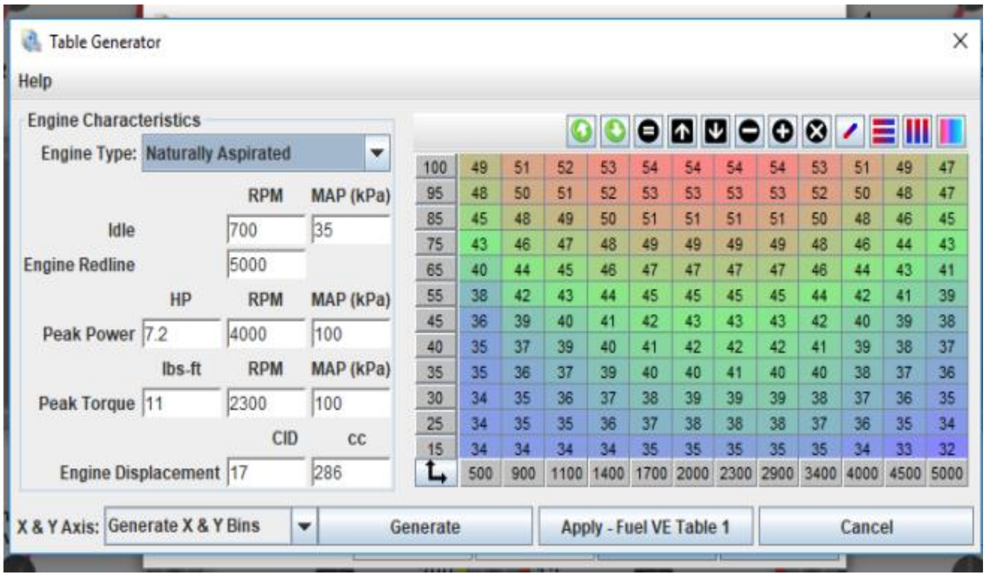

Figure F.4: Fuel VE table generator [21]

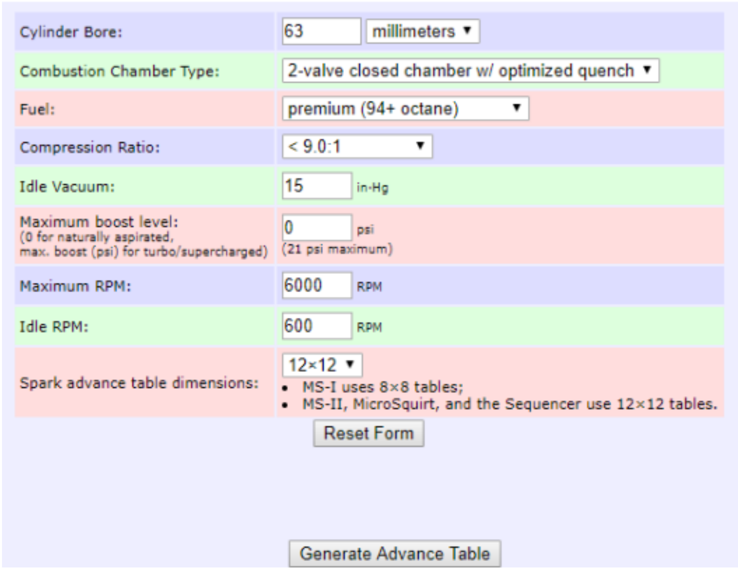

Figure F.5: Ignition table generator [21] [24]

Annex G – Head Gaskets Design

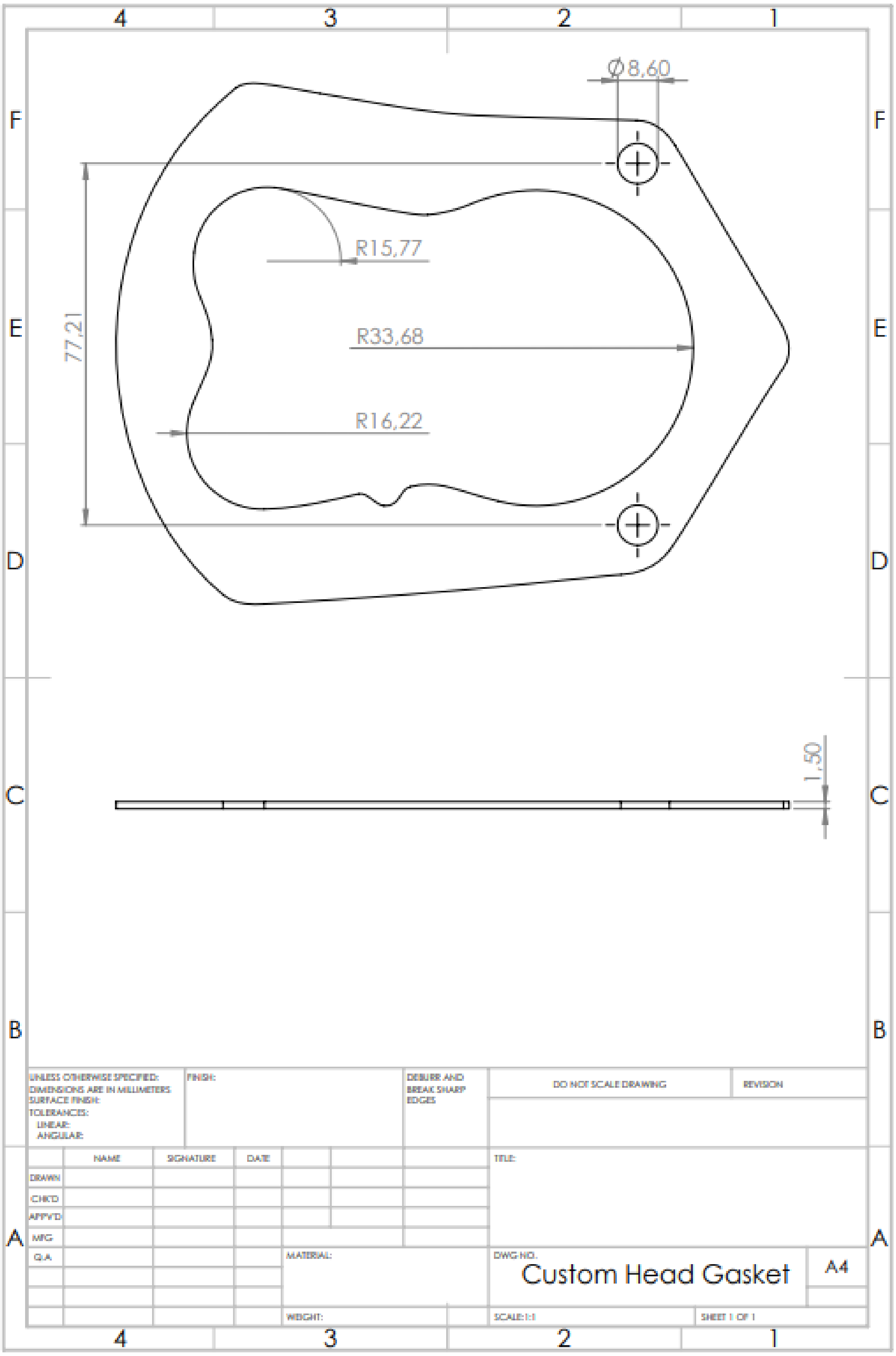

Figure G.1: Custom head gasket 2D design

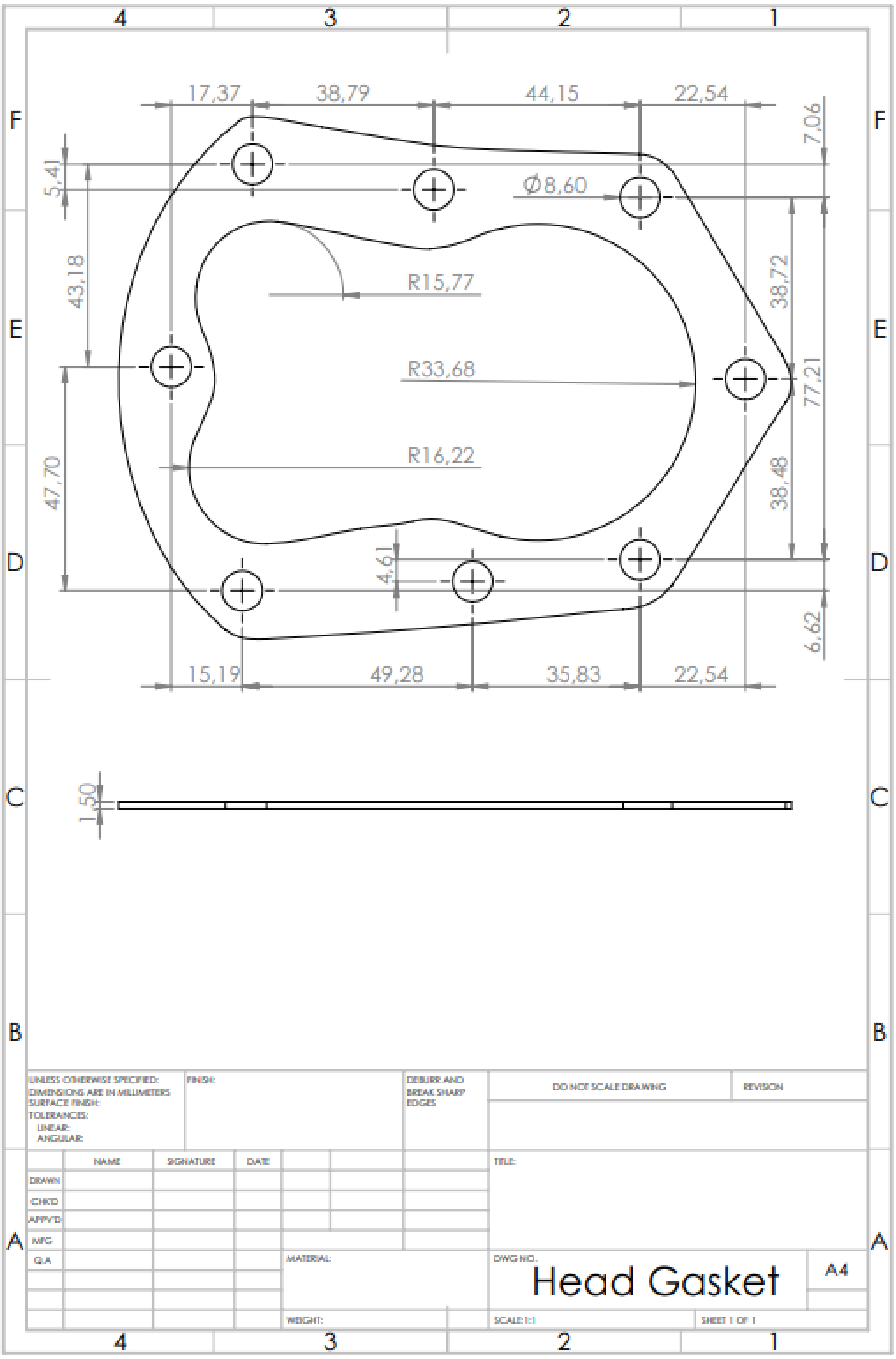

Figure G.2: Head gasket 2D design

Annex H – Copper Head Gaskets Fabrication

Both gaskets were cut into a 1.5 mm copper plate using a CNC waterjet cutting machine from FabLab. After, an annealing process was performed to eliminate the hardness of the piece and normalize the structure of the copper.

This process consists of three steps:

- Heating
- Temperature maintenance
- Cooling

To anneal the material, an oxyacetylene welding equipment was used as it could reach and maintain a temperature close to the 1000 ºC necessary.

As seen in Figure H.1, a neutral flame (blue-white) was used to anneal the soft copper piece. This flame is the result of a mixture of Oxygen and Acetylene in equal parts.

Figure H.1: Annealing of the head gasket

At this stage, the piece must remain heated for a period so that the modification affects the overall quality of the material. At the end of the annealing process, the piece is submerged in water to allow it to cool abruptly as seen in Figure H.2.

Figure H.2: Cooling of the head gasket

After cooling, the piece is subjected to cleaning to remove some impurities. To eliminate the burrs and excesses existing after the water jet cutting and thus ensure that when applied to the engine, it guarantees an adequate seal, a grinding process was accomplished. In Figure H.3 it is possible to see the differences of the cleaning process.

Figure H.3: Comparison between before and after cleaning