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Tesi di Laurea Magistrale

Development of a numerical model for injection control system of an ECU for single-cylinder 4-stroke motorcycle engines with different displacements

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Abstract

The project has been realised in cooperation with professor José Hernández Grau of the thermal and fluids engineering department, at the Polytechnic University of Cartagena through the programme Erasmus plus.

The master's thesis treats the development of a numerical model, with aid of MATLAB and Simulink, for the injection control system of an engine control unit (ECU) for two single-cylinder 4-stroke motorcycle engines with different displacements. The commercial displacement is 125 cm^3 and subsequently, through a bore modification, a new engine size of 170 cm^3 has been assembled.

The test bench has been made for 125 cm^3 engine with conventional ECU for measuring and calculating its operating parameters; successively, an experimental signal analysis by means of oscilloscope was carried out for studying the injection, ignition, and the pressure in the thermodynamic cycle. The same trials were made for 170 cubic capacity, and then, the model for the fuel injection control has been developed, and validated through the test bench, with the aid of the MATLAB and Simulink software. The model has the particularity that none parameter has been considered constant, paying mainly attention on the injector operating, on the air mass flow rate in the intake manifold and on the lambda air excess.

Summary

Chapter 1: Introduction

Before the apparition of electronic control unit, the ignition has been changed in a continuous way by a centrifugal advance mechanism and a vacuum unit; therefore, the influence of manifold pressure and of engine speed on the ignition angle were separated, and the characteristics of that control action have been given by the shape of the mechanical cam mechanisms. Hence, fuel was brought into the cylinder by carburettors that evaporated fuel into the air stream. Then, the air-fuel ratio was variable in the operating region, which caused drivability problems, high pollutant emissions, fuel economy problems. Some modifications have been made to solve these problems, but with high cost of carburettors and not always reliable. The introduction of emission legislation required the introduction of more catalytic converters. At a later time, carburettors with some electronically controllable elements were brought to obtain an air-fuel ratio near the stoichiometric ratio. The next step was the integration of the electronic system; and in this way, it was possible controlling injection and ignition timing simultaneously [1].

The engine control unit, also known by the acronym of ECU, or also called engine control module (ECM), is an electronic device and is born in the nineties with the main purpose of the electronic management of the formation of the air-fuel mixture and its combustion, for the containment of polluting emissions in an internal combustion engine. Therefore, without prejudice that the main task of the ECU is inject the right fuel amount inside the intake manifold or inside the cylinder and assure the ignition of the mixture in the combustion chamber at the wanted position of crank angle, in the modern motorcycles of ours days, and in general in vehicles equipped by internal combustion engines, this electronic devise also supervises the ignition timing (and even variable cam timing), the injection timing, and all peripherals of the control system. Then, there are also electronic controls that supervise the performance of the engine motorcycle in terms of torque and power; and all these systems follow the control strategies dictated by the engine management system (EMS), for the spark ignition engines (SI engines) [2].

1.1 Objectives of the work

The main task of this final project is the development of a numerical model, with aid of MATLAB and Simulink, for the injection control system of an engine control unit (ECU) for two single-cylinder 4-stroke motorcycle engines with a displacement of 125 cm^3 and 170 cm^3 respectively. The injection timings will be computed through the elaborated model by equations available in literature.

First of all, an experimental signal analysis is planned with conventional ECU for engine of 125 cm^3, to obtain all parameters that will be useful to calculation and to create the model. In addition to analysis of injection and ignition timings (and the respective crank angles), it will be done the pressure signal analysis in the thermodynamic cycle, to be able to appreciate the pressure peaks inside the cylinder. An important thing to keep in mind is that none values in this model it is considered constant, because due to different rpm and different throttle position of the engine, the engine parameters are going to change. Special attention will be given an important parameter, namely λ, that is the air excess in the combustion and it will be considered variable in the model; because as it is going to be seen, the air excess coefficient λ will change the model results if it keeps constant or not.

What has been said so far, it will be done for the 125 cm^3 engine. Subsequently, these steps will be done for the 170 cm^3 engine with the same ECU of the 125 cm^3 engine with an injector signal adapter or with a programmable ECU. At the end of the analysis and model results, it will be made a validation of the model by comparing the test bench results with model results, such as, for example, the fuel consumption and injection timings.

1.2 Development phases of the project

The various steps for the work development will be treated in the following chapters, as well as it proceeded to do this master thesis, in wide open throttle condition first and in partial load condition then. The project development can be subdivided in seven macro-phases how illustrated in the following steps, and visible in the *Figure 1.1*.

- First phase: test program measurements on test bench for 125 cm^3 engine.
- Second phase: experimental signal analysis with conventional ECU of 125 cm^3 and parameters of interest will be calculated, namely ignition and injection timing (and the respective crank angles as well), and finally the cylinder pressure analysis in the thermodynamic cycle.

- Third phase: MATLAB numerical model development for injection control system. In this model each value will not be constant, and the excess of air λ as well, to be as real as possible.
- Fourth phase: numerical model results and validation for engine of 125 cm^3.
- Fifth phase: test program measurements on test bench for 170 cm^3 engine and experimental signal analysis.
- Sixth phase: model application for a displacement of 170 cm^3 with an injector signal adapter or with a programmable ECU.
- Seventh phase: numerical model results and validation for engine of 170 cm^3.

it can note that in the model phases the arrow can came back to the previous phase to produce some modification, if it is necessary.

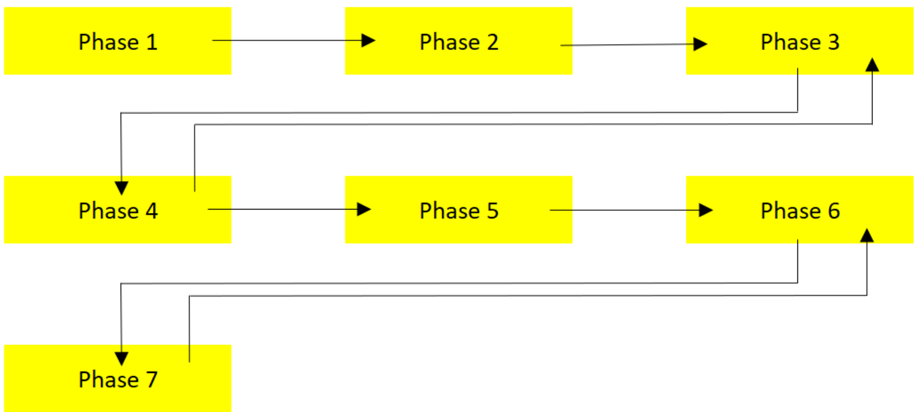

Figure 1.1: all phases of the project.

Chapter 2: Bibliography revision

The engine management system (EMS) for SI engine with injection in the intake manifold, as mentioned previously, consist of various electronic devices and actuators to control at the real time the operating conditions of the engine to supervise the injector, spark plug, throttle, and so on. To have clearer how it is made the EMS in simple sketch, it is possible to see it on the *Figure 2.1* [2], where it can note that some of the fundamental modules are coordinated with the torque module: air fuel rate (AFR) control, electronic throttle control, idle speed control, injection timing control, knock control, diagnostics control. Each of these modules work just like the same time of the torque control structure to obtain the desired engine output as required by the driver.

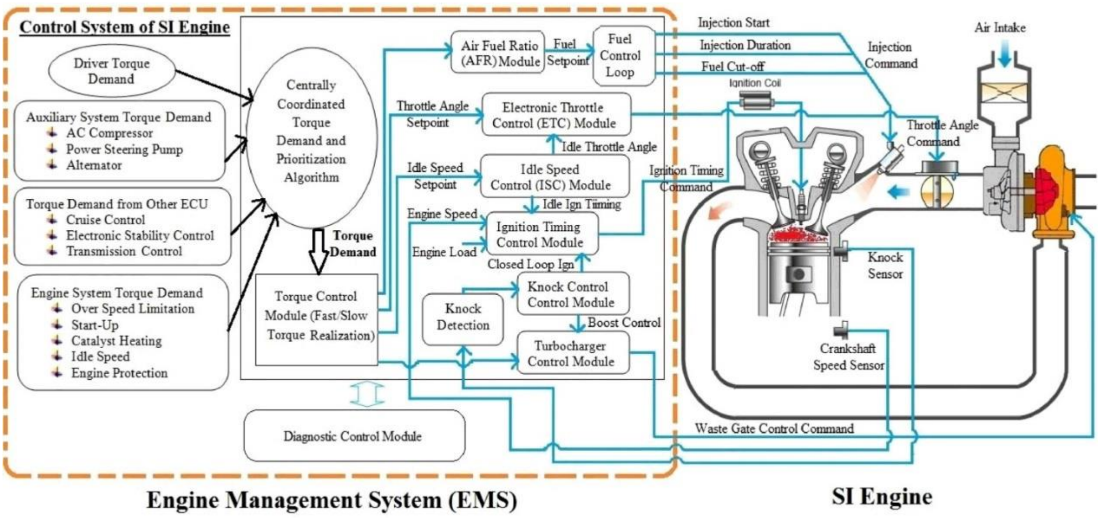

Figure 2.1: simple sketch to represent the EMS control in a SI engine, with injection in the intake manifold.

There are other modules which work simultaneously to the torque control module to satisfy the required engine output for the torque. Therefore, it can be deduced that torque control module is fundamental part of the architecture of the engine control system.

2.1 Air Fuel Ratio (AFR) control

One of the most important control in the EMS is the Air Fuel Ratio (AFR) control since this ratio can be varied through the torque demand originated by the torque structure considering vehicle and engine demand. Moreover, it is known that the three way catalytic converter (TWC) has more efficiency when the engine works around the stoichiometric air fuel rate. Because the TWC has the capacity to store in its surface oxygen and carbon monoxide, it is possible that short excursion of the air/fuel be tolerated as long as they do not exceed the remaining storage capacity. On the *Figure 2.2* there are some of the elements of the AFR control, that is mass air flow estimator, fuel film compensator, Air Fuel Ratio observer, Air Fuel Ratio controller that uses all the information to produce the correct amount of injection pulse width.

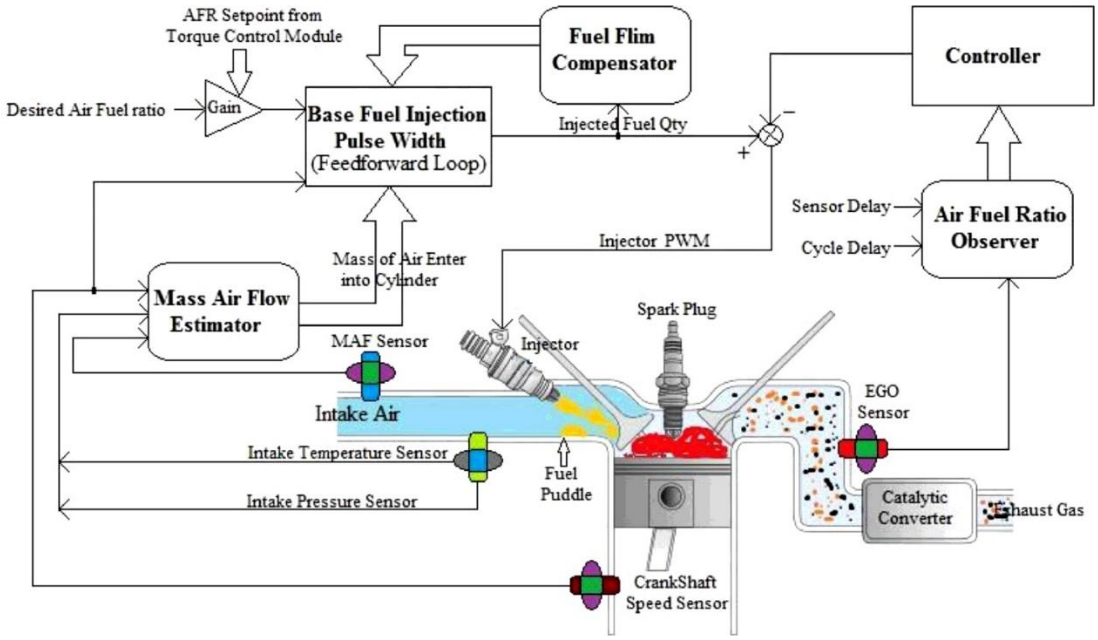

Figure 2.2: air fuel ratio control, the fuel film compensator, the air fuel ratio observer, air fuel ratio controller.

2.1.1 Mass air flow estimator

The fuel mass required to maintain the stoichiometric combustion is computed by engine control unit, based on the air flow and manifold air pressure. To know the air mass inducted inside the cylinder it is necessary meter the correct amount of fuel, and there are two methods to calculate the air flow into the cylinder of an SI engine: the first method uses the manifold absolute pressure (MAP) and temperature sensors and the engine speed (speed-density method) and the second, more used, is a method based on the Mass Air Flow (MAF) sensor which meters directly the air mass. The MAP sensor method uses an equation to compute the air flow inside the cylinder (and so allows to calculate the needed amount of fuel as well):

$$W_{cyl} = \eta_v \frac{n_e}{2} V_d \frac{p}{RT}$$

Where W_{cyl} is the value of air flow that arrives inside the cylinders, η_v is the volumetric efficiency (VE) which is mapped in a table in 2D format during the calibration of the ECU[1]; n_e is the engine speed, V_d is the engine displacement, p is the intake manifold absolute pressure or MAP (because of this the method it is called MAP, but also known in literature as speed-density method), T is the temperature in the , how the mass air flow estimator works and computes the air flow, namely by receiving signals from the sensors of crankshaft speed, intake temperature and pressure.

Regarding the second method, or rather the MAF method[2], looking the *Figure 2.2* it is possible note that this approach works using only a sensor inside the intake manifold, that estimates in an accurate way the air flow into the cylinder; usually it is used an hot wire anemometer such as sensor.

2.1.2 Fuel film compensator

During the injection some of the injected fuel goes to settle on the intake manifold, on the stem and backside of the intake valve forming a fuel film, producing a difference between the injected mass and that which should be entirely injected into the cylinder. Therefore, there is part of fuel that remain in the manifold and is mixed with air mass before entering in the combustion chamber. Hence, a part of fuel, not being compensated, induces a variation of air-fuel ratio that slows down the realization of the already set air-fuel ratio. So, there are models that describes how compute the necessary fuel mass to inject, considering also the fuel that does not enter in the cylinder. One of the most known is the Aquino model, that is a first-order model useful to compute, macroscopically, the settled fuel dynamic in the intake manifold. So, this is a model that compensates the fuel transport lag.

2.1.3 Air-fuel ratio sensor

Most of the approaches, for the air-fuel ratio observer, are based on the development for exhaust transport delay, mixing phenomena and sensor dynamic. The exhaust transport delay is divided in two parts: the cycle delay due to the four strokes

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[1] Generally in this table engine speed and MAP appear. Anyway it is possible commit errors for the calculation of air flow, by a variation of VE due to engine aging and wear, deposits in the combustion chamber, ect.

[2] The MAF method could be called also direct method because of it measures directly the pressure, while the first method carries out an estimation.

of the engine and exhaust gas transport due to exhaust gas flowing from the exhaust valve to exhaust gas oxygen (EGO) sensor in the tailpipe. A model it is used by controller to compute the compensation of the required time delay. The time delay depends by the operating engine conditions established by engine speed and throttle opening. A 2D map produces the time delay given by the variations of engine speed and load, and the delay can be variable more or less for different operating conditions of the engine.

To estimate an observer for the lambda control loop, it is necessary modelling he dynamics injection seen from the exhaust dynamic. In the air-fuel ratio observer the information required is the equivalence ratio of the air-fuel of the individual cylinders. So, the measurement is provided by the EGO sensor signal, mixing phenomena and sensor characteristics by a proper model.

2.1.4 Air fuel ratio controller

The AFR module estimates the path dynamic of the air and fuel combined with the needed compensations. The controller computes the injection pulse width (IPW) based on air flow estimation, by manifold absolute pressure (MAP) or mass air flow (MAF) sensors according to the engine speed and driver demand. The air mass is used to compute the fuel mass, and then to calculate the desired air-fuel ratio in advance to the driver torque demand. To compute this fuel mass, a model, with the injector and fuel characteristics, and its dynamic, is used to estimate the appropriate injection pulse width. In general, to obtain the fuel injection map in the feedforward loop means the tuning of the feedforward for the various operating conditions of the engine to construct the final injection map. Therefore, the feedforward loop generates an AFR value for the different torque control module and using the controller the feedback loop maintains the air-fuel ratio as close as possible to the desired air-fuel ratio by using the feedback output obtained by the EGO sensor signal. Anyway, most of our vehicles have the controllers of fuel injection system that consist in an open-loop feedforward control with look-up table and closed loop feedback control.

2.2 Ignition control module

The ignition timing is elaborated by a set of value of the torque and other parameters such as engine speed and engine air flow. The ignition control module can change its operating due to, for example, the idling, cranking, overrun and normal conditions. When cranking the engine, the ignition signal is produced by the crankshaft signal for a value of ignition advance and at the same way for the high speed, in order to limit the engine speed. The ignition timing is controlled by open-loop control and closed-loop control. The open-loop scheme for the ignition timing is based on look-up tables for the different spark crank angle on the engine operating. A first optimization is made during the engine calibration to obtain the best values of the ignition advance angle, and these values are stored in the ECU such as table and indicate the basic map of the ignition advance angle. To establish the ignition timing to obtain the required torque by the ignition control loop, it is necessary knowing the mass air flow, the engine speed and its temperature; so, the ignition advance crank angle is elaborated by map, for different engine speed and different loads, such as is shown on the *Figure 2.3*:

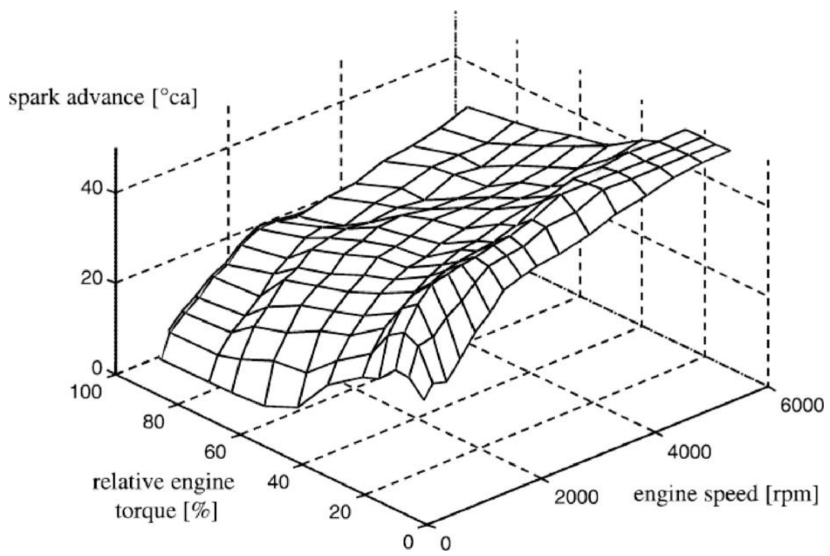

Figure 2.3: the ignition advance crank angle for different rpm and different loads

In the closed-loop control of ignition timing, the feedback control is added to the open-loop control module, which uses the knock detection system output to adapt the ignition angle for a safe value, despite the environmental conditions variations, the fuel quality, ect. The two kind of module are combined even to obtain a faster response and better control accuracy.

2.3 The numerical model

The numerical model for the injection control system of an ECU is going to be obtained by starting from “Modelling a fault-tolerant fuel control system”. This model will be elaborated to obtain a model as close as possible to reality, where filling coefficient is considered not constant. The starting model can be synthesized with the following words. Physical and empirical relationships form the basis for the throttle and intake manifold dynamics of this model. The air-fuel ratio is computed by dividing the air mass flow rate (pumped from the intake manifold) by the fuel mass flow rate (injected at the valves). The ideal (i.e. stoichiometric) mixture ratio provides a good compromise between power, fuel economy, and emissions. The target air-fuel ratio for this system is the stoichiometric one, i.e. around 14.6. Typically, a sensor determines the amount of residual oxygen present in the exhaust gas (EGO). This gives a good indication of the mixture ratio and provides a feedback measurement for closed-loop control. If the sensor indicates a high oxygen level, the control law increases the fuel rate. When the sensor detects a fuel-rich mixture, corresponding to a very low level of residual oxygen, the controller decreases the fuel rate [3]. Anyway, the numerical model will be explained better in its chapter.

Chapter 3: Hints on the reciprocating internal combustion engines

In general, it can say that motor is a machine who can transform any kind of energy in mechanical energy. One of the most useful motor is the thermal engine, that has the ability to transform the thermochemical energy in mechanical energy, through a compressible fluid. The general idea of the thermal engines is the following: converting the chemical energy in thermal energy with the combustion process that work fluid carries out, and successively convert the thermal energy in usable mechanical energy with aid of a mechanical system, namely in useful work.

Then, it is possible identify two kind of engines according to the different way to transform the thermal energy into mechanical energy compared to the type of motion: the rotative engines and reciprocating engines; moreover, the combustion can be external or internal. But, the engines in this project are reciprocating internal combustion engines, namely those engines that achieve the combustion process in discontinuous way, and the work fluid is a mixture of air-fuel. Therefore, the internal combustion engines (ICE) are by definition volumetric machines and driving machines[3].

A first distinction can be made according to the different kind of ignition. The most common land vehicles are equipped with compression ignition (CI) engines and spark ignition (SI) engines. In the CI engines the mixture ignition of air-fuel takes place after a compression of the fluid inside the cylinder through the climb buck up of piston, and due to the high pressure and high temperature the fuel auto-ignition takes place, since it is using the diesel as fuel (in fact they are called diesel engine as well, that is diesel cycle engines). In the SI engines, just like that of the project, there is a spark plug, which through a spark, takes place the fuel ignition already mixed with air, since in these engines the used fuel is the gasoline (in fact they are called gasoline engines). So, in the following chapters it is going to be talked only of SI engines.

[3] Because of this it is achieved, compared to the external temperature and the dosage or air/fuel ratio correct, an increase of temperature by obtaining useful work.

3.1 Classification by the work cycle

An important distinction of the ICE can be made according to the work cycle, namely two-stroke engines and four-stroke engines. There are two substantial differences. The first is that two-stroke engines, at the same conditions, can ideally produce twice the useful power of the four-stroke engines (in lack of higher losses); since one work cycle of four-stroke engines corresponds at two work cycle of two-stroke engines (even if the work phases are the same for both). The second is that four-stroke engines have the intake and exhaust valves[4] to the intake and exhaust of the fluid from cylinder, while the two-stroke engines do not have valves but rather fluid passage "openings" (transfer port, intake port and exhaust port) which opens and closes during the motion of the piston; consequently the two kind of engines have different construction, as shown on the *Figure 3.1* [4].

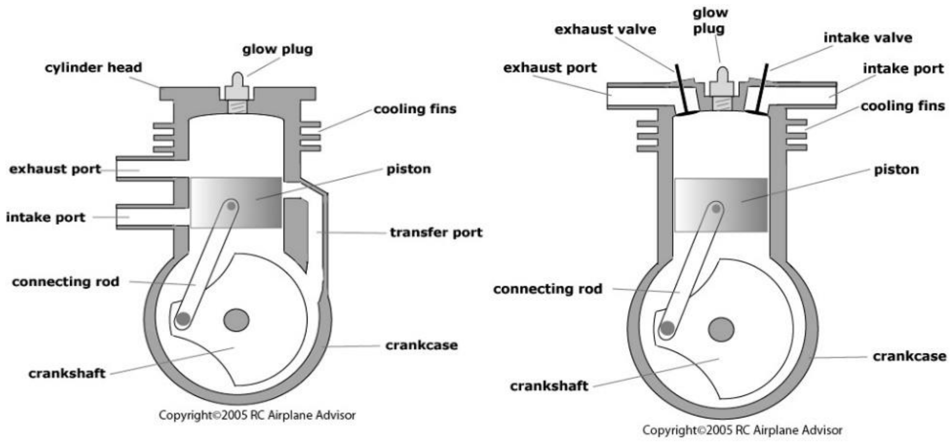

Figure 3.1: the differences visible for the first constructive approach of the cylinder.

However, nowadays the two-stroke engines are not so used and their manufacturing is very lower than four-stroke engines, due to bad control of the polluting emissions. This master thesis will talk about of four-stroke engines.

3.2 Thermodynamic cycle of four-stroke engines

The four-stroke engines achieve one thermodynamic cycle in four strokes, that correspond to the four alternative motions of the piston (as shown in *Figure 3.2*):

- *Intake stroke*, that starts from the top dead centre or TDC, with the intake valve opening, and the piston begins its descent to get at the bottom dead

[4] Valves can be one or more for intake and exhaust phase.

centre (BDC) with the intake valve closing. The air can enter inside the cylinder or the preformed mixture (if it is an engine with direct injection or injection in the intake manifold), ideally in ambient conditions[5], due to the depression that it is formed during the piston descent. It is important to keep in mind that intake valve begins its opening some degrees before that piston reaches into the TDC, and the same valve delays its closing some degrees after that piston reaches into the BDC[6].

- *Compression stroke*, which begins when the piston starts the climb back up along the chamber and intake valve closing takes place some degree after the BDC. So, both valves are closed and pressure and temperature increase until that piston reaches into the TDC, where before the BDC, for the engines with injection in the intake manifold the mixture ignition starts due to the spark plug ignition, or for the direct injection engines before the spark ignition the fuel will inject in the combustion chamber. At this point an important phase it is starts, namely the combustion.
- *Expansion stroke* or *power stroke*, where due of the combustion the gases inside the cylinder have achieved high temperatures and high pressures and tend to expand by causing the cylinder movement from TDC to BDC. This is the stroke where thermo-dynamical energy transforms in mechanical energy and the only phase where the engine produces the useful work or power, because of this it is also called power stroke. Moreover, the exhaust valve is going to be open few degrees before the piston achieves the BDC.
- *Exhaust stroke*, in which the exhaust valve is already open when the piston reaches to the BDC and the spontaneous exhaust of gases can start, since that out of the cylinder there are temperatures and pressures lower. When the cylinder moves to the TDC, the gases expulsion take place and just after the TDC the exhaust valve closes and the cycle starts again. As the intake

[5] Because of the engine heating, the intake manifold warms up and it heats also the air or mixture entering into the cylinder, decreasing the density to the detriment of cylinder filling by having less useful power.

[6] Due to the high engine speed, the air mass (or mixture) has very little time to enter inside the cylinder during the valve opening (it is talking about of few milliseconds) even because of the inertia. So, to improve the cylinder filling it is used the opening advance and closing postponement of the intake valve. Some time it is possible to have the variable cam timing which modifies the valves timing by the different engine rpm.

valve, the exhaust valve remains a little bit more open than the exhaust phase precisely to have the maximum emptying possible of the flue gas.

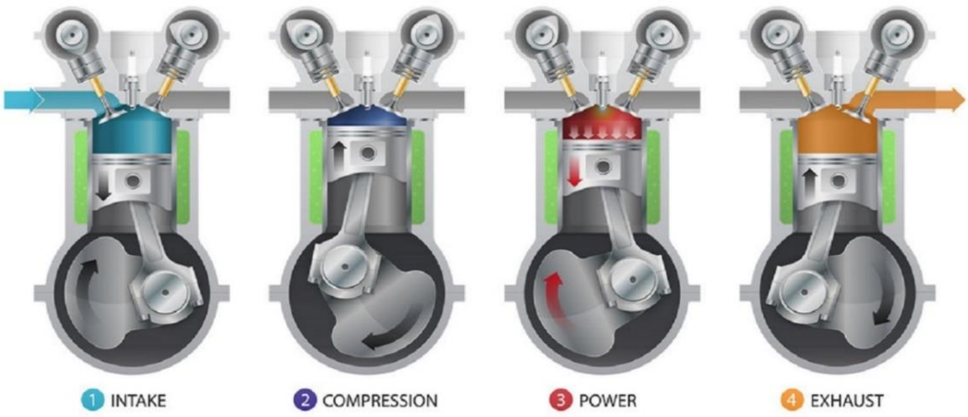

Figure 3.2: the main phases of the four-stroke engines are clearly visible on this figure.

3.4 Injection system for SI engines

The carburettors carry out the preparation of the air-fuel mixture to send into combustion chamber, and nowadays for most of vehicles they are no longer produced by leaving enough space for the injection system. There are two kind of injection:

- Indirect injection, in which the fuel admission takes place in the intake manifold and injector can be located in different points of the manifold, before or after the throttle, or nearness the intake valve. In this system, the greater the distance of injector from valves and longer time fuel will take to evaporate and mix with air; but at the same time the fuel has more time to mix with air, ideally namely admission and compression phases, and injection pressures are low.
- Direct injection, where the fuel injection takes place "directly" in the combustion chamber or in the cylinder. This system has more precision of the previous one, with less fuel consumption and more control of polluting, but less time for the fuel to mix with air, ideally only the compression, and consequently the injection pressures are very high.

In the *Figure 3.3* it is possible to see the differences between the two kind of injection [7].

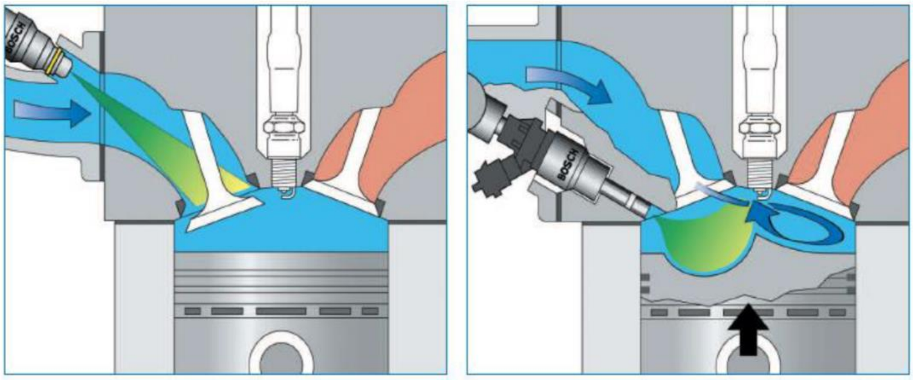

Figure 3.3: differences between indirect injection, on the left, and direct injection, on the right.

Therefore, not all modern vehicles equipped by internal combustion engines have the direct injection due to complex design and high production costs.

3.4.1 Abnormal combustion in SI engines

For the spark ignition engines it is necessary that air-fuel mixture is lighted by spark plug that should discharge some degrees before the TDC, in this way it is possible to optimize the flame front. The ideal front flame is realized when after the spark discharge, the combustion is propagated by creating two distinct zones, one with burnt gases and the other one with unburnt gases. However, the anomalies can appear and during the various combustion, and in this way, it can make worse.

The most important is the knock or detonation, where the mixture fractions farther from the spark plug can auto-ignite it in a spontaneous way due to high chamber temperature and before that flame front achieves them. The pressure increases suddenly inside the cylinder and takes place a pressures oscillating by taking in vibration the walls of the combustion chamber and causing a characteristic metallic noise. The knock restricts the maximum compression ratio and the supercharging, but it can be controlled by reducing the ignition advance. Moreover, the knock arising can be identified through the pressure signal analysis metered in the combustion chamber, or from inducted vibration from crankcase by an accelerometer [5], such as shown in *Figure 3.4*.

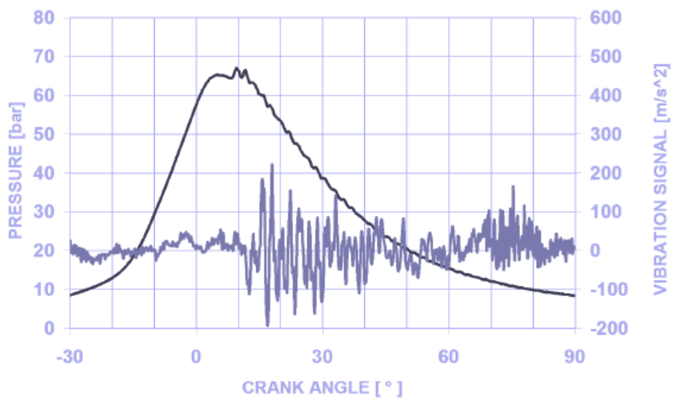

Figure 3.4: method to evaluation the knock by pressure in the combustion chamber or vibration of crankcase.

The knock is responsible to the damages of mechanical organs due to warm up fatigue. Sometime can take place a knock degeneration where it can achieve 200-250 bar of pressure peak, with an immediately damaging of the engine.

Another common phenomena is the pre-ignition that can happen from a different source than spark, like high temperature in the chamber, a spark plug that works too hot for the application, heated carbonaceous deposits in the combustion chamber. Therefore it is possible to have the flame front before the spark ignition. Since the pre-ignition produces an increasing of the temperature, it can give birth the knock; hence, these two phenomena are connected.

The combustion anomalies can be reduced by an advance ignition, or also the regulable amount of injected fuel, because of big power losses with an incorrect injection.

Chapter 4: General characteristics of the engine

The tests have been carried out with the help of ITALKIT company which procured one Yamaha YZF engine of 2008, that is a single-cylinder four-stroke motorcycle engine with a displacement of 125 cm^3 by having cylinder inclined forward. This motorcycle underwent an accident and could be interesting an engine performance evaluation. The *Figure 4.1* shown a motorcycle model in object of study.

Figure 4.1: an example of Yamaha YZF-R125 of 2008.

On the *Table 4.1* it is possible to see the more important reachable data of the engine from the respective manual [6].

Engine type	Liquid cooled 4-stroke
Number of cylinders	1
Displacement	124,7 cm^3
Bore × Stroke	52,0 mm × 58,6 mm
Compression ratio	11,2:1

Table 4.1: general data of the engine with displacement of 125 cm^3.

The ITALKIT company is in charge to develop kit for racing bikes, and provides the same kit for the Polytechnic University of Cartagena racing team; in particular, it provided new displacement, where only the bore has gone through a change by becoming 61 mm. Therefore, the general engine will be the same except for the displacement, as it is possible to see in the *Table 4.2*.

Engine type	Liquid cooled 4-stroke
Number of cylinders	1
Displacement	171,3 cm^3
Bore × Stroke	61,0 mm × 58,6 mm

Compression ratio	11,2:1

Table 4.2: general data of the engine with displacement of 170 cm^3.

The *Figure 4.2* shows the assembly of the 170 cm^3 displacement, where is clearly visible the cavity inside the cylinder for the liquid cooling system, which is going to be discussed successively.

Figure 4.2: the displacement of 170 cm^3.

4.1 Fuel injection system

The injection systems are advanced during the years becoming more and more precise to obtain bigger performances, in respect to the legislations which become more restrictive with the passing years. The main purpose of the injection system is to inject the correct amount of fuel inside the combustion chamber to form a correct air-fuel ratio thus the engine can work in an optimal way per each operating point. It is useful to remember the general formula to compute the air excess in the combustion, that is:

$$\lambda = \frac{AFR}{AFR_{st}} = \frac{m_a/m_f}{\left(m_a/m_f\right)_{st}}$$

Namely indicates the air-fuel ratio divided by stoichiometric air-fuel ratio, where in both cases the air-fuel ratio is equal to air mass divided by fuel mass. It is talked about rich mixture if $\lambda < 1$ (fuel excess), stoichiometric mixture if $\lambda = 1$, poor mixture if $\lambda > 1$ (air excess). Generally, for low engine rpm the λ coefficient is near the stoichiometric ratio, but for high engine rpm the λ coefficient reduces to be richer. Because the fuel excess can aid the lubrification of the internal organs inside cylinder and does not increase too much the temperature, with increasing engine speed. Therefore, the air-fuel ratio can vary for the different engine speed and different loads, but also with external loads acting on the motorcycle and with acceleration and deceleration produced by driver.

Anyway, the motorcycle is equipped with an injection system with electronic control (Fast Idle or FI) and receives the signal from ECU of the fuel amount to inject, for each different operating point of the engine. The FI system allows to inject accurately the fuel and consequently an emissions reduction occurs.

4.1.2 Injector of the motorcycle

The injector is a needle model with the AISAN 1100-87k00 description code. The fuel is injected through a channel that opens or closes due to an electro-valve activated by electric pulse which is generated from engine control unit. Following, the injector elements are explained to understand the mode of operation, referring to the *Figure 4.4* that shows an injector section with each of its, and to the *Figure 4.3* which shown the studied injector.

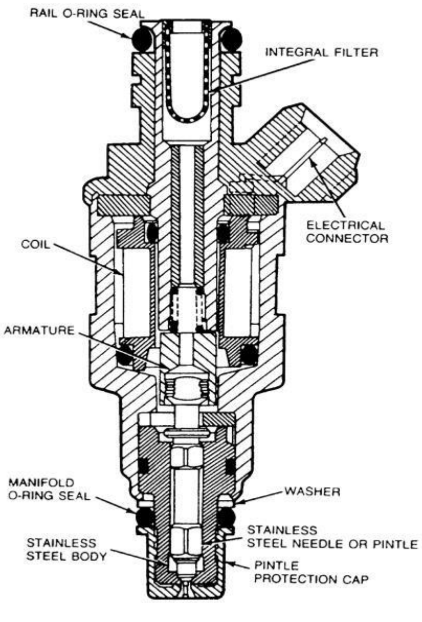

Figure 4.4: injector section to see the most important elements.

Figure 4.3: injector of this study.

First of all, an O-ring seal can be observed and it is located between the intake manifold and injector, and it works by sealing to avoid the fluid comes out (visible in

Figure 4.4). In the superior part, a filter can be notated which does not allow impurities to pass, otherwise the needle could damage. The ECU send electric pulse to the injector through electric connection (visible on the right of *Figure 4.4*), which in turn activates the coil that puts in motion the needle, and this way the fuel comes out from injector. In the inferior part, the fuel is pressurized for a better operating of the injector and thus needle can position itself in an adequate position for the injection. The needle is brought back in his beginning position by a spring, when injector stopped the electric signal sent by ECU for the injection.

4.1.3 FI injection system

An analogous circuit at injection system of the 125 cm^3 engine is represented in *Figure 4.5* [7], where the numbers and letters indicate: 1. Fuel pump, 2. Fuel injector, 3. Ignition coil, 4. Coolant temperature sensor, 5. ECU (engine control unit), 6. Lean angle sensor, 7. Crankshaft position sensor, 8. FID (fast idle solenoid), 9. Air filter case, 10. Throttle body, 11. Throttle body sensors assembly, 12. Intake air temperature sensor, 13. Throttle position sensor, 14. Intake air pressure sensor, A. Fuel system, B. Air system, C. Control system.. There is an injection pump (1) which provides the fuel through a filter to possibly pick contaminants inside fuel. Then, there is a pressure regulator that sends the fuel to the injector at a relative pressure of 250 kPa. As said, the injector opens, injecting fuel into the intake manifold, as soon as ECU elaborates the electric signal; in addition, the ECU verifies the ignition timing by signals taken from a magnetic pick-up sensor that measures the crankshaft position (7) to inject fuel in the correct moment.

Therefore, the signals coming from sensor of throttle, sensor of crankshaft position, sensor of pressure and temperature of intake air, sensor of angle inclination and sensor of coolant temperature, permits at ECU to compute the injection pulse-width.

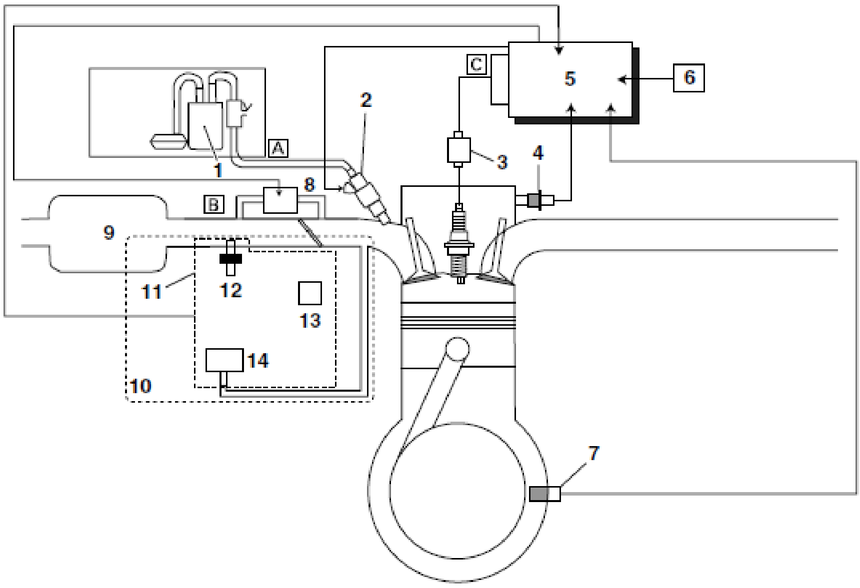

Figure 4.5: injection system representation of the Yamaha YZF R-125 engine.

4.2 Admission air system

To realize the engine tests, it has been substituted the air admission filter with an air deposit connected to intake system by a pipe and nozzle, obtaining an air flow with an intake pressure equal that overflowed from nozzle. It must be kept in mind that using this system, it should be used a pipe with a length as low as possible and at the same time as straight as possible, by trying to avoid curves, factors which lead to load losses of the air flow during intake phase, and so at losses power in the engine. In the *Figure 4.6* it is possible to see the engine with the original air filter admission, and in *Figure 4.7* the pipe with the deposit.

Figure 4.6: the air filter admission of the engine.

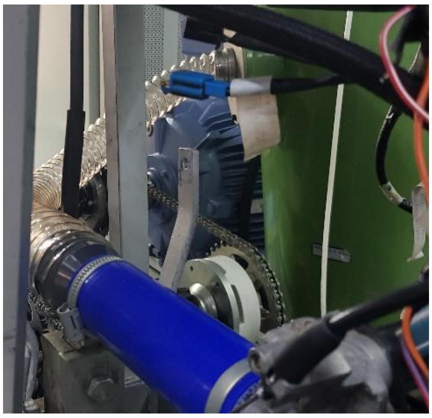

Figure 4.7: the pipe and the deposit behind in green.

4.3 Cooling system

The cooling process has a big relevance on the engine operation, because it does not allow to the most critical areas the high temperature achievement which can cause the bad operation of the engine due to the high quantity of absorbed heat. In this engine there are two kind of cooling system: one by oil and one by water.

The cooling system by oil (*Figure 4.9*) is used for each mechanical element inside the engine. The oil is introduced in a closed circuit and sent in every interested area through a pump. This oil circuit has also the purpose to reduce the friction between mechanical elements when high revolutions are reached. The mechanical element which need a good lubrication for correct operation are: crankshaft, camshaft, primary and secondary shaft. The cylinders, valves and their positions are lubricated by the oil in the oil sump. In the

For the water cooling circuit (also closed, *Figure 4.8*), the most important element is the radiator that has the task to dissipate the heat produced by engine; then, there is a pump which has the task to send into the pipes the cooling fluid formed by water and cooling liquid. This fluid flows into the conduits that encloses the cylinder because here the engine achieves the highest temperatures due to combustion, and thus it must be cooled. Because of the differences of temperature between engine and cooling fluid, the acts such as cold source and the engine such as hot source; hence, the engine gives heat to the cooling fluid by reducing the uncontrolled increasing of the temperatures. Subsequently, the fluid is done moved into the radiator to be cooled. Here, the fluid acts as hot source and gives heat to the radiator which acts such as cold source, that in turn is cooled by external air. Sometime can be expected a cooling fan in the radiator in case the temperature is too much high. So, once that fluid is cooled, the cooling cycle restarts again in the engine.

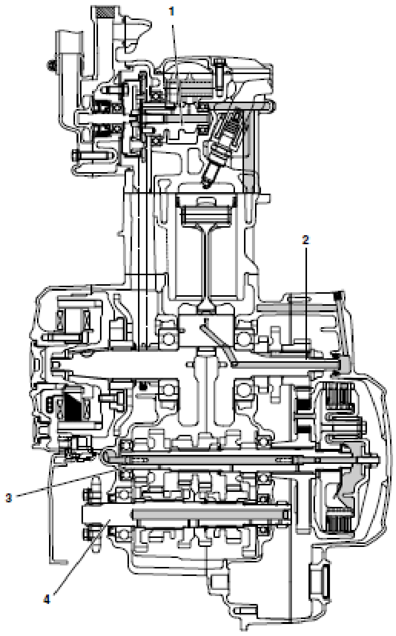

Figure 4.9: oil cooling circuit.

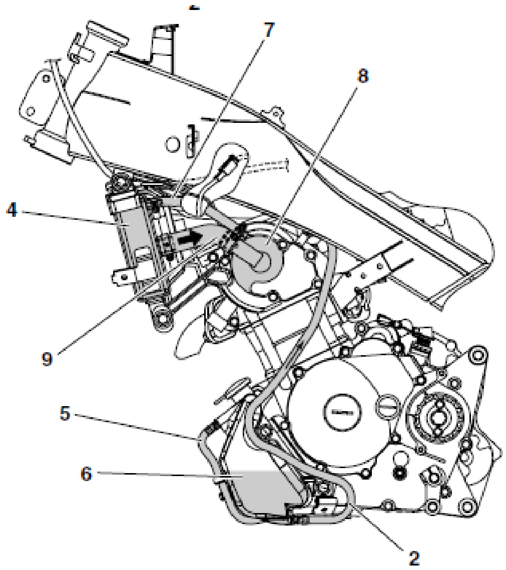

Figure 4.8: water cooling circuit.

4.4 Transmissions

The study of engine transmission is very important and it is necessary to know all transmission relations from engine to wheel because some data acquisition tools as torque meter, they measure physical quantities into the wheel and from here, by transmission relations it is possible to elaborate physical quantities into the crankshaft just like engine speed, torque, power, and so on. It has to be kept in mind that wheel and torque meter are not connected directly but through a double chain system, therefore between wheel and torque meter there will be a certain transmission ratio. The double chain system of the test bench can be seen on *Figure 4.10.*

The general expression of transmission ratio, expressible by different parameters but for ease it will be made in function of the teeth number, is:

$$R = \frac{Z_1}{Z_2}$$

Namely the ratio of the teeth number of the driven gear wheel divided by the teeth number of the driving gear wheel.

Figure 4.10: the double chain system of the test bench in movement.

The *Table 4.3* shows all transmission ratios. The first step is inside the engine to connect it to the gearbox, or rather there is a speed reducer with an only transmission ratio that is $R_1 = \frac{z_{Gearbox}}{zcrankshaft}$. After that, there is the gearbox that for the different gears guarantees different speed to the motorcycle, or rather providing different transmission ratio to the pinion axis, which is $R_2 = \frac{z_{Pinion}}{z_{Gearbox}}$. The last transmission ratio is that from pinion to the wheel and it is only one just like R_1, that is $R_3 = \frac{z_{Wheel}}{z_{Pinion}}$. Finally, to obtain the final or total transmission ratio for each driving wheel, it is necessary multiply all values of transmission ratios for each single gear, which is $R_T = R_1 \times R_2 \times R_3$. For R_1, R_2, R_3, there are two columns where on the left there is the teeth ratio and on the right the result.

<table>
<tr><th>Gear</th><th colspan="2">R_1 (Crankshaft - Gearbox)</th><th colspan="2">R_2 (Gearbox - Pinion)</th><th colspan="2">R_3 (Pinion - Wheel)</th><th>R_T</th></tr>
<tr><td>1</td><td rowspan="6">73÷24</td><td rowspan="6">3,042</td><td>34÷12</td><td>2,833</td><td rowspan="6">48÷14</td><td rowspan="6">3,429</td><td>29,548</td></tr>
<tr><td>2</td><td>30÷16</td><td>1,875</td><td>19,554</td></tr>
<tr><td>3</td><td>30÷22</td><td>1,364</td><td>14,221</td></tr>
<tr><td>4</td><td>24÷21</td><td>1,143</td><td>11,918</td></tr>
<tr><td>5</td><td>22÷23</td><td>0,957</td><td>9,975</td></tr>
<tr><td>6</td><td>21÷25</td><td>0,840</td><td>8,760</td></tr>
</table>

Table 4.3: the value of the transmission ratios for each gear and the total transmission ratio.

4.5 Engine control unit

The engine control unit used is a conventional ECU of the Yamaha 125 motorcycle which has different mapping declared by manufacturer to obtain the maximum performances respecting the limits dictated by polluting regulations.

Subsequently, it is going to be elaborated a programmable ECU following the project development, or rather through the injection and ignition control by obtaining performances acceptable.

Chapter 5: Trial cell and engine instrumentation

5.1 The test bench room

The test bench room is located Vehicle System Research Laboratory (Laboratorio de Investigación de Sistemas para Vehículos, SiVeLab) in the basaments of Center for Development and Technological Research (Centro de Desarrollo e Investigación Tecnológica, CEDIT) of the Technological Park of Fuente Álamo (Parque Tecnológico de Fuente Álamo). In the *Figure 5.1* is shown the SiVeLab.

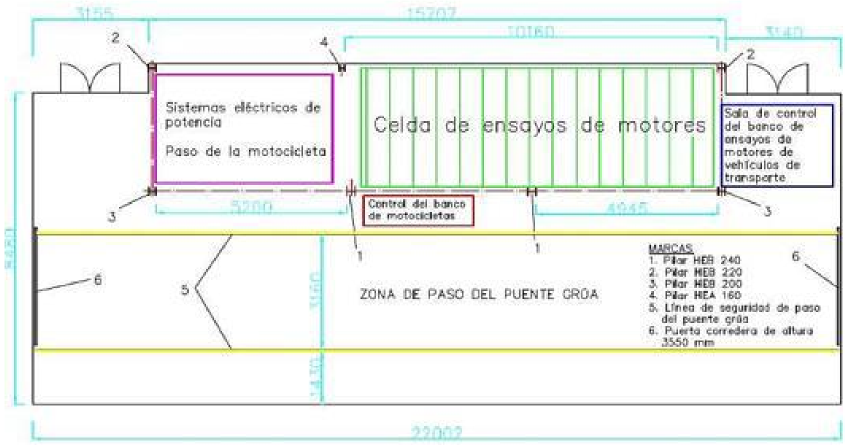

Figure 0.1: layout of the vehicle system laboratory.

The test bench cell is divided in two different areas: the right part for industrial engines and the left part for the engine motorcycles tests, both visible in the *Figure 4.2*.

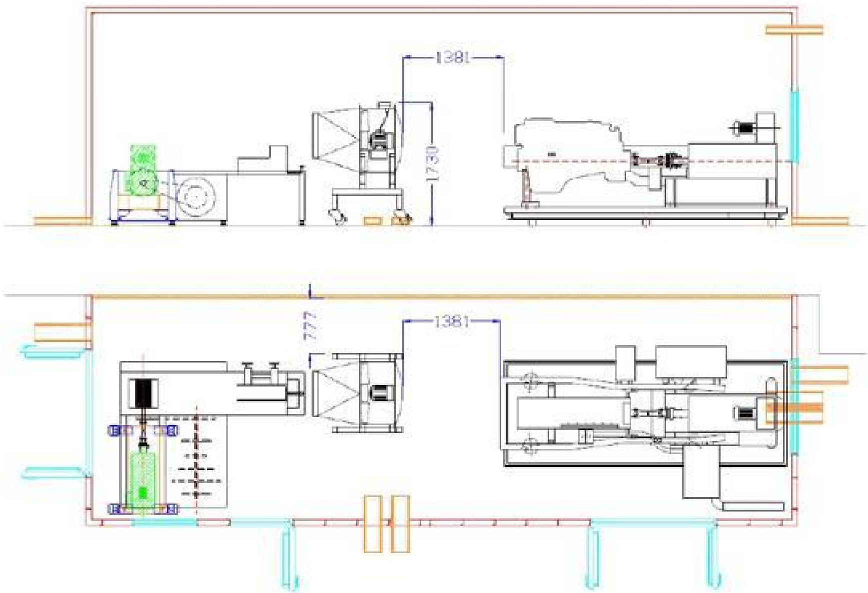

Figure 5.2: test bench cell.

5.1.1 Gas extraction system

The exhaust gases of the motorcycle engine, once come out from engine, must be directed towards the exterior of the building through a series of pipes (of green colour as in the *Figure 5.3*) and with aid of a fan and others auxiliary equipment.

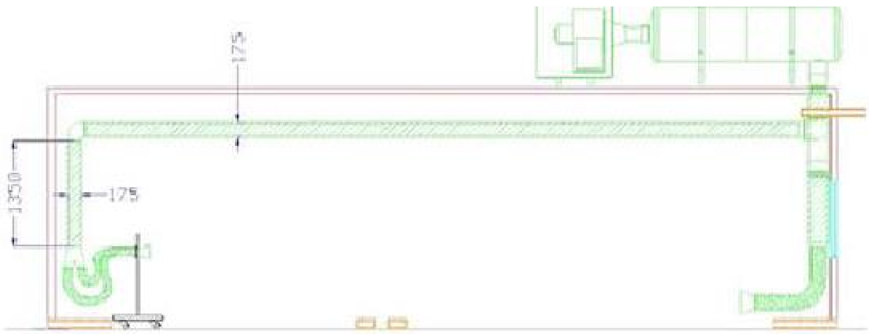

Figure 5.3: the layout of the gas extraction system.

The flow of the coming out gases from the cell must be lower than that value which fan can discharge, that is 11100 m^3/h. Therefore, the fan speed must be adjusted such a way the exhaust gases can come out from cell for both test bench.

5.1.2 Engine brake

The motor brake is an electric element that has the task to equilibrate the torque and to absorb the engine power. This kind of electric brake, visible in the *Figure 5.4*, is used for the motorcycle engine but it must also be suitable for the other kind of engines to tested in the cell.

Figure 5.4: engine brake.

5.1.3 Measuring instrument for the environmental conditions

The engine parameters, as the power, are influenced by the environmental conditions just like temperature, pressure, humidity relative, thus it is very important meter them by an instrument such as shown in the *Figure 5.5*.

Figure 5.5: environmental conditions inside the test bench cell.

Moreover, these parameters are very important for the calculation of the air mass which enters into the cylinder that will be implemented in the numerical model.

5.2 Measuring instruments and their calibration

5.2.1 Torque meter

The values of torque and speed will be taken by a Kistler sensor which works in the following way: there is a flange coupled to the electric motor so that these elements can be the same torque and speed, and very close to the flange there is the Kistler sensor (the distance between sensor and flange is two mm) by measuring the values of torque and speed. The sensor and its schematic representation are visible in the *Figure 5.7* and *Figure 5.6*. For a correct data acquisition and to prevent the sensor from failing, must be taken into consideration that the line of reference of the data evaluation must be aligned with the flange. Then, the program will furnish the values of torque and speed taken by measurement device, and torque will be multiplied by the transmission ratio according to the different work gear.

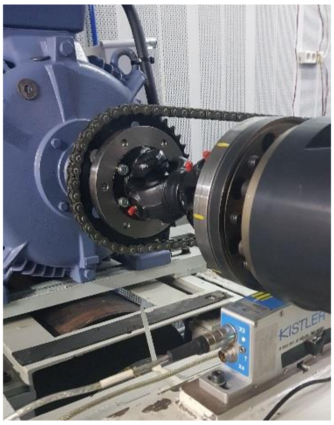

Figure 5.7: Kistler sensor and flange coupled to the electric motor.

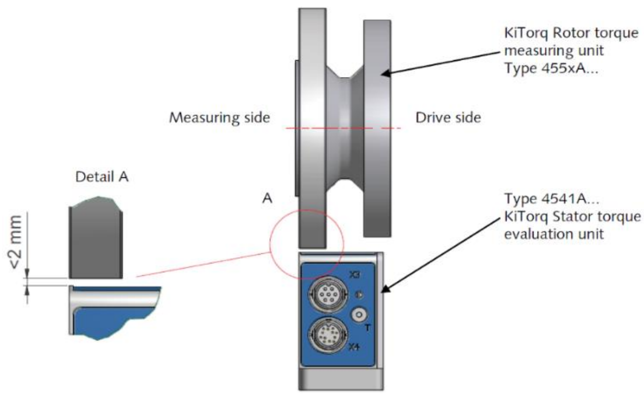

Figure 5.6: detail of the distance between flange and sensor.

5.2.2 Measuring instrument of gases

The exhaust gases have been analysed through a sensor inside the exhaust manifold, namely the manifold has been drilled where a shell was positioned to connect to the sensor which takes the measurements. These gases are inserted inside the pipes by a cooling device to decrease the exhaust gas outlet temperature. Once the gases have cooled down, they are inserted into the Horiba device measurement (*Figure 5.8*).

Figure 5.8: Horiba device measurement.

Initially this device had a pump which allowed the gases intake, but because of its break, an external pump (*Figure 5.9*) has placed so that exhaust gases are conducted along the circuit of the measurement equipment. After having verified that gases circulate inside the analyzer, it is necessary to realize the calibration for the tuning of the system.

Figure 5.9: external pump.

The analyzer needs to be connected one hour before the calibration process to obtain a correct calibration. For measurements very accurate, it is recommended that flow rate through the gases analyzer be 5 l/min; before to take each parameter, it is necessary to establish a calibration equal to zero since this equipment has drift of zero during the analysis processes. Therefore, to avoid such a derivation, the stabilization times of the measurements do not have to be excessive since they could give rise to the appearance of errors which generate an inappropriate measurement.

Moreover, it is important to keep in mind the regulation UNE 82501:2004, that is a Spanish regulation which recommend how measuring the emissions of the exhaust gases of the internal combustion engines for vehicles through various tests, and which indicates the maximum permissible errors for each independent measure, and in case of these errors be greater than permissible, a new calibration will be necessary. The *Table 5.1* shows the regulation UNE 82501:2004.

CLASS	KIND OF ERROR	ERRORS	MAXIMUM	PERMITTED	
	INDICATION	CO	CO	O	HC
II	Absolute	± 0,2 % Vol	± 1 % Vol	± 0,2 % Vol	± 30 ppm Vol
	Relative	± 10 %	± 10 %	± 10 %	± 10 %

Table 5.1: Spanish regulation UNE 82501:2004.

For the calibration of the measurement equipment, it will be verified that reference voltages and the voltages assigned to each measure device maintain a value of zero Volt, and otherwise the potentiometer has to be regulated respect to specified value.

After that, plastic containers with different gases composition are used to observe if shown data satisfy the characteristic of examined container. These characteristics and those of the Horiba equipment measurement, have to be as close as possible, so that there will not be any deviation in the effective measure. The followings tables, *Table 5.2* and *Table 5.3*, show the characteristics.

ITV CONTAINER UNIT OF MEASUREMENT	PERCENTAGE ITV [%]
C_3H_8	100 ppm
CO	0,2 %
O_2	0,2 %
CO_2	15 %

Table 0.2: gases characteristic of the reference container.

HORIBA CONTAINER UNIT OF MEASUREMENT	PERCENTAGE HORIBA [%]
C_3H_8	48 ppm
CO	0,35 %
O_2	0,26 %
CO_2	15,62 %

Table 5.3: gases characteristic of the Horiba container.

As it is possible to see, there are small different between the Horiba and ITV measurements, but to obtain a greater accuracy it will be necessary access in the Horiba calibration menu and insert the effective value of the container, so that the equipment takes into account the fact that inserted fluid has the same values of the container characteristics. Moreover, each measurement device has operating range which define the measures quality. The measures unit must be incorporated inside this operating range by corresponding to a signal value, as shown on the following *Table 5.4*.

MEASURED DATA	MEASURED RANGE	SIGNAL SYSTEM
Oxygen concentration [%]	0 - 25 %	0 – 5 V
Carbon dioxide concentration [%]	0 - 20 %	0 – 5 V
Carbon monoxide concentration [%]	0 - 10 %	0 – 5 V
Hydrocarbon concentration [%]	0 – 10000 ppm	0 – 5 V

Table 5.4: concentration of the exhaust gases in relation to measured range and signal system.

It is important remember that a dilution factor has been used equal to 1,1 for the data treatment of the exhaust gases since the measurement equipment have had a small inserted of air during the test.

5.2.4 Lambda probe

It is a sensor which is going to be put in the exhaust pipe by connecting it through a simple threaded bushing, to have for each operating point of the engines the different lambda values or relative dosage. In the *Figure 5.10* the lambda probe is visible.

Figure 5.10 the lambda probe on the left and the connection through a bushing in the exhaust pipe on the right.

In order that this probe can operate, it has to be connected to 12 Volt voltage source so by transmitting sufficient energy to make the measurement of the relative dosage, which will be represented in the screen of the MOTEC signal transformation equipment. Moreover, this device has to be calibrated to a proper operating; therefore the MOTEC software will be installed and connected on a computer to start the system calibration. Once the relation between lambda probe, computer and MOTEC device, has been set, the probe will be inserted in a closed container (as shown on the *Figure 5.11*) who contains a gas with a knew composition. When this gas composition is known, the composition of the oxygen inside the container is also knew which has to coincide with the screen value for a proper calibration. If there is not this equivalence with the container, it is possible to change the oxygen value directly in the program by completing the calibration process.

Figure 5.11: the container to calibrate the lambda probe.

For the probe, it is possible to observe a direct relation between voltage and emitted signal in voltage unit, as visible in the following *Table 5.5*.

MEASURED DATA	MEASURED RANGE	SIGNAL SYSTEM
Oxygen concentration [dimensionless]	0 - 5	0 – 5 V

Table 5.5: direct relation between voltage and emitted signal in voltage unit.

5.2.5 Temperature sensors

The temperatures control in an internal combustion engine is very important since a temperatures achievement very high could cause the engine malfunction and damage it; therefore, there are a temperatures range where the engine can work optimally, and these temperatures are not even low.

For measuring the temperatures in the exhaust manifold, three K-type thermocouples are used to take the measurements of the exhaust gases temperatures just off the engine, the exhaust gases temperatures before and immediately after the catalytic. For taking the measurements of the gases in the intake manifold, a PT100 device is used since it has more precision for lower temperature values. These temperature sensors to take the measurements do not need of calibration because the used data purchase system has the possibility to indicate which kinds of thermocouples regulate automatically the calibration. The *Figure 5.12* shows the temperature sensors in the exhaust manifold.

Figure 5.12: the temperature sensors in the exhaust pipe.

5.2.6 Pressure sensor of the exhaust gases

For measuring the pressure in the exhaust manifold is used the 4075-A Kistler sensor, who is connected to the manifold through a threaded bushing. Therefore, by using this device is possible to obtain the pressures regarding the exhaust gases for the different operating points of the engine. By means of SMAR LD 301 pressure transmitter, the signal coming from the Kistler sensor is transformed in voltage, and subsequently through a constant in mbar so that to be used for the computes. The *Table 5.6* shows the correspondence between measured values and signal voltage. The pressure sensor is visible in *Figure 5.13* below.

MEASURED DATA	MEASURED RANGE	SIGNAL SYSTEM

pressure of the exhaust gases [bar]	0 – 2 bar	0 – 10 V

Table 5.6: correspondence between measured values and signal voltage, of the exhaust gases pressure.

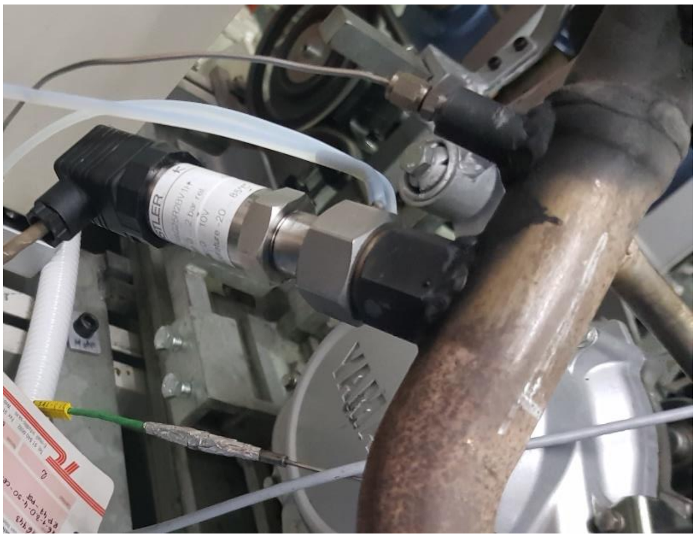

Figure 5.13: the Kistler pressure sensor connected by threaded bushing.

5.2.7 Pressure sensor of the intake manifold

The pressure in the intake manifold (MAP) is an important factor regarding the engine operating, because is closely related to the throttle body opening; furthermore, the MAP is also an important factor regarding the model for computing the injection timings, therefore a very precise measurements equipment will be needed. In fact, it has been decided to use a SMAR LD 301 pressure transmitter placed near the intake of air. This device, unlike the previous ones, operates by different pressure intensities and has a final pressure value in the relative pressure values. In the *Figure 5.14* shows the pressure transmitter. Instead, the *Table 5.7* shows the correspondence between measured values and signal in mA.

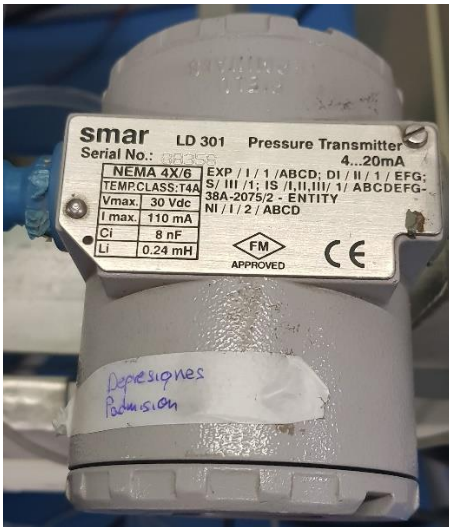

Figure 5.14: the SMAR LD 301 pressure transmitter.

MEASURED DATA	MEASURED RANGE	SIGNAL SYSTEM
intake pressure [mbar]	0 – 2.5 bar	4 – 20 mA

Table 5.7: correspondence of the pressure transmitter between measured values and signal in mA, of the intake pressure.

5.2.8 Atmospheric pressure sensor

For taking the measurements of the atmospheric pressure, another pressure transmitter has been used which operates by voltage values and its voltage ranges are suitable to the atmospheric pressure values, as shown on the *Table 5.8*.

MEASURED DATA	MEASURED RANGE	SIGNAL SYSTEM
atmospheric pressure [mbar]	0 – 1100 mbar	0 – 10 V

Table 5.8: correspondence of the pressure transmitter between measured values and signal in mA, of the atmospheric pressure.

Or alternatively, the measure of the atmospheric pressure can be taken through the screen in the trial cell, that beyond the atmospheric pressure, there are the atmospheric temperature and relative humidity, which are useful for the computations of air density. It has to be kept in mind that measurements of the atmospheric pressure are absolutes, all the others are relative.

5.2.9 Air flow sensor

For the trial system, that is a nozzle-deposit system as intake system, it has been decided to implement a low flow pressure transmitter to measure the depressions generated during intake phase and subsequently by means a spreadsheet it is possible to obtain the air flow related to an equation to nozzle-deposit pressure.

But, the selected pressure transmitter is very sensitive to the vibration generated during the engine operating; therefore, it has been decided to incorporate this flow gauge with portable measurement devices for obtaining the flow in two way. The pressure transmitter has the following operating range (*Table 5.9*):

MEASURED DATA	MEASURED RANGE	SIGNAL SYSTEM
deposit pressure [mbar]	0 – 30 mbar	4 – 20 mA

Table 5.9: operating range of the pressure transmitter for the tank.

5.2.9 Fuel consumption measurement device

The fuel consumption is one of the most important parameters and so its measurements needs an high accuracy. The measurement equipment for this parameter is the Coriolis ELITE CMF025, that a flow gauge with Coriolis effect. This devise has an ample measurement range, therefore for small measures, the measurements precision could be not very accuracy. Since, it is required an high precision, it has been decided to realize a totalizator by means of these device results. This system operates such as flue flow meter, by giving the result of total fuel consumption which is going to increase with time. The *Figure 5.15* shows the graph trend of the totalizator at 3500 rpm and WOT (wide open throttle); it has been used for stabilising the fuel consumption measurement with an higher precision, since the measurements range is till two orders of magnitude larger.

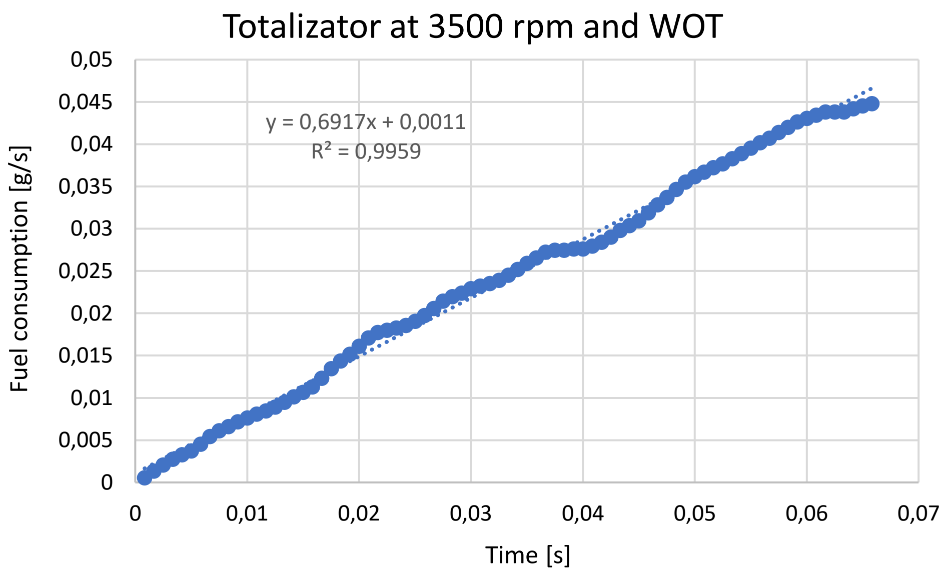

Figure 5.15: example of totalizator at 3500 rpm and WOT.

By applying a trend line on the graph of the *Figure 5.15* it is possible to obtain the equation that characterizes it and the line incline, or the angular coefficient, indicates the fuel consumption for that engine operating point in kg/h, after various measures. The *Table 5.10* shows the system signal of the fuel consumption in a defined operating range.

MEASURED DATA	MEASURED RANGE	SIGNAL SYSTEM
fuel consumption [kg/h]	0 – 250 kg/h	4 – 20 mA

Table 5.10: the signal system of fuel consumption.

5.2.10 Equipment for the signal analysis

The fundamental parameter to be measured in this project is the injection timing for each engine operating point, because the injection timings will be compared with the injection timings computed by model elaborated through Matlab and Simulink, so that model can be validated. Other measurements as rotation timings, the cylinder pressure and spark ignition timings, have been incorporated in different channel so that to be able to be viewed at their best.

The device used to measure the signals is the YOKOGAWA oscilloscope, shown on *Figure 5.16*.

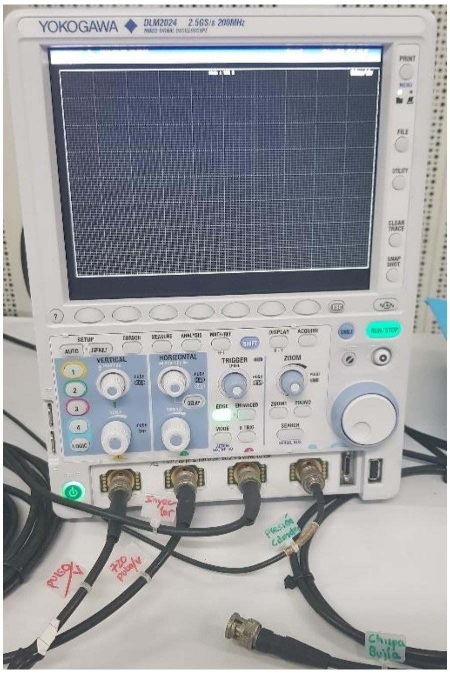

Figure 5.16: YOKOGAWA oscilloscope.

This instrument has four input channels and through it is possible to establish the proper measures for each case; furthermore, there is screen who shows a graphic image of the electric signal which is repeated during time, or more simply waveform. Regulating time and voltage, the regulation more suitable is going to be made for each test for being saved by means of signal for being studied and analysed through the Xviewer program; this program permits to study and observe in detail all part of the graphs, and so of the waveforms, in all times range. Subsequently, the signals for all channels will be explained.

5.2.10.1 Channel 1: Injection pulse

For obtaining the injection timings output from oscilloscope, it is necessary to obtain an output BNC, that is a connector used for coaxial cables, from injector. To make this, the connectors must be disconnected from injector and after making that follow a connection logic scheme. Once that cables are without connection, the injector negative cable has been punctured by an auxiliary cable which on one side has two terminals that show the positive and negative, and on the other side a BNC output. The positive terminal of this cable auxiliary is going to be punctured by the negative terminal of the injector, while the black terminal of the auxiliary cable is connected directly to the ground by having so the connection of injector with BNC output in order to be connected directly to the oscilloscope. What has been said so far is illustrated in the *Figure 5.17*.

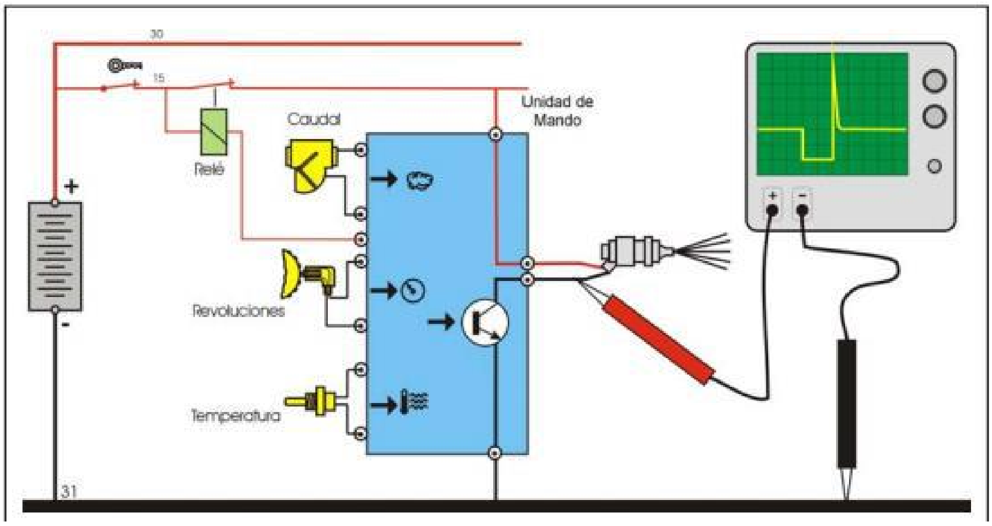

Figure 5.17: connection scheme from injector to oscilloscope.

Once that oscilloscope is able to receive the signal of the injection pulse, the following step will be to make a measurement.

The injector is powered on one side by a positive terminal, while on the other side it receives a pulse from the ground by means of the engine control unit. The terminal of injector furnishes fuel during the time where it remains to ground, so that the flow fuel depends by injector signal which remain a ground level, and right in this moment the injection pulse will take place. The *Figure 5.18* shows the injection pulse signal obtained by oscilloscope, at 3500 rpm and WOT condition. The *Figure 5.19* shows the pulse injection signal compared with crank angle signal and top dead center signal. As it is possible to see with both figures, each point of the graphs can be analysed by making a zoom (visible in the bottom of the figures), and subsequently comparing the oscilloscope signal values with those of the numerical model. Furthermore, at the end of all injection pulses, it is possible to observe a pick due to self-induction of the coil injector to came back to a voltage of 12 V where the injector is closed.

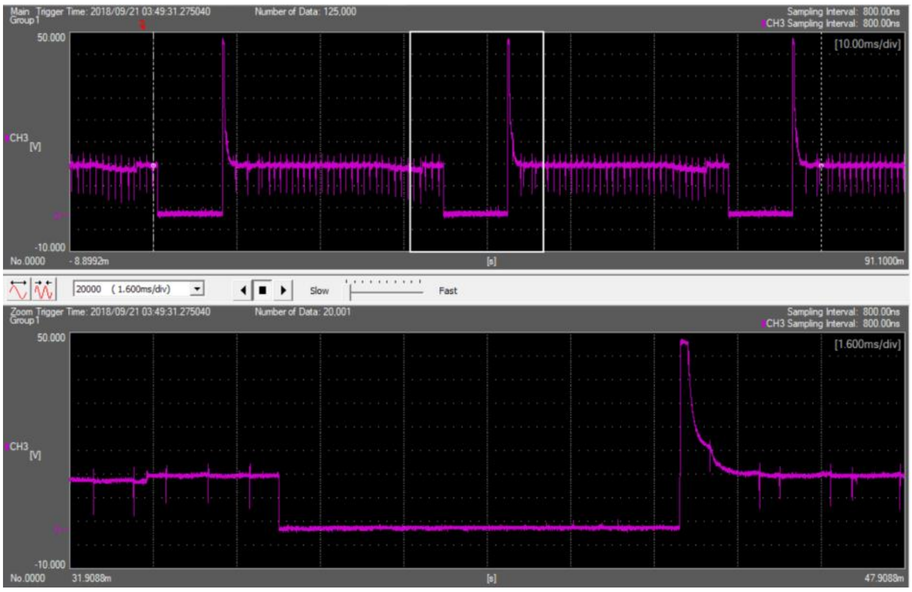

Figure 5.18: pulse injection at 3500 rpm and WOT condition.

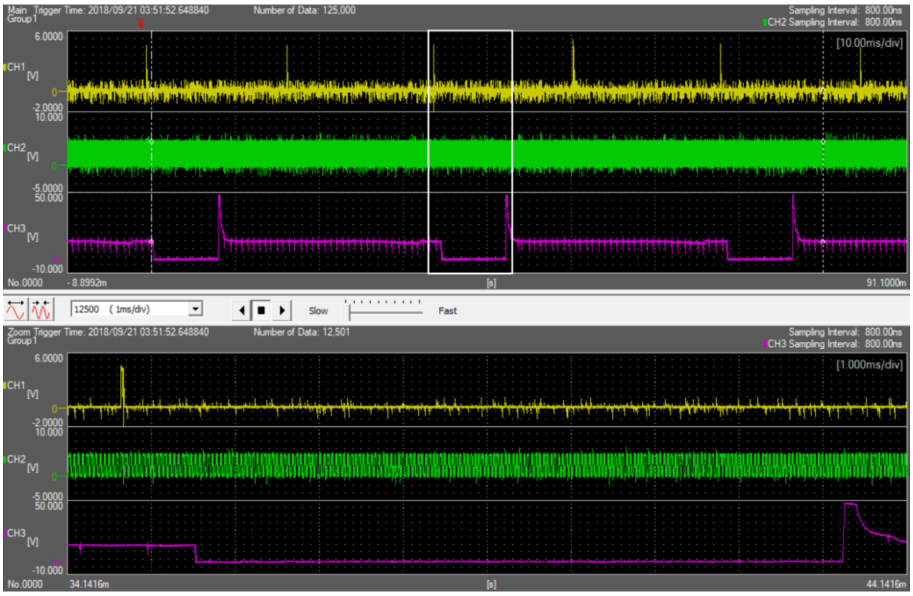

Figure 5.19: injection pulse signal compared with crank angle signal and TDC signal at 3500 and WOT.

5.2.10.2 Channel 2: Rotation pulse

The measurements of rotation pulse can be made, for example, by implemented an encoder who is connected to the crankshaft in revolution and so by rotating together, the encoder generates a transformation from mechanical movement to digital pulse. With the crankshaft movement, the infrared emitting source emits light that is red through an optic sensor which in turn is able to generate digital pulse according to the light if is able to across the disk or not able to across the disk due to the obscured region. In the *Figure 5.20* is shown the encoder integral to the crankshaft.

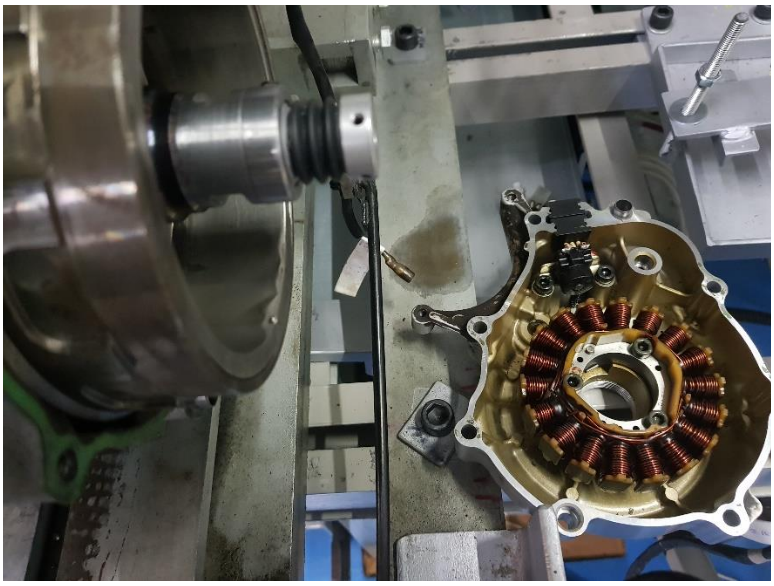

Figure 5.20: the encoder connected to the crankshaft.

For the operating process of the pulse, the source power must be a voltage of 5 V, because of it is implemented a device connected to a 5 V small source which receives the signal from the encoder to be powered by source. This device also has two BNC output so that to be able to establish a connection to BNC rotation pulse and the other output 360 degrees for one rotation. Therefore, by connecting both BNC terminal it is possible to obtain the following signals, visible in the *Figure 5.21*: the yellow line represents for each rotation a crankshaft pulse (and so the Top Dead Center where there is jump), the green line represents 360 degrees for each complete lap of crankshaft namely crank angle, and the purple line the injection signal. The example shown below is at 3500 rpm and WOT condition.

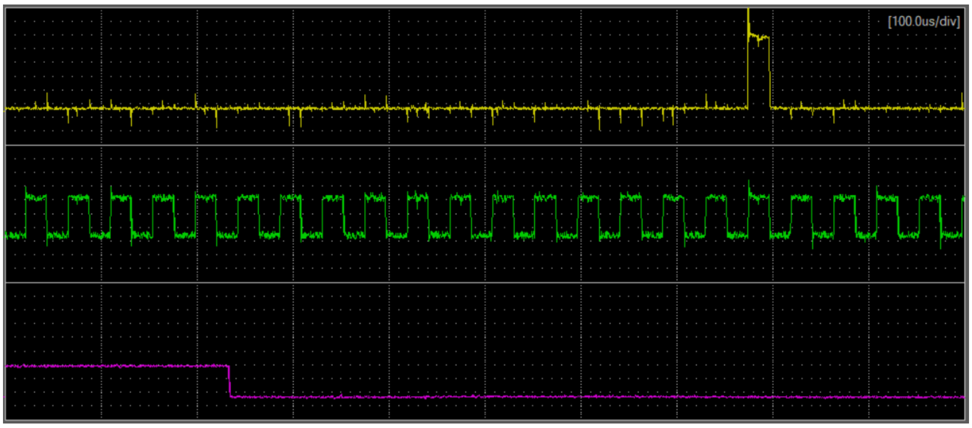

Figure 5.21. the signal of TDC, crank angle, injection pulse.

5.2.10.3 Channel 3: cylinder pressure

Another measure that will be inserted in the oscilloscope input is the cylinder pressure measurement. To establish this measurement, a charge amplifier (visible in the *Figure* 5.22) is used which is responsible of the amplification and filtering of received signal.

This measure will be great measure to establish if the encoder is aligned or not, because of the pulse rotation measurement marked by encoder corresponding to the cylinder pressure, then it is possible to say that there is not degree misalignment. Instead, if it is observed that cylinder pressure is not equal with rotation pulse, then it is possible to say that the encoder is measuring a rotation pulse that does not coincide with TDC; therefore, this situation has to be analysed to know the deviation degree which will have to be subtracted to all crank angle computations regarding for example the injection start or spark ignition, especially if it is going to make a map modification of the ECU, namely an ECU programmable.

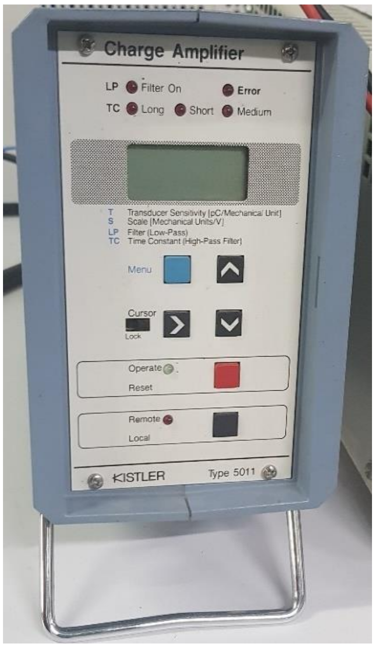

Figure 5.23: charge amplifier.

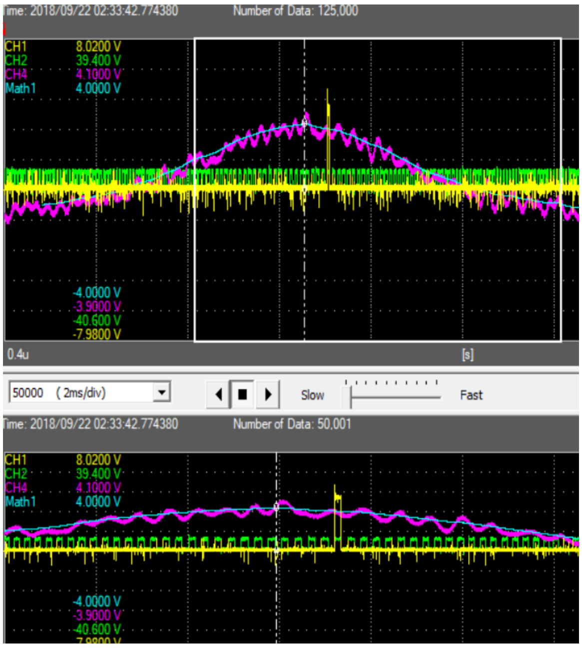

Figure 5.22: correction of the encoder deviation.

The *Figure 5.23* above shows the encoder deviation compared to the TDC and for its measure has been made cylinder pressure measurement obtained by oscilloscope. An amplifier filtering of 1 kHz is done to make more flowing the signal of cylinder pressure, while in the oscilloscope is made an IR filtering of 150 Hz for obtaining the blue line who eases the identification of the maximum point. Therefore, it is possible to note that the pressure maximum is out-of-phase compared to the TDC, by counting the difference through the crank angles, of 4,5 degrees regarding the displacement of 125 cm^3. Instead, for the displacement of 170 cm^3, after having assembled it, the out-of-phase crank angle is 39 degrees.

Regarding the pressure, the oscilloscope furnishes the signal in V which has to be converted in bar by using a simple equation:

$$K\left(\frac{bar}{s}\right) = S\left(\frac{bar}{V}\right) * \left(\frac{T}{Trang}\right)$$

The factor regarding the temperatures is a correction factor of the calibration to measure a time range low than calibration that in this case will be equal to 1. The S factor is the scale configure by amplifier and is equal to 10, therefore to convert the signal of pressure from V to bar is necessary to multiply by 10. The *Figure 5.24* shows the pressure signal at 3500 rpm and WOT condition, where the lower part shows the pressure signal trend in little more than a work cycle, that is little more than 720° of crank angle (in fact there are two TDC).

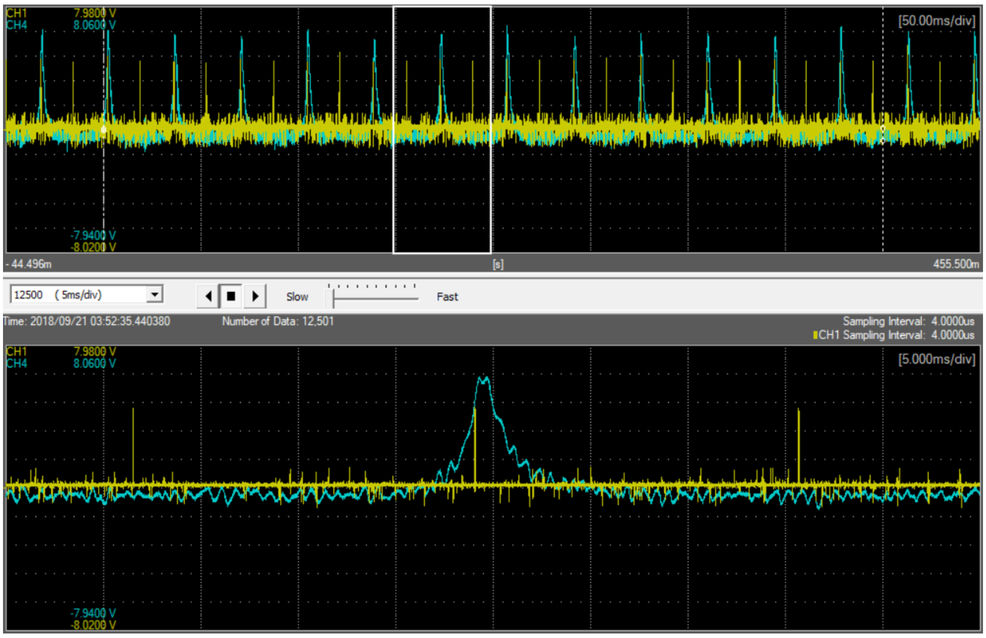

Figure 5.24: pressure (light blue line) and TDC (yellow line) signals.

5.2.10.4 Channel 4: Spark ignition

For obtaining the spark produced by the spark plug is necessary to substitute the conventional spark plug with another spark plug with character piezoelectric which incorporates a sensor who furnishes the characteristic signal wave of the spark plug. The *Figure 5.25* shows the spark plug for the tests.

Figure 5.25: the spark plug for the tests with the incorporated sensor.

Initially, the spark plug signal is zero, when the ECU send the command for the spark ignition there will be a voltage release which activates the play of spark plug electrode. Once this opening has carried out, the voltage becomes constant, while the mixture is burning until achieving the final fluctuations where the coil has been discharged and so the voltage drop has carried out at zero. Subsequently, the coil will be recharged and cycle is going to start. With the aid of crank angle signal, the crank angle where there is a jump between two electrodes can be computed, namely the crank angle of the spark ignition, by considering the encoder correction of 4,5. The *Figure 5.26* shows the spark ignition signal (light blue line), at 3500 rpm and WOT condition, where is clearly visible the ignition, the combustion TDC (through TDC signal in yellow), and the injection start.

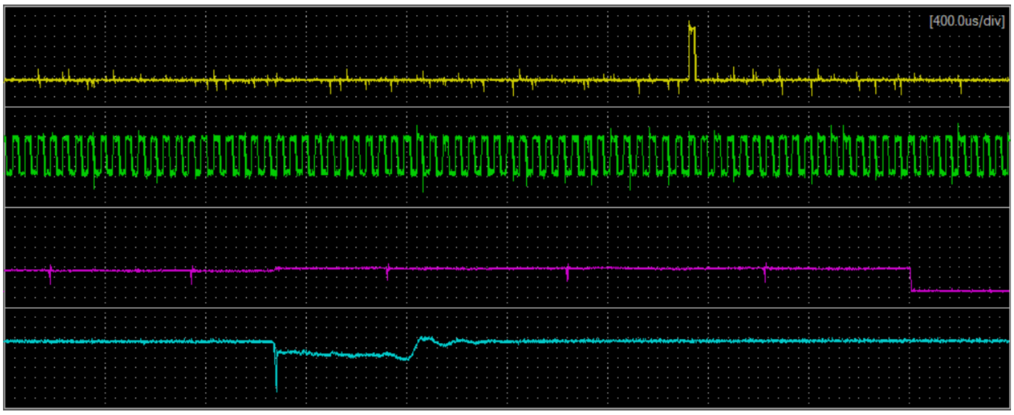

Figure 5.26: the spark ignition signal (light blue), injection signal (purple), crank angle signal (green), TDC signal (yellow).

Chapter 6: Engine test bench

Once that measurement equipment has been prepared with their calibrations, the following step will be to define a plan to make the tests. During a trial, the engine parameters measurements are made for computing all values during the engine operating (for example the useful power, the lost power, exchanged heats, and so on), and successively an analysis of the exhaust gases concentration will be done to know the emissions of the studied engine.

In according to the ISO 8665/1994 normative, the parameters which have to be measured during the tests are:

- Speed and torque of the engine;
- Temperature and pressure of the intake fluid;
- Temperature at which the fuel is found;
- Temperature and pressure of the lubrification oil;
- Temperatures, pressures and flow rate of water cooling system;
- Temperature of the exhaust gases;
- Fuel consumption;
- Emissions analysis, in particular of O_2, CO, CO_2 and HC concentration;
- Environmental conditions of the trial cell, such as pressure, temperature and relative humidity.

6.1 Procedure and protocol of the tests

The followed procedure, according to the normative above enunciated, consists of the following phases:

1. Fulfilment of measuring instrumentation for the necessaries data collection of the engine to analyse its operation.
2. Fulfilment of trial cell which provides for preparation of all measuring devices.
3. Check of the seal, proper sensors operating and gases analysers through gases with known properties (standard gas).
4. Gauges and analysers calibration process by inserting the same gas.
5. Resolution of points which must be tested.
6. Fuel analysing through its sample.

7. Data collection for stationary points during engine operating and during these measurements the following steps must be made:
    a. Engine speed and throttle position have been established and after a certain time the measures were taken, so that to be stabilised.
    b. The data acquisition system is activated for taking continuously the different parameters, during an established measuring time of 3 s.
8. After that all parameters have been taken by instrumentations, and data such as environmental conditions and exhaust gases have been taken by hands.
9. End of the tests. The zero has been set for all measurement devices for a good regulation of made measures and intake gases has been verified through the standard gas to verify if dilution has been correct during measuring process.

6.2 125 cm^3 engine tests with conventional ECU

The protocol of the tests for the 125 cm^3 engine has been carried out by following four steps:

1. The measurement range is from 3500 to 90 rpm[7], therefore the crankshaft speed has been set to 3500 rpm in wide open throttle and then the engine speed is gradually increased for measuring the stationary points until 9000 rpm to obtain the full load curve of the engine.
2. The maximum torque has been taken for each point of the previous step, that is in WOT condition, thus being able to carry out the acquisition of the data in partial load condition.
3. Five speeds uniformly distributed inside measurements range are taken and for those speeds the tests are made for 87.5%, 75%, 62.5% and 50% of the maximum torque. Each percentage of maximum torque corresponds to a certain throttle opening, therefore the throttle opening will be set to obtain the desired torque for studying each stationary point. It is going to be started

[7] The range is from 3500 to 9000 rpm because both before 3500 rpm and after 9000 rpm the combustion would not have been appreciable with measurable values not accurate, due to knocking problems. Moreover, a lot of measurements have been made until 7500 rpm because of after 7500 rpm the test bench started to vibrate by obtaining data not accurate.

with the highest speed and maximum torque, to do then the decreasing torque percentage, and so on with the others speeds in decreasing order.

4. It will proceed to measure the 37.5%, 25 and 12.5% of the maximum torque by starting from the highest speeds to measure then each point with decreasing torque percentage of the lower speeds.

After deciding the process to follow, namely as already discussed, the curve in full load, that is in wide open throttle condition, is going to be elaborated with fixed gear as for example fifth gear for the range 3500 ÷ 9000 rpm, reminding that for each point the data will have to be stabilised in order that being validates.

In the test bench it is impossible having a direct coupling to the crankshaft axes, thus a mechanical transmission is necessary. The electric motor used for measuring the torque values has a velocity limited around 6000 rpm; therefore for all tests, the fifth gear of the engine has been used, which has a transmission ratio of 9,975 (from crankshaft to wheel).

6.2.1 Full load tests

It has to remind that, the intake system during the tests is a nozzle-deposit system (in particular for the graphs analysed below); in fact the values of power and torque are a little different compared with values declared by manufacturer. For example, the maximum torque is 9.04 Nm in the crankshaft, who multiplied by transmission ratio of the fifth gear, corresponds to 90.2 Nm in the output wheel. Anyway, the manufacturer declares 12.41 Nm of maximum torque which is greater than measured value of 9.04 Nm. Regarding the maximum power, according to the manufacturer is 11 kW, and the maximum measured value during the test is 6.78 kW, namely a significant difference. However, the parameters just discussed are the actual parameters, by not considering the environmental conditions. If environmental conditions were considered, the parameters would be a little different since a correction factor must be considered who depends by environmental conditions, by obtaining the real parameters in that conditions of test bench. The correction factor is [9]:

$$\alpha = \left(\frac{99\ kPa}{(p_{test} - \Phi \times p_{s,test})kPa}\right)^{1.2} \left(\frac{T_{test}}{298.15K}\right)^{0.6}$$

Where p_{test} and T_{test} are the pressure and temperature of the cell during the tests respectively of 1022 kPa and 301.15 K (28 + 273.15), Φ is the relative humidity and $p_{s,test}$ is the saturation pressure at the T_{test} temperature. Therefore, the corrected values of power and torque, depending on trial cell conditions, will be:

$$P_{corr} = P \times \alpha, \; T_{corr} = T \times \alpha$$

Moreover, α depends by different operating points of the engine. The *Table 5.1* shows the actual values of power and torque compared by the corrected values of power and torque in the trial cell, thus with the correction factor.

n	rpm	3500	4000	4250	4500	4750	5000	5250	5500	6000	6500	7000	7500	8000
α	-	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99
P	kW	3,25	3,78	4,02	4,15	4,27	4,49	4,75	5,01	5,54	6,11	6,60	6,78	6,59
T	Nm	8,88	9,03	9,04	8,80	8,58	8,57	8,63	8,69	8,81	8,98	9,00	8,63	7,87
P_{corr}	kW	3,22	3,74	3,98	4,10	4,22	4,44	4,69	4,95	5,48	6,04	6,52	6,70	6,52
T_{corr}	Nm	8,78	8,93	8,94	8,70	8,49	8,48	8,54	8,59	8,71	8,88	8,90	8,53	7,78

Table 0.1: differences between actual values of power and torque, and corrected values of power and torque, for different rpm and WOT.

Thus, the curves of power and torque (with corrected values) have been developed in full load for each operating point of the 125 cm³ engine, that is of the crankshaft, as shown in the *Figure 6.1* and such as previously discussed, and successively these curves will be compared with the manufacture data.

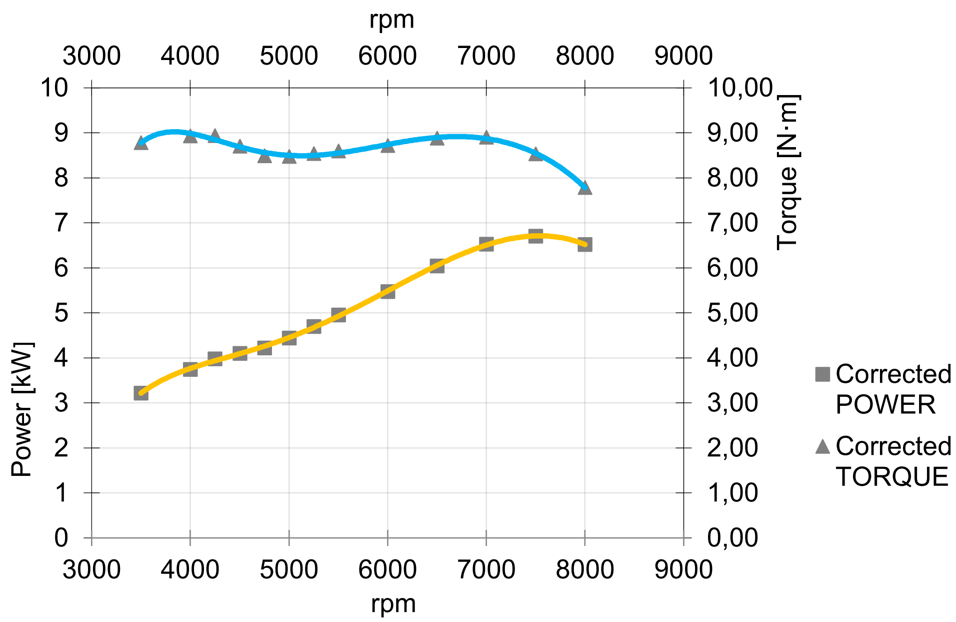

Figure 6.1: the 125 cm³ engine curves of power and torque made during the tests.

Anyway, each calculation which indicate the engine operating have been developed in a spreadsheet visible in the APPENDIX 1, where from these computations some of the most important graphs have been made.

The *Figure 6.2* shows the specific fuel consumption of the engine referring to the mechanical power that the engine is able to develop, or rather, referring to the mechanical energy. For indirect injection engines, the specific fuel consumption is approximately lower than 300 g/kWh; sure enough the fuel consumption of the 125 cm^3 is lower than 300 g/kWh, apart from a few values.

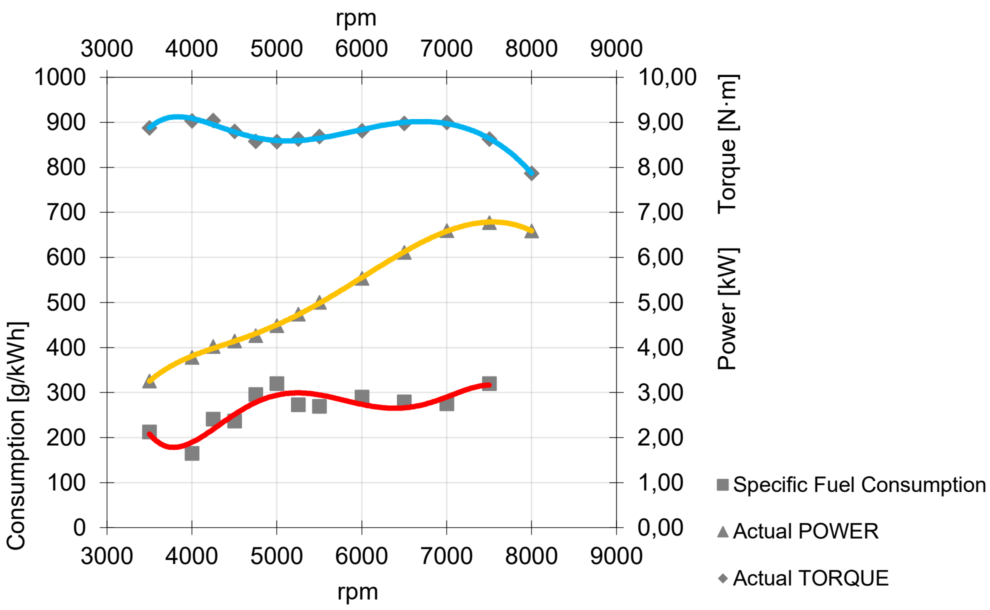

Figure 6.2: 125 cm^3 engine fuel consumption compared by power and torque.

The fuel consumption is also compared in the graph with the torque. It has been decided to make this comparation because, as it is possible to see on the graph, an increasing of torque involves a fuel consumption decreasing, instead a decreasing torque implicates an increasing of fuel consumption, by having so a close correlation. Because, the maximum torque implicates the maximum efficiency, ideally achieving lower consumption. Furthermore, in the same graph there are the actual power and actual torque compared with actual specific fuel consumption, and not real power and real torque; since, the ISO 1585 normative does not provide for a correction fuel consumption. Sure enough, the specific fuel consumption is never calculated with correction, since it is inversely proportional to the engine efficiency, whose variation is very low with environmental conditions.

It is also interesting to compare the mean effective pressure (since it is one of the most important parameters to consider during an engine design) that is p_{me}, with the actual torque, because of talking of mean effective pressure or actual torque is the same thing, since they are directly proportional, namely:

$$p_{me} \propto T \;\Rightarrow p_{me} = K \times T$$

Where $K = \frac{2\pi * n_r}{V_d}$, V_d is the total engine displacement and n_r is the number of crank revolutions for each power stroke per cylinder, which is 2 for four-stroke engines. In fact, during the design, the engine displacement required to provide a given torque or power, at a certain speed, can be estimated by assuming appropriate values of p_{me} for that determined application. Typical values of mean effective pressure for naturally aspired spark ignition engines are in the range 850 to 1050 kPa, at the engine speed where maximum torque is obtained (generally around 3000 rpm) [10]. The *Figure 6.3* shows the mean effective pressure trend compared with the actual torque for the 125 cm^3 engine in full load.

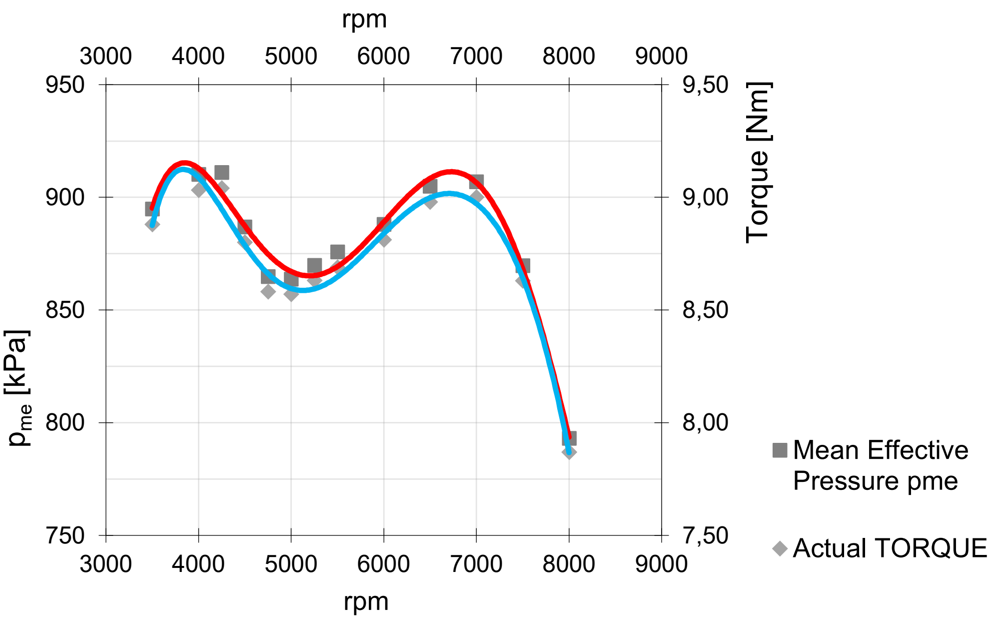

Figure 6.3: p_{me} compared to the torque in full load for the 125 cm^3 engine.

Another comparison can be made between air mass flow rate and volumetric efficiency η_v, which for each operating engine point is known as:

$$\eta_v = \frac{\dot{m}_a}{\dot{m}_{a,ref}} = \frac{m_a}{m_{a,ref}}$$

Where m_a is air mass contained in the cylinder and $m_{a,ref}$ is the air mass which cylinder could be contain at a certain temperature and pressure. The air mass taken such as reference at a given temperature and pressure, it can be considered approximately constant since the considered air is inside the trial cell, where environmental conditions are unvaried. Therefore, the volumetric efficiency is correlated to the air mass which achieves the cylinder because the air mass of reference is approximately constant; thus, the volumetric efficiency trend is similar to the trend of air mass flow rate, namely when the air mass flow rate increases or reduces, the volumetric efficiency tends to increase or reduce respectively. The *Figure 6.4* shows the trend of volumetric efficiency related to the air mass flow rate for the 125 cm^3 engine in full load.

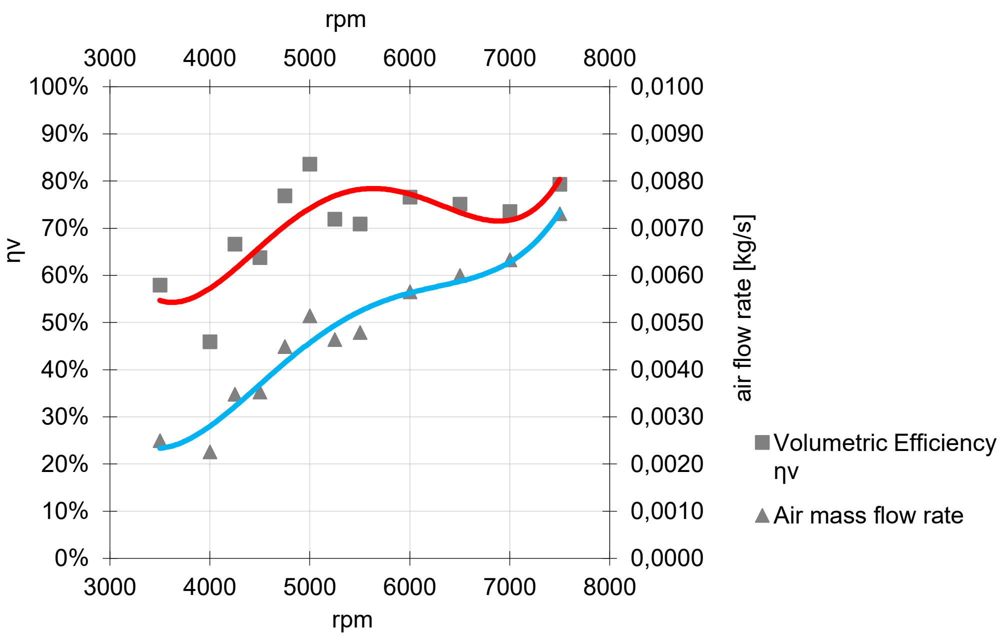

Figure 6.4: volumetric efficiency compared to air mass flow rate for 125 cm^3 engine in WOT conditions.

When it is talked about emissions, an important factor to consider is lambda or air-fuel ratio (AFR), that as already reveal in advance in the subchapter 3.1, defines the air-fuel ratio during the engine operating related to the air-fuel ratio stoichiometric, namely a relation between the amount of intaked air compared to the air theoretically needed for fuel burning. Furthermore, a lambda value equal to 1 entails smallest emissions of CO and HC, and if little by little that CO and HC concentrations increase, the lambda value decreases. The *Figure 6.5* shows the AFR compared to the intake air mass flow rate, in full load for the 125 cm^3 engine, where it is possible to see that λ decreases with increasing of rpm and air flow rate.

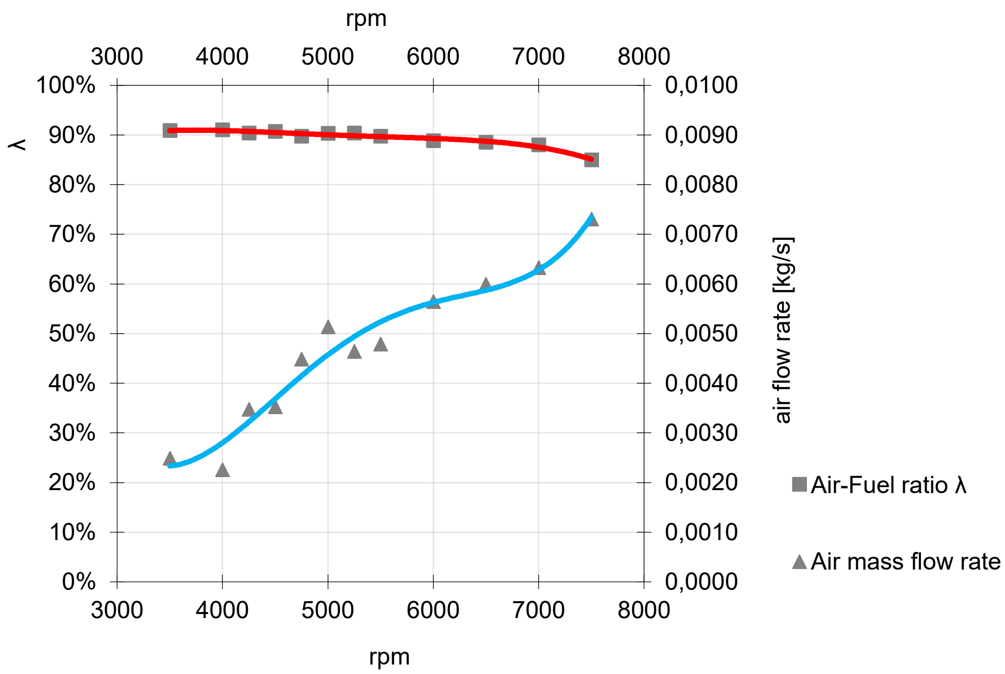

Figure 6.5: λ compared to the intake air mass flow rate in full load for 125 cm^3 engine.

The relation visible above between the trend of λ and air flow rate is a proper relation since an increasing of rpm entails an increasing of CO and CH concentrations (such as visible in the APPENDIX 1), by having a lambda decreasing related to the revolution increasing.

It is also useful to highlight the fuel consumption compared to the air mass flow rate, as shown in the *Figure 6.6*, for 125 cm^3 engine in full load.

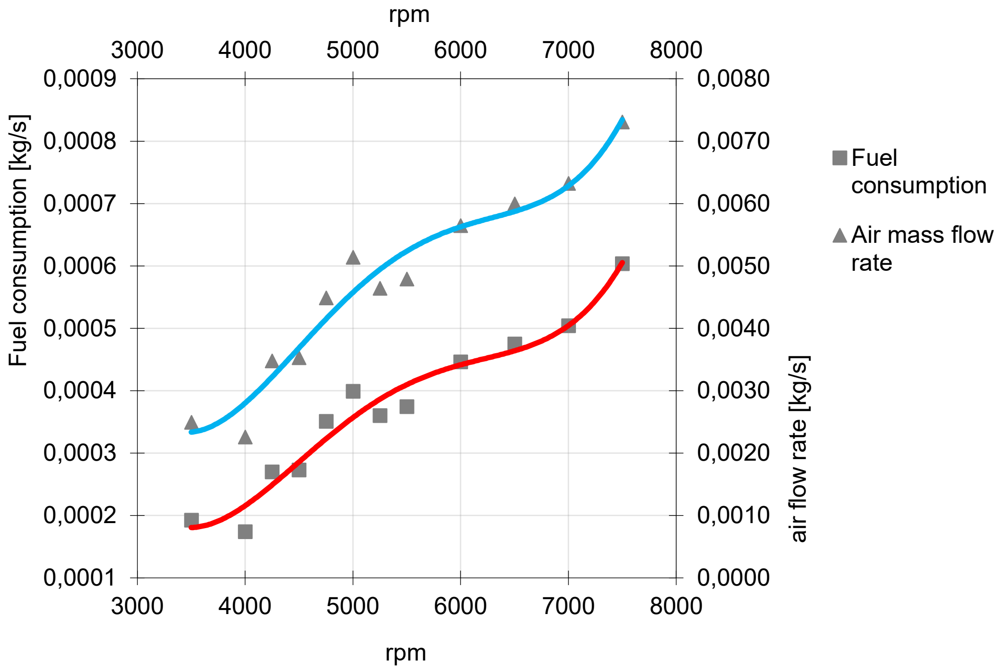

Figure 6.6: fuel consumption and air flow rate in WOT conditions for 125 cm³ engine.

It has been decided to highlight the fuel consumption in kg/s for having homogeneity with the air flow rate in kg/s, even if it would be more suitable have both of them in g/s or in kg/h. Anyway, for the engines analysed such as that in this project, it is tended to have the AFR as close as possible to the stoichiometric; thus, there must be a close correlation between them to get this. Therefore, if one of them increases, the second one must increase to compensate for their difference, and the same thing carries out if there had to be a decrease. In the graph above, this relation is clearly visible.

Probably, the most important parameter is the measuring of the engine fuel conversion efficiency since it measures the ratio between the engine useful power and the thermal power produced during the combustion, namely:

$$\eta_e = \frac{P_u}{LHV \cdot \dot{m}_f} = \frac{P_u}{LHV \cdot \frac{\dot{m}_a}{A/F}}$$

Where P_u is the engine useful power, *LHV* is the lower heating values that for the gasoline engine is around 43 MJ/kg, $\dot{m}_a$ is the intaked air mass flow rate and A/F is the ratio air-fuel. It is interesting compares the engine efficiency with the volumetric efficiency and the mean effective pressure, as shown in the *Figure 6.7*.

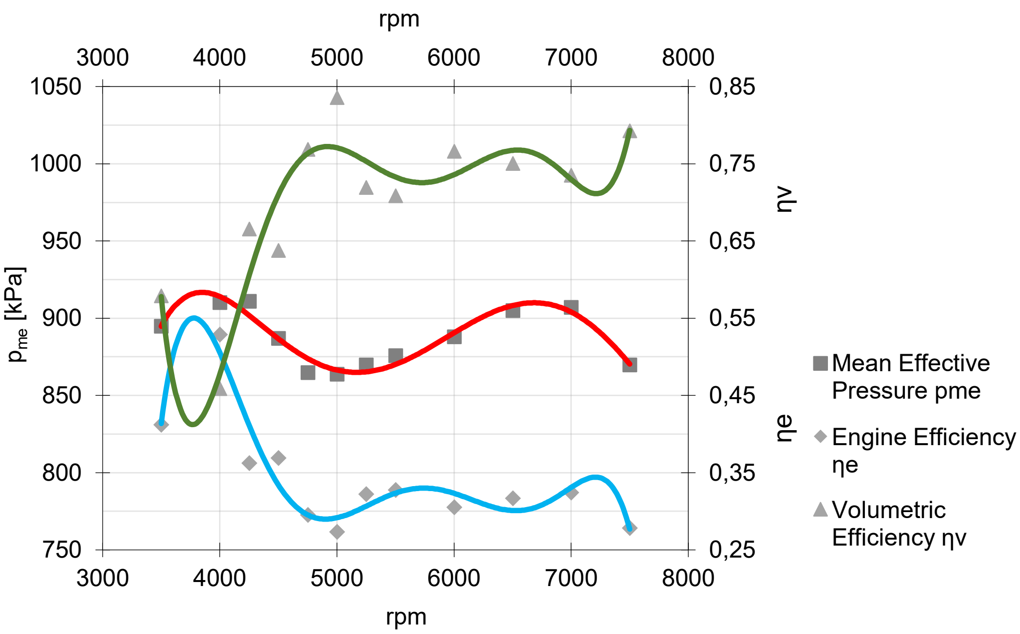

Figure 6.7: volumetric efficiency, mean effective pressure and volumetric efficiency in full load for 125 cm³ engine size.

As discussed, the volumetric efficiency tends to increase since the intaked air mass flor rate tends to increase with the rpm and the environmental conditions of the trial cell are not subjected to variations. The interesting thing is that, the engine efficiency has a trend almost symmetric to the volumetric efficiency because of the engine useful power tends to have an increasing with the rpm, but even the thermal power since the air flow rate increases with the rpm and the air-fuel ratio tends to be lower but near the stoichiometric one without being subjected to great variations. Therefore, the engine efficiency is not able to compensate the volumetric efficiency, thus the engine useful power increases, since it linearly depends on both of them. Moreover, another interesting thing is that, the mean effective pressure, because of linearly depends on the engine efficiency and volumetric efficiency, has a maximum between the two maximums of their trends, namely 911 kPa at 4250 rpm.

6.2.2 Partial load tests

The partial load tests can be made once the full load tests have completed. The tests will start from the highest speeds by maintaining one of them fixed and by decreasing the torque until 50%, and so on with the lowest speeds, by obtaining so the superior load map. For the lower load map, it is always proceeded in the same way but starting from 50% of the torque, by reducing it until 0, at the highest speeds, and so on with the lowest speeds. In this way, the engine operating mapping has been achieved with the 87.5%, 75%, 62.5%, 50%, 37.5%, 25% and 12.5% of the real torque and operating curves respectively.

The *Figure 6.8* and *Figure 6.9* show respectively the curve of power and torque with the partial loads, for the 125 cm^3 engine.

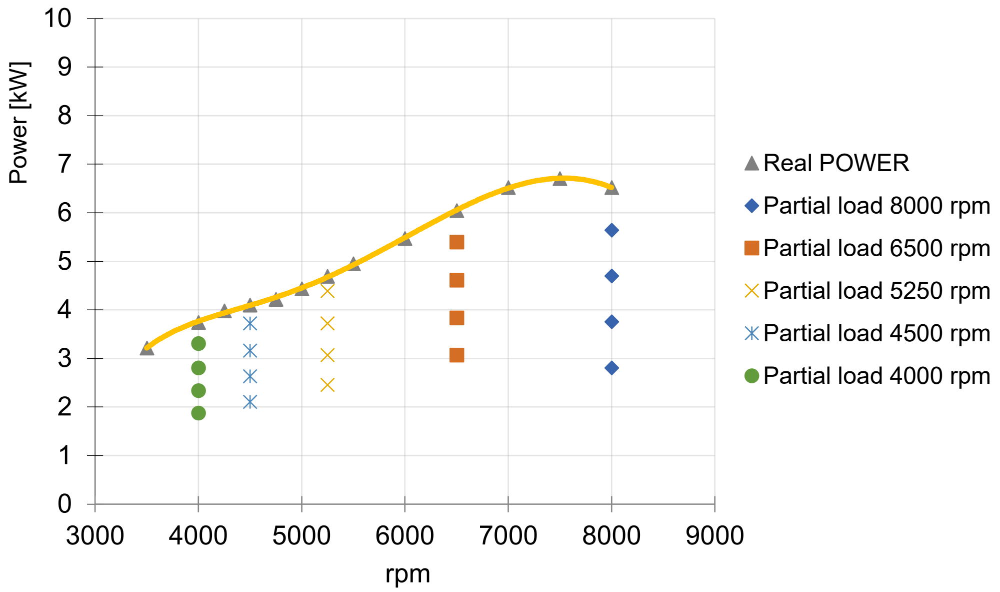

Figure 6.8: the power trend with the partial loads, for the 125 cm^3 engine.

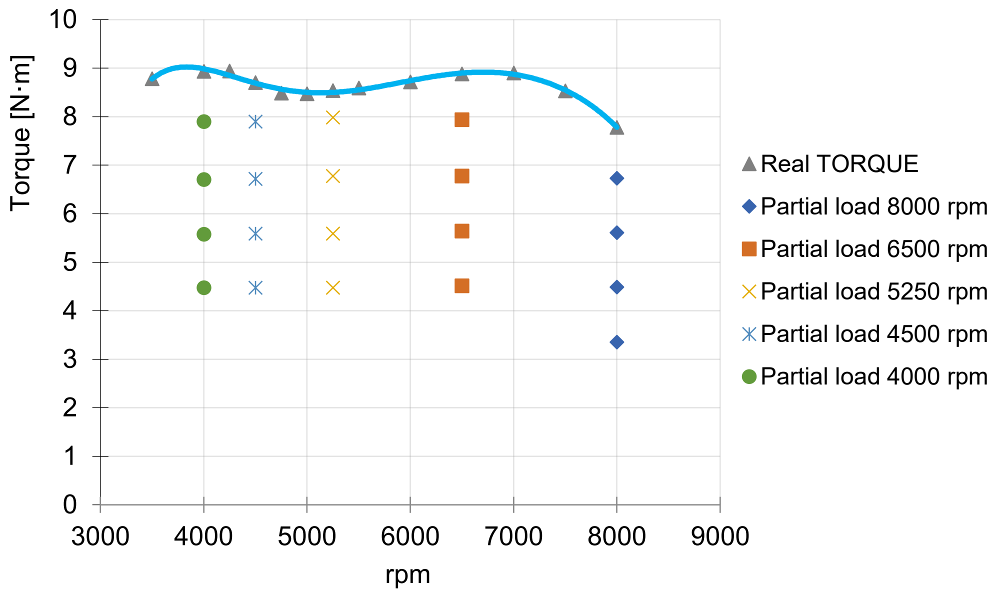

Figure 0.9: the torque trend with the partial loads, for the 125 cm^3 engine.

The speeds, where the partial load tests have been made, are 8000 rpm, 6500, 5250 rpm, 4500 rpm and 4000 rpm such as visible in the graphs above. Anyway, the other graphs with some of fundamental engine parameters as done in the previous paragraph, that is in full load conditions, are not reported because of they would be the same than partial load conditions. All calculations achieved, even for the partial load conditions, are visible in the spreadsheet of the APPENDIX 1.

6.2.3 Comparison with the manufacturer's data by conventional ECU

Once made all tests, it is interesting to compare the obtained engine parameters with the manufacturer's data, by doing two trials in full load of which one of them will be done with the conventional filter of the motorcycle, and the second by means of the nozzle-deposit system as intake system.

As previously said, the test bench has been carried out by putting the fifth gear of the motorcycle engine which will be connected to the electric motor by means of a torque wrench; moreover, the measurements made are referred to the wheel torque, thus the crankshaft torque can be obtained by dividing for the transmission ratio of the fifth gear, which is 9,975. In the APPENDIX 1 it is possible to see all data regarding these two trials in comparison with the manufacturer's data to appreciate the numerical values.

The *Figure 6.10* shows the torque trends of the trials obtained on the crankshaft[8].

[8] Reminding to divide by 9,975 which is the transmission ratio of the fifth gear, to have the crankshaft torque.

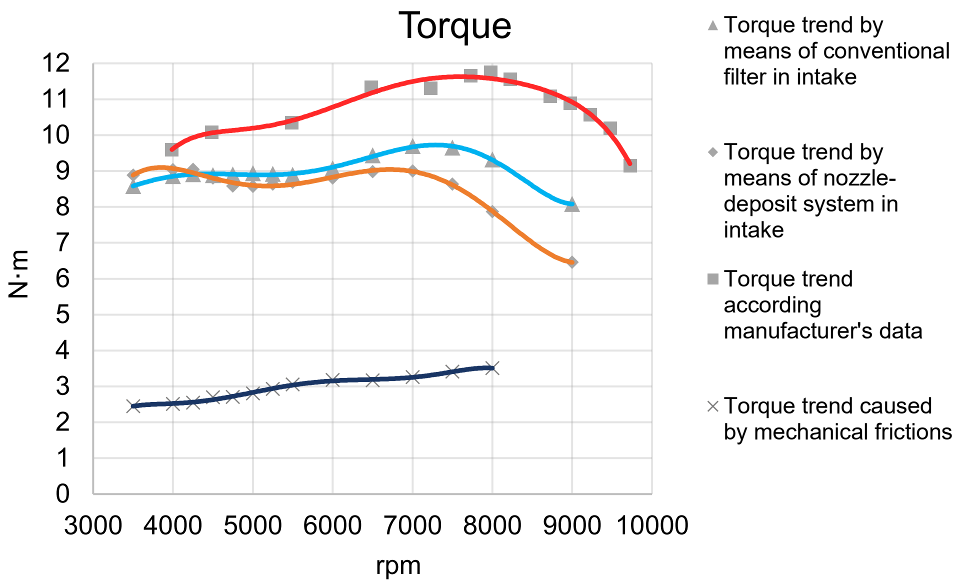

Figure 6.10: torque trend on the crankshaft by means of conventional filter in intake (light blue), by means of nozzle-deposit system in intake (orange), according manufacturer's data (red), and braking torque (blue), for 125 cm^3 engine size.

The torque trend obtained by manufacture's data is the curve with the highest values which are unlikely reachable since manufacturer's trials are made in advantageous environmental conditions and because motorcycle engine does not have wears yet (after years of using), that can be found on the other tests. The torque trend in orange is referred to the trial made by means of flow rate measurer without filter, and torque trend in light blue by means of conventional filter. As it is possible to see, the full load tests with conventional filter gives better performances than tests made without filter, by achieving a maximum torque value of 9,69 Nm at 7000 rpm (which corresponds at 96,7 Nm on the wheel) that is higher than 9,04 Nm at 4250 rpm (90,2 Nm on the wheel) obtained from tests without filter. It is also interesting to represent the braking torque in blue, understood as a resistance that opposes to the engine motion, caused by mechanical frictions; obviously, it increases with the rpm increasing since the mechanical frictions increases and the crankshaft torque must be higher than braking torque in order that crankshaft can rotate.

Another important graph is regarding the power trends (on the crankshaft) of the trials, as shown on the *Figure 6.11*.

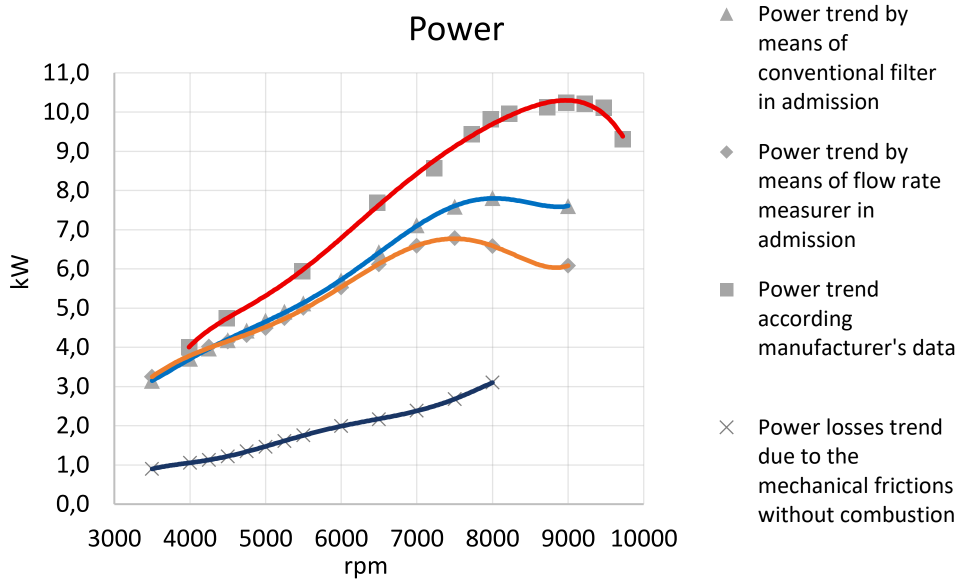

Figure 6.11: power trend on the crankshaft by means of conventional filter in admission (light blue), by means of flow rate measures in admission (orange), according manufacturer's data (red), and power losses due to mechanical frictions (blue).

The results of the power are very similar to the results of the torque, namely the curve according manufacturer's data has the highest values of power for the same reasons said above. But there is an additional power loss by using the conventional filter in admission and subsequently the flow rate measurer, thus without any filter. The main reason of these power losses is caused by the pipe which connects the tank to the intake system, that is if the pipe length is higher with curves and direction changes, there will be an higher load loss of the air mass flow rate that entails losses in the useful power calculation.

Moreover, as it can be seen on the graph, a trial has been made by dragging the engine through the electric motor, thus without combustion, for measuring the mechanical power losses, which increases obviously with rpm increasing due to higher mechanical frictions. The engine losses an amount of 3,1 kW only for the mechanical frictions at 8000 rpm (visible better in the APPENDIX 1).

6.3 170 cm^3 engine tests with programmable ECU

Regarding the engine with the displacement of 170 cm^3, the trials carried out are not the same compared to the displacement of 125 cm^3, because a new programmable ECU has been assembled and it is produced by the Polini Manufacturer. This ECU was calibrated for having an higher injection than the conventional ECU, since the displacement of 170 cm^3 needs to more fuel injection than 125 cm^3, but the ignition is the same.

Even in this case two trials have been carried out, one by means of the nozzle-deposit system as intake system, and the other using the conventional filter of the engine in the intake manifold. The first has been made only in wide open throttle conditions from 3000 to 9750 rpm (including the engine idle speed for experimental signal analysis of the oscilloscope), while the second in full load and in partial load from 3000 to 9750 rpm[9]. The engine operating points measured are similar to the test bench of 125 displacement, but for the trials made with conventional filter there are less points because the more significant points have been chosen for measuring the torque; moreover regarding the partial load, the tests have not been made in the same way of 125 displacement, that is by choosing different torque values for obtaining different throttle opening, but rather by choosing different values of throttle opening, namely 20% of WOT, 40% of WOT,50% of WOT, 60% of WOT, 80% of WOT, since these are the more significant percentages of full load for the programmable ECU.

6.3.1 Full load tests

As the lower displacement, the data analysed in this paragraph will be regarding the trials with the nozzle-deposit system as intake system. Therefore, for the 170 cm^3 engine it is possible to obtain (as the 125 cm^3 engine) the correction factor α depending by the environmental conditions of the trial cell for calculating the corrected power and corrected torque, and comparing them with the actual power and actual torque, as it is shown on the *Table 6.2*:

n	rpm	3000	3500	3750	4000	4250	4500	4750	5000	5250	5500
T	Nm	12,95	13,38	13,60	13,44	13,53	13,45	13,49	13,41	13,17	12,95
P	kW	4,07	4,90	5,34	5,62	6,01	6,33	6,70	7,01	7,23	7,45
α	-	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99
T_{corr}	Nm	12,81	13,23	13,45	13,29	13,38	13,30	13,34	13,26	13,02	12,80
P_{corr}	kW	4,02	4,85	5,28	5,56	5,95	6,26	6,63	6,93	7,15	7,36
n	rpm	6000	6500	7000	7500	8000	8500	9000	9250	9500	9750
T	Nm	12,74	12,99	13,27	12,97	12,49	11,57	10,44	9,64	9,20	8,88
P	kW	8,01	8,84	9,72	10,18	10,45	10,30	9,83	9,33	9,14	9,05
α	-	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99

[9] For the displacement of 170 cm^3, the test bench has been enlarged until 9750 rpm because of the mechanical transmission was subjected to a better alignement in relation to its axis, so that the problems regarding the vibrations after 7500 rpm were minimized.

T_{corr}	Nm	12,60	12,85	13,12	12,83	12,35	11,45	10,33	9,53	9,10	8,78
P_{corr}	kW	7,92	8,75	9,61	10,07	10,34	10,18	9,73	9,23	9,04	8,95

Table 6.2: the actual power and actual torque, the correction factor α and the corrected power and corrected torque in relation of rpm, for displacement of 170 cm³.

Thus, the curves of power and torque (with corrected values) have been developed in full load for each operating point of the 170 cm^3 engine, that is of the crankshaft, as shown in the *Figure 6.12*.

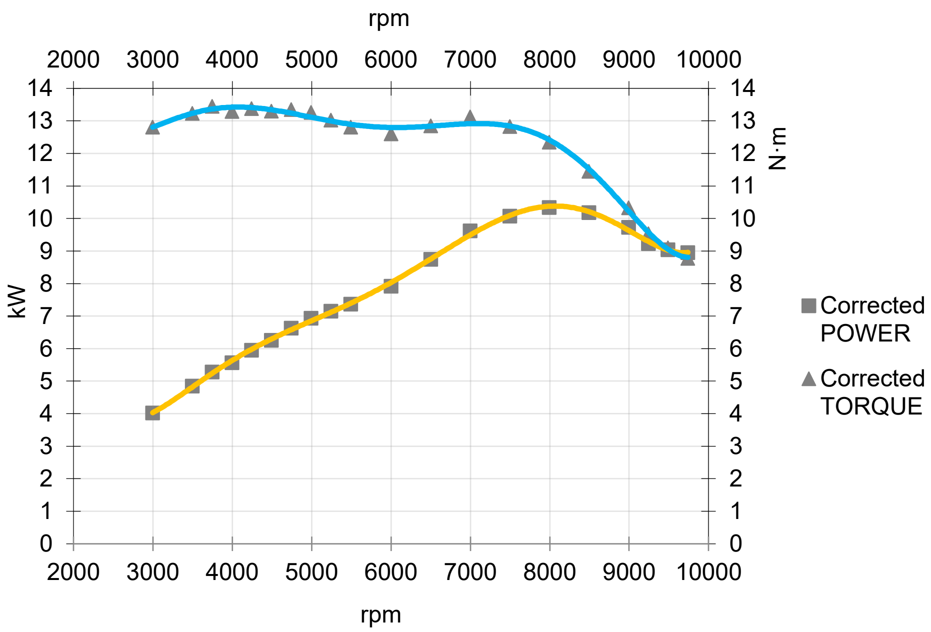

Figure 6.12: the curves of corrected power and corrected torque for displacement of 170 cm³.

As expected, the curves of power and torque have an higher values compared to the lower engine size. In particular, the maximum power is 10,34 kW at 8000 rpm. Regarding the torque, the maximum value is 13,45 Nm at 3750 rpm, reminding that, by multiplying this value for the transmission ratio of 9,975, the torque value on the wheel will be obtained, namely 134,14 Nm.

The *Figure 6.13* shows the specific fuel consumption of the engine referring to the mechanical power that the engine is able to develop, or rather, referring to the mechanical energy. Even in this case the specific fuel consumption does not approximately exceed the 300 g/kWh, except few values.

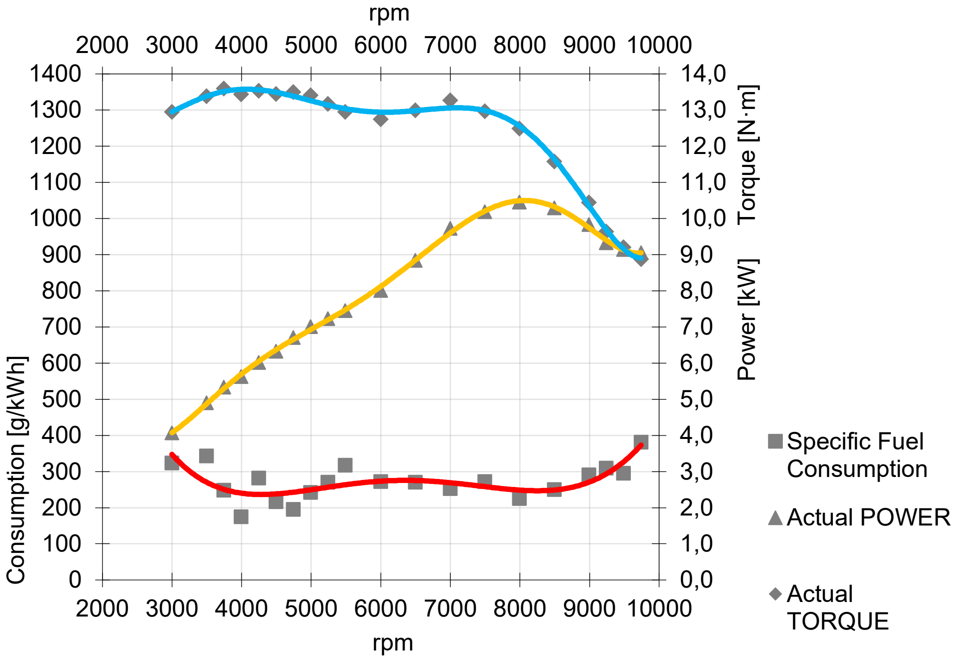

Figure 6.13: 170 cm³ engine fuel consumption compared by power and torque.

Even in this case the fuel consumption is compared in the graph with the torque because of, as already discussed previously, there is so a close correlation, reminding that the fuel consumption does not have to be corrected in function the environmental conditions such as established by the ISO 1585 normative.

The *Figure 6.14* below, shows the mean effective pressure trend in comparison to the actual torque.

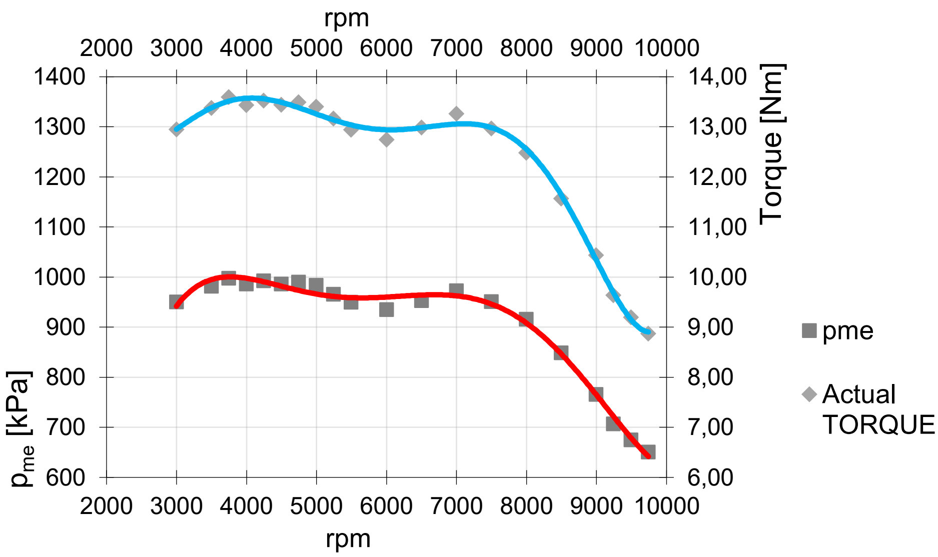

Figure 6.14: the p_{me} and torque for the engize size of 170 cm³ in full load.

The p_{me} is one of the most important parameters during the design phases of an engine, and as already anticipated, talking about of the p_{me} or the torque is the same thing, due to their direct proportionality through a constant which contains the total engine size and the number of crank revolutions for each thermodynamic cycle.

The *Figure 6.15* shows the volumetric efficiency compared to the air mass flow rate.

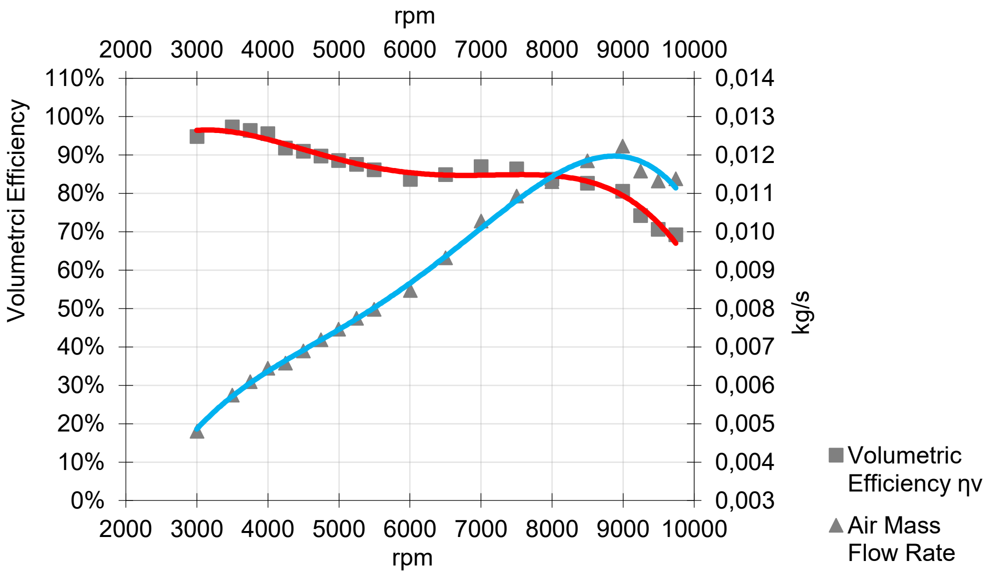

Figure 6.15: the volumetric efficiency in comparison to the air mass flow rate for displacement of 170 cm³ in full load.

This time there is a difference from the graph of the lower engine size. The air mass flow rate increases because of it is close correlated to the rpm. But the volumetric efficiency does not follow the same trend. First of all, the connecting pipe of the nozzle-deposit system in intake regarding the 125 cm^3 engine size is more length than the engine size of 170 cm^3; thus, there will be an higher pressure drops which are going to decrease the cylinder filling. In fact, the volumetric efficiency of the 170 cm^3 displacement has higher values, which anyway are going to decrease with the rpm increasing due to the higher pressure losses. Afterwards, the volumetric efficiency is maximum approximately in the same point the maximum torque (in fact the maximum volumetric efficiency is 0,97 at 3500 rpm and the highest values of torque are between 3500 and 4500 rpm).

The *Figure 6.16* shows the air-fuel ratio λ in comparison to the volumetric efficiency. As it is possible to see, the AFR tends to decrease little by little with the rpm increasing because of the CO and HC concentrations increase while the air flow mass increases with the rpm increasing.

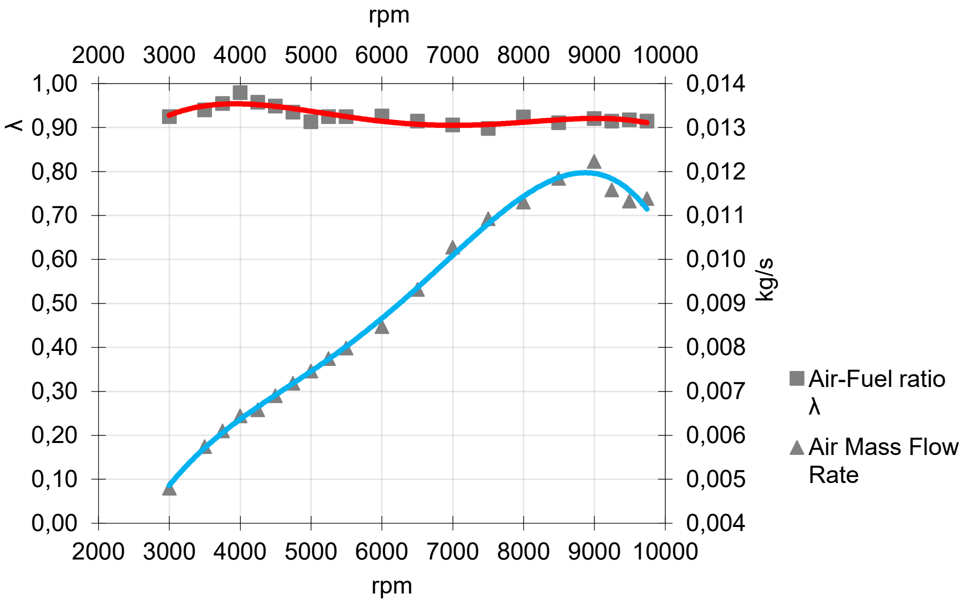

Figure 6.16: the AFR in comparison to the air flow rate for the engine size of 170 cm³ in full load.

The *Figure 6.17* shows the fuel consumption compared to the air mass flow rate. It has been decided to draw both the curves in kg/s for having more homogeneity, even if would be better having the fuel consumption in kg/h to be quantified better.

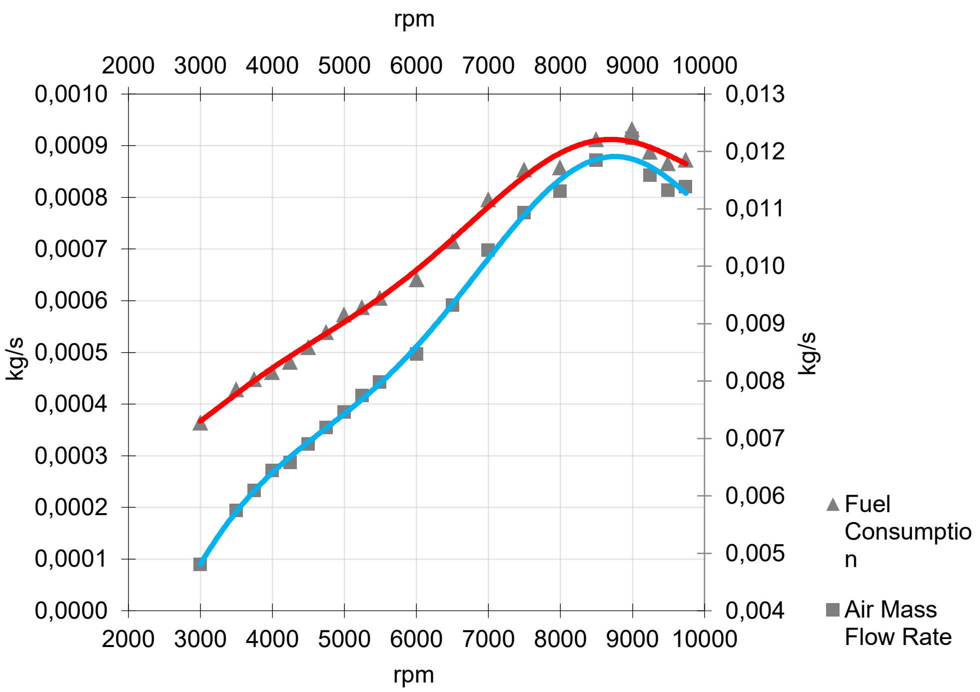

Figure 6.17: the fuel consumption and the air mass flow rate in full load for the engine size of 170 cm³.

Anyway, there is a close correlation between them, because of:

$$\dot{m}_f = \frac{\dot{m}_a}{A/F}$$

And since the term A/F tends to be slightly lower but close to the stoichiometric one for each engine operating point, the fuel consumption and the air flow rate have a trend very similar, because an increasing of one of them with the rpm increasing, entails an increasing of the other and vice versa.

The last graph analysed is shown in the *Figure 6.18*. Regarding the engine efficiency, compared with the lower displacement, is more linear without any particular peak, being included between 20% and 30%, and at the same time is lower than the efficiency of the 125 cm^3 engine size, since there was not a power increasing directly proportional to the displacement increasing.

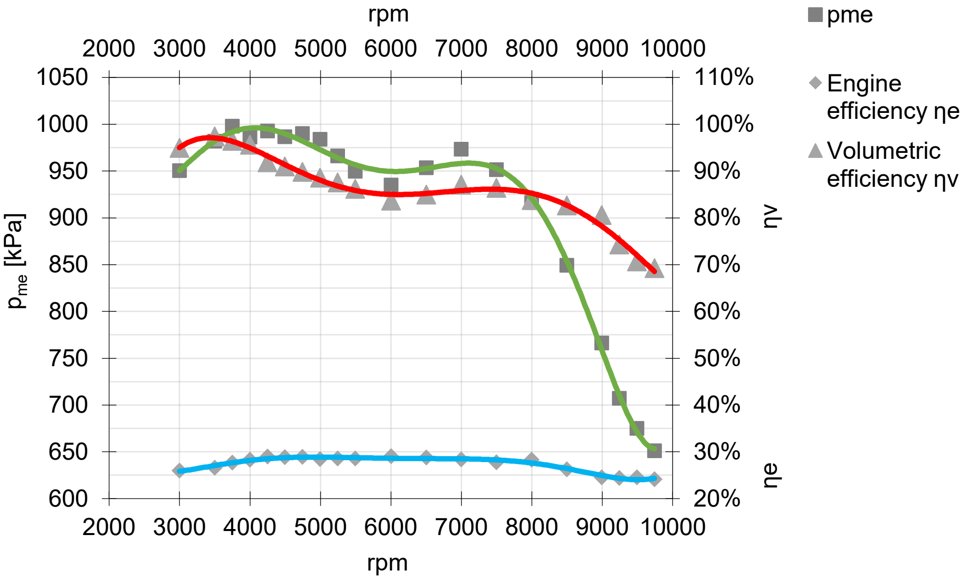

Figure 6.18: the p_{me}, the volumetric efficiency and the engine efficiency in full load for the displacement of 170 cm^3.

As well as the volumetric efficiency is more linear and also higher compared to the filling of the lower displacement. The interesting thing is that, the mean effective pressure has a maximum point between the highest value of filling and the highest value of efficiency, due to the their directly proportionality to the p_{me}, namely 997,8 kPa at 3750 rpm.

Anyway, all engine parameters can be seen in the APPENDIX 1.

6.3.2 The comparison of the two trials

The test bench has been made with two kind of intake system: the first by means of nozzle-deposit system using the flow rate meter (the Coriolis ELITE CMF025 device) and the second through the conventional filter; reminding that, the all trials have been carried out with the fifth gear of the engine. The measured torque is on the wheel, thus for obtaining the crankshaft torque it is necessary to divide for 9,975 (which is the fifth gear transmission ratio). The *Figure 6.19* shows the torque curves for the two kind of trials in full load, as well as the *Figure 6.20* shows the power trends.

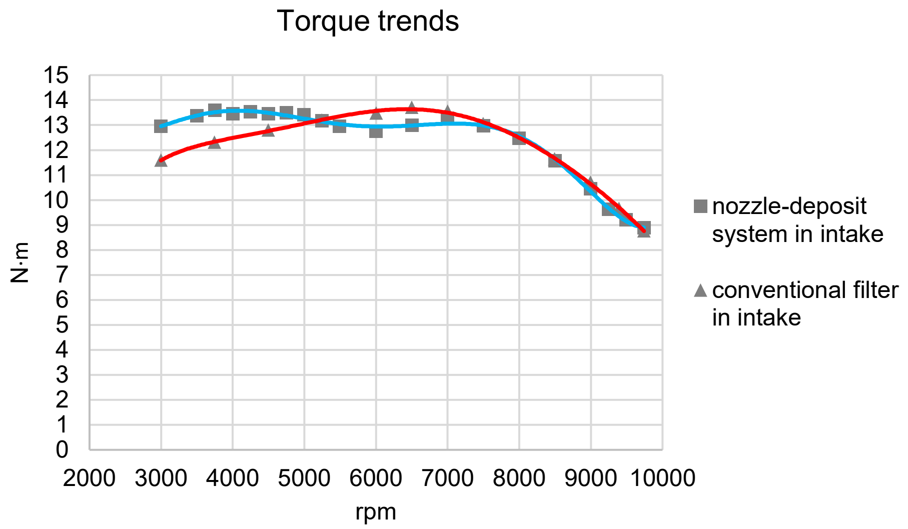

Figure 6.19: torque trends for the two kind of trials for the 170 cm³ displacement in full load.

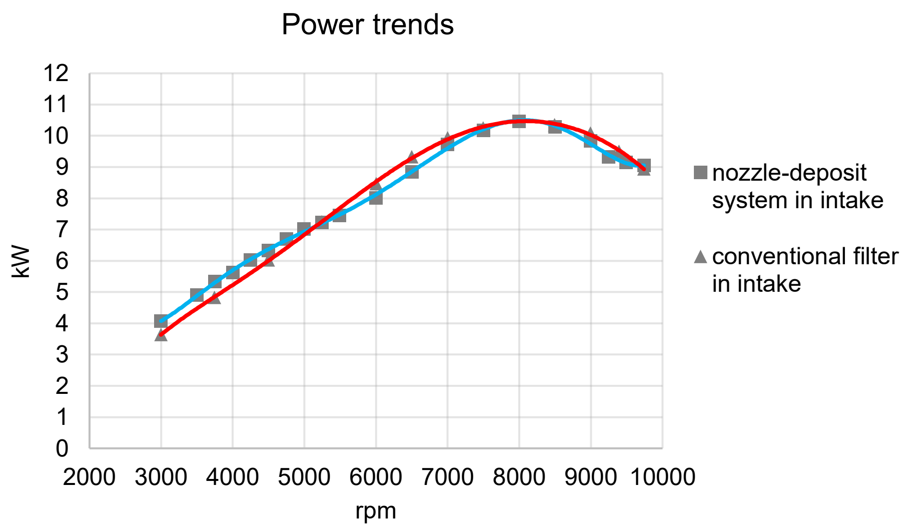

Figure 6.20: power trends for the two kind of trials for the 170 cm³ displacement in full load.

First of all, it is possible note that both curves of power and torque have closely trends compared to the trials carried out for the lower engine size of 125 cm^3. The substantial difference is caused to the different connecting pipe of the two trials, because of in the 125 cm^3 displacement has been assembled with an higher length of the pipe to the deposit, thus there will more pressure drops than the engine with the higher cubic capacity. For this reason the performances of the 170 cm^3 displacement are very similar for the two kind of tests. Moreover, for low rpm until approximately 5250 rpm the performances are better for test made with the nozzle-deposit system, because of the air flow can be considered such as incompressible flow, namely the fluid density is not subjected to appreciable variations by having a subsonic regime with a Mach number lower than 0,3. This is the reason why the torque measured with this intake system is higher and consequently also the calculated power; in fact the maximum is 135,6 Nm on the wheel and 13,6 Nm on the crankshaft at 3750 rpm. For high rpm, instead, the nozzle-deposit system has an higher pressure drops since the air flow can be considered such as compressible flow, that is the fluid density is subjected to appreciable variations by having a supersonic regime with a Mach number higher than 1. For this reason after 5250 rpm the engine performances with the conventional filter are higher or at maximum equals compared to the engine performances with the nozzle-deposit system in intake. In fact, the maximum torque measured during the test for the engine with conventional filter is 136,7 Nm on the wheel and 13,7 Nm on the crankshaft at 6500 rpm. The maximum power is for both trials at 8000 rpm (since depends by the rpm): for the test made with the nozzle-deposit system is 10,45 kW, instead regarding the test made with conventional filter is 10,46 kW.

Chapter 7: Results of the oscilloscope experimental signal analysis

For the model validation, the two kind of ECUs will be analysed and described step by step in this chapter, namely the estimation of injection timings, of the ignition timings and of the pressure in the thermodynamic cycle, for the two type of engine cubic capacity. The experimental signal analysis of the oscilloscope are referred to the test bench made with the flow rate meter, that is through the nozzle-deposit system as intake system. The analysis provides for the study of oscilloscope signal in full load and in partial load.

7.1 Displacement of 125 cm^3 with conventional ECU in WOT

7.1.1 Injection timings analysis

The injection timings analysis deals the measuring the injection pulse width, but also the injection start timings in crank angle and respective times in ms.

Regarding the start of injection, the data have been taken by the YOKOGAWA oscilloscope and with the Xviewer program have been reported on the computer. In the following step it is shown as it is proceeded, by considering the example of 3500 rpm. In the *Figure 7.1* it is shown the signal of TDC (the yellow line), crank angle (the green line), injection (purple line), more or less in one thermodynamic cycle.

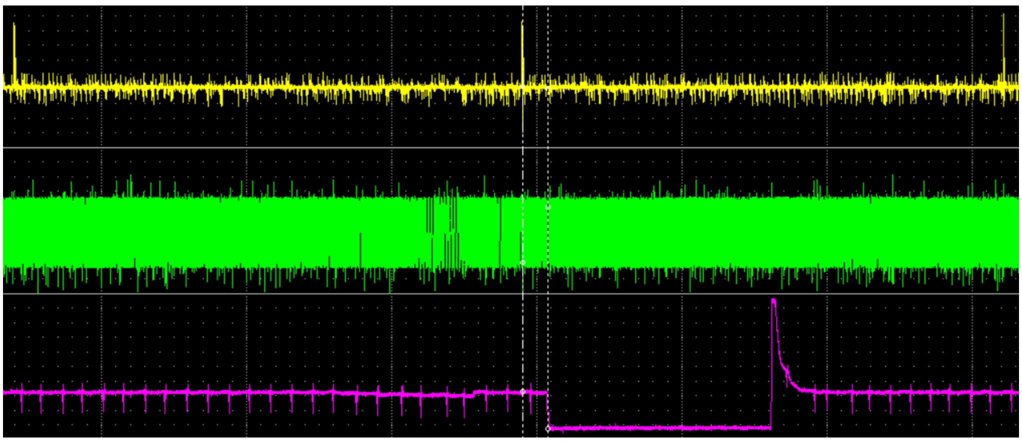

Figure 7.1: the signal of TDC, crank angle and injection at 3500 rpm in WOT for the engine size of 125 cm^3.

With more zoom it is possible analyse the crank angle signal of injection, as shown in the *Figure 7.2*, around the combustion TDC.

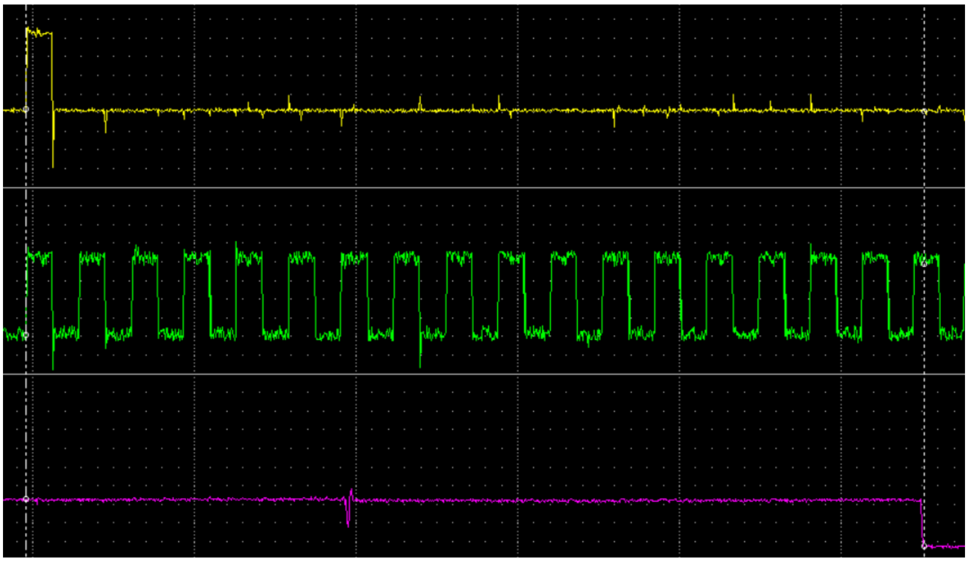

Figure 7.2: signal analysis at 3500 rpm with more zoom for cubic capacity of 125.

The peak of yellow line, on the left part, indicates the TCD and when the purple line takes a leap the injection has started. It is possible to see better the full injection on the *Figure 1*. For calculating the crank angle, it is necessary counting one cycle of the green line, that is after one above line and down line the crankshaft has carried out one degree of rotation. In this case, at 3500 rpm, the crank angle of injection start is 17°, after combustion TDC; but, it needs to consider and to subtract the encoder deviation which is to 4,5°. Therefore, the correct crank angle of injection start is 12,5°, that is 372,5° at after intake TDC and the respective time is approximately 17,7 ms. The *Table 7.1* shows, from left to right: the rpm from 3500 to 9000, the crank angles around the combustion TDC already corrected, the crank angles after the intake TDC and respective times.

speed	CA around combust.TDC	CA intake TDC	timings
rpm	degrees	degrees	ms
3500	12,5	372,5	17,738
4000	-16,5	343,5	14,313
4250	-16,5	343,5	13,471
4500	-46,0	314,0	11,630
4750	-45,5	314,5	11,035
5000	-75,5	284,5	9,483
5250	-75,5	284,5	9,032
5500	-75,0	285,0	8,636

6000	-105,0	255,0	7,083
6500	-134,0	226,0	5,795
7000	-164,0	196,0	4,667
7500	-196,0	164,0	3,644
8000	-213,5	146,5	3,052
9000	-263,5	96,5	1,787

Table 7.1: injection start timings in crank angles and times in WOT for engine size of 125 cm^3.

It can note that for high rpm the injection starts before the BDC.

Regarding the injection timings, they have been calculated through the oscilloscope software, and the respective crank angle, namely an interval that indicates the crank angle duration in degrees. These values are indicated on the *Table 7.2*, from 3500 to 9000 rpm:

speed	injection timings	crank angles
rpm	ms	degrees
3500	7,683	161,3
4000	7,953	190,9
4250	7,863	200,5
4500	7,913	213,6
4750	7,878	224,5
5000	7,889	236,7
5250	7,748	244,1
5500	7,778	256,7
6000	7,898	284,3
6500	8,018	312,7
7000	8,408	353,1
7500	8,502	382,6
8000	8,700	417,6
9000	8,600	464,4

Table 7.2: injection timings in WOT for engine size of 125 cm^3.

It is also interesting reporting the injection timings in crank angle degrees in function of the rpm, so that it is possible to have clearer how the injection can variate in function of these kind of parameters, namely the rpm and the crank angle degrees, as shown in the *Figure 7.3*.

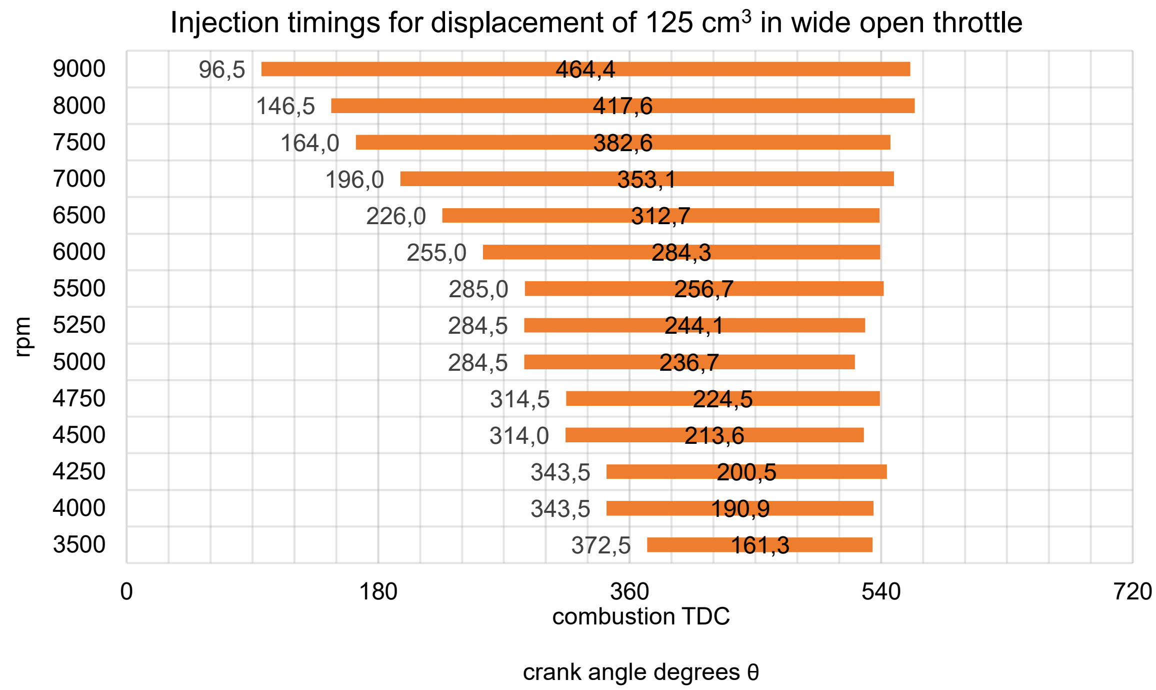

Figure 7.3: injection timings in crank angle degrees in function of the rpm for 125 cubic capacity in WOT.

It can be noted that the injection finishes for all rpm around the BDC of the power stroke phase, but the beginning of the injection is subjected to increase with the rpm increasing, as can be expected. Some manufactures, for the engines with the indirect injection, that is in the intake manifold, carry out this choice for making simpler the design, namely settling the injection end around a constant point and then variating the start of the injection. Despite a part of injected fuel will remain in the intake manifold since the valve opening intake is about 180° (something extra considering the opening advance and the closing delay), it is going to begin to mix with the air mass in the manifold, and once the intake valve opens and the piston moves down, the mixture not completely formed is going to move into the cylinder due to the lower pressure created inside the chamber.

7.1.2 Spark ignition timings analysis

The analysis of spark timings has been made in the same way of the others analysis. The crank angle values have been taken by the oscilloscope signals with Xviewer, and then respective times have been calculated. The process is the same as injection start timings analysis; it has been taken into consideration the example at 3500 rpm visible through the graphs of *Figure 7.4*, approximately in one work cycle, where the yellow line indicates the TDC signal, the green line is the crank angle signal and the light blue line is the signal of the spark.

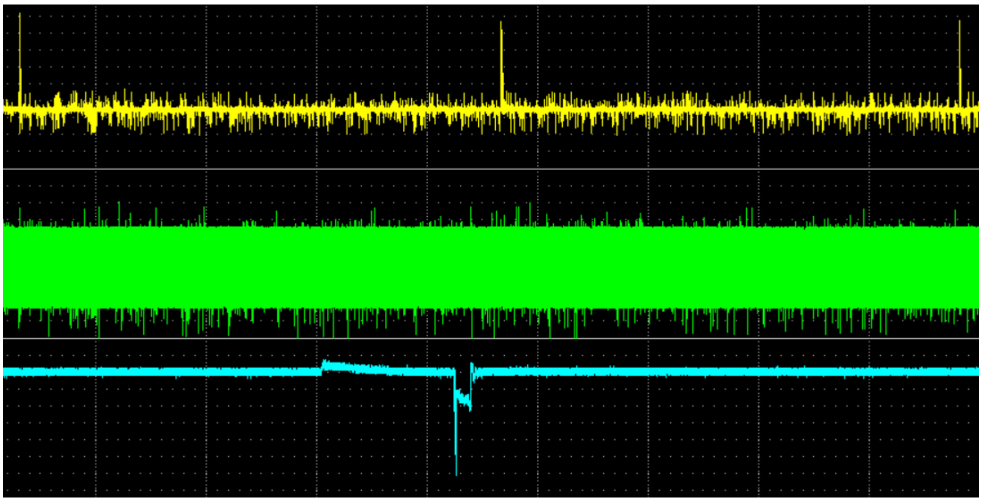

Figure 7.4: TDC signal, crank angle signal, spark signal at 3500 rpm for the cubic capacity of 125.

To see with more precision, it is possible making a zoom of the *Figure 7.4* for obtaining the *Figure 7.5* where the crank angle appears clearer and thus, it can be calculated.

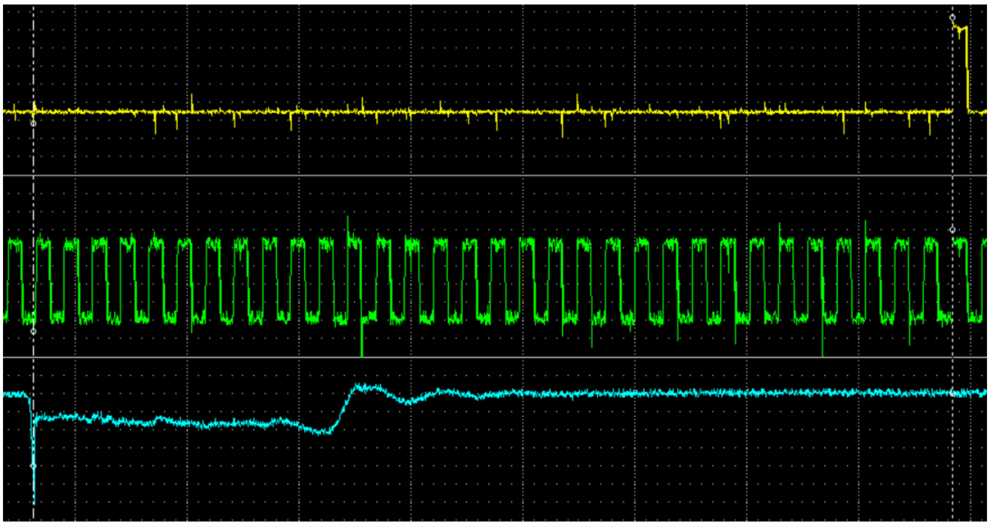

Figure 7.5: spark analysis at 3500 rpm with more zoom.

The spark ignition achieves when the light blue line (spark signal) carries out a peak, and it is before the combustion TDC. Hence, by counting the green cycle from left yellow peak to light blue peak, namely from right to left, the crank angle, with correction of 4,5°, is 27,5°. On the *Table 7.3*, from left to right: the speed in rpm from 3500 to 7500 rpm, the crank angles calculated before the combustion TDC and with the encoder deviation, the crank angles after the intake TDC, and respective time in ms that crankshaft carries out to those rotations. It can be noted that measured data of crank angle are more or less constants, since the spark ignitions are not subjected to high variations.

speed	CA before combust. TDC	CA intake TDC	ignition timings
rpm	degrees	degrees	ms
3500	36,5	323,5	15,405
4000	36,5	323,5	13,479
4250	40,5	319,5	12,529
4500	41,5	318,5	11,796
4750	40,5	319,5	11,211
5000	40,5	319,5	10,650
5250	40,5	319,5	10,143
5500	40,5	319,5	9,682
6000	41	319	8,861
6500	36,5	323,5	8,295
7000	36,5	323,5	7,702
7500	37	323	7,178

Table 7.3: the ignitions timings in ms and in crank angles in full load for displacement of 125 cm^3.

7.1.3 Pressure signal analysis

The pressures analysis is a bit different from the analysis treated so far. To calculate the maximum pressure inside cylinder expressed in bar, it can be used the following formula:

$$p_{max,cyl} = \left(p_{int,abs} - p_{int,rel}\right) + k \cdot p_{peak}$$

where $p_{int,abs}$ is the intake absolute pressure measured in mbar in the intake manifold, $p_{int,rel}$ is the intake relative pressure measured around the BDC after the intake phase in a 20 degreed interval and measured in mVolt, p_{peak} is the pressure peak inside cylinder. To convert the pressures values in bar, it is used a constant $k = 10\ \frac{bar}{Volt}$ (as explained in the paragraph 5.2.10.3). In the *Figure 7.6*, where the purple line is pressure signal, it is possible to see how it has been measured the value of $p_{int,rel}$ at 3500 rpm, that is by making an average of pressures data in a 20 degrees range around the BDC.

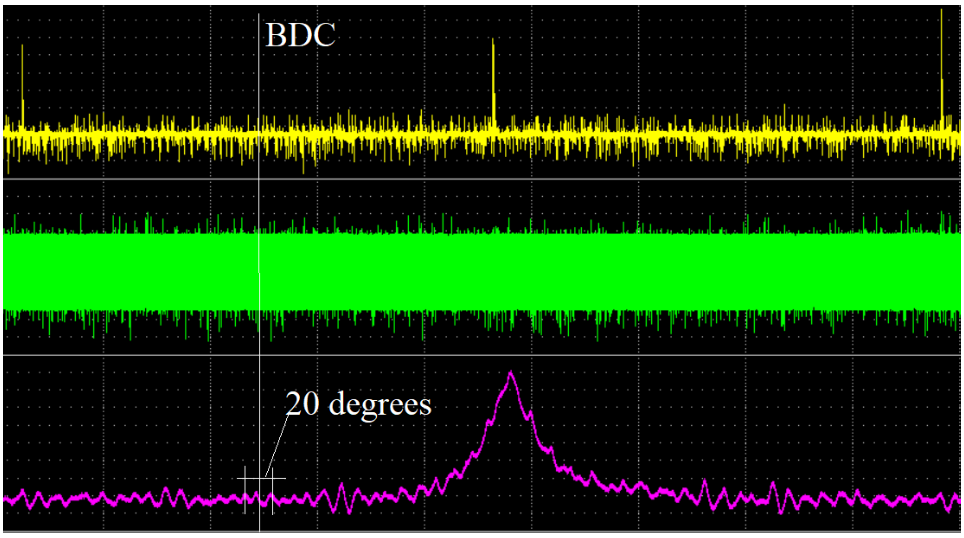

Figure 7.6: pressure trend at 3500 rpm in one work cycle, with the $p_{int,rel}$ value, for 125 cm^3 displacement.

Moreover, it can be noted that the pressure peak is a bit on the right of combustion TDC just as it should be. The values of pressure peak are calculated through an average of all data of pressure peaks; since in one graph there are more thermodynamic cycles, and then more values of pressure peaks (green line) such as shown in *Figure 7.7* at the engine speed of 3500 rpm, where the yellow line is the TDC signal.

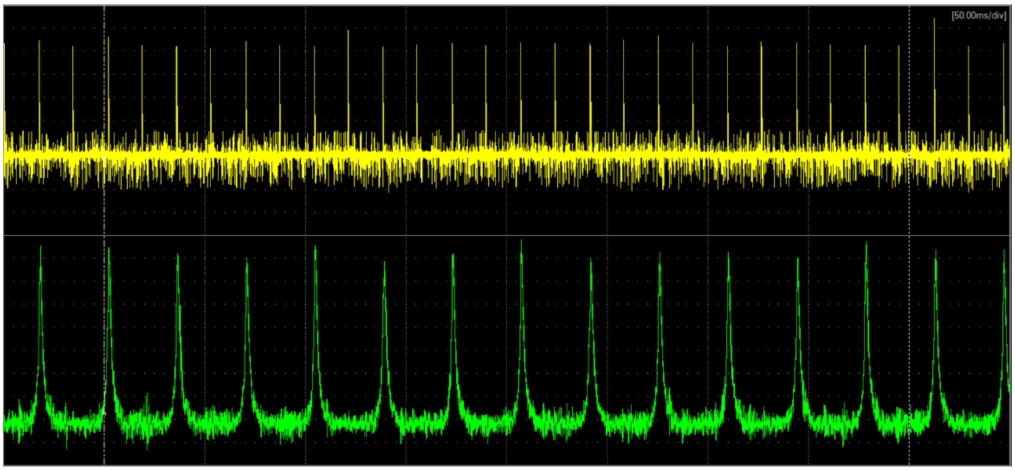

Figure 7.7: all pressure peaks signals at 3500 rpm regarding the engine size of 125 cm^3.

For example at 3500 rpm, it is possible calculating the cylinder pressure maximum $p_{max,cyl}$:

$$p_{max,cyl}(3500rpm) = \left(\frac{1015,11}{1000} - 10 \cdot \frac{-300}{1000}\right) + 10 \cdot 6,391 \approx 67,93\text{bar}$$

In the *Table 7.4* are shown all values of the pressure signal analysis, thus from left to right: the engine speed in rpm, the maximum values of pressure peak in Volt, the minimum values of pressure peak in Volt, the averages of pressure peaks that is p_{peak} in Volt, the intake absolute pressure in the intake manifold $p_{int,abs}$ in mbar, the intake relative pressure $p_{int,rel}$ in mVolt, the maximum mid-pressure which achieves during the combustion inside the cylinder in one thermodynamic cycle namely $p_{max,cyl}$ in bar.

speed	$p_{peak,max}$	$p_{peak,min}$	p_{peak}	$p_{int,abs}$	$p_{int,rel}$	$p_{max,cyl}$
rpm	V	V	V	mbar	mV	bar
3500	6,825	5,975	6,391	1.015	-300	67,928
4000	7,175	6,406	6,779	1.015	-600	74,809
4250	7,329	6,131	6,942	1.016	-350	73,934
4500	7,338	5,881	6,722	1.019	-400	72,242
4750	6,913	5,574	6,393	1.020	-800	72,945
5000	7,571	6,533	7,201	1.018	100	72,028
5250	6,698	5,238	6,022	1.013	-700	68,238
5500	7,176	5,728	6,474	1.010	-25	66,005
6000	7,179	5,736	6,518	1.009	-500	71,186
6500	7,013	5,129	6,293	1.008	-50	64,439
7000	7,256	5,639	6,409	1.005	-100	66,095
7500	7,175	5,723	6,479	1.009	250	63,301

Table 7.4: pressure values relatives to the signal analysis in full load for the displacement of 125 cm^3.

Moreover, it is interesting to represent the graphs which show the correlation between the maximum pressure achieves in the thermodynamic cycle and the torque to the rpm variate, as shown in the *Figure 7.9*, and the other correlation between the maximum pressure and the ignition timings as shown in the *Figure 7.10*, by making an interpolation of the operating points using a polynomial equation.

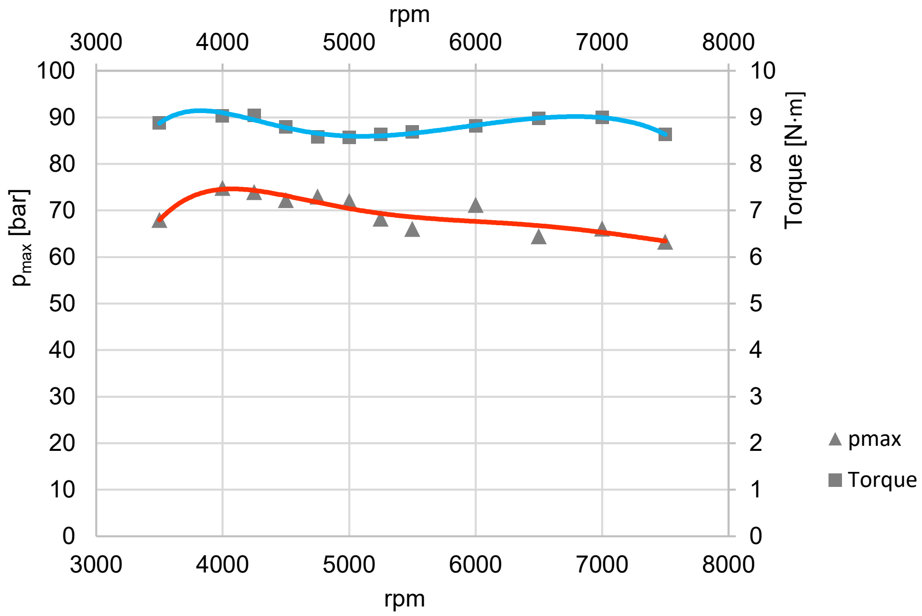

Figure 7.8: comparison between the maximum pressure and the torque in WOT for 125 cubic capacity.

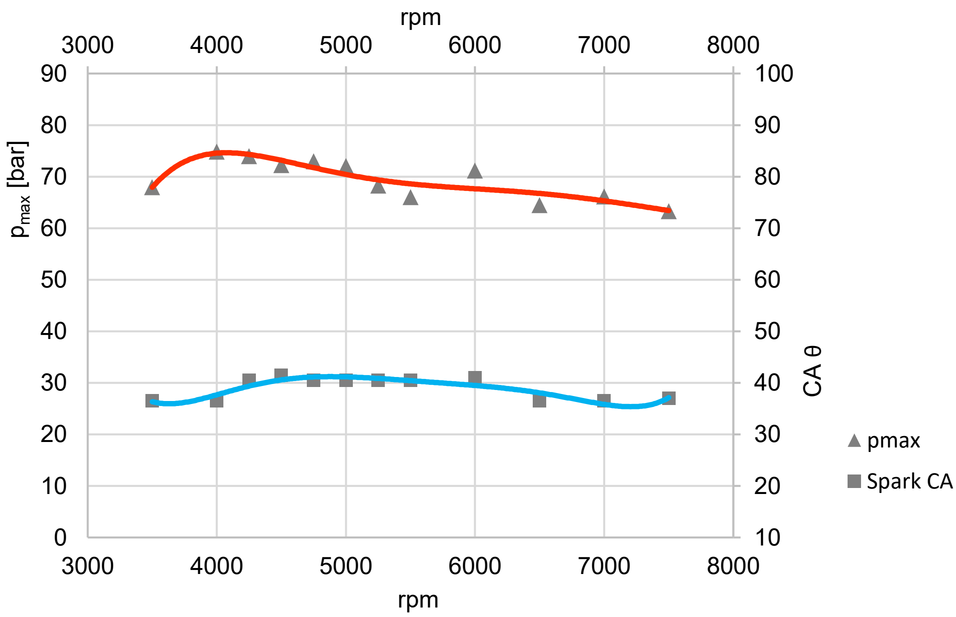

Figure 7.9: comparison between the maximum pressure and the spark ignition in crank angles in WOT for 125 cubic capacity.

The curves have been obtained for both figures by using an interpolations which have a similar trends. As known, the mean effective pressure has the same trend of the torque because of the volume considered is a constant since is equal to the engine cubic capacity; but the maximum pressure does not achieve the highest value in the same piston position since the spark ignition timings are not the same for each operating engine points (in fact for the spark ignition advances there will be a lower pressure, and for spar ignition delays there will be an higher pressure, which is clearly visible in the *Figure 7.10* above), and the volume considered is not a constant because carries out a variation between 14,8 cm^3 and 17,5 cm^3, calculated from the maximum pressures. Hence, there is a cubic capacity variation equal approximately to 2,7 cm^3 due to the different maximum pressures, caused also from the different ignition timings, in crank angles. For this reason the graphs of the maximum pressure and torque have not exactly the same trend.

7.2 Displacement of 125 cm^3 with conventional ECU in partial load

7.2.1 Injection timings analysis

Even for the partial load, the injection timings analysis deals the measuring the injection pulse width, but also the injection start timings in crank angle and respective times in ms. The torque varies for 37,5% to 87,5% of the maximum torque, from 4000 to 8000 rpm, by having different percentages of WOT in bases of the rpm.

In the *Figure 7.9* the full injection analysis is shown at 4000 rpm and 50% of the maximum load, where the yellow line is the TDC signal, the green line is the crank angle signal and the purple line is the injection signal. For the calculation of crank angle, it is proceeded such as the subchapter 7.1, and it is possible to see better the crank angle signal around the combustion TDC with more zoom in the *Figure 7.10* to calculate it.

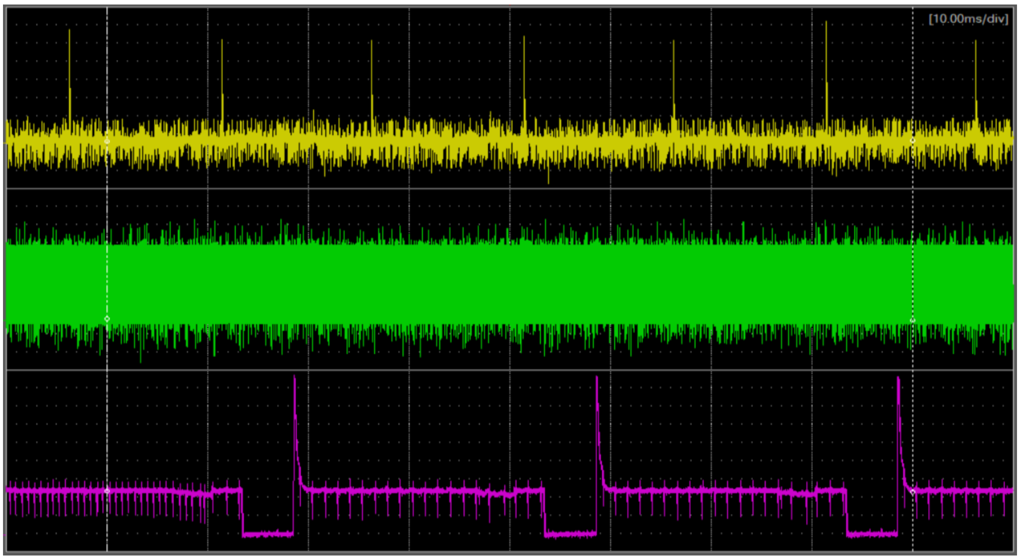

Figure 7.10: full injection analysis at 4000 rpm and 50% of the load for 125 cubic capacity.

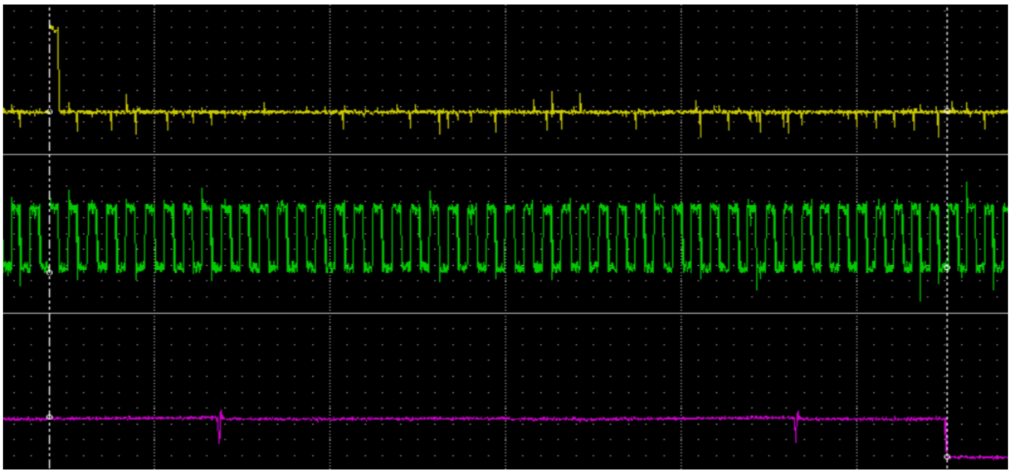

Figure 7.11: TDC signal, crank angle signal, injection signal at 4000 rpm and 50 % of the load for 125 cubic capacity.

In the *Figure 7.10* the TDC and injection signal are clearly visible, hence by counting the cycle of crank angle it is possible to measures it. Therefore, at 4000 rpm and 50% of partial load the crank angle, with correction, is 43,5° after the combustion TDC, namely 403,5° after the intake TDC. The values of all injection timings in ms and degrees are shown in the *Table 7.5*, where from left to right it is possible to see: the percentage of load, the percentage of open throttle, the rpm, the crank angles around the combustion TDC with correction, the crank angle after the intake TDC, the timings in ms.

%Torque max	%WOT	speed	CA around combust.TDC	CA after intake TDC	timings
-	-	rpm	degrees	degrees	ms

50,0	21,2	4000	43,5	403,5	16,813
62,5	27,9	4000	13,0	373,0	15,542
75,0	43,3	4000	-17,0	343,0	14,292
87,5	54,4	4000	-16,5	343,5	14,313
50,0	25,2	4500	43,5	403,5	14,944
62,5	33,0	4500	13,5	373,5	13,833
75,0	42,4	4500	-16,5	343,5	12,722
87,5	57,4	4500	-16,5	343,5	12,722
50,0	29,8	5250	14,0	374,0	11,873
62,5	37,9	5250	-15,5	344,5	10,937
75,0	48,1	5250	-46,0	314,0	9,968
87,5	61,6	5250	-75,5	284,5	9,032
37,5	25,2	6500	15,5	375,5	9,628
50,0	31,4	6500	-14,5	345,5	8,859
62,5	37,5	6500	-74,0	286,0	7,333
75,0	42,5	6500	-104,0	256,0	6,564
87,5	59,2	6500	-134,5	225,5	5,782
37,5	28,2	8000	-38,5	321,5	6,698
50,0	36,3	8000	-133,5	226,5	4,719
62,5	47,0	8000	-163,5	196,5	4,094
75,0	59,6	8000	-193,5	166,5	3,469

Table 7.5: injection start timings in crank angles and times in partial load for engine size of 125 cm^3.

As previously done for the wide open throttle conditions, it is also interesting reporting the injection timings in crank angle degrees in function of the rpm in a graph as shown in the *Figure 7.11*.

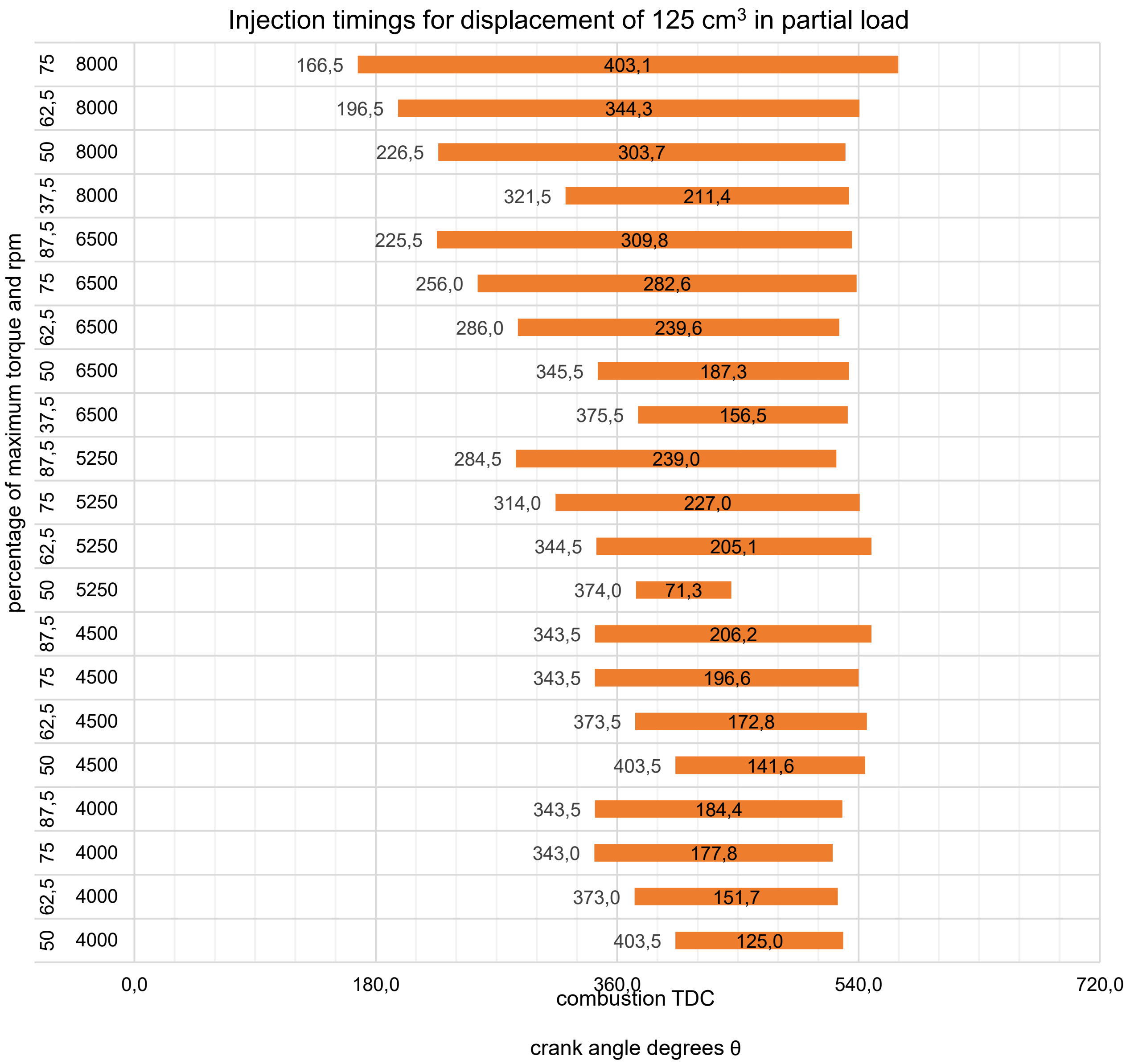

Figure 7.12: injection timings in crank angle degrees in function of the rpm for 125 cubic capacity in partial load.

Even in this case, it can be noted that the injection finishes for all rpm around the BDC of the power stroke phase, and the beginning of the injection variable, as expected.

7.2.2 Spark ignition timings analysis

The spark analysis has been made, as the other steps, with Xviewer and the crank angles, so that the spark began, has been evaluated (in this case each angle is before combustion TDC) and respective timings as well. In the *Table 7.6*, the results can be seen:

%Torque max	%WOT	speed	CA before combust.TDC	CA after intake TDC	timings
-	-	rpm	degrees	degrees	ms
50,0	21,2	4000	28,5	331,5	13,813
62,5	27,9	4000	32,5	327,5	13,646
75,0	43,3	4000	31,5	328,5	13,688
87,5	54,4	4000	35,5	324,5	13,521
50,0	25,2	4500	38,5	321,5	11,907
62,5	33,0	4500	36,5	323,5	11,981
75,0	42,4	4500	32,5	327,5	12,130
87,5	57,4	4500	38,5	321,5	11,907
50,0	29,8	5250	46,5	313,5	9,952
62,5	37,9	5250	42,0	318,0	10,095
75,0	48,1	5250	37,5	322,5	10,238
87,5	61,6	5250	37,5	322,5	10,238
37,5	25,2	6500	52,0	308,0	7,897
50,0	31,4	6500	50,0	310,0	7,949
62,5	37,5	6500	47,0	313,0	8,026
75,0	42,5	6500	43,0	317,0	8,128
87,5	59,2	6500	38,5	321,5	8,244
37,5	28,2	8000	49,5	310,5	6,469
50,0	36,3	8000	47,0	313,0	6,521
62,5	47,0	8000	42,5	317,5	6,615
75,0	59,6	8000	38,5	321,5	6,698

Table 7.6: the ignitions timings in ms and in crank angles in partial load for displacement of 125 cm^3.

7.1.3 Pressure signal analysis

The pressures analysis was done by calculating the maximum pressure inside the combustion chamber for each rpm in different partial load, such as in the subchapter 7.1.3, with the formula:

$$p_{max,cyl} = (p_{int,abs} - p_{int,rel}) + k \cdot p_{peak}$$

where $p_{max,cyl}$, $p_{int,abs}$, $p_{int,rel}$, p_{peak}, k have been already explained in subchapter 7.1.3. The way to calculate these parameters is the same, but of course in partial load condition. In the *Figure 7.14* the pressure peak signal is shown at 4000 rpm and 62,5% of maximum torque, where the yellow line is the TDC signal and the light blue line is the pressure peak signal.

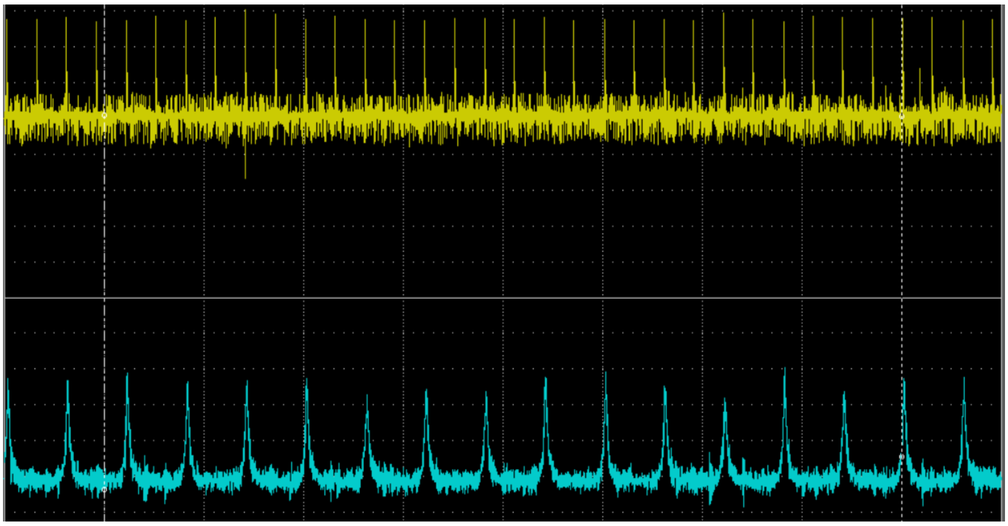

Figure 7.13: pressure signal at 4000 rpm and 62,5% of the load, for 125 cubic capacity.

In the *Table 7.7* the full calculation of the each pressure value can be seen, where the first column is the percentage of load, the second is the percentage of open throttle, the third the rpm, the fourth the maximum values of pressure peak in Volt, the fifth the minimum values of pressure peak in Volt, the sixth the averages of pressure peaks that is p_{peak} in Volt, the seventh column is the intake absolute pressure in the exhaust manifold $p_{int,abs}$ in mbar, the eighth is the intake relative pressure $p_{int,rel}$ in mVolt, the ninth is the maximum pressure inside the combustion chamber in one work cycle $p_{max,cyl}$ in bar; moreover, the maximum pressure inside the cylinder (the last column on the right) is lower than the wide open throttle condition as one should expect because it is a partial load condition and throttle is not all open, thus by having pressure drops.

%Torque max	%WOT	speed	$p_{peak,max}$	$p_{peak,min}$	p_{peak}	$p_{int,abs}$	$p_{int,rel}$	$p_{peak,max}$
-	-	rpm	V	V	V	mbar	mV	V
50,0	21,21	4000	4,552	3,175	3,864	956,3	-100	37,149
62,5	27,86	4000	4,928	3,573	4,294	983,2	-200	45,919
75,0	43,25	4000	5,731	4,615	5,080	996,4	50	51,294
87,5	54,35	4000	7,883	6,133	7,058	1.004,6	400	67,584
50,0	25,21	4500	4,139	3,089	3,562	958,5	-100	37,579
62,5	33,04	4500	4,655	3,419	4,037	989,5	-200	43,358
75,0	42,39	4500	5,573	4,131	4,845	1.000,7	-200	51,452
87,5	57,37	4500	7,475	6,296	6,878	1.008,7	100	68,793
50,0	29,80	5250	4,996	3,390	4,003	958,5	50	40,489
62,5	37,90	5250	5,055	4,039	4,482	988,5	-50	46,309

75,0	48,10	5250	5,943	4,550	5,380	999,7	250	52,300
87,5	61,60	5250	6,276	5,238	5,848	1.009,0	-100	60,487
37,5	25,21	6500	3,634	2,278	2,845	938,2	50	28,890
50,0	31,40	6500	4,258	3,120	3,514	949,1	-100	37,089
62,5	37,50	6500	5,406	3,778	4,546	981,9	-50	46,943
75,0	42,50	6500	6,479	4,231	5,062	997,0	-250	54,115
87,5	59,20	6500	7,113	5,411	6,245	1.003,7	-450	67,955
37,5	28,20	8000	2,757	1,256	2,030	889,1	-200	23,192
50,0	36,30	8000	4,638	3,593	4,153	986,7	-100	42,518
62,5	47,00	8000	4,319	4,070	4,256	1.002,0	-200	43,558
75,0	59,60	8000	4,465	3,673	4,152	1.008,8	50	42,525

Table 7.7: pressure values relatives to the signal analysis in partial load for the displacement of 125 cm³.

Once again, the *Figure 7.15* and the *Figure 7.16* show the correlation between the maximum pressure achieves in the thermodynamic cycle and the torque to the rpm variate, and the correlation between the maximum pressure and the ignition timings, but this time for the partial load.

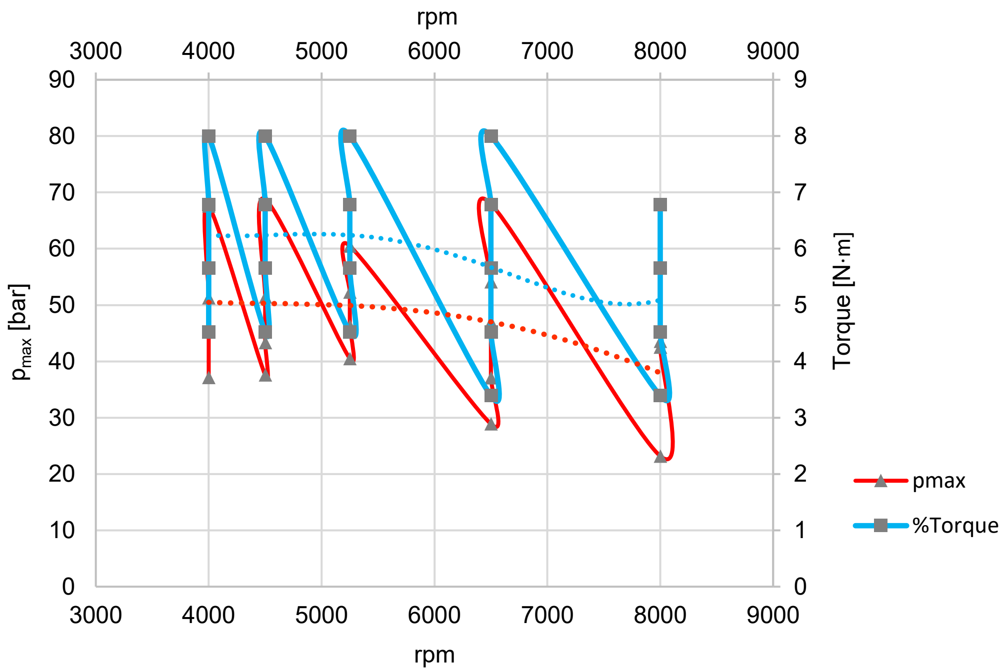

Figure 7.14: comparison between the maximum pressure and the torque in partial load for 125 cubic capacity.

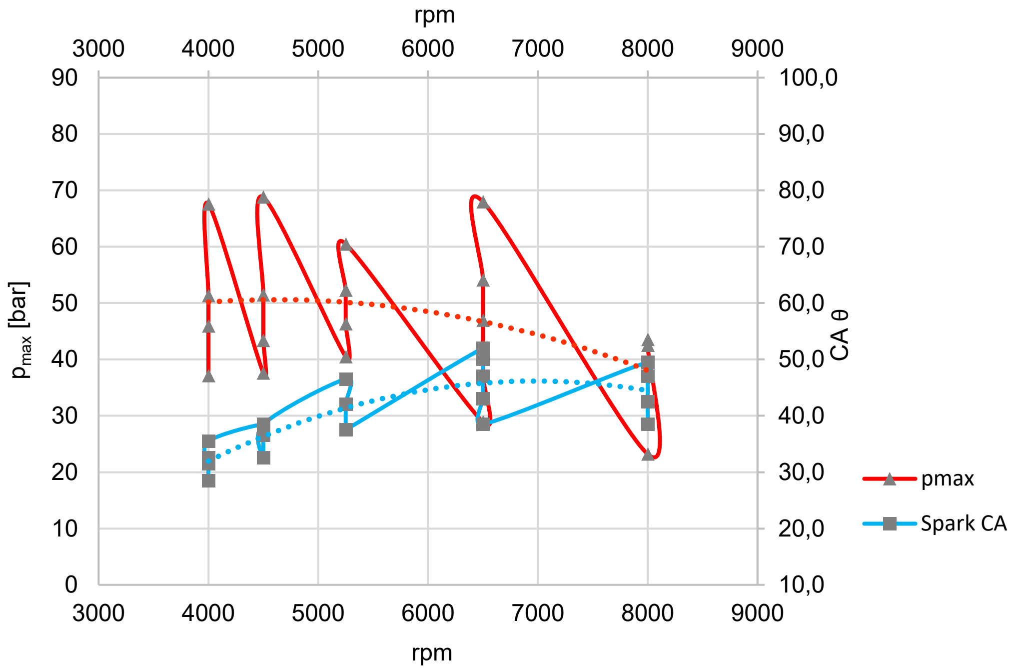

Figure 7.15: comparison between the maximum pressure and the spark ignition in crank angles in partial load for 125 cubic capacity.

Even in this case, as for the full load, the curves obtained have a similar trend. The curves of maximum pressure and torque have not exactly the same trend because of the maximum pressure does not achieve the highest value ever in the same piston position since the spark ignition timings are not ever the same for each engine operating points, and the volume considered carries out a variation calculated from the maximum pressures 13,4 cm^3 and 20,1 cm^3, by having a cubic capacity variation equal approximately to 6,7 cm^3 due to the different maximum pressures. Regarding the spark ignition timings, they have not approximately the same trend of the maximum pressure, except for the high rpm where the maximum pressure in the thermodynamic cycle start to decrease as well as the spark ignition timings.

7.3 Displacement of 170 cm^3 with programmable ECU in WOT

The main characteristic of the 170 cubic capacity programmable ECU is that it has been calibrated for having an higher fuel injection compared to the conventional ECU and the same spark ignition of the 125 cubic capacity. The trials for the displacement have been carried out with an higher range of rpm, namely from 3000 rpm to 9750 rpm (including the engine idle speed of 1533 rpm as well); but the experimental signal analysis of the oscilloscope could made only in wide open throttle condition, because the partial load test was made only for the test carried out with the conventional filter in intake, without making the oscilloscope analysis.

7.3.1 Injection timings analysis

Even for the higher cubic capacity, the injection timings analysis deals the measuring the injection pulse width, but also the injection start timings in crank angle and respective times in ms. Regarding the start of injection, the data have been taken by the YOKOGAWA oscilloscope and with the Xviewer program have been reported on the computer. In the following step will be considered the example at 3000 rpm. In the *Figure 7.17* it is shown the signal of crank angle (the yellow line), TDC (the green line), injection (purple line), more or less in one thermodynamic cycle.

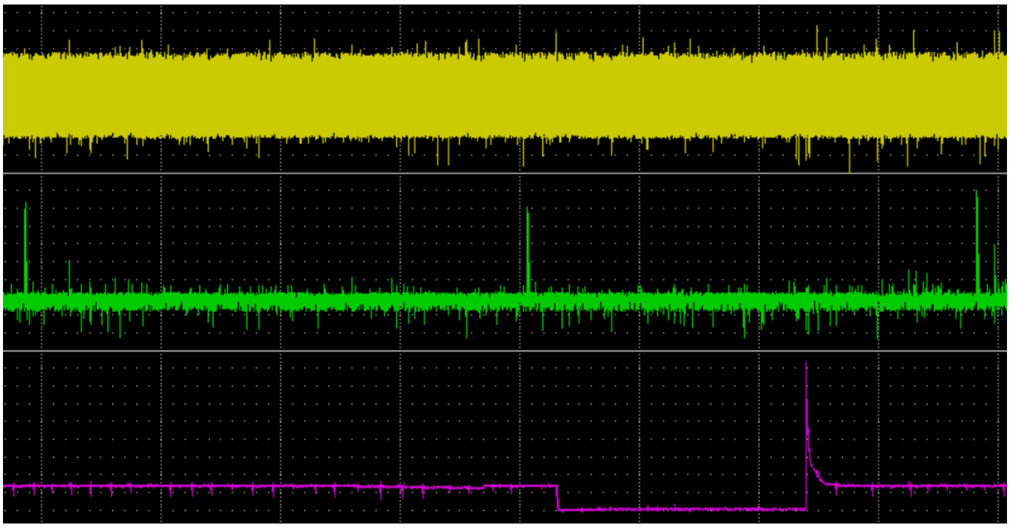

Figure 7.16: the signal of crank angle, TDC and injection at 3000 rpm in WOT for the engine size of 170 cm³.

Making a zoom it is possible analyse the crank angle signal of injection, as shown in the *Figure 7.18*, around the combustion TDC.

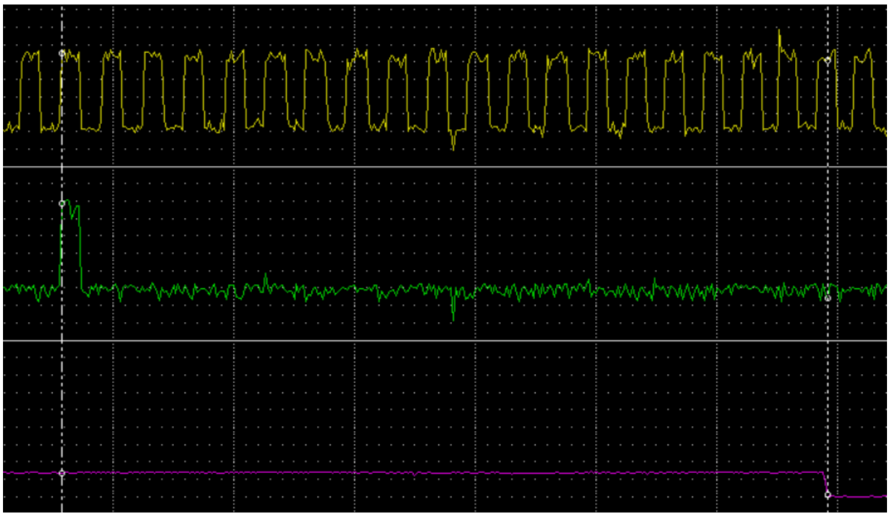

Figure 7.17: signal analysis at 3000 rpm with more zoom for cubic capacity of 170.

Therefore, by counting the cycle of crank angle it is possible to measures it. Thus, at 3000 rpm in wide open throttle the crank angle, with correction, is 19,5° before the combustion TDC, namely 340,5° after the intake TDC. The values of all injection timings in ms and degrees are shown in the *Table 7.8*, where from left to right it is possible to see: the percentage of load, the percentage of open throttle, the rpm, the crank angles around the combustion TDC with correction, the crank angle after the intake TDC, the timings in ms.

speed	CA around combust.TDC	CA intake TDC	timings
rpm	degrees	degrees	ms
1533	175	535	58,165
3000	-19,5	340,5	18,917
3250	-19,5	340,5	17,462
3500	-48,5	311,5	14,833
3750	-48,5	311,5	13,844
4000	-77,5	282,5	11,771
4250	-77	283	11,098
4500	-76,5	283,5	10,500
4750	-76,5	283,5	9,947
5000	-106	254	8,467
5250	-105,5	254,5	8,079
5500	-105,5	254,5	7,712
6000	-134,5	225,5	6,264
6500	-165	195	5,000
7000	-194	166	3,952
7500	-221,5	138,5	3,078
8000	-282	78	1,625
8500	-310	50	0,980

9000	-339,5	20,5	0,380
9250	-309,5	50,5	0,910
9500	-309,5	50,5	0,886
9750	-338	22	0,376

Table 7.8: injection start timings in crank angles and times in WOT for engine size of 170 cm^3.

Then, it is interesting reporting the injection timings in crank angle degrees in function of the rpm, so that it is possible to have clearer how the injection can variate in function of these kind of parameters, namely the rpm and the crank angle degrees, as shown in the *Figure 7.19*.

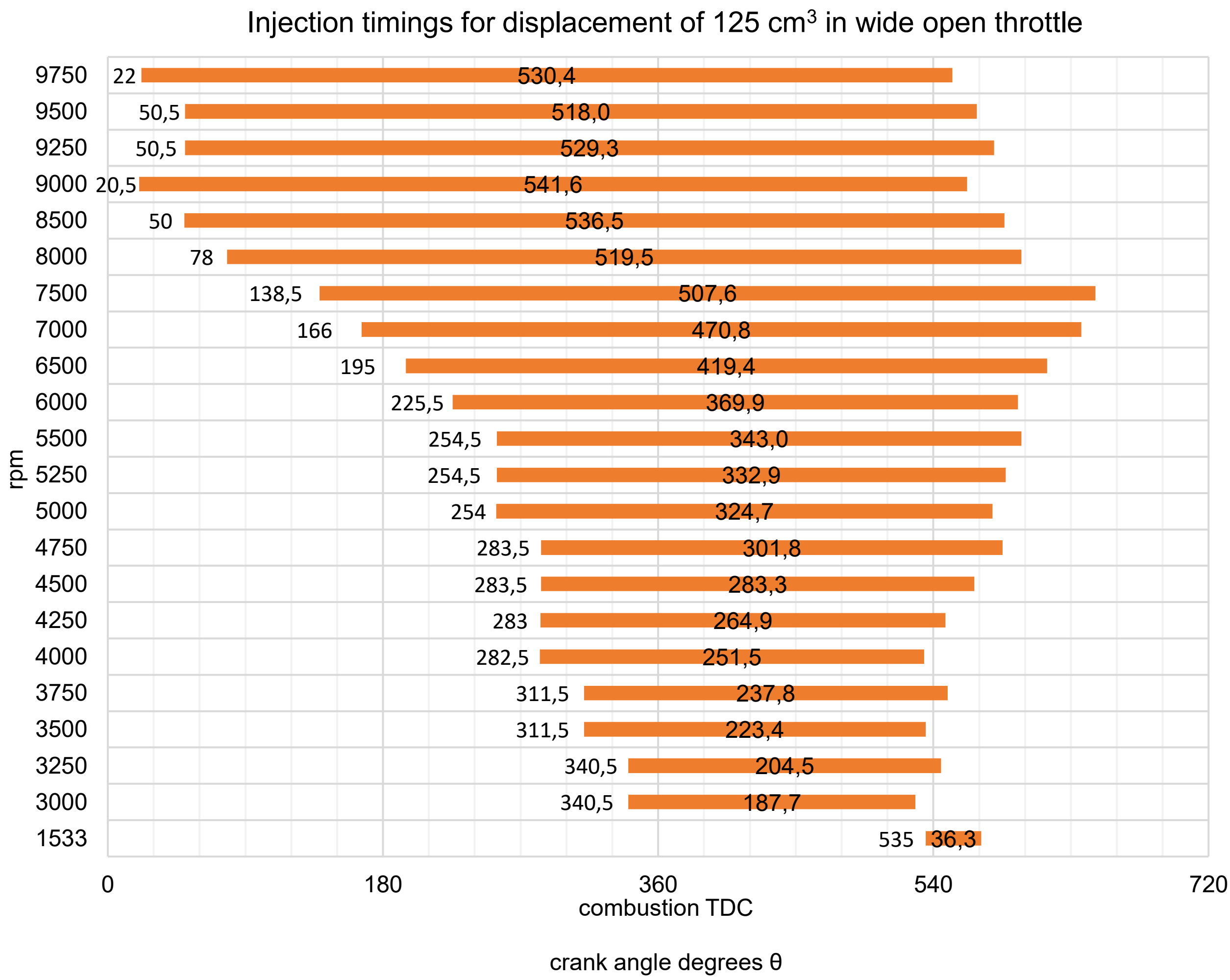

Figure 7.18: injection timings in crank angle degrees in function of the rpm for 170 cubic capacity in WOT.

In this case the injection does not finish ever around the BDC of the power stroke as the displacement of 125 cm^3, since the engine control unit is not the same. For example, at 6500 rpm the injection finishes at 646,1° of crank angles after the intake TDC, or 286,1° of crank angles after the combustion TDC.

7.3.2 Spark ignition timings analysis

The spark analysis has been made with the software Xviewer (as previously) and the crank angles, so that the spark began, has been evaluated and the respective timings as well; the example at 3000 rpm has been taken into consideration visible through the graphs of *Figure 7.19*, approximately in one work cycle, where the yellow line indicates the crank angle signal, the green line is the TDC signal, the purple line the injection and the light blue line is the signal of the spark.

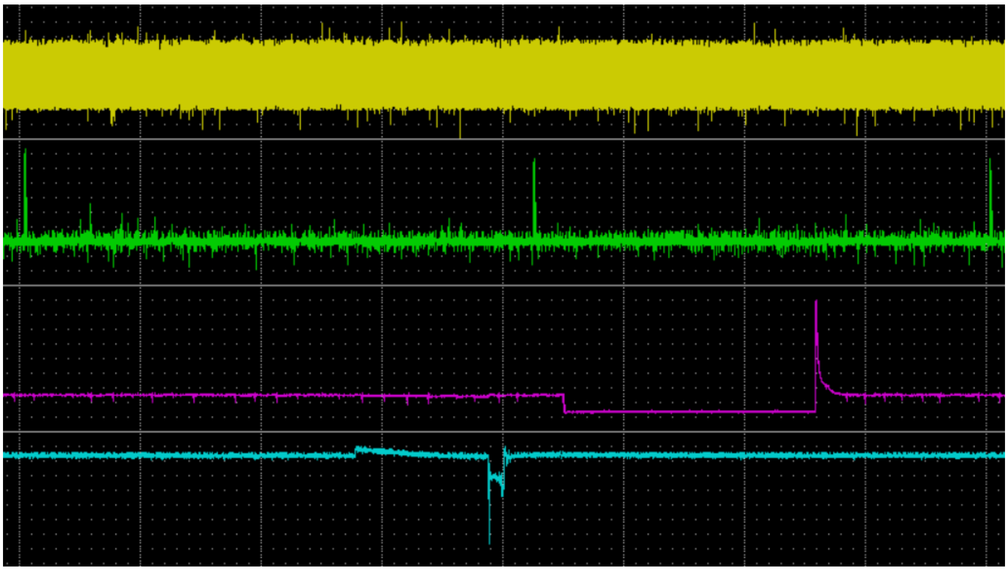

Figure 7.19: the signal of TDC, crank angle and injection at 3500 rpm in WOT for the engine size of 170 cm^3.

The *Figure 7.20* shows the signals with more precision.

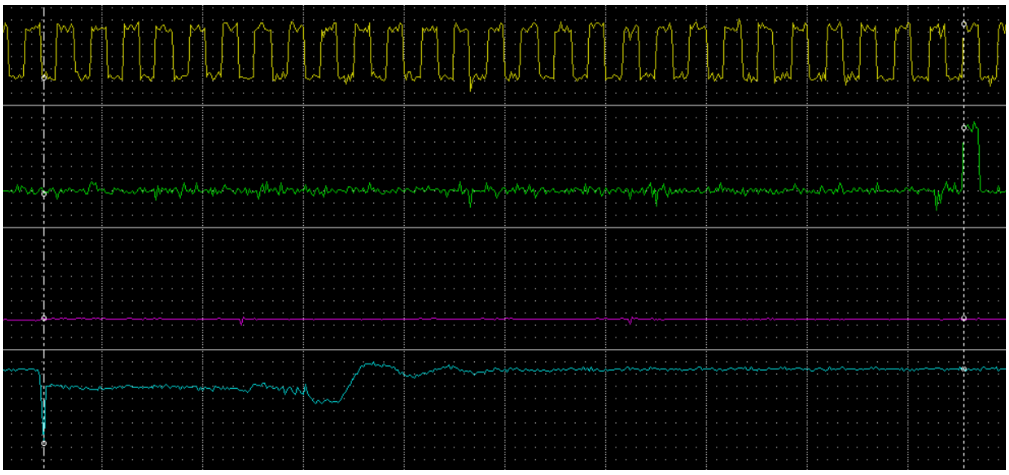

Figure 7.20: spark analysis at 3000 rpm with more zoom.

Thus, the spark ignition achieves when the light blue line (spark signal) carries out a peak, and it is before the combustion TDC. By counting the green cycle from left yellow peak to light blue peak, namely from right to left, the crank angle, with correction of 39°, is 27,5°. On the *Table 7.9*, from left to right: the speed in rpm from 3500 to 7500 rpm, the crank angles calculated before the combustion TDC and with the encoder deviation, the crank angles after the intake TDC, and respective time in ms that crankshaft carries out to those rotations. It can be noted that measured data of crank angle are more or less constants, since the spark ignitions are not subjected to high variations.

speed	CA before combust. TDC	CA intake TDC	ignition timings
rpm	degrees	degrees	ms
1533	5,5	354,5	38,541
3000	66,5	293,5	16,306
3250	67,5	292,5	15,000
3500	69,0	291,0	13,857
3750	70,0	290,0	12,889
4000	71,0	289,0	12,042
4250	74,0	286,0	11,216
4500	75,0	285,0	10,556
4750	74,0	286,0	10,035
5000	74,0	286,0	9,533
5250	75,0	285,0	9,048
5500	74,0	286,0	8,667
6000	73,0	287,0	7,972
6500	71,5	288,5	7,397
7000	70,0	290,0	6,905
7500	70,5	289,5	6,433
8000	70,5	289,5	6,031
8500	71,0	289,0	5,667
9000	70,5	289,5	5,361
9250	71,0	289,0	5,207
9500	72,0	288,0	5,053
9750	72,5	287,5	4,915

Table 7.9: the ignitions timings in ms and in crank angles in full load for displacement of 170 cm^3.

7.3.3 Pressure signal analysis

As made previously, the formula to compute the maximum pressure in the cylinder is:

$$p_{max,cyl} = \left(p_{int,abs} - p_{int,rel}\right) + k \cdot p_{peak}$$

where $p_{max,cyl}$, $p_{int,abs}$, $p_{int,rel}$, p_{peak}, k have been already explained. The way to calculate these parameters is the same, but of course in wide open throttle condition for the cubic capacity of 170. In the *Figure 7.21* it is shown the pressure peak signal at 3000 rpm in full load, where the yellow line is the crank angle signal, the green line is the TDC signal and the light blue line is the pressure peak signal.

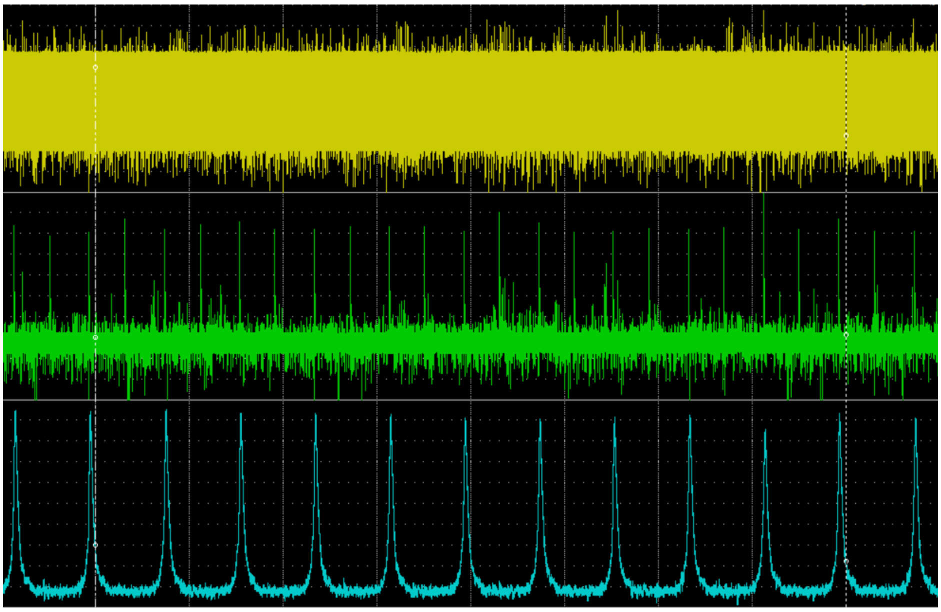

Figure 7.21: pressure peaks signals at 3000 rpm regarding the engine size of 170 cm^3.

In the *Table 7.10* are shown all values of the pressure signal analysis, thus from left to right: the engine speed in rpm, the maximum values of pressure peak in Volt, the minimum values of pressure peak in Volt, the averages of pressure peaks that is p_{peak} in Volt, the intake absolute pressure in the intake manifold $p_{int,abs}$ in mbar, the intake relative pressure $p_{int,rel}$ in mVolt, the maximum mid-pressure which achieves during the combustion inside the cylinder in one thermodynamic cycle namely $p_{max,cyl}$ in bar.

speed	$p_{peak,max}$	$p_{peak,min}$	p_{peak}	$p_{int,abs}$	$p_{int,rel}$	$p_{max,cyl}$
rpm	V	V	V	mbar	mV	bar
1533	0,840	0,808	0,827	1023,22	-50	9,790
3000	8,793	7,918	8,524	1022,00	900	77,265
3250	7,959	7,075	7,488	1022,00	120	74,702
3500	8,356	7,436	7,896	1022,00	30	79,679
3750	8,196	7,221	7,760	1022,00	-350	82,125
4000	8,079	7,094	7,730	1020,93	-50	78,825
4250	8,278	6,716	7,569	1019,87	-200	78,710
4500	8,028	6,883	7,582	1018,80	-100	77,842
4750	7,839	6,719	7,195	1017,60	150	71,466

5000	7,839	6,073	7,120	1016,40	-250	74,720
5250	7,749	6,583	7,305	1015,20	200	72,067
5500	7,030	5,736	6,422	1014,54	-700	72,233
6000	5,780	4,953	5,432	1013,20	-1100	66,336
6500	6,831	6,074	6,449	1011,16	300	62,504
7000	6,555	5,756	6,281	1008,05	-130	65,114
7500	6,789	5,565	6,239	1006,20	250	60,899
8000	5,056	4,156	4,630	1005,05	-1230	59,609
8500	5,829	4,536	5,325	1005,57	-400	58,252
9000	5,884	4,991	5,569	1003,80	180	54,891
9250	5,595	4,631	5,022	1003,28	180	49,422
9500	4,238	3,916	4,111	1002,75	-700	49,109
9750	5,273	3,746	4,533	1001,50	-50	46,827

Table 7.10: pressure values relatives to the signal analysis in full load for the displacement of 170 cm^3.

the *Figure 7.22* and the *Figure 7.23* show the correlation between the maximum pressure achieves in the thermodynamic cycle and the torque to the rpm variate, and the correlation between the maximum pressure and the ignition timings, but this time for the partial load.

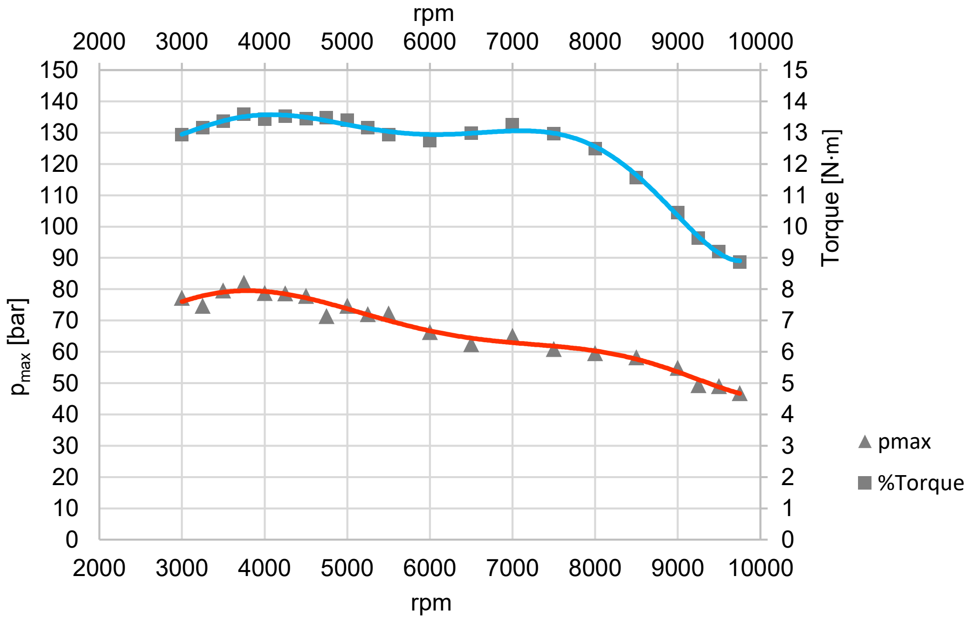

Figure 7.22: comparison between the maximum pressure and the torque in WOT for 170 cubic capacity.

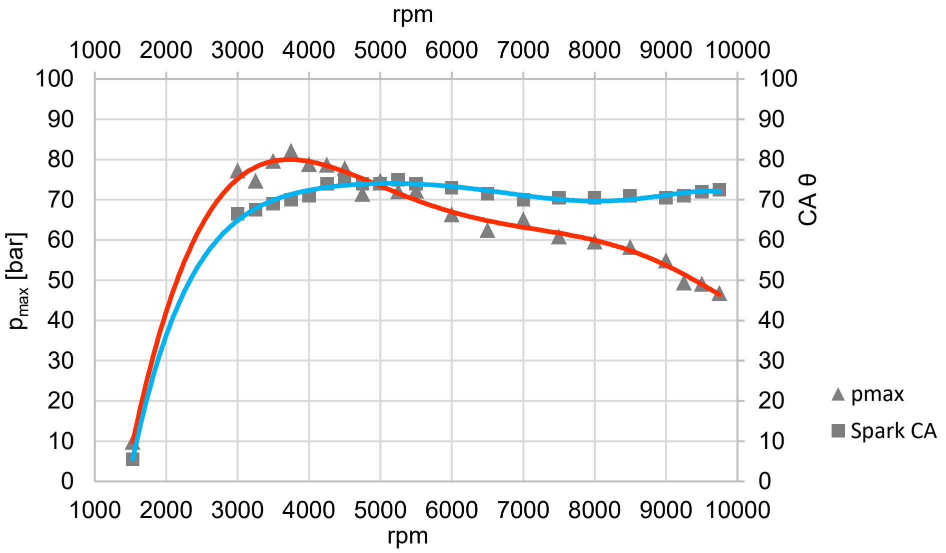

Figure 7.23: comparison between the maximum pressure and the spark ignition in crank angles in WOT for 125 cubic capacity.

Again, the obtained curves have a similar trends. The curves of maximum pressure and torque have not exactly the same trend because the maximum pressure does not achieve the highest value ever in the same piston position since the spark ignition timings are not ever the same for each engine operating points. In fact for the spark ignition advances there will be a lower pressure, and for spark ignition delays there will be an higher pressure, which are clearly visible in the *Figure 7.23* above. Therefore, the considered cubic capacity carries out a variation calculated from the maximum pressures of 20,8 cm^3 and 26,8 cm^3, by having a cubic capacity variation equal approximately to 8 cm^3 due to the different maximum pressures. Due to these reasons, the trend of the maximum pressure and of the torque are similar but not equals.

Chapter 8: Numerical model for the injection control

For the definition of the numerical model, the procedure followed by the ECU for calculating the parameters for its proper operating needs to be known. Once that this calculation process has been defined, it is possible to establish with the aid of Simulink a model to compare the results with the engine control unit results.

The main task for the calculation method of the ECU is based on the obtaining of the injector pulse width (that will be called PW), which is going to depend by the measurements of the different sensors connected to the ECU that, by receiving these information, will carry out more computations for obtaining the injection pulse width desired which will be sent to the injector.

8.1 Calculation model of fuel injection

The method followed is that of the "Fuel Injection Equations" proposed by Al Grippo, which establishes the following correlation for computing the injection pulse width:

$$\mathrm{PW} = 0{,}67 \cdot (InjTurnON + BattComp) + AccPw + \frac{ReqdFuel}{InjFlowRate}$$

Where:

- PW is the fuel injector pulse width in microseconds;
- InjTurnON is the time in microseconds for injector to open fully when a 12 Volts is applied, and a typical value is 1,5 ms = 1500 µs;
- BattComp is the compensation (+ or – microseconds) of turn on time for fact that battery voltage may not be exactly 12 volts;
- AccPw is the temporary extension of pulse width (in microseconds) during acceleration, defined by user, and may be left 0.0, as there is also an acceleration enrichment term in ReqdFuel, which is a function of throttle rate;
- InjFlowRate (IFR) is the injector flow rate at operating pressure in g/(µs·);
- ReqdFuel is the total fuel required for a particular cylinder firing in g.

The factor of 0,67 accounts for the fact that the average fuel flow rate is only 1/3 the max ate during turn on time. The 1/3 factor comes from $F = m \cdot a$ applied to a solenoid, and assuming flow is proportional to the opening of solenoid:

$$TurnON + \frac{ReqdFuel\ - \frac{1}{3}IFR \cdot TurnON}{IFR} = 0{,}67TurnON + \frac{ReqdFuel}{IFR}$$

The parameters InjTurnON, BattComp and InjFlowRate depend by the different type of injector, thus these values are considered constants.

Obviously, the total fuel required depends by the needed air mass for each thermodynamic cycle and by the lambda air-excess:

$$ReqdFuel = MassAirFa \cdot \lambda \cdot \left(1 + \frac{PmFuel}{100}\right)$$

Where the MassAirFa is the air mass for the desired air-fuel ratio in lb/cycle. The lambda value will be implemented in the model for each engine operating point. Regarding the PmFuel, it is a controls parameter which by calculation purpose takes on a value of 1, thus not having visible effect in the final results.

The needed air mass for each work cycle is computed by the ECU which needs of the information measured through the sensors, namely the parameters inside the following equation:

$$MassAirFa = AirDen \cdot \left[\frac{CID}{1728}\right] \cdot \left[\frac{1}{14{,}7 \cdot NCYL}\right] \cdot \left[\frac{MAP}{MAP_{WOT}}\right] \cdot \left[\frac{VE}{\frac{AirFuel}{14{,}7}}\right]$$

Where the parameters measured through the sensors are clearly visible. The $\frac{CID}{1728}$ term is the cubic inch displacement of the engine, converted into cubic feet, and it does not need to be measured. The $\frac{1}{14{,}7 \cdot NCYL}$ term indicates the cylinders number of the engine normalised in relation to the stoichiometric Air/fuel ratio. The $\frac{MAP}{MAP_{WOT}}$ term is referred to the measurement of the intake manifold absolute pressure (MAP) in the considered operating point of the engine, compared to the intake manifold absolute pressure in wide open throttle condition (MAP_{WOT}) which is generally around 100 kPa. The VE term is the volumetric efficiency for the considered operating point of the engine, which mainly depends on the engine rpm and on the pressure in the intake manifold. For the computation of the AirDen term, namely the air density at the considered environmental conditions, the correlation is:

$$AirDen = \frac{0{,}0391568 \cdot (\mathrm{BaroPress} - 31)}{\left(\frac{MatTemp}{10}\right) + 459{,}7}$$

Where BaroPress is the barometric pressure in KPa*10, 31 = 3,1 Kpa*10 is the correction for the vapor pressure by assuming an humidity of 75 percent at 85 F temperature. The MatTemp term is the air temperature in degrees F*10 in the intake manifold.

What has been discussed so far is an example for computing the injection pulse width of a conventional ECU, which lends well for the model development.

8.2 Introduction to the simulation model

For the development of the numerical model for the fuel injection control, the MathWorks model was used, namely the "Modeling a Fault-Tolerant Fuel Control System" such as beginning point, which works with the software Simulink [3]. The *Figure 8.1* shows the beginning model.

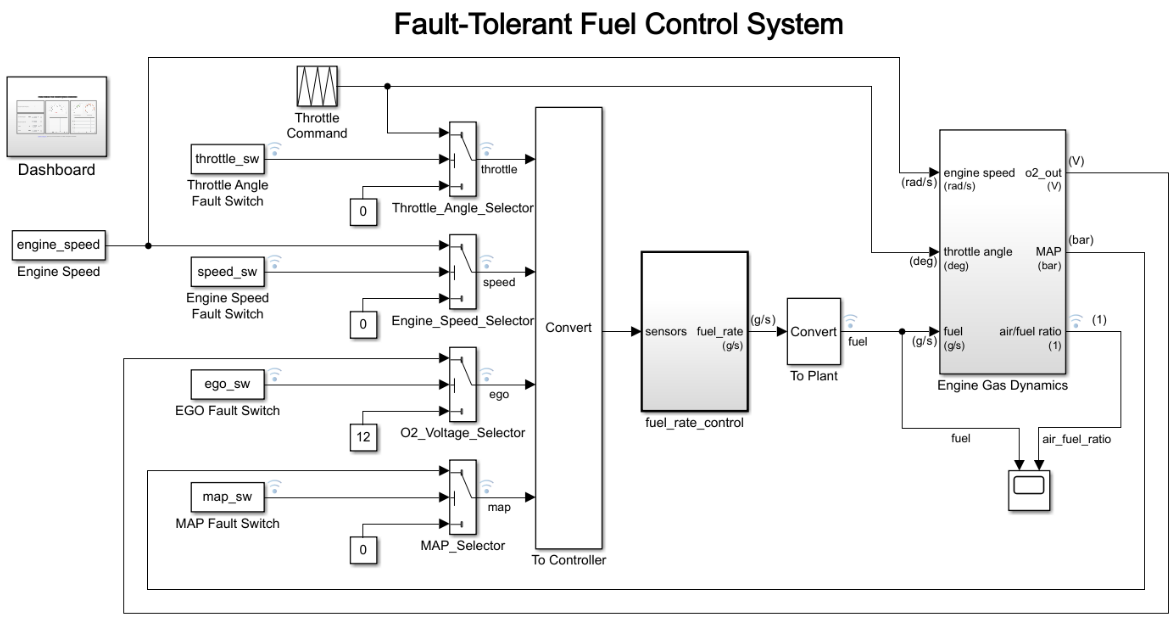

Figure 8.1: the Fault-Tolerant Fuel Control System as beginning model.

The main key of this model is the estimation of the air mass flow rate in the intake manifold (in the fuel rate control subsystem), and thus on the fuel flow rate calculation through the air-fuel ratio stoichiometric and with the lambda air-excess. The intake airflow can be computed for different engine speed and different throttle opens.

8.2.1 Fuel rate control subsystem

This block computes the fuel mass flow rate (in g/s) as close as possible to the stoichiometric mixture by using the signal sensors. The *Figure 8.2* shows system where there are other three blocks: the control logic subsystem, the airflow calculation subsystem and the fuel calculation subsystem.

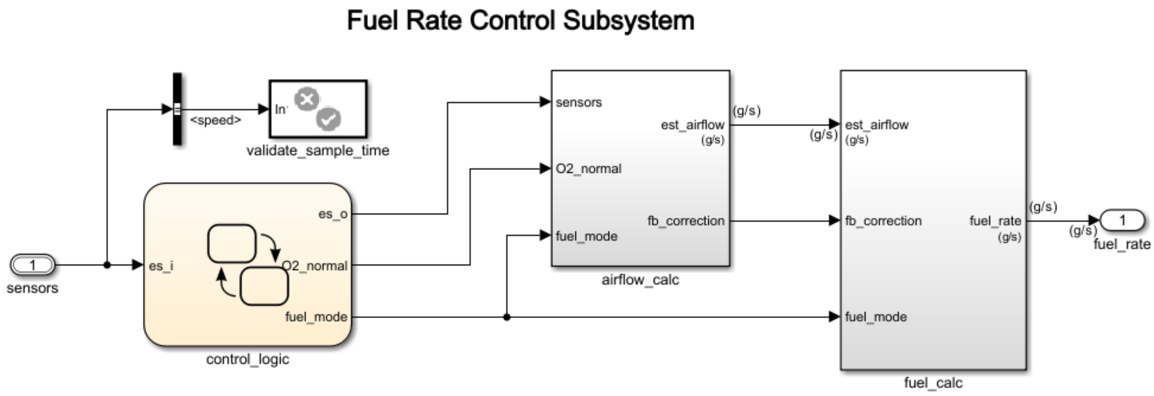

Figure 8.2: the fuel rate control subsystem.

8.2.2 Control logic subsystem

The subsystem is shown in the *Figure 8.3*.

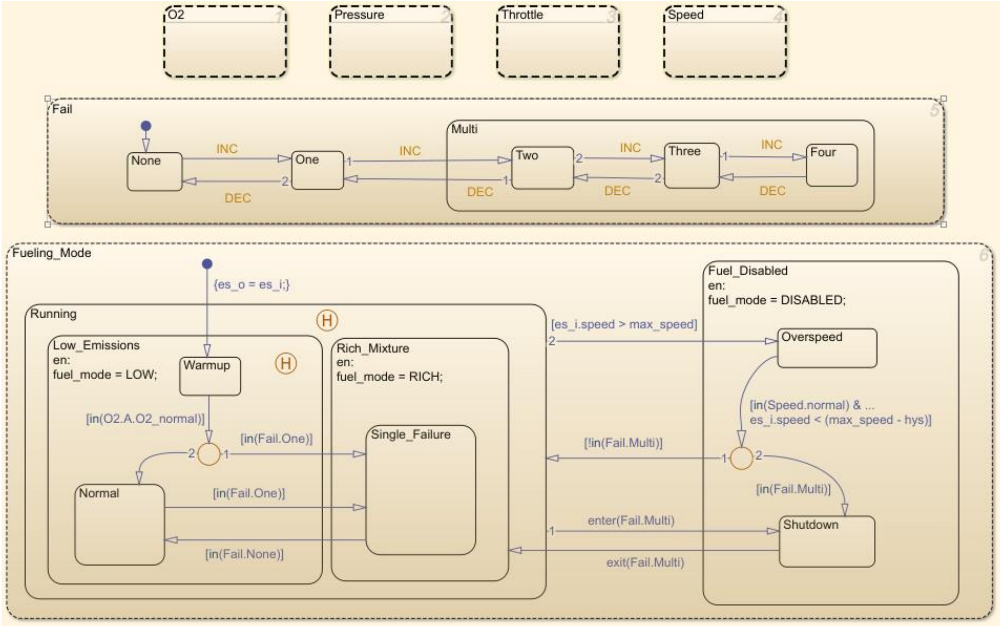

Figure 8.3: the control logic subsystem.

The engine operating can be changed during a fault and with this subsystem the engine safety can be made higher during its operating for example after a malfunctioning. The lower parallel state represents the fueling mode of the engine. If a single sensor fails, the operation continues but the air/fuel mixture is richer to allow smoother running at the cost of higher emissions. If more than one sensor has failed, the engine shuts down as a safety measure, since the air/fuel ratio cannot be controlled reliably. The four parallel blocks shown at the top of *Figure 8.3* correspond to the four individual sensors; every time that measurements exceed the established maximum values, they begin automatically to give an estimation value which are inside of the auxiliary matrices that are contained in the four parallel blocks at the top. Therefore, with these sensors it is possible making an error every time the measurements exceed ma maximum values, but in the developed model these error estimations have not been expected and the matrices have all values equal to zero.

8.2.3 Intake airflow estimation

The whole subsystem is shown in the *Figure 8.4*.

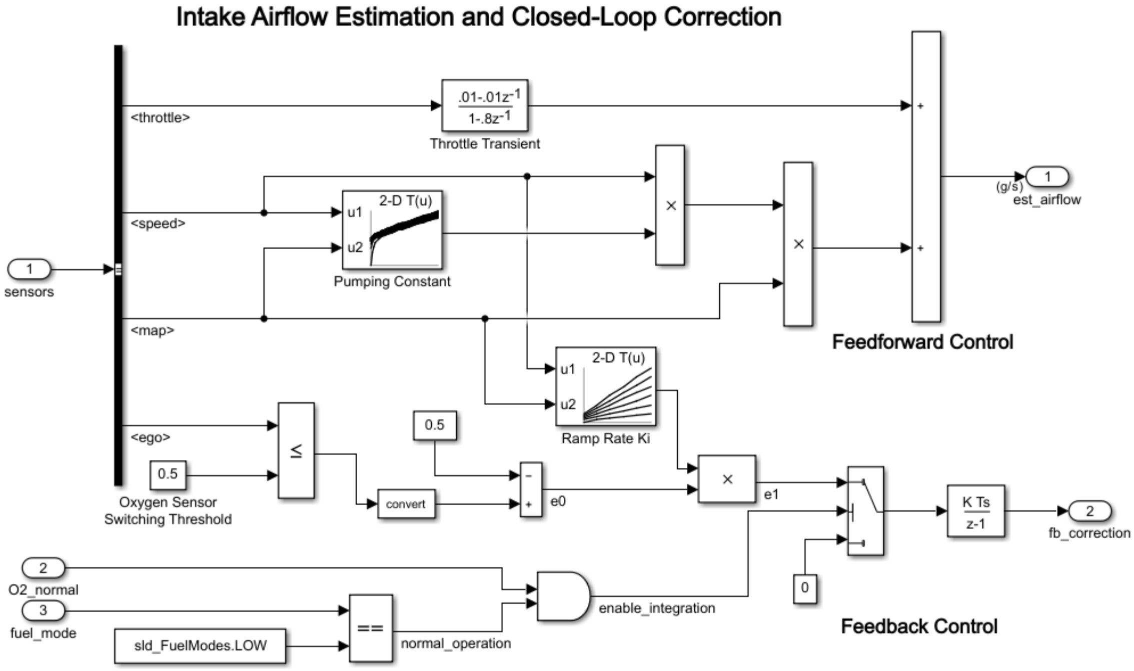

Figure 8.4: the intake airflow estimation subsystem.

The block estimates the intake air flow to determine the fuel rate which gives the appropriate air/fuel ratio. The air mass flow rate is given by the formula in g/s:

$$\dot{m}_a = \frac{n}{4\pi} \cdot V_d \cdot \eta \cdot \frac{MAP}{R \cdot T} = \frac{n}{4\pi} \cdot MAP \cdot C_{pump}$$

With $C_{pump} = \frac{V_d \cdot \eta}{R \cdot T}$ (pumping constant), where V_d is the engine cubic capacity in m^3, η is the volumetric efficiency, R is the specific gas constant in J/(kg·K), T is the intake temperature in K, MAP is the intake manifold absolute pressure in bar and n is the engine speed in rad/s. The C_{pump} is implemented such as matrix by using a lookup table. The air mass flow rate equation must be reported in the international system of units, or better for reporting both the terms (at left and right to the equal) in the international system of units and in particular in g/s, it can be obtained:

$$\dot{m}_a\left(\frac{g}{s}\right) = \frac{n}{4\pi}\left(\frac{rad}{s}\right) \cdot MAP(bar) \cdot C_{pump}\left(\frac{m^3}{\frac{J}{kg \cdot K}K}\right)$$

$$\dot{m}_a\left(\frac{g}{s}\right) = \frac{n}{4\pi}\left(\frac{rad}{s}\right) \cdot MAP \cdot 10^5\left(\frac{kg\frac{m}{s^2}}{m^2}\right) \cdot C_{pump}\left(\frac{m^3}{\frac{kg\frac{m^2}{s^2}}{kg \cdot K}K}\right)$$

$$\dot{m}_a\left(\frac{g}{s}\right) = \frac{n}{4\pi}\left(\frac{rad}{s}\right) \cdot MAP \cdot 10^8\left(\frac{g}{m \cdot s^2}\right) \cdot C_{pump}(m \cdot s^2)$$

Now, both terms are express in the same units. Thus, rewriting the equation and by including the numerical coefficients inside C_{pump}:

$\dot{m}_a = n \cdot MAP \cdot C_{pump}$, with $C_{pump} = \frac{10^8}{4\pi}\frac{V_d \cdot \eta}{R \cdot T}$.

Instead, the closed-loop control, it adjusts the estimation according to the residual oxygen feedback in order to maintain the mixture ratio precisely. Even when a sensor failure mandates open-loop operation, the most recent closed-loop adjustment is retained to best meet the control objectives. But, anyway this closed-loop control was not used in the model because the estimation of residual oxygen in the exhaust gases is not an easy computation, since it could be present only if it is contained in an eventual additive (contained in the fuel) and that could still react with the fuel; because of the air excess lower than 1 in the considered engine, for both displacements, namely variable but around 0,9. In fact the oxygen sensor (EGO) is not used for the purposes of calculating the injection pulse width.

Regarding the part on the top of the *Figure 8.4*, it is the transient part of the throttle openings by computing a correction due to the acceleration or deceleration of the throttle itself, namely making the fuel injection more fluid and not improvise. Even this part is not considered since the transient points have not been tested.

8.2.4 Fuel rate calculation

The subsystem of fuel rate calculation is shown in the *Figure 8.5*.

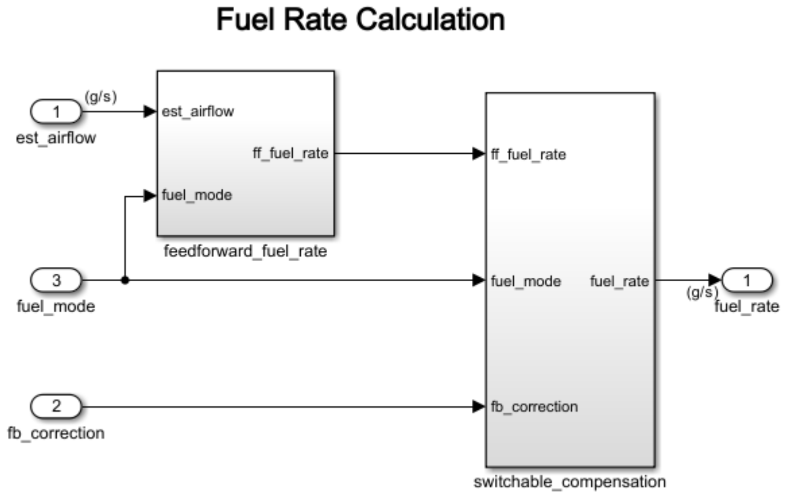

Figure 8.5: the subsystem of fuel rate calculation.

The estimation of the fuel consumption is computed in the "feedforwards fuel rate" block in the *Figure 8.6*.

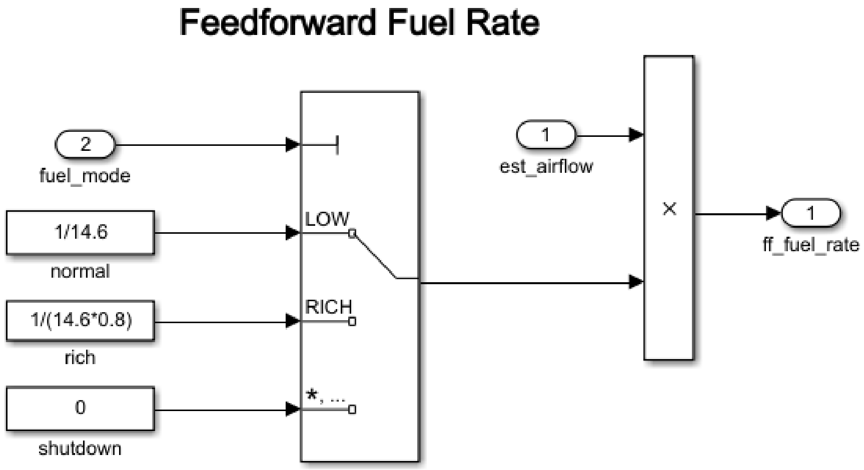

Figure 8.6: the feedforward fuel rate block.

The first input is the estimation of the computed airflow, which is multiplied for the air fuel ratio for obtaining the feedforward fuel rate (ff_fuel_rate) in base of the engine operating point, that is defined by the control logic. The air excess corresponds to the stoichiometric, i.e. equals the optimal air to fuel ratio of 14,6; The lambda values will be implemented with different values for the different engine operating points.

The computed feedforward fuel rate (ff_fuel_rate) can have a correction factor calculated from the estimation of the oxygen excess, which guarantees a better adjustment in the engine operating during the transient, namely a compensation, as shown on the *Figure 8.7*.

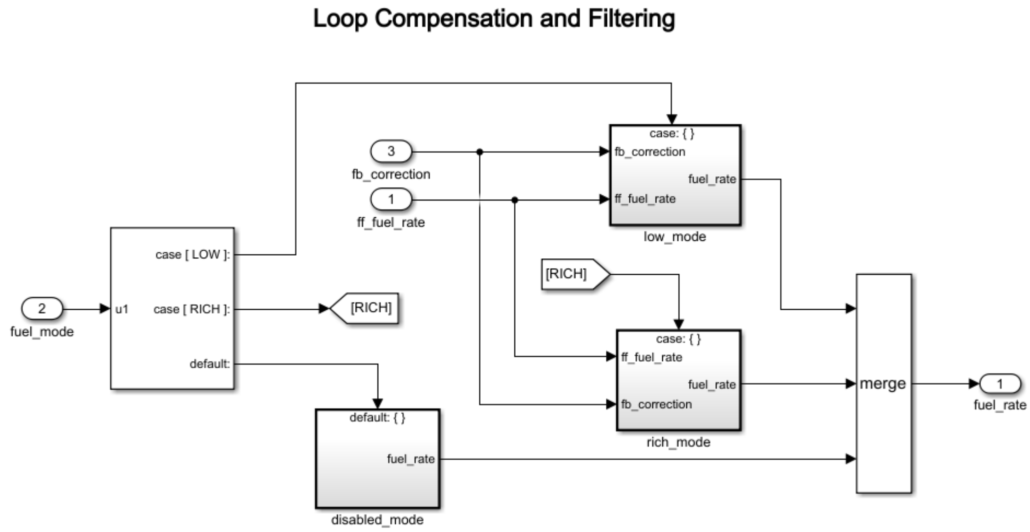

Figure 8.7: fuel rate calculation with the factor correction.

But, the transient points were not considered, and the correction factor will be equal to zero. Hence, at the value of feedforward fuel rate (ff_fuel_rate) will be summed for both cases, according to the different operating, the factor equal to zero (the block of low mode and rich mode), and the value called fuel_rate will be the final value, i.e. the fuel consumption.

8.2.5 Engine gas dynamic system

The block of the engine gas dynamic is shown in the *Figure 8.8*.

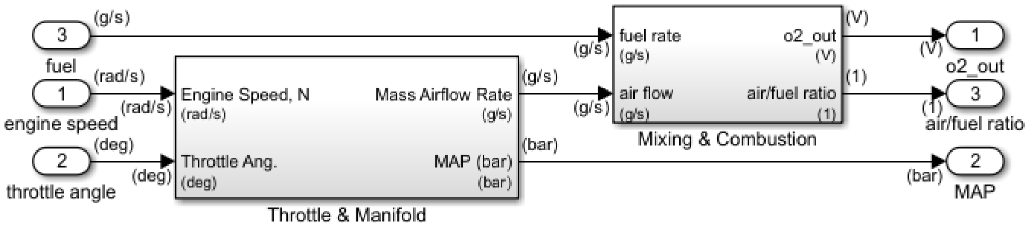

Figure 8.8: the engine dynamic system.

This block will not used because is based on the oxygen sensor operating (EGO) for studying the dynamic of the mixing and combustion.

8.3 Model development for the control of fuel injection

The model for fuel injection control will be developed after making the measurements and calculations for both displacements by means of the test bench. For the model definition, it is important taking into consideration which are the input and output parameters; in particular the input parameters will be the engine speed and the throttle opening percentage, and the output will be the injection pulse width computed by the model which is going to be compared with oscilloscope measure signal.

For obtaining this information, the intake manifold absolute pressure (MAP) must be implemented in the model through a matrix, namely by creating a lookup table in function of the throttle opening (on the columns) and engine speed (on the rows). The developed model can be seen in the *Figure 8.9*; in the example the simulation has been ran at 5000 rpm and wide open throttle conditions for the cubic capacity of 125. The left part of the model, namely the management and treatment of the input data through the various sensors, is remained unchanged except the MAP implementation, and the sensor for the oxygen residuals in the exhaust gases, as anticipated, has not been considered. The fuel rate control subsystem has been modified and implemented with the operating engine parameters and two subsystems on the right part have been implemented for the injection pulse width computation, i.e. fuel rate per cycle and injection calculation. Moreover, the most important parameters of the considered engine (as displacement, speed, MAP, throttle opening, ect) can be visualised in the control display with the yellow color.

The all MAP values implemented as mapping are visible in the *Figure 8.10* for 125 cm^3 displacement, in function of different throttle opening obtained during the test for different load percentages (rows) and in function of the engine speed in rad/s (columns).

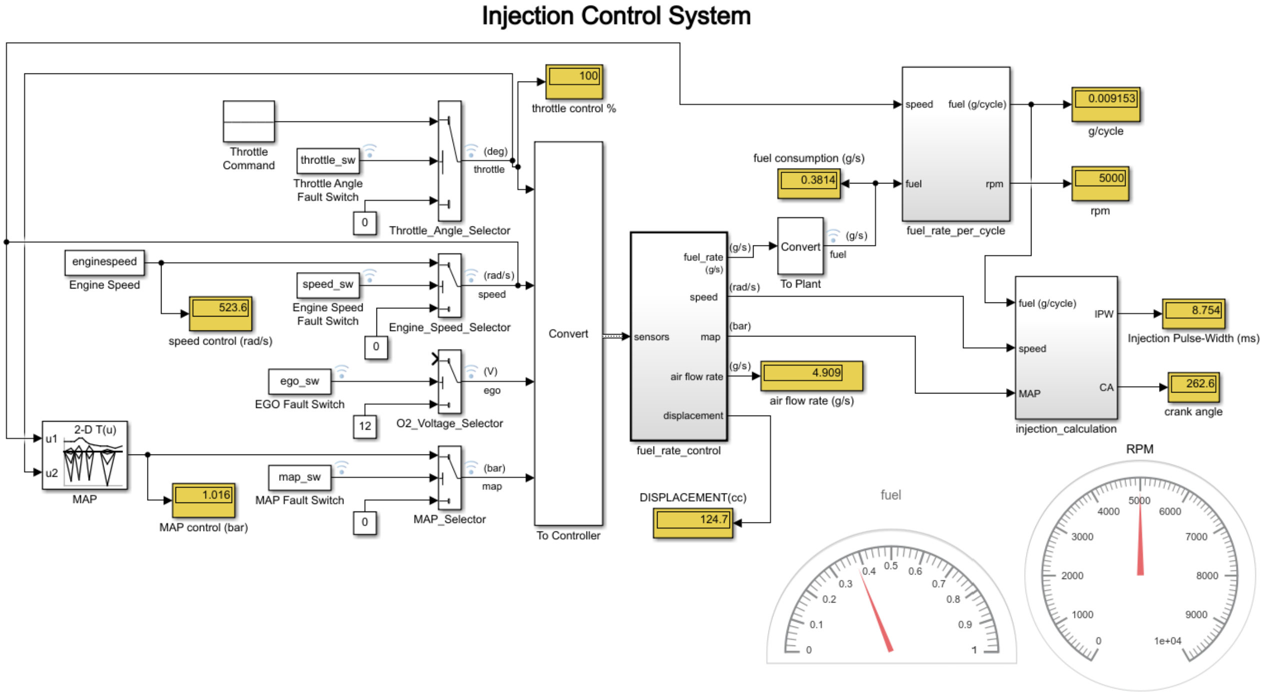

Figure 8.9: the injection control system model developed. Example at 5000 rpm and WOT, for 125 cc.

Row		21.2	25.2	27.9	29.8	31.4	33	37.5	37.9	42.4	42.5	43.3	48.1	54.4	57.4	59.6	61.6	100
(1)	***366.519***	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1.015
(2)	***418.879***	0.9563	1	0.9832	1	1	1	1	1	1	1	0.9964	1	1.0046	1	1	1	1.015
(3)	***445.059***	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1.016
(4)	***471.239***	1	0.9585	1	1	1	0.9895	1	1	1.0007	1	1	1	1	1.0087	1	1	1.02
(5)	***497.419***	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1.02
(6)	***523.6***	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1.016
(7)	***549.779***	1	1	1	0.9585	1	1	1	0.9895	1	1	1	0.9997	1	1	1	1.009	1.013
(8)	***575.959***	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1.01
(9)	***628.319***	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1.009
(10)	***680.678***	1	1	1	1	0.9491	1	0.9819	1	1	0.997	1	1	1	1	1.0037	1	1.008
(11)	***733.038***	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1.005
(12)	***785.398***	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1.008

Figure 8.10: MAP table in function of throttle opening and engine speed for the displacement of 125 cm^3.

The different throttle openings can be changed using the block called throttle command in the left top part of the model main page. It has been set in a vector time from zero 20 seconds (thus 3 values of time, namely 0, 10, 20) and the percentages of throttle openings can be changed in a vector with three inputs, which they do not need be constants, but they can variate as well, for looking how the fuel consumption and so the injection timings can change during a time of 20 seconds.

It is important noting that there is not a pressure value for each slot, because the throttle openings are not the same for different engine speeds and different load percentages, and where the engine operating point was not measured there is ideally the pressure value equal to 1 (even if these points were not measured). The other MAP table, for the cubic capacity of 170, is represented in the APPENDIX 2.

8.3.1 Fuel rate control subsystem

The modified subsystem of the fuel rate control can be seen in the *Figure 8.11*.

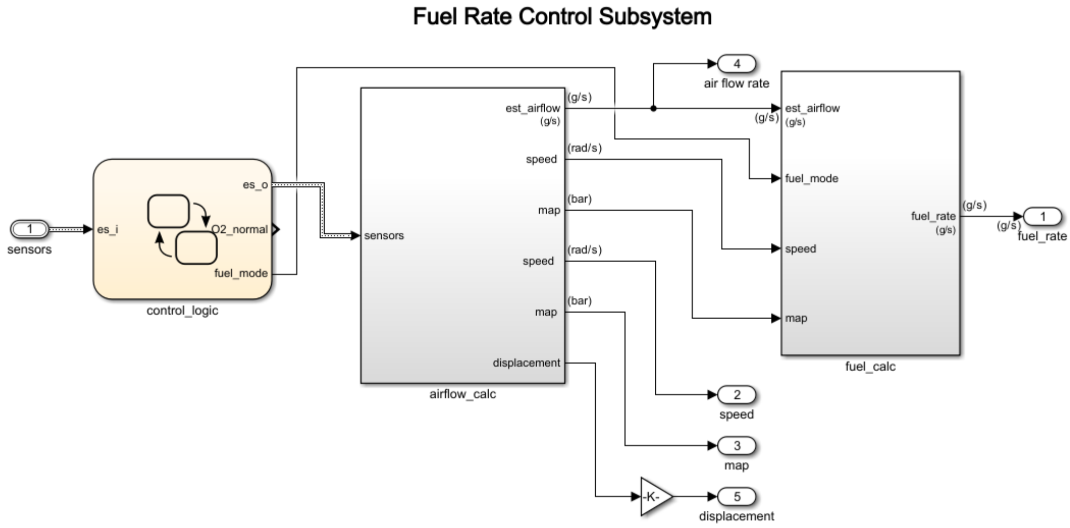

Figure 8.11: the fuel rate control implemented.

As previously, there are the blocks of control logic, airflow calculation and fuel calculation. The first has not been subjected to modifications, except the oxygen residuals lack and some modification on the programming language, because the model workspace was written in two MATLAB scripts (for both cubic capacity). In the second and third block, more connecting output and input have been added, and in particular, the displacement has been multiplied per 10^6 for obtaining the displacement self in centimetre cubic.

8.3.2 Intake airflow estimation

The intake airflow subsystem has been modified, as shown in the *Figure 8.12* (ever at 5000 rpm and wide open throttle conditions for the cubic capacity of 125).

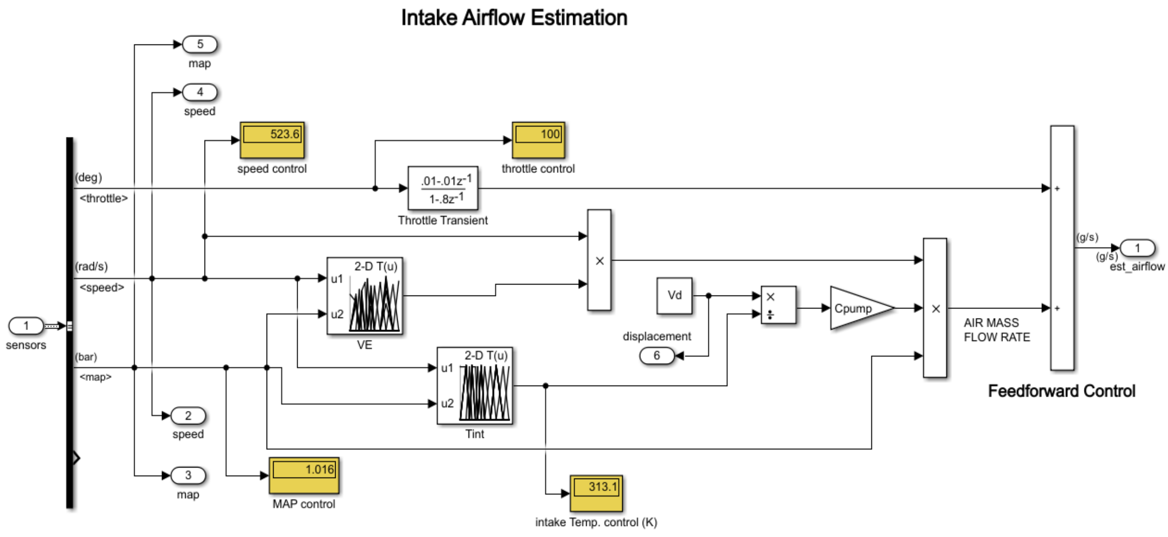

Figure 8.12: the intake airflow subsystem implemented. Example at 5000 rpm and WOT, for 125 cc.

For the model development, in order to obtain results as close as to reality, each term in the air mass flow rate equation has been extrapolated so that all parameters inside the equation can variate for the different engine operating conditions, and in the pumping constant the specific gas constant is going to remain, namely:

$$\dot{m}_a = \frac{n}{4\pi} \cdot V_d \cdot \eta \cdot \frac{MAP}{T} \cdot C_{pump}$$

Where $C_{pump} = \frac{10^8}{4\pi \cdot R}$, as already computed in the 8.2.3 paragraph, is expressed in the international system of units; thus, the pumping constant is going to be equal for each mapping point. The V_d displacement term is referred to the different cubic capacities; the volumetric efficiency (VE) and the temperature in the intake manifold (T_{INT}) are implemented with similar mapping of the MAP, but in function of the MAP and engine speed. The *Figure 8.13* and the *Figure 8.14* show the volumetric efficiency and the intake temperature respectively in function of the MAP (on the rows) and of the speed (columns). The volumetric efficiency has been computed through the air mass flow rate computed starting by air mass flow rate calculated from the lambda values (which has been computed through the exhaust gases percentages) for the 125 cm^3 displacement and for the, and for the displacement of 170 cm^3 the volumetric efficiency was computed through the estimation of the air mass flow rate computed by environmental conditions and conditions in the deposit (because of the intake system considered is the nozzle-deposit) because the it resulted more accurate.

	0.9491	0.9563	0.9585	0.9819	0.9832	0.9895	0.9964	0.997	0.9997	1.0007	1.0037	1.0046	1.005	1.008	1.0087	1.009	1.01	1.013	1.015	1.016	1.02
366.519	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0.57893	0	0
418.879	0	0.40489	0	0	0.50636	0	0.58379	0	0	0	0	0.53853	0	0	0	0	0	0	0.45899	0	0
445.059	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0.66555	0
471.239	0	0	0.39744	0	0	0.45706	0	0	0	0.56493	0	0	0	0	0.62239	0	0	0	0	0	0.63749
497.419	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0.76863
523.6	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0.83581	0
549.779	0	0	0.37878	0	0	0.53669	0	0	0.63129	0	0	0	0	0	0	0.67276	0	0.71907	0	0	0
575.959	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0.70853	0	0	0	0
628.319	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0.76588	0	0	0	0	0
680.678	0.42044	0	0	0.64809	0	0	0	0.82301	0	0	0.87664	0	0	0.75035	0	0	0	0	0	0	0
733.038	0	0	0	0	0	0	0	0	0	0	0	0	0.73514	0	0	0	0	0	0	0	0
785.398	0	0	0	0	0	0	0	0	0	0	0	0	0	0.79251	0	0	0	0	0	0	0

Figure 8.13: the volumetric efficiency mapped in function of the MAP and engine speed for 125 cm^3 displacement.

	0.9491	0.9563	0.9585	0.9819	0.9832	0.9895	0.9964	0.997	0.9997	1.0007	1.0037	1.0046	1.005	1.008	1.0087	1.009	1.01	1.013	1.015	1.016	1.02
366.519	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	312.69	0	0
418.879	0	324.58	0	0	322.45	0	320	0	0	0	0	317.02	0	0	0	0	0	0	312.36	0	0
445.059	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	312.65	0
471.239	0	0	324.52	0	0	322.16	0	0	0	319.56	0	0	0	0	316.2	0	0	0	0	0	313.24
497.419	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	313.52
523.6	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	313.15	0
549.779	0	0	321.88	0	0	319.6	0	0	317.41	0	0	0	0	0	0	314.97	0	313.64	0	0	0
575.959	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	313.56	0	0	0	0
628.319	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	314.09	0	0	0	0	0
680.678	322.58	0	0	321.13	0	0	0	322.09	0	0	319.48	0	0	314.58	0	0	0	0	0	0	0
733.038	0	0	0	0	0	0	0	0	0	0	0	0	315.62	0	0	0	0	0	0	0	0
785.398	0	0	0	0	0	0	0	0	0	0	0	0	0	316.5	0	0	0	0	0	0	0

Figure 8.14: the intake temperature mapped in function of the MAP and engine speed for 125 cm^3 displacement.

The zero indicates that in these points the trials were not carried out; in fact these points were not subjected to the simulation and validation. The mapping of volumetric efficiency and intake temperature regarding the 170 cm^3 are show in the APPENDIX 2 (directly inside the script).

8.3.3 Fuel rate calculation

The block of fuel rate calculation subjected to modifications is shown in the *Figure 8.15* (ever at 5000 rpm and wide open throttle conditions for the displacement of 125cc).

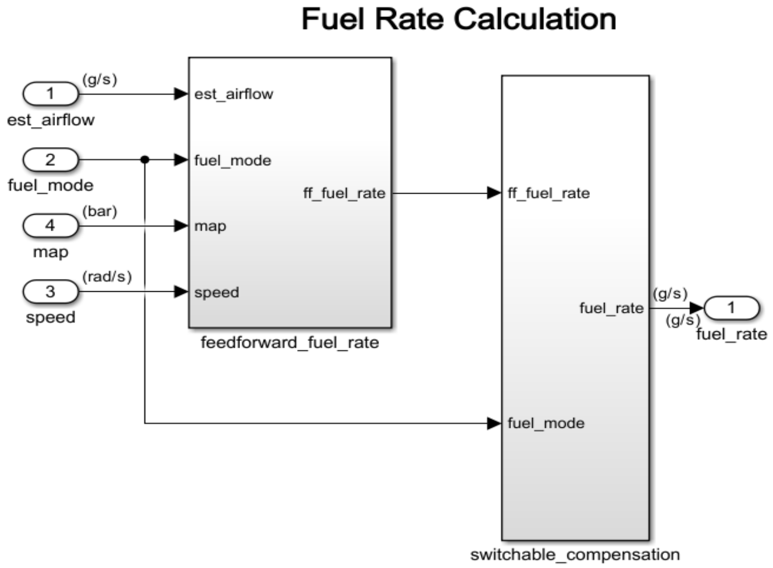

Figure 8.15: the subsystem f fuel rate calculation after the implementation.

As it is possible to see, there are more input signal, to obtain a lambda matrix in function of the engine speed and the MAP, visible in the *Figure 8.16*. Regarding the residual oxygen correction, it was set equal to zero, since as already said there are not information about the oxygen residual.

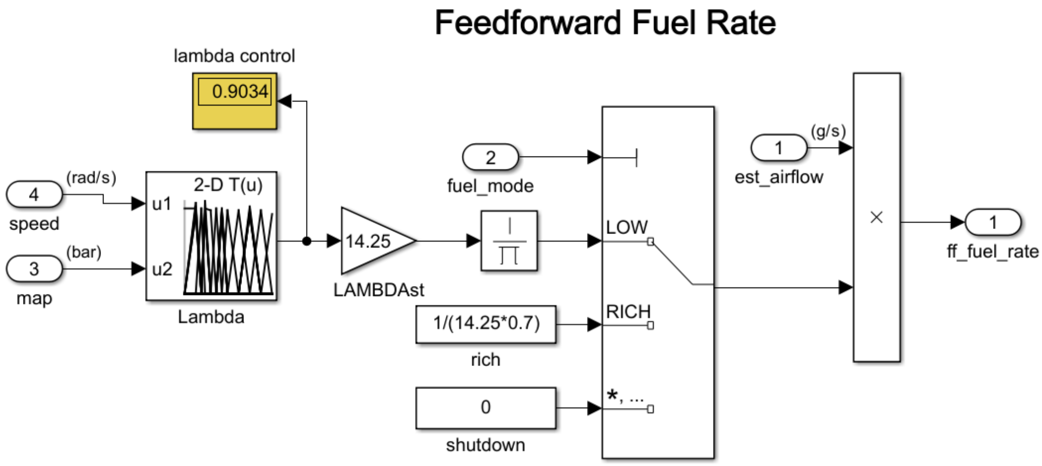

Figure 8.16: the feedforward fuel rate system implemented, shown for the displacement of 125cc in WOT at 5000 rpm.

The most important parameter here is the implementation of the lambda, but variable for the different engine operating point, and with the estimation of air flow rate, the fuel consumption can be computed (since the corrections are not considered), which will be an important parameter for the computation of injection timings. The air excess was metered with different way, and other the measurements done with devices, it has been decided to compute (and use for the model) the lambda through the estimation of the exhaust gases percentages (since this is an accurate estimation, but with the simplifications of fuel without nitrogen, dry intake air and lack of NO and NO_2 in the exhaust gases) using the Brettschneider method [9]:

$$\lambda = \frac{[\mathrm{CO_2}] + \frac{[\mathrm{CO}]}{2} + [\mathrm{O_2}] + \left(\frac{H_{CV}}{4} \cdot \frac{\mathrm{K}}{\mathrm{K} + \frac{[\mathrm{CO_D}]}{[\mathrm{CO_{2_D}}]}} - \frac{O_{CV}}{2} \right) \cdot ([\mathrm{CO_2}] + [\mathrm{CO}])}{\left(1 + \frac{H_{CV}}{4} - \frac{O_{CV}}{2}\right) \cdot ([\mathrm{CO_2}] + [\mathrm{CO}] + \mathrm{K_1} \cdot [\mathrm{HC}] \cdot 10^{-4})}$$

where the square brackets indicate the gas concentration in volume, H_{CV} indicates the atomic ratio of hydrogen to carbon in fuel, O_{CV} indicates the atomic ratio of oxygen to carbon in fuel, K is 3,5 the balance constant of the reaction $H_2O + CO \leftrightarrow H_2 + CO_2$, and K_1 is the number of the carbon atoms in the selected HC molecule, namely 6. Therefore, the matrix obtained for the air excess values in function of the MAP and engine speed is shown in the *Figure 8.17* for the displacement of 125 cm^3. For the higher displacement of 170 cm^3 the lambda mapping is visible in the APPENDIX 2 (directly inside the script).

	0.9491	0.9563	0.9585	0.9819	0.9832	0.9895	0.9964	0.997	0.9997	1.0007	1.0037	1.0046	1.005	1.008	1.0087	1.009	1.01	1.013	1.015	1.016	1.02
366.519	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0.90919	0	0
418.879	0	0.92281	0	0	0.85395	0	0.83911	0	0	0	0	0.9642	0	0	0	0	0	0	0.91056	0	0
445.059	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0.90384	0
471.239	0	0	0.98488	0	0	0.84186	0	0	0	0.83906	0	0	0	0	0.89251	0	0	0	0	0	0.90732
497.419	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0.89726
523.6	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0.90336	0
549.779	0	0	0.87808	0	0	0.84654	0	0	0.84638	0	0	0	0	0	0	0.8918	0	0.90386	0	0	0
575.959	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0.8973	0	0	0	0
628.319	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0.88839	0	0	0	0	0
680.678	0.92866	0	0	0.87378	0	0	0	0.85116	0	0	0.85161	0	0	0.88253	0	0	0	0	0	0	0
733.038	0	0	0	0	0	0	0	0	0	0	0	0	0.88012	0	0	0	0	0	0	0	0
785.398	0	0	0	0	0	0	0	0	0	0	0	0	0	0.84946	0	0	0	0	0	0	0

Figure 8.17: the lambda values implemented in function of MAP and engine speed, for 125cc cubic capacity.

Even in this case the zero indicates that the measurement was not carried out in that point, thus even the simulations. The dosage is slightly rich since the air excess is closely lower than 1.

8.3.4 Implementation of the injector pulse width correlation

Once that the fuel consumption was computed, the injection pulse width can be calculated with the aid of the equation described in the subchapter 8.1. Firstly, the fuel consumption, must be converted into grams per cycle, namely the required fuel (ReqdFuel):

$$ReqdFuel\left({}^{g}/_{cycle}\right) = \frac{\dot{m}_f\left({}^{g}/_{s}\right)}{n\left({}^{rev}/_{min}\right)\cdot\frac{1}{2}(cycle)} = \frac{\dot{m}_f\left({}^{g}/_{s}\right)}{\frac{n}{60}\left({}^{rev}/_{s}\right)\cdot\frac{1}{2}(cycle)}$$

In particular, the block called fuel rate per cycle is used for making this conversion, as the *Figure 8.18* shows.

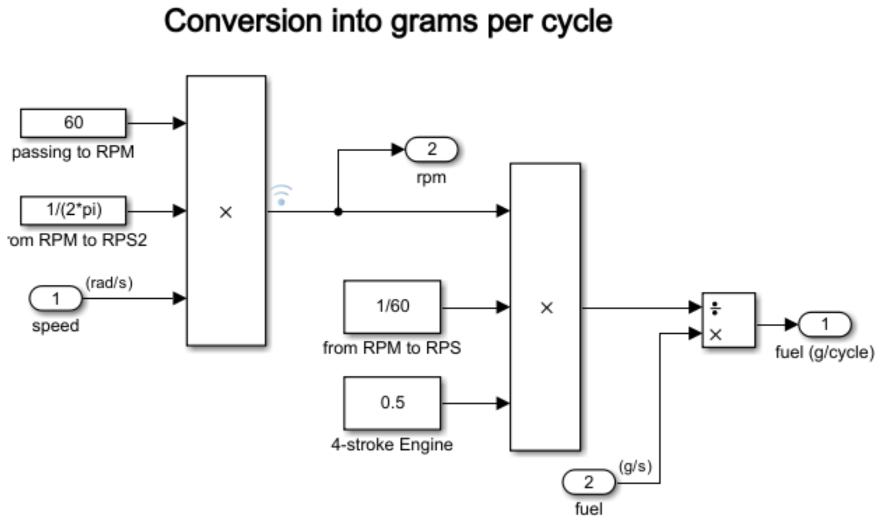

Figure 8.18: the conversion of fuel consumption into grams per cycle.

The first multiplication is used to convert the speed in rpm so that can be visualised on the main page of the model, and even used for the fuel consumption conversion. After calculating the fuel required for each engine operating point, the injection pulse width can be computed through the block called injection calculation, which is visible in the *Figure 8.19*:

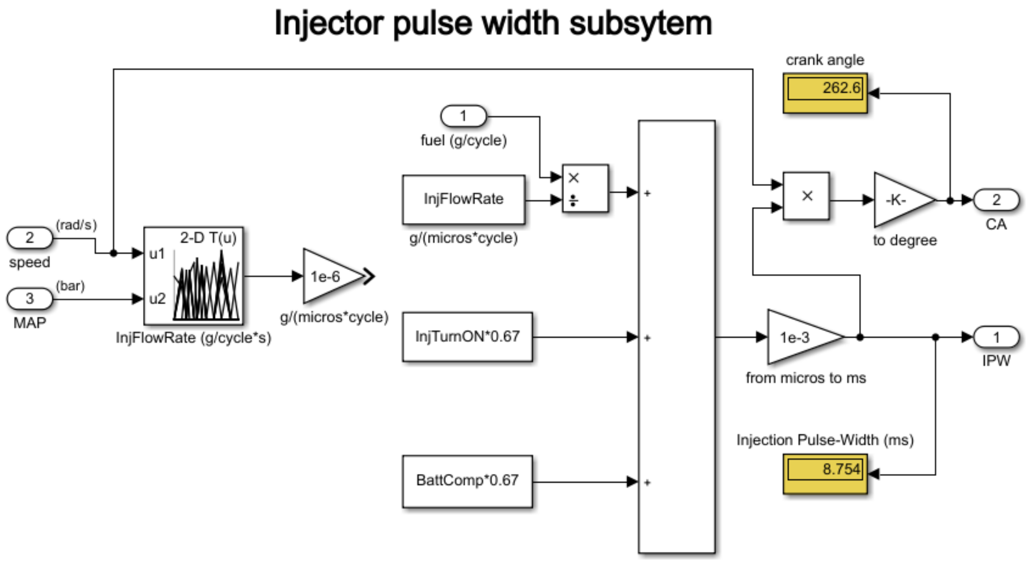

Figure 8.19: the subsystem of the injector pulse width.

As discussed, there are the terms useful for computing the injection timings, namely the injector flow rate (InjFlowRate), injector turn on (InjTurnON), battery compensation (BattComp) and the required fuel (ReqdFuel); reminding that the InjFlowRate, InjTurnON, BattComp are constants for each type of injector, but they are not known because are belong to the manufacturer.

Hence, the InjTurnON can be approximated to a typical value of 1,5 ms = 1500 micros. The battery compensation term has been computed starting from the oscilloscope injection analysis, measuring the leap carried out by the injection signal for passing from zero to the injector full opening, as shown in the *Figure 8.20* (example made at 3500 rpm in WOT for the 125 cubic capacity)

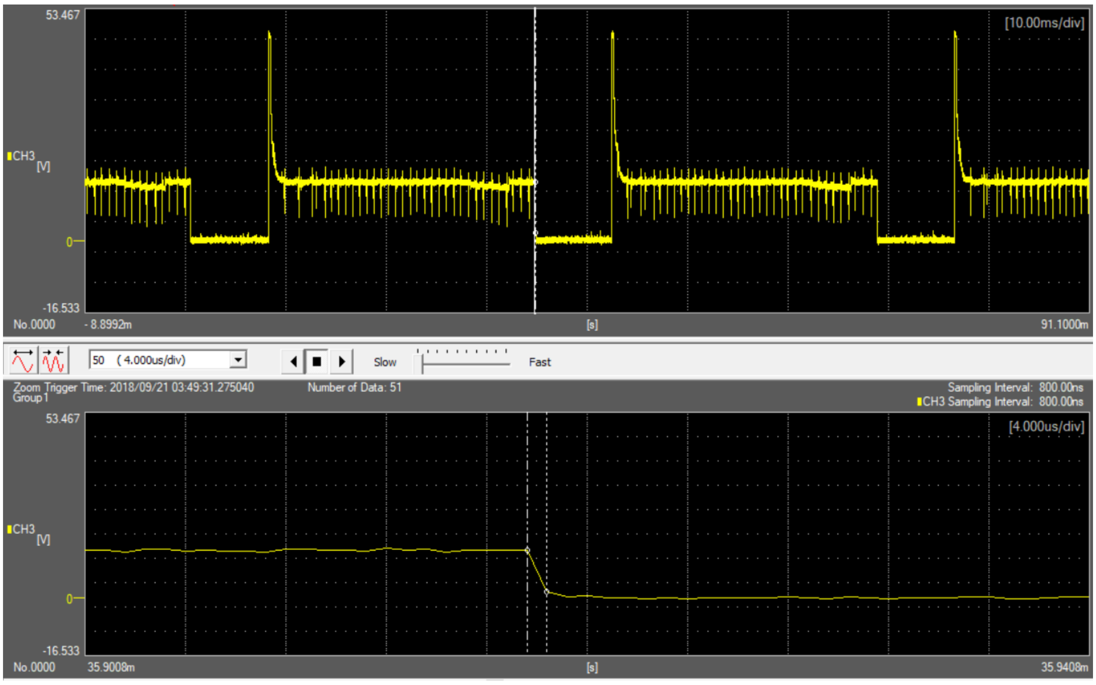

Figure 8.20: the battery compensation measurement at 3500 rpm and WOT condition for 125 cm³ displacement.

Therefore, by referring this length of the injection signal to the crank angle and the engine speed, it is possible to calculate the BattComp in time; afterwards, by doing a linear regression with all injection signal analysis, the battery compensation is around 0,8 μs for the displacement of 125 cm^3, and around 1,28 μs for the displacement of 170 cm^3.

In general, it is possible to affirm that the injectors have a flow rate of 1 gram each second of operating, but for better simulations the InjFlowRate term has been computed through the injection timings formula by doing a linear regression with all the engine operating points, already having the lacking terms of the equation. For the 125cc engine size the injector flow rate computed is around 1,18 g/s and for the 170cc engine size is around 1,39 g/s. Finally, the injection timings can be computed and being visualised in the control display, even in the respective crank angles.

For the two displacement the terms of BattComp and IFR (InjFlowRate) are not equals, even if they should be, because in the test bench with the engine size 170 cm^3, the fuel pipe had a lower length; afterwards, the fuel injected through the injector had an higher pressure, thus these terms result higher.

Moreover, an IFR table has been implemented in the model for computing the injection timings, to verify which the differences could be between the injection timings computed with IFR constant and the injection timings ideally computed with IFR variable. The IFR mapping for the 125 is shown in the *Figure 8.21* for the engine size of 125cc, and as previously is in function of the MAP (on the columns) and of the engine speed (on the rows).

	0.9491	0.9563	0.9585	0.9819	0.9832	0.9895	0.9964	0.997	0.9997	1.0007	1.0037	1.0046	1.005	1.008	1.0087	1.009	1.01	1.013	1.015	1.016	1.02
366.519	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0.986515...	0	0
418.879	0	1.079606...	0	0	1.158378...	0	1.127951...	0	0	0	0	0.868218...	0	0	0	0	0	0	0.750655...	0	0
445.059	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1.110879...	0
471.239	0	0	0.987487...	0	0	1.043980...	0	0	0	1.112149...	0	0	0	0	1.091376...	0	0	0	0	0	1.052354...
497.419	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1.289635...
523.6	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1.390619...	0
549.779	0	0	1.022306...	0	0	1.194297...	0	0	1.237490...	0	0	0	0	0	0	1.178986...	0	1.220692...	0	0	0
575.959	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1.206150...	0	0	0	0
628.319	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1.294080...	0	0	0	0	0
680.678	1.226782...	0	0	1.485507...	0	0	0	1.593980...	0	0	0.955104...	0	0	1.250449...	0	0	0	0	0	0	0
733.038	0	0	0	0	0	0	0	0	0	0	0	0	1.167371...	0	0	0	0	0	0	0	0
785.398	0	0	0	0	0	0	0	0	0	0	0	0	0	1.287471...	0	0	0	0	0	0	0

Figure 8.21: the injector flow rate table mapped in function of the MAP and the engine speed, for the engine size of 125cc.

8.3.5 Model workspace

It has been decided to implement all input parameters in two different scripts for the different displacements using the MATLAB workspace, instead that the Simulink workspace, for making the simulations more effectives and intuitive. So that once that the script will be open, it will have to be ran by choosing a different values of engine speed and of the throttle openings inside the script (through the vectors "enginespeed" in rad/s and "throttle_opening",) and successively the Simulink simulation can be ran, having already set the maximum values and minimum values of the engine parameters such as pressure, throttle openings and engine speed, which will be read by the sensors. The scripts can be seen in the APPENDIX 2.

8.4 Model results and validation

Once that the model has been completed, the following step is its validation by comparing the model results with the experimental data. The validation is going to be made for both displacement in full load and partial load. Regarding the displacement of 170cc the injection timings validation can be made only in full load, because in partial load there are not data regarding the oscilloscope signal analysis; but some data, as the air mass flow rate can be compared with the model results, and the injection timing will be computed but only for educational purpose.

8.4.1 Displacement of 125 cm^3

Firstly, the results will be analysed in wide open throttle conditions. The *Table 8.1* shows the comparison between the experimental analysis and the model results in full load, and the relative errors. In particular, from left, the engine speed; the injection pulse width, the fuel rate, the air flow rate regarding the experimental analysis; the injection pulse width, the fuel rate, the air flow rate regarding the model results; and the respective errors for the injection pulse width, the fuel rate and the air flow rate.

rpm	Experimental analysis			Model results			Error %		
	PW	Fuel rate	Air flow R	PW	Fuel rate	Air flow R			
	ms	g/s	g/s	ms	g/s	g/s	PW	Fuel rate	Air flow R
3500	7,683	0,192	2,490	6,341	0,184	2,382	17,47	4,34	4,55
4000	7,953	0,174	2,256	5,234	0,167	2,160	34,19	4,22	4,44
4250	7,863	0,270	3,476	7,182	0,258	3,328	8,66	4,23	4,43
4500	7,913	0,273	3,525	6,911	0,262	3,382	12,66	4,03	4,22
4750	7,878	0,351	4,486	8,200	0,336	4,308	4,09	4,11	4,14
5000	7,889	0,399	5,135	8,754	0,381	4,909	10,97	4,37	4,60
5250	7,748	0,360	4,639	7,638	0,343	4,415	1,42	4,80	5,07
5500	7,778	0,374	4,788	7,571	0,355	4,545	2,67	5,08	5,36
6000	7,898	0,446	5,647	8,154	0,422	5,345	3,25	5,32	5,64
6500	8,018	0,475	5,993	8,038	0,450	5,659	0,24	5,27	5,91
7000	8,408	0,504	6,323	7,871	0,473	5,933	6,39	6,15	6,58
7500	8,502	0,603	7,304	8,676	0,566	6,854	2,04	6,14	6,56

Table 8.1: the experimental analysis compared to the model results, in full load for the 125cc engine size.

The table shows clearly that regarding the air fuel rate and fuel rate, there are few percentages of error which tend to increase with the rpm increasing. Regarding the injector pulse width, there are relevant errors for low rpm which tends to decrease with the rpm increasing.

The *Figure 8.22* shows the trends of the injection timings regarding the oscilloscope signal analysis, the injection timings computed by the model with IFR = 1,18 and injection timings computed by the model with IFR ideally not constant (since it is a constant for each injector), for visualising better the differences and the percentage errors, in full load.

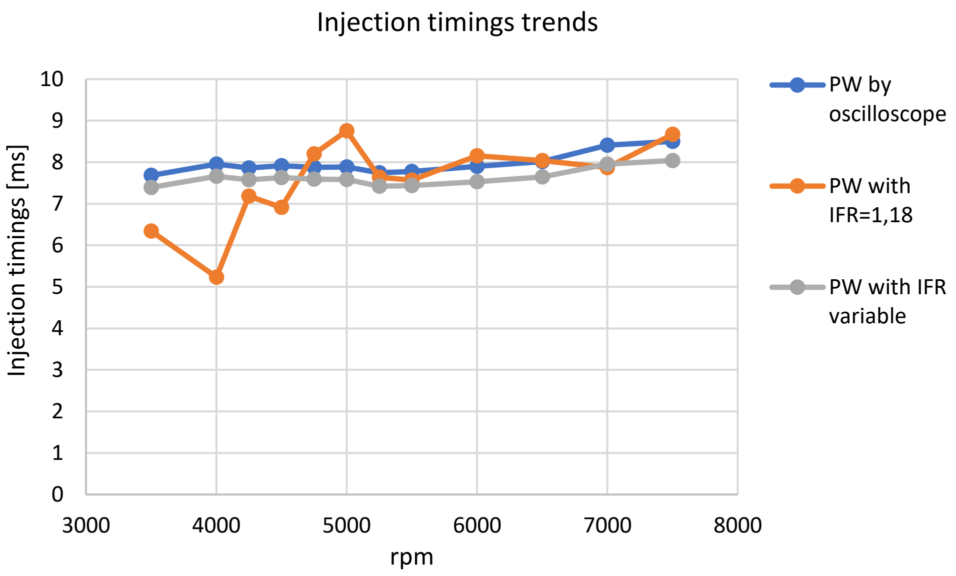

Figure 8.22: trends of the injection timings for the displacement of 125cc, in WOT conditions.

The injection pulse width computed with IFR not constant are visible in the APPENDIX 2, only in case of wide open throttle.

Regarding the partial load, the *Table 8.2* shows the comparison between experimental analysis and model results, where from right to left: the throttle opening percentage, the engine speed; the injection pulse width, the fuel rate, the air flow rate regarding the experimental analysis; the injection pulse width, the fuel rate, the air flow rate regarding the model results; and the respective errors for the injection pulse width, the fuel rate and the air flow rate.

Throttle %	rpm	Experimental analysis			Model results			Error %		
		PW	Fuel rate	Air flow R	PW	Fuel rate	Air flow R			
		ms	g/s	g/s	ms	g/s	g/s	PW	Fuel rate	Air flow R
21,2	4000	5,210	0,151	1,870	4,342	0,131	1,728	16,66	13,16	7,60
27,9	4000	6,319	0,205	2,312	5,673	0,184	2,236	10,23	10,41	3,28
43,3	4000	7,408	0,241	2,822	6,598	0,220	2,633	10,94	8,53	6,70
54,4	4000	7,683	0,193	3,148	5,574	0,180	2,472	27,45	6,91	21,49
25,2	4500	5,243	0,157	2,029	4,082	0,136	1,913	22,13	13,14	5,72
33	4500	6,398	0,211	3,066	5,310	0,191	2,287	17,01	9,67	25,39
42,4	4500	7,283	0,262	3,173	6,448	0,241	2,883	11,47	7,91	9,15
57,4	4500	7,638	0,271	3,826	6,748	0,254	3,235	11,66	6,28	15,45
29,8	5250	5,381	0,196	2,394	4,321	0,171	2,144	19,69	12,41	10,42
37,9	5250	6,510	0,288	3,216	6,072	0,262	3,159	6,73	8,97	1,79
48,1	5250	7,206	0,336	3,508	7,070	0,313	3,780	1,90	6,64	7,75
61,6	5250	7,588	0,340	3,516	7,244	0,322	4,097	4,54	5,04	16,53
31,4	6500	4,802	0,252	2,819	4,444	0,220	2,911	7,45	12,79	3,28

37,5	6500	6,142	0,413	4,007	6,860	0,375	4,664	11,68	9,36	16,40
42,5	6500	7,246	0,539	5,075	8,732	0,494	5,996	20,50	8,26	18,13
59,6	6500	7,943	0,577	5,878	9,354	0,534	6,482	17,76	7,41	10,27

Table 8.2: the experimental analysis compared to the model results, in partial load for the 125cc engine size.

In partial load, the errors achieved are higher than in full load, due to the not full opening of the throttle, which produces pressure drops.

8.4.1 Displacement of 170 cm^3

The *Table 8.3* shows the experimental analysis compared to the model results in full load, and the relative errors committed. From left, the engine speed; the injection pulse width, the fuel rate, the air flow rate regarding the experimental analysis; the injection pulse width, the fuel rate, the air flow rate regarding the model results; and the respective errors for the injection pulse width, the fuel rate and the air flow rate.

rpm	Experimental analysis			Model results			Error %		
	PW	Fuel rate	Air flow R	PW	Fuel rate	Air flow R			
	ms	g/s	g/s	ms	g/s	g/s	PW	Fuel rate	Air flow R
3000	10,428	0,364	4,802	11,403	0,361	4,753	9,35	1,00	1,01
3250	10,486	0,396	5,274	11,432	0,392	5,206	9,03	1,22	1,29
3500	10,640	0,429	5,746	11,462	0,423	5,668	7,72	1,35	1,36
3750	10,568	0,448	6,099	11,204	0,442	6,013	6,02	1,38	1,40
4000	10,478	0,462	6,447	10,839	0,455	6,347	3,45	1,53	1,55
4250	10,387	0,482	6,580	10,654	0,474	6,468	2,57	1,68	1,70
4500	10,493	0,510	6,902	10,627	0,500	6,768	1,28	1,91	1,93
4750	10,588	0,539	7,189	10,617	0,528	7,031	0,27	2,18	2,20
5000	10,823	0,573	7,464	10,679	0,559	7,278	1,33	2,48	2,49
5250	10,567	0,588	7,748	10,426	0,572	7,533	1,33	2,76	2,78
5500	10,395	0,606	7,986	10,245	0,587	7,741	1,44	3,06	3,07
6000	10,274	0,641	8,472	9,946	0,620	8,187	3,19	3,33	3,35
6500	10,753	0,715	9,321	10,184	0,690	8,989	5,29	3,55	3,56
7000	11,209	0,796	10,281	10,460	0,765	9,875	6,68	3,94	3,95
7500	11,279	0,854	10,931	10,441	0,818	10,471	7,43	4,20	4,22
8000	10,823	0,858	11,307	9,920	0,824	10,857	8,34	3,96	3,98
8500	10,519	0,913	11,848	9,891	0,873	11,331	5,97	4,34	4,36
9000	10,030	0,932	12,234	9,573	0,891	11,689	4,56	4,44	4,46
9250	9,537	0,889	11,584	8,911	0,845	11,015	6,56	4,90	4,91
9500	9,087	0,866	11,325	8,474	0,820	10,720	6,75	5,32	5,34
9750	9,066	0,873	11,386	8,354	0,828	10,795	7,85	5,16	5,18

Table 8.3: the experimental analysis compared to the model results, in full load for the 170 cm^3 displacement.

In comparison to the 125cc, in the case of 170cc the errors are slightly lower, especially regarding the fuel rate and the air mass flow. Because the experimental analysis of the oscilloscope has been carried out only in full load, thus the injector parameters have been computed without consider the partial load.

The *Figure 8.23* shows the injection timings trends regarding the oscilloscope signal analysis, the injection timings computed by the model with IFR = 1,39 and injection timings computed by the model with IFR ideally not constant (since it is a constant for each injector), for visualising better the differences and the percentage errors, in full load.

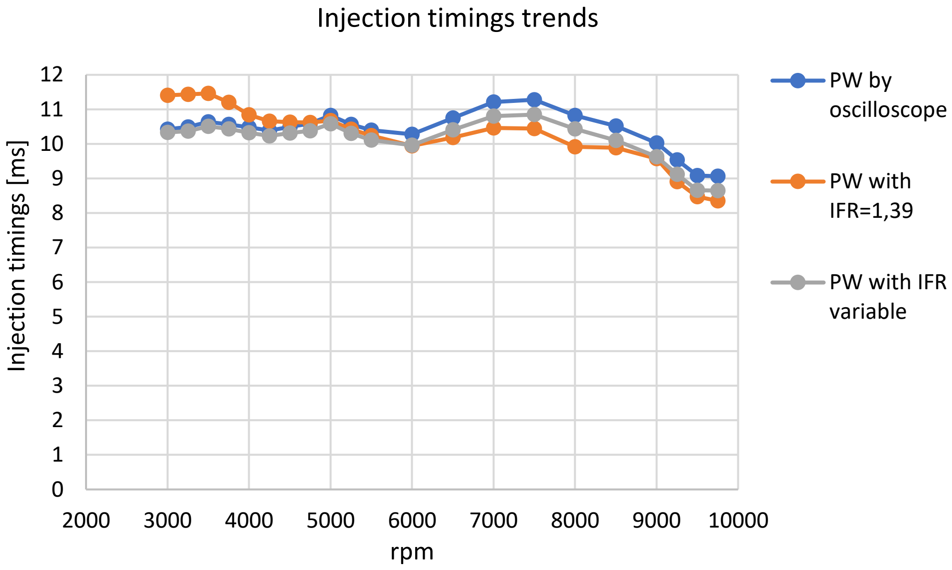

Figure 8.23: trends of the injection timings for the displacement of 170cc, in WOT conditions.

The differences between the various trends are lower compared to the 125cc engine size. The injection pulse width calculated with IFR not constant are visible in the APPENDIX 2, only in case of full load.

The *table 8.4* shows the model results, only compared to the experimental results of the air mass flow rate where from right to left it is possible to see: the throttle opening percentage, the engine speed; the air flow rate regarding the experimental analysis; the injection pulse width, the fuel rate, the air flow rate regarding the model results; and the respective errors for the air flow rate. As anticipated, these data have been obtained as an educational purpose, since there are not the experimental data regarding the injection timings. Moreover, the air flow rate and fuel rate, regarding the partial load, did not estimated with a good approximation; thus for this reason the error is higher than it should be.

Throttle %	rpm	Experimental analysis		Model results			Error %	
		Fuel rate	Air flow R	PW	Fuel rate	Air flow R		
		g/s	g/s	ms	g/s	g/s	Fuel rate	Air flow R
20	3000	0,192	2,643	5,908	0,170	2,336	11,56	11,60
20	3750	0,255	3,541	6,059	0,219	3,037	14,21	14,24
20	4500	0,264	3,691	5,160	0,216	3,021	18,11	18,15
20	5250	0,242	3,362	4,169	0,192	2,669	20,59	20,60
20	6000	0,261	3,613	3,880	0,199	2,758	23,67	23,68
20	6500	0,233	3,237	3,343	0,176	2,345	24,74	27,55
20	7000	0,258	3,609	3,334	0,188	2,630	27,11	27,11
20	7500	0,129	1,794	2,060	0,091	1,268	29,32	29,31
20	8000	0,383	5,344	3,850	0,263	3,666	31,39	31,41
20	8500	0,358	4,986	3,445	0,240	3,334	33,13	33,13
20	9000	0,275	3,811	2,797	0,186	2,581	32,25	32,27
20	9500	0,358	5,062	3,202	0,241	3,405	32,72	32,72
20	9750	0,376	5,397	3,533	0,285	4,088	24,32	24,26
40	3000	0,227	3,163	7,226	0,216	3,000	5,10	5,14
40	3750	0,233	3,244	6,113	0,221	3,081	5,00	5,04
40	4500	0,363	5,049	7,559	0,341	4,737	6,17	6,18
40	5250	0,350	4,847	6,367	0,325	4,504	7,06	7,07
40	6000	0,354	4,881	5,646	0,322	4,441	8,99	9,02
40	6500	0,408	5,575	5,973	0,373	5,094	8,63	8,64
40	7000	0,464	6,347	6,200	0,420	5,748	9,42	9,43
40	7500	0,072	0,991	1,754	0,065	0,890	10,14	10,22
40	8000	0,269	3,739	3,604	0,240	3,332	10,85	10,87
40	8500	0,333	4,620	4,004	0,295	4,087	11,52	11,54
40	9000	0,474	6,622	5,018	0,418	5,835	11,85	11,90
40	9500	0,539	7,532	5,353	0,477	6,669	11,43	11,46
40	9750	0,460	6,433	4,567	0,401	5,614	12,71	12,73
50	3000	0,208	2,930	6,733	0,199	2,792	4,67	4,71
50	3750	0,344	4,856	8,710	0,334	4,706	3,06	3,08
50	4500	0,408	5,719	8,549	0,392	5,493	3,93	3,94
50	5250	0,425	5,872	7,662	0,404	5,579	4,96	4,99
50	6000	0,461	6,426	7,276	0,435	6,057	5,71	5,73
50	6500	0,442	6,107	6,504	0,413	5,711	6,49	6,49
50	7000	0,486	6,740	6,589	0,452	6,261	7,08	7,11
50	7500	0,547	7,512	6,819	0,504	6,916	7,92	7,94
50	8000	0,431	5,943	5,258	0,393	5,425	8,70	8,71
50	8500	0,342	4,772	4,190	0,313	4,365	8,45	8,53
50	9000	0,342	4,794	3,981	0,309	4,341	9,44	9,45
50	9500	0,464	6,518	4,892	0,427	5,994	8,04	8,05
50	9750	0,428	6,031	4,432	0,386	5,443	9,74	9,76
60	3000	0,408	5,668	12,488	0,398	5,525	2,51	2,52
60	3750	0,409	5,667	10,157	0,397	5,495	3,01	3,02

60	4500	0,372	5,189	7,901	0,359	4,999	3,66	3,67
60	5250	0,435	6,095	7,845	0,415	5,816	4,56	4,56
60	6000	0,492	6,807	7,726	0,466	6,450	5,22	5,25
60	6500	0,581	7,975	8,278	0,546	7,503	5,90	5,92
60	7000	0,592	8,038	7,858	0,554	7,528	6,32	6,33
60	7500	0,165	2,239	2,770	0,153	2,076	7,23	7,27
60	8000	0,603	8,187	7,027	0,557	7,560	7,64	7,67
60	8500	0,606	8,234	6,660	0,556	7,551	8,27	8,29
60	9000	0,492	6,665	5,324	0,449	6,084	8,66	8,72
60	9500	0,933	12,660	8,826	0,859	11,644	8,01	8,02
60	9750	0,675	9,150	6,426	0,611	8,277	9,51	9,54
80	3000	0,281	3,867	8,803	0,270	3,725	3,66	3,66
80	3750	0,344	4,715	8,678	0,333	4,550	3,47	3,49
80	4500	0,378	5,167	7,993	0,363	4,969	3,81	3,83
80	5250	0,464	6,381	8,333	0,445	6,114	4,16	4,18
80	6000	0,463	6,331	7,374	0,442	6,033	4,69	4,71
80	6500	0,594	8,066	8,507	0,564	7,646	5,21	5,21
80	7000	0,697	9,499	9,131	0,657	8,954	5,73	5,74
80	7500	0,497	6,704	6,386	0,466	6,287	6,22	6,22
80	8000	0,568	7,606	6,738	0,530	7,097	6,68	6,70
80	8500	0,711	9,547	7,750	0,663	8,895	6,82	6,84
80	9000	0,611	8,344	6,438	0,565	7,714	7,54	7,56
80	9500	0,922	12,720	8,808	0,857	11,813	7,12	7,13
80	9750	0,686	9,549	6,581	0,628	8,743	8,43	8,45

Table 8.4: the experimental analysis compared to the model results, in partial load for the 170cc engine size.

The percentage errors are higher compared with the full load because the estimation of the fuel consumption by means of the air mass flow and lambda has not been carried out through the nozzle-deposit system as intake system by using the trial cell-deposit differential pressure, since the partial load tests were made only by means of the conventional filter.

Chapter 9: Conclusion

The model developed is proved to have a good approximation for calculating the injection pulse width (including the intake air flow rate and the fuel consumption), since the percentage errors computed turned out moderately lows, although the injector data did not known and some parameters have not been adequately measured.

The parameters which conditions mainly the computations is the fuel consumption, because the Coriolis ELITE CMF025 device used for the has a measurements range of 0-250 kg/h, and the fuel consumption measured during the test bench is in the range of 0,5-3,5 kg/h (the fuel consumption of 170cc engine size achieves 3,5 kg/h in WOT condition and high rpm); hence, it possible that the fuel rate measurements have not been so accurate. The *Figure 9.1* shows an example of the fuel consumption measured for the displacement of 125 cm^3 in full load at 6500 rpm, in function of the time.

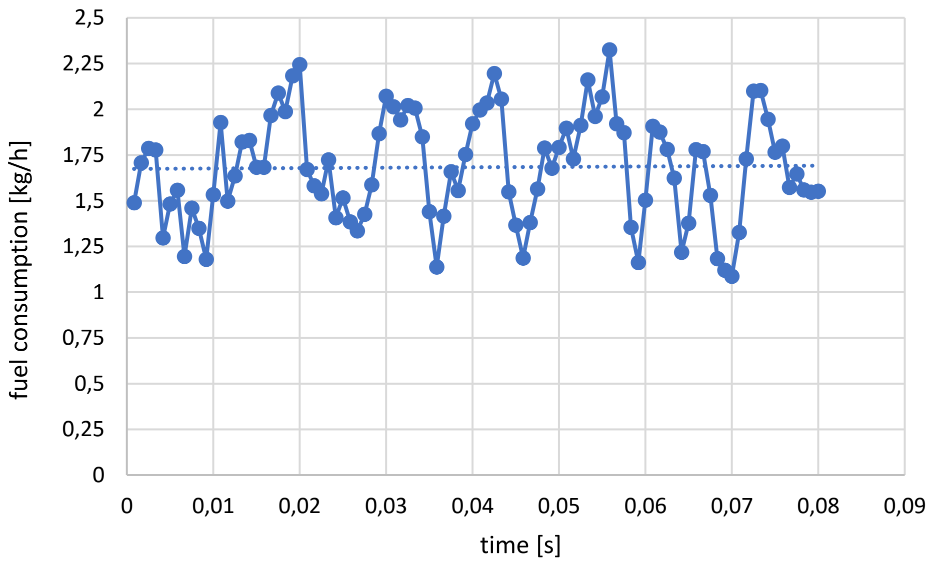

Figure 9.1: the fuel consumption measured for the engine size of 125cc in WOT.

The fuel consumption fluctuation is even higher than 1 kg/h. In fact, it has been decided to use the “totalizator” (explained in the paragraph 5.2.9) for stabilising the fuel consumption measurement with an higher precision, because the measurements range is till two orders of magnitude larger, otherwise the committed errors would have been higher. For this reason for the displacement of 170cc it has been decided to use the fuel consumption computed through the air mass flow estimation calculated by means of the trial cell-deposit differential pressure (for the trials made with the nozzle-deposit system as intake system), and with the lambda estimation.

For example, for the displacement of 125 cm^3, by using the air mass flow rate computed through the differential pressure of the trial cell-deposit system for computing the fuel consumption, the injector flor rate (IFR) resulting is equal to 1,067 (instead of 1,18), by using only the data regarding the wide open throttle (as done for the 170 engine size). Therefore, the *Table 9.1* shows these results regarding the injection timings.

rpm	Experimental analysis			Model results			Error %		
	PW	Fuel rate	Air flow R	PW	Fuel rate	Air flow R			
	ms	g/s	g/s	ms	g/s	g/s	PW	Fuel rate	Air flow R
3500	7,683	0,184	2,379	7,394	0,184	2,382	3,77	0,11	0,10
4000	7,953	0,244	3,169	7,659	0,167	2,160	3,70	31,82	46,71
4250	7,863	0,267	3,438	7,573	0,258	3,328	3,69	3,17	3,29
4500	7,913	0,275	3,557	7,634	0,262	3,382	3,52	4,89	5,16
4750	7,878	0,282	3,603	7,595	0,336	4,308	3,59	19,39	16,36
5000	7,889	0,342	4,400	7,587	0,381	4,909	3,82	11,60	10,37
5250	7,748	0,289	3,717	7,424	0,343	4,415	4,19	18,81	15,81
5500	7,778	0,291	3,724	7,435	0,355	4,545	4,41	22,04	18,05
6000	7,898	0,431	5,453	7,531	0,422	5,345	4,64	1,96	2,01
6500	8,018	0,429	5,417	7,649	0,450	5,659	4,61	4,80	4,27
7000	8,408	0,464	5,817	7,952	0,473	5,933	5,42	2,03	1,96
7500	8,502	0,472	5,721	8,042	0,566	6,854	5,41	19,83	16,54

Table 9.1: the experimental analysis compared to the model results, in full load for the 125cc engine size, with IFR=1,067.

The percentages errors of the injector pulse width have decreased mainly at 3500 and 4000 rpm (where before were 17,5% and 34,2%), but at the same time the errors regarding the fuel rate and the air mass flow rate have increased because the volumetric efficiency is not calculated through the air flow rate computed by the differential pressure of the trial cell-deposit system.

Moreover, it could be interesting expand the mapping of the engine parameters implemented in the model, in order to consider as well, the throttle transient, and making an estimation of the oxygen residual, if participating, of the exhaust gases, in order to obtain an higher precision of the injection pulse width, even if these parameters should little condition the final results.

APPENDIX 1

1.1 125 cm^3 in full load (nozzle-deposit system)

Engine data													
Number of cylinders, z	-	1											
Volumen desplazado por cilindro, V_cil	cm^3	124,7											
Thermodynamic cycle per revolution, i	-	0,5											
Datos del combustible													
number of C atoms, alfa	-	1											
number of H atoms, beta	-	1,861											
number of O atoms, gamma	-	0,016											
number of N atoms, delta	-	0											
molecular weight, Mf	$kg_f/kmol_f$	17,08											
Air Fuel Ratio stoiciometric, AFR_stq	kg_a/kg_f	14,25											
lower heating value, LHV	MJ/kg	43,0	43,0	43,0	43,0	43,0	43,0	43,0	43,0	43,0	43,0	43,0	43,0
Measured parameters													
point	-	1	2	3	4	5	6	7	8	9	10	11	12
Throttle opening	grados	100	100	100	100	100	100	100	100	100	100	100	100
Engine speed, torque, fuel consumption													
Engine speed, n	rpm	3.500	4.000	4.250	4.500	4.750	5.000	5.250	5.500	6.000	6.500	7.000	7.500
Actual Torque, T_e	Nm												
Fuel consumption, m_f	kg/h	0,6917	0,6258	0,9713	0,9813	1,2629	1,4358	1,2963	1,3479	1,6054	1,7100	1,8147	2,1717
	Kg/s	0,00019	0,00017	0,00027	0,00027	0,00035	0,00040	0,00036	0,00037	0,00045	0,00048	0,00050	0,00060
Environmental conditions of the trial cell													
barometric pressure, p_cell	mbar	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1.022
Temperature, t_cell	ºC	28	28	28	28	28	28	28	28	28	28	28	28
Humidity relative, phi_rel	%	47	47	47	47	47	47	47	47	47	47	47	47
Data for obtaining the intake air mass flow													
Nozzle discharge coefficient	-	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93
Differential pressure, Dp_atm-dep	mbar	0,102	0,181	0,213	0,228	0,234	0,349	0,249	0,250	0,536	0,529	0,610	0,590
Intake and exhaust pressure													
Intake manifold absolute pressure, Dp_atm-adm	mbar	1.015	1.015	1.016	1.019	1.020	1.018	1.013	1.010	1.009	1.008	1.005	1.009
Exhaust manifold absolute pressure, Dp_esc-atm	mbar	977,59	973,83	971,30	971,35	967,90	963,83	965,38	963,83	959,50	953,15	945,44	942,15
Intake temperature	ºC	39,54	39,21	39,50	40,09	40,37	40,00	40,49	40,41	40,94	41,43	42,47	43,35
Results of the exhaust gases analysis (HC < 100 ppm y NOx < 1000 ppm)													
Exhuast gases temperature, t_esc	ºC	525	553	560	568	573	586	597	607	619	639	656	650
Vol. Correction of O2 in the dry exhaust gases, O2_D	%	0,19	0,19	0,19	0,19	0,19	0,17	0,15	0,15	0,17	0,18	0,17	0,16
Vol. Correction of CO2 in the dry exhaust gases, CO2_D	%	12,54	11,64	12,14	12,08	10,93	11,15	11,35	11,15	10,78	11,40	11,70	10,80
Vol. Correction of CO in the dry exhaust gases, CO_D	%	3,08	2,93	3,31	3,19	3,37	3,08	3,07	3,28	3,55	3,92	3,83	4,85
Vol. Correction of HC in the dry exhaust gases, HC_D	ppm C6	315	224	238	211	147	202	207	198	196	194	462	471
Vol. Correction of NOx in the dry exhaust gases, NOx_D	ppm NO+NO2	0	0	0	0	0	0						
Dilution factor, DF		1,10	1,10	1,10	1,10	1,10	1,10	1,10	1,10	1,10	1,10	1,10	1,10
measured lambda HORIBA	-	1,02	1,10	1,03	1,05	1,11	1,12	1,11	1,11	1,11	1,03	1,01	1,00
Vol. relation of O2 in the dry exhaust gases, O2_D	%	2,14	2,12	2,11	2,13	2,09	1,92	1,69	1,64	1,82	1,95	1,84	1,77
Vol. relation of CO2 in the dry exhaust gases, CO2_D	%	11,40	10,58	11,04	10,98	9,94	10,14	10,32	10,14	9,80	10,36	10,64	9,82
Vol. relation of CO in the dry exhaust gases, CO_D	%	2,80	2,66	3,01	2,90	3,06	2,80	2,79	2,98	3,23	3,56	3,48	4,41
Vol. relation of HC in the dry exhaust gases, HC_D	ppm C6	286	204	216	192	134	184	188	180	178	176	420	428
Temperature before catalyst	ºC	200,02	223,75	227,58	237,77	242,91	252,49	259,56	262,47	274,13	293,59	314,70	322,18
Temperature after catalyst	ºC	203,33	221,54	228,13	235,36	240,57	246,17	253,60	258,87	271,28	292,67	308,43	320,64

Calculated parameters														
point	-	1	2	3	4	5	6	7	8	9	10	11	12	13
engine speedr, n	rpm	3500	4000	4250	4500	4750	5000	5250	5500	6000	6500	7000	7500	8000
actual Torque, actual Power, mean effective pressure, specific fuel consumption														
actual Torque, M_e	Nm	8,88	9,03	9,04	8,80	8,58	8,57	8,63	8,69	8,81	8,98	9,00	8,63	7,87
actual Power, W_e	kW	3,25	3,78	4,02	4,15	4,27	4,49	4,75	5,01	5,54	6,11	6,60	6,78	6,59
mean effective pressure, pme	kPa	894,9	910,2	911,0	886,8	864,8	863,6	869,8	875,7	888,0	904,9	907,0	869,7	793,0
specific fuel consumption, g_ef	g/kWh	212,5	165,4	241,4	236,6	295,9	320,0	273,2	269,3	290,0	279,8	275,1	320,4	
Power and Torque corrected by environmental conditions														
Saturated steam pressure, p_s,ens (kPa)	mbar	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8
Dry air pressure, p_ens-phi*p_s,ens (kPa)	mbar	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2
Factor de corrección, alfa-a	-	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99
Power corrtd, W_e,corr	kW	3,22	3,74	3,98	4,10	4,22	4,44	4,69	4,95	5,48	6,04	6,52	6,70	6,52
Torque corrct, M_e,corr	Nm	8,78	8,93	8,94	8,70	8,49	8,48	8,54	8,59	8,71	8,88	8,90	8,53	7,78
Relative dosage, volumetric efficiency and emissions (starting from measurements of air flow rate and fue consumption)														
AFR, lambda and relative dosage														
number of C atoms, alfa	-	1,000	1,000	1,000	1,000	1,000	1,000	1,000	1,000	1,000	1,000	1,000	1,000	
number of H atoms, beta	-	1,861	1,861	1,861	1,861	1,861	1,861	1,861	1,861	1,861	1,861	1,861	1,861	
number of O atoms, gamma	-	0,016	0,016	0,016	0,016	0,016	0,016	0,016	0,016	0,016	0,016	0,016	0,016	
number of N atoms, delta	-	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	
Air Fuel Ratio stoiciometric, AFR_stq	kg_a/kg_f	14,25	14,25	14,25	14,25	14,25	14,25	14,25	14,25	14,25	14,25	14,25	14,25	
air density of the trial cell, rho_adm	kg/m^3	1,182	1,182	1,182	1,182	1,182	1,182	1,182	1,182	1,182	1,182	1,182	1,182	
air mass flow rate, m_a	kg/s	0,0024	0,0032	0,0034	0,0036	0,0036	0,0044	0,0037	0,0037	0,0055	0,0054	0,0058	0,0057	
air fuel ratio, AFR	kg_a/kg_f	12,38	18,23	12,74	13,05	10,27	11,03	10,32	9,95	12,23	11,40	11,54	9,48	
air excess, lambda	-	0,87	1,28	0,89	0,92	0,72	0,77	0,72	0,70	0,86	0,80	0,81	0,67	
relative dosage, 1/lambda	-	1,15	0,78	1,12	1,09	1,39	1,29	1,38	1,43	1,17	1,25	1,24	1,50	
Volumetric Efficiency														
Volumetric Efficiency, nhu_v,ens	-	0,55	0,64	0,66	0,64	0,62	0,72	0,58	0,55	0,74	0,68	0,68	0,62	
air density in the intake manifold, rho_adm	kg/m^3	1,131	1,132	1,132	1,134	1,133	1,132	1,125	1,123	1,119	1,117	1,109	1,111	
Volumetric Efficiency, nhu_v,adm	-	0,58	0,67	0,69	0,67	0,64	0,75	0,61	0,58	0,78	0,72	0,72	0,66	
Emissions of polluting gases														
exhuast gases mass flow rate, m_esc,w = m_f+m_a	kg/s	0,0026	0,0033	0,0037	0,0038	0,0040	0,0048	0,0041	0,0041	0,0059	0,0059	0,0063	0,0063	
bs_CO	g/kWh	81,3	86,5	101,9	98,4	104,1	110,0	88,0	89,6	126,4	126,0	122,5	151,0	
bs_HC1	g/kWh	2,5	2,5	2,8	2,5	1,7	2,7	2,3	2,1	2,6	2,4	5,6	5,6	
bs_NOx	g/kWh	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	
bs_CO2	g/kWh	519,8	540,3	587,3	585,4	531,1	625,7	511,4	478,9	602,6	576,2	588,2	528,3	

Relative dosage, volumetric efficiency and emissions (starting from the gases analysis and fuel consumption)													
AFR, lambda and relative dosage													
AFR with Brettschneider method													
K balance constant of the reaction H2O gases	-	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5
carbon atoms in the selected HC molecule	-	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0
(beta/alpha)/4	-	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653
(gamma/alpha)/2	-	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040
1+((beta/alpha)/4)-((gamma/alpha)/2)	-	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613
Lambda Brettschneider	-	0,909	0,911	0,904	0,907	0,897	0,903	0,904	0,897	0,888	0,885	0,880	0,849
relative dosage Brettschneider	-	1,100	1,098	1,106	1,102	1,115	1,107	1,106	1,114	1,126	1,130	1,136	1,177
AFR Brettschneider	kg_a/kg_f	12,96	12,98	12,88	12,93	12,79	12,88	12,88	12,79	12,66	12,62	12,54	12,11
Volumetric Efficiency													
air fuel rate from lambda, m_a	kg/s	0,0025	0,0023	0,0035	0,0035	0,0045	0,0051	0,0046	0,0048	0,0056	0,0060	0,0063	0,0073
	kg/h	8,9633	8,1216	12,5125	12,6899	16,1505	18,4864	16,6996	17,2382	20,3276	21,5750	22,7638	26,2929
fuel consumption computed	kg/s	0,00018	0,00024	0,00027	0,00028	0,00028	0,00034	0,00029	0,00029	0,00043	0,00043	0,00046	0,00047
	kg/h	0,661	0,879	0,961	0,990	1,014	1,230	1,039	1,048	1,550	1,546	1,669	1,701
Volumetric Efficiency, nhu_v,ens	-	0,579	0,459	0,666	0,637	0,769	0,836	0,719	0,709	0,766	0,750	0,735	0,793
Volumetric Efficiency, nhu_v,adm for air density in intake manifold	-	0,605	0,479	0,695	0,665	0,802	0,873	0,756	0,746	0,809	0,794	0,784	0,843
Composition of exhaust gases and emissions													
[H2O] wet	%	12,0	11,2	11,8	11,7	10,9	10,9	11,1	11,0	10,9	11,5	11,7	11,4
[O2] wet	%	0,2	0,2	0,2	0,2	0,2	0,2	0,1	0,1	0,1	0,2	0,1	0,1
[CO2] wet	%	11,0	10,3	10,7	10,7	9,7	9,9	10,1	9,9	9,6	10,1	10,3	9,6
[CO] wet	%	2,7	2,6	2,9	2,8	3,0	2,7	2,7	2,9	3,2	3,5	3,4	4,3
[HC1] wet	%	0,2	0,1	0,1	0,1	0,1	0,1	0,1	0,1	0,1	0,1	0,2	0,3
[NO2] wet	%	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0
[H2] wet	%	0,8	0,8	0,9	0,9	1,0	0,9	0,9	0,9	1,0	1,1	1,1	1,5
B1	-	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457
B2	-	7,185	7,664	7,267	7,354	7,800	7,821	7,732	7,724	7,764	7,323	7,161	7,085
B3	-	1,100	1,099	1,107	1,102	1,115	1,107	1,107	1,115	1,126	1,130	1,137	1,178
[N2] wet	%	69,6	65,3	68,4	67,8	63,2	63,5	64,3	63,9	62,9	66,4	67,5	65,9
Summation [i]	%	96,4	90,6	95,0	94,1	88,1	88,2	89,3	88,9	87,8	92,9	94,4	93,0
Molecular weight of the exhaust gases, M_esc,w	kg/kmol	27,47	25,76	26,99	26,75	24,94	25,05	25,36	25,21	24,85	26,25	26,80	26,22
mass flow rate of damp fumes m_esc,w	g/h	9655	8747	13484	13671	17413	19922	17996	18586	21933	23285	24578	28465
mass flow rate of damp fumes m_esc,w	kg/s	0,0027	0,0024	0,0037	0,0038	0,0048	0,0055	0,0050	0,0052	0,0061	0,0065	0,0068	0,0079
Mol in the exhaust gases, n_esc,w = m_esc,w/M_esc,w	mol/h	3,51E+02	3,40E+02	5,00E+02	5,11E+02	6,98E+02	7,95E+02	7,10E+02	7,37E+02	8,83E+02	8,87E+02	9,17E+02	1,09E+03
bs CO	g/kWh	82,0	65,3	101,6	97,3	137,4	136,2	114,3	120,3	141,4	140,9	131,7	192,8
bs HC1	g/kWh	15,2	9,1	13,3	11,7	10,9	16,3	14,0	13,2	14,2	12,7	28,9	34,0
bs NOx	g/kWh	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0
bs CO2	g/kWh	524,7	408,0	585,6	578,6	701,2	774,8	664,4	643,4	674,1	644,2	632,5	674,4

Composition and specific heat of exhaust gases													
Composition of exhaust gases													
y O2 wet	-	0,002	0,002	0,002	0,002	0,002	0,002	0,001	0,001	0,001	0,002	0,001	0,001
y CO2 wet	-	0,110	0,103	0,107	0,107	0,097	0,099	0,101	0,099	0,096	0,101	0,103	0,096
y CO wet	-	0,027	0,026	0,029	0,028	0,030	0,027	0,027	0,029	0,032	0,035	0,034	0,043
y H2 wet	-	0,008	0,008	0,009	0,009	0,010	0,009	0,009	0,009	0,010	0,011	0,011	0,015
y H2O wet	-	0,120	0,112	0,118	0,117	0,109	0,109	0,111	0,110	0,109	0,115	0,117	0,114
y N2 wet	-	0,696	0,653	0,684	0,678	0,632	0,635	0,643	0,639	0,629	0,664	0,675	0,659
sum yi	-	0,963	0,904	0,949	0,940	0,880	0,881	0,891	0,888	0,877	0,928	0,942	0,928
Specific heat in the exhaust gases													
Differential temperature gases atmosphere, tesc-ta	ºC	497	525	532	540	545	558	569	579	591	611	628	622
Average temperature of gases and environment, (tesc+ta)/2	ºC	276	290	294	298	301	307	312	318	323	334	342	339
cp_esc,w	kJ/kg K	1,2	1,2	1,2	1,2	1,2	1,2	1,2	1,2	1,2	1,2	1,2	1,2
Energy balance													
Thermal Power of the combustion, m_f H_f	kW	8,26	7,47	11,60	11,72	15,08	17,15	15,48	16,10	19,18	20,43	21,68	25,94
	%	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00
Actual engine Power, We	kW	3,25	3,78	4,02	4,15	4,27	4,49	4,75	5,01	5,54	6,11	6,60	6,78
	%	0,39	0,51	0,35	0,35	0,28	0,26	0,31	0,31	0,29	0,30	0,30	0,26
Sensible heat of exhaust gases, H_esc,w	kW	1,53	2,11	2,37	2,48	2,59	3,22	2,78	2,85	4,18	4,32	4,76	4,72
	%	0,19	0,28	0,20	0,21	0,17	0,19	0,18	0,18	0,22	0,21	0,22	0,18
Energy due to the unburned content, H_inq	kW	0,98	0,91	1,51	1,49	2,18	2,26	2,00	2,23	2,91	3,21	3,23	4,93
	%	0,12	0,12	0,13	0,13	0,14	0,13	0,13	0,14	0,15	0,16	0,15	0,19
Heat dissipated in cooling system, Q_ref+Q_lub	kW	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
	%	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
Other energy flows, E_otras	kW	2,49	0,67	3,70	3,60	6,05	7,19	5,95	6,01	6,54	6,78	7,08	9,51
	%	0,30	0,09	0,32	0,31	0,40	0,42	0,38	0,37	0,34	0,33	0,33	0,37

1.2 125 cm^3 in partial load (nozzle-deposit system)

Engine data																					
Number of cylinders, z	-	1																			
Volumen desplazado por cilindro, V_cil	cm^3	124,7																			
Thermodynamic cycle per revolution, i	-	0,5																			
Fuel data																					
number of C atoms, alfa	-	1																			
number of H atoms, beta	-	1,861																			
number of O atoms, gamma	-	0,016																			
number of N atoms, delta	-	0																			
molecular weight, Mf	$kg_f/kmol_f$	17,08																			
Air Fuel Ratio stoiciometric, AFR_stq	kg_a/kg_f	14,25																			
lower heating value, LHV	MJ/kg	43,0																			
Measured parameters																					
point	-	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
Throttle opening	grados	59,6	47,0	36,3	59,2	42,5	37,5	31,4	61,6	48,1	37,9	29,8	28,2	25,2	33,0	42,4	57,4	54,4	43,3	27,9	21,2
Engine speed, torque, fuel consumption																					
Engine speed, n	rpm	8.000	8.000	8.000	6.500	6.500	6.500	6.500	5.250	5.250	5.250	5.250	8.000	4.500	4.500	4.500	4.500	4.000	4.000	4.000	4.000
Actual Torque, T_e	Nm																				
Fuel consumption, m_f	kg/h	2,2266	1,7611	1,5167	2,0766	1,9397	1,4879	0,9082	1,2223	1,2085	1,0354	0,7045	0,3800	0,5649	0,7600	0,9425	0,9772	0,6957	0,8666	0,7386	0,5447
	kg/s	0,000619	0,000489	0,000421	0,000577	0,000539	0,000413	0,000252	0,000340	0,000336	0,000288	0,000196	0,000106	0,000157	0,000211	0,000262	0,000271	0,000193	0,000241	0,000205	0,000151
Environmental conditions of the trial cell																					
barometric pressure, p_cell	mbar	1.024	1.024	1.024	1.024	1.023	1.023	1.023	1.023	1.023	1.024	1.024	1.024	1.024	1.024	1.024	1.022	1.022	1.022	1.022	1.022
Temperature, t_cell	°C	28,6	28,6	28,6	28,6	30,0	30,0	30,0	30,0	30,0	27,9	27,9	27,9	27,9	27,9	27,9	27,0	27,0	27,0	27,0	28,0
Humidity relative, phi_rel	%	55	55	55	55	55	55	55	55	55	55	55	55	55	55	55	51	51	51	51	51
Data for obtaining the intake air mass flow																					
Nozzle discharge coefficient	-	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93
Differential pressure, Dp_atm-dep	mbar	0,824	0,596	0,402	0,623	0,467	0,291	0,144	0,224	0,223	0,186	0,103	0,149	0,074	0,169	0,181	0,263	0,178	0,143	0,096	0,063
Intake and exhaust pressure																					
Intake manifold absolute pressure, Dp_atm-adm	mbar	1.008,8	1.002,0	986,7	1.003,7	997,0	981,9	949,1	1.009,0	999,7	988,5	958,5	889,1	958,5	989,5	1.000,7	1.008,7	1.004,6	996,4	983,2	956,3
Exhaust manifold absolute pressure, Dp_esc-atm	mbar	951,64	959,36	970,38	963,17	968,79	976,57	980,62	963,83	959,50	953,15	945,44	974,57	984,00	984,00	980,00	973,45	973,02	980,96	985,60	988,66
Intake temperature	°C	48,95	52,45	53,60	46,33	48,94	47,98	49,43	41,83	44,26	46,45	48,73	49,86	51,37	49,01	46,41	43,05	43,87	46,85	49,30	51,43
Results of the exhaust gases analysis (HC < 100 ppm y NOx < 1000 ppm)																					
Exhuast gases temperature, t_esc	°C	616	615	605	609	592	601	633	592	564	546	549	683	553	545	556	574	555	542	541	585
Vol. Correction of O2 in the dry exhaust gases, O2_D	%	0,10	0,11	0,15	0,24	0,41	0,30	0,13	0,13	0,15	0,17	0,20	0,20	0,19	0,17	0,11	0,17	0,20	0,17	0,16	0,16
Vol. Correction of CO2 in the dry exhaust gases, CO2_D	%	9,17	9,75	9,24	10,58	9,22	10,16	13,29	12,10	10,93	10,71	11,66	12,50	117,70	10,47	10,47	11,86	44,55	10,47	11,04	13,16
Vol. Correction of CO in the dry exhaust gases, CO_D	%	7,63	7,06	6,75	5,03	4,81	3,98	2,34	3,69	5,30	5,20	4,19	2,99	4,07	5,28	5,31	3,60	3,87	5,36	4,87	2,59
Vol. Correction of HC in the dry exhaust gases, HC_D	ppm C6	370	359	273	326	286	330	293	249	304	323	308	205	332	330	326	290	299	370	392	312
Vol. Correction of NOx in the dry exhaust gases, NOx_D	ppm NO+NO2	0	0	0	0	0	0														
Dilution factor, DF		1,10	1,10	1,10	1,10	1,10	1,10	1,10	1,10	1,10	1,10	1,10	1,10	1,10	1,10	1,10	1,10	1,10	1,10	1,10	1,10
measured lambda HORIBA	-	1,02	1,10	1,03	1,05	1,11	1,12	1,11	1,11	1,11	1,03	1,01									1,00
Vol. relation of O2 in the dry exhaust gases, O2_D	%	1,05	1,20	1,62	2,64	4,54	3,27	1,42	1,42	1,66	1,92	2,15	2,20	2,05	1,83	1,25	1,89	2,22	1,85	1,78	1,75
Vol. relation of CO2 in the dry exhaust gases, CO2_D	%	8,34	8,86	8,40	9,62	8,38	9,24	12,08	11,00	9,94	9,74	10,60	11,36	107,00	9,52	9,52	10,78	40,50	9,52	10,04	11,96
Vol. relation of CO in the dry exhaust gases, CO_D	%	6,94	6,42	6,14	4,57	4,37	3,62	2,13	3,35	4,82	4,73	3,81	2,72	3,70	4,80	4,83	3,27	3,52	4,87	4,43	2,35
Vol. relation of HC in the dry exhaust gases, HC_D	ppm C6	336	326	248	296	260	300	266	226	276	294	280	186	302	300	296	264	272	336	356	284
Temperature before catalyst	°C	311,52	295,84	279,53	286,85	266,53	250,30	252,37	253,53	230,11	214,44	205,60	285,12	239,10	245,93	259,94	278,96	261,81	243,86	231,42	239,02
Temperature after catalyst	°C	289,15	274,42	253,14	269,63	253,60	246,69	243,05	249,86	229,96	210,91	193,87	257,24	178,95	194,36	206,12	220,41	205,85	192,33	180,87	177,96

Calculated parameters																					
point	-	1	2	3	4	5	6														
engine speedr, n	rpm	8.000	8.000	8.000	6.500	6.500	6.500	6.500	5.250	5.250	5.250	5.250	8.000	4.500	4.500	4.500	4.500	4.000	4.000	4.000	4.000
actual Torque, actual Power, mean effective pressure, specific fuel consumption																					
actual Torque, M_e	Nm	6,79	5,65	4,52	8,00	6,79	5,65	4,52	8,00	6,79	5,65	4,52	3,39	4,52	5,65	6,79	8,00	8,00	6,79	5,65	4,52
actual Power, W_e	kW	5,7	4,7	3,8	5,4	4,6	3,8	3,1	4,4	3,7	3,1	2,5	2,8	2,1	2,7	3,2	3,8	3,4	2,8	2,4	1,9
mean effective pressure, pme	kPa	683,9	569,8	455,6	806,2	684,2	569,4	455,5	806,2	684,2	569,4	455,5	341,4	455,5	569,4	684,2	806,2	806,2	684,2	569,4	455,5
specific fuel consumption, g_ef	g/kWh	391,6	371,8	400,4	381,3	419,7	386,9	295,2	277,9	323,7	333,3	283,5	133,9	265,2	285,4	294,6	259,2	207,6	304,7	312,1	287,7
Power and Torque corrected by environmental conditions																					
Saturated steam pressure, p_s,ens (kPa)	mbar	39,2	39,2	39,2	39,2	42,5	42,5	42,5	42,5	42,5	37,6	37,6	37,6	37,6	37,6	37,6	35,7	35,7	35,7	35,7	37,8
Dry air pressure, p_ens-phi*p_s,ens (kPa)	mbar	1002,5	1002,5	1002,5	1002,5	999,6	999,6	999,6	999,6	999,6	1003,3	1003,3	1003,3	1003,3	1003,3	1003,3	1003,8	1003,8	1003,8	1003,8	1002,7
Factor de corrección, alfa-a	-	0,99	0,99	0,99	0,99	1,00	1,00	1,00	1,00	1,00	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99
Power corrtd, W_e,corr	kW	5,6	4,7	3,8	5,4	4,6	3,8	3,1	4,4	3,7	3,1	2,5	2,8	2,1	2,6	3,2	3,7	3,3	2,8	2,3	1,9
Torque corrct, M_e,corr	Nm	6,7	5,6	4,5	7,9	6,8	5,6	4,5	8,0	6,8	5,6	4,5	3,4	4,5	5,6	6,7	7,9	7,9	6,7	5,6	4,5
Relative dosage, volumetric efficiency and emissions (starting from measurements of air flow rate and fue consumption)																					
AFR, lambda and relative dosage																					
number of C atoms, alfa	-	1,000	1,000	1,000	1,000	1,000	1,000	1,000	1,000	1,000	1,000	1,000	1,000	1,000	1,000	1,000	1,000	1,000	1,000	1,000	1,000
number of H atoms, beta	-	1,861	1,861	1,861	1,861	1,861	1,861	1,861	1,861	1,861	1,861	1,861	1,861	1,861	1,861	1,861	1,861	1,861	1,861	1,861	1,861
number of O atoms, gamma	-	0,016	0,016	0,016	0,016	0,016	0,016	0,016	0,016	0,016	0,016	0,016	0,016	0,016	0,016	0,016	0,016	0,016	0,016	0,016	0,016
number of N atoms, delta	-	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000
Air Fuel Ratio stoiciometric, AFR_stq	kg_a/kg_f	14,25	14,25	14,25	14,25	14,25	14,25	14,25	14,25	14,25	14,25	14,25	14,25	14,25	14,25	14,25	14,25	14,25	14,25	14,25	14,25
air density of the trial cell, rho_adm	kg/m^3	1,182	1,182	1,182	1,182	1,176	1,176	1,176	1,176	1,176	1,185	1,185	1,185	1,185	1,185	1,185	1,186	1,186	1,186	1,186	1,182
air mass flow rate, m_a	kg/s	0,0068	0,0057	0,0047	0,0059	0,0051	0,0040	0,0028	0,0035	0,0035	0,0032	0,0024	0,0029	0,0020	0,0031	0,0032	0,0038	0,0031	0,0028	0,0023	0,0019
air fuel ratio, AFR	kg_a/kg_f	10,93	11,75	11,21	10,19	9,42	9,69	11,17	10,35	10,45	11,18	12,23	27,27	12,93	14,52	12,12	14,10	16,29	11,72	11,27	12,36
air excess, lambda	-	0,77	0,82	0,79	0,71	0,66	0,68	0,78	0,73	0,73	0,78	0,86	1,91	0,91	1,02	0,85	0,99	1,14	0,82	0,79	0,87
relative dosage, 1/lambda	-	1,30	1,21	1,27	1,40	1,51	1,47	1,28	1,38	1,36	1,27	1,17	0,52	1,10	0,98	1,18	1,01	0,87	1,22	1,26	1,15
Volumetric Efficiency																					
Volumetric Efficiency, nhu_v,ens	-	0,69	0,58	0,48	0,74	0,64	0,50	0,35	0,55	0,55	0,50	0,37	0,29	0,37	0,55	0,57	0,69	0,64	0,57	0,47	0,38
air density in the intake manifold, rho_adm	kg/m^3	1,091	1,072	1,052	1,095	1,079	1,065	1,025	1,116	1,097	1,078	1,038	0,959	1,029	1,070	1,091	1,111	1,104	1,085	1,062	1,027
Volumetric Efficiency, nhu_v,adm	-	0,75	0,65	0,54	0,79	0,70	0,56	0,41	0,58	0,59	0,55	0,42	0,36	0,42	0,61	0,62	0,74	0,69	0,63	0,52	0,44
Emissions of polluting gases																					
exhuast gases mass flow rate, m_esc,w = m_f+m_a	kg/s	0,0074	0,0062	0,0051	0,0065	0,0056	0,0044	0,0031	0,0039	0,0038	0,0035	0,0026	0,0030	0,0022	0,0033	0,0034	0,0041	0,0033	0,0031	0,0025	0,0020
bs_CO	g/kWh	330,3	310,2	305,8	198,6	194,0	152,0	77,7	107,3	181,4	196,0	145,9	105,3	139,5	217,2	190,5	130,9	129,3	193,0	173,4	92,2
bs_HC1	g/kWh	4,8	6,0	4,7	4,9	4,4	4,8	3,7	2,8	3,9	4,6	4,1	2,7	4,3	5,2	4,4	4,0	3,8	5,1	5,3	4,2
bs_NOx	g/kWh	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0
bs_CO2	g/kWh	623,6	672,6	657,4	656,9	584,4	609,8	692,7	553,7	587,8	634,1	637,6	691,1	6340,4	676,9	589,9	678,0	2337,8	592,7	617,4	736,9

Relative dosage, volumetric efficiency and emissions (starting from the gases analysis and fuel consumption)																					
AFR, lambda and relative dosage																					
AFR with Brettschneider method																					
K balance constant of the reaction H2O gases	-	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5
carbon atoms in the selected HC molecule	-	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0
(beta/alpha)/4	-	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653
(gamma/alpha)/2	-	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040
1+((beta/alpha)/4)-((gamma/alpha)/2)	-	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613
Lambda Brettschneider	-	0,78	0,80	0,80	0,85	0,85	0,87	0,93	0,89	0,85	0,85	0,88	0,92	0,98	0,84	0,84	0,89	0,96	0,84	0,85	0,92
relative dosage Brettschneider	-	1,29	1,26	1,25	1,17	1,17	1,14	1,08	1,12	1,18	1,18	1,14	1,09	1,02	1,19	1,19	1,12	1,04	1,19	1,17	1,08
AFR Brettschneider	kg_a/kg_f	11,08	11,34	11,38	12,14	12,13	12,45	13,24	12,71	12,06	12,07	12,52	13,04	14,04	12,00	11,96	12,72	13,74	11,96	12,17	13,15
Volumetric Efficiency																					
air fuel rate from lambda, m_a	kg/s	0,0069	0,0055	0,0048	0,0070	0,0065	0,0051	0,0033	0,0043	0,0040	0,0035	0,0024	0,0014	0,0022	0,0025	0,0031	0,0035	0,0027	0,0029	0,0025	0,0020
	kg/h	24,6602	19,9769	17,2640	25,2054	23,5311	18,5299	12,0209	15,5362	14,5784	12,4926	8,8168	4,9562	7,9296	9,1191	11,2713	12,4307	9,5606	10,3642	8,9896	7,1642
fuel consumption computed	kg/s	0,00061	0,00051	0,00041	0,00048	0,00042	0,00032	0,00021	0,00028	0,00029	0,00027	0,00019	0,00022	0,00014	0,00026	0,00027	0,00030	0,00023	0,00024	0,00019	0,00014
	kg/h	2,197	1,825	1,494	1,743	1,506	1,158	0,767	0,996	1,047	0,960	0,689	0,795	0,520	0,920	0,955	1,083	0,825	0,849	0,684	0,512
Volumetric Efficiency, nhu_v,ens	-	0,70	0,56	0,49	0,88	0,82	0,65	0,42	0,67	0,63	0,54	0,38	0,14	0,40	0,46	0,56	0,62	0,54	0,58	0,51	0,40
Volumetric Efficiency, nhu_v,adm for air density in intake manifold	-	0,76	0,62	0,55	0,95	0,90	0,72	0,48	0,71	0,68	0,59	0,43	0,17	0,46	0,51	0,61	0,66	0,58	0,64	0,57	0,47
Composition of exhaust gases and emissions																					
[H2O] wet	%	11,2	11,5	11,0	11,3	10,2	10,6	12,2	11,9	11,7	11,5	11,8	11,9	52,9	11,4	11,4	11,7	30,5	11,4	11,6	12,2
[O2] wet	%	0,1	0,1	0,1	0,2	0,4	0,3	0,1	0,1	0,1	0,2	0,2	0,2	0,1	0,1	0,1	0,2	0,1	0,1	0,1	0,1
[CO2] wet	%	8,1	8,6	8,2	9,4	8,3	9,1	11,7	10,7	9,7	9,5	10,3	11,0	55,5	9,3	9,3	10,5	30,9	9,3	9,8	11,6
[CO] wet	%	6,8	6,3	6,0	4,5	4,3	3,6	2,1	3,2	4,7	4,6	3,7	2,6	1,9	4,7	4,7	3,2	2,7	4,7	4,3	2,3
[HC1] wet	%	0,2	0,2	0,1	0,2	0,2	0,2	0,2	0,1	0,2	0,2	0,2	0,1	0,1	0,2	0,2	0,2	0,1	0,2	0,2	0,2
[NO2] wet	%	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0
[H2] wet	%	2,7	2,4	2,3	1,5	1,5	1,2	0,6	1,0	1,6	1,6	1,2	0,8	0,5	1,6	1,6	1,0	0,8	1,7	1,5	0,7
B1	-	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457
B2	-	6,614	6,636	6,951	7,137	7,844	7,797	7,203	7,124	6,899	7,014	7,070	7,270	1,740	7,073	7,060	7,245	2,962	7,031	7,006	7,149
B3	-	1,288	1,257	1,253	1,175	1,175	1,145	1,077	1,122	1,182	1,182	1,139	1,093	1,015	1,189	1,192	1,121	1,037	1,192	1,172	1,084
[N2] wet	%	64,5	65,9	63,1	65,6	59,6	61,6	70,9	68,8	67,4	66,3	68,3	69,2	311,2	65,4	65,3	67,7	179,0	65,6	67,0	71,0
Summation [i]	%	93,6	94,9	90,9	92,7	84,5	86,5	97,6	95,9	95,4	93,8	95,6	95,8	422,2	92,7	92,6	94,4	244,1	93,0	94,5	97,9
Molecular weight of the exhaust gases, M_esc,w	kg/kmol	25,83	26,32	25,18	26,04	23,67	24,42	27,94	27,19	26,77	26,34	27,03	27,27	121,78	25,99	25,96	26,77	70,17	26,08	26,61	27,99
mass flow rate of damp fumes m_esc,w	g/h	26887	21738	18781	27282	25471	20018	12929	16758	15787	13528	9521	5336	8495	9879	12214	13408	10256	11231	9728	7709
mass flow rate of damp fumes m_esc,w	kg/s	0,0075	0,0060	0,0052	0,0076	0,0071	0,0056	0,0036	0,0047	0,0044	0,0038	0,0026	0,0015	0,0024	0,0027	0,0034	0,0037	0,0028	0,0031	0,0027	0,0021
Mol in the exhaust gases, n_esc,w = m_esc,w/M_esc,w	mol/h	1,04E+03	8,26E+02	7,46E+02	1,05E+03	1,08E+03	8,20E+02	4,63E+02	6,16E+02	5,90E+02	5,14E+02	3,52E+02	1,96E+02	6,98E+01	3,80E+02	4,71E+02	5,01E+02	1,46E+02	4,31E+02	3,66E+02	2,75E+02
bs CO	g/kWh	347,5	305,4	331,8	240,2	281,5	212,6	86,7	127,4	207,1	213,2	146,8	50,9	17,6	187,1	194,0	118,2	32,9	201,3	186,3	92,5
bs HC1	g/kWh	30,6	28,2	24,4	28,3	30,4	32,0	19,7	15,6	21,6	24,1	19,6	6,3	2,6	21,3	21,6	17,3	4,6	25,2	27,2	20,3
bs NOx	g/kWh	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0
bs CO2	g/kWh	656,1	662,2	713,1	794,5	848,0	852,6	772,6	657,4	671,2	689,8	641,7	334,0	799,4	583,1	600,6	612,3	594,0	618,2	663,4	739,6

Composition and specific heat of exhaust gases																					
Composition of exhaust gases																					
y O2 wet	-	0,001	0,001	0,001	0,002	0,004	0,003	0,001	0,001	0,001	0,002	0,002	0,002	0,001	0,001	0,001	0,002	0,001	0,001	0,001	0,001
y CO2 wet	-	0,081	0,086	0,082	0,094	0,083	0,091	0,117	0,107	0,097	0,095	0,103	0,110	0,555	0,093	0,093	0,105	0,309	0,093	0,098	0,116
y CO wet	-	0,068	0,063	0,060	0,045	0,043	0,036	0,021	0,032	0,047	0,046	0,037	0,026	0,019	0,047	0,047	0,032	0,027	0,047	0,043	0,023
y H2 wet	-	0,027	0,024	0,023	0,015	0,015	0,012	0,006	0,010	0,016	0,016	0,012	0,008	0,005	0,016	0,016	0,010	0,008	0,017	0,015	0,007
y H2O wet	-	0,112	0,115	0,110	0,113	0,102	0,106	0,122	0,119	0,117	0,115	0,118	0,119	0,529	0,114	0,114	0,117	0,305	0,114	0,116	0,122
y N2 wet	-	0,645	0,659	0,631	0,656	0,596	0,616	0,709	0,688	0,674	0,663	0,683	0,692	3,112	0,654	0,653	0,677	1,790	0,656	0,670	0,710
sum yi	-	0,934	0,947	0,907	0,925	0,843	0,863	0,975	0,958	0,952	0,937	0,954	0,957	4,221	0,925	0,924	0,942	2,440	0,928	0,943	0,978
Specific heat in the exhaust gases																					
Differential temperature gases atmosphere, tesc-ta	°C	588	586	577	581	562	571	603	562	534	518	521	655	525	517	528	547	528	515	514	557
Average temperature of gases and environment, (tesc+ta)/2	°C	322	322	317	319	311	315	331	311	297	287	289	356	290	286	292	300	291	285	284	307
cp_esc,w	kJ/kg K	1,2	1,2	1,2	1,2	1,2	1,2	2,2	3,2	4,2	5,2	6,2	7,2	8,2	9,2	10,2	11,2	12,2	13,2	14,2	15,2

Energy balance																					
Thermal Power of the combustion, m_f H_f	kW	26,60	21,04	18,12	24,80	23,17	17,77	10,85	14,60	14,43	12,37	8,41	4,54	6,75	9,08	11,26	11,67	8,31	10,35	8,82	6,51
	%	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00
Actual engine Power, We	kW	5,69	4,74	3,79	5,45	4,62	3,85	3,08	4,40	3,73	3,11	2,48	2,84	2,13	2,66	3,20	3,77	3,35	2,84	2,37	1,89
	%	0,21	0,23	0,21	0,22	0,20	0,22	0,28	0,30	0,26	0,25	0,30	0,63	0,32	0,29	0,28	0,32	0,40	0,27	0,27	0,29
Sensible heat of exhaust gases, H_esc,w	kW	5,20	4,39	3,56	4,50	3,78	3,03	4,07	6,93	8,62	9,44	8,37	14,08	9,41	15,58	18,48	25,09	21,54	20,83	18,36	17,11
	%	0,20	0,21	0,20	0,18	0,16	0,17	0,38	0,47	0,60	0,76	0,99	3,10	1,39	1,72	1,64	2,15	2,59	2,01	2,08	2,63
Energy due to the unburned content, H_inq	kW	7,75	5,62	4,88	4,95	4,95	3,07	0,97	2,08	2,93	2,51	1,36	0,53	0,13	1,89	2,36	1,65	0,40	2,18	1,66	0,64
	%	0,29	0,27	0,27	0,20	0,21	0,17	0,09	0,14	0,20	0,20	0,16	0,12	0,02	0,21	0,21	0,14	0,05	0,21	0,19	0,10
Heat dissipated in cooling system, Q_ref+Q_lub	kW	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
	%	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
Other energy flows, E_otras	kW	7,96	6,29	5,89	9,91	9,81	7,83	2,72	1,19	-0,85	-2,69	-3,80	-12,91	-4,92	-11,06	-12,78	-18,84	-16,98	-15,50	-13,57	-13,14
	%	0,30	0,30	0,32	0,40	0,42	0,44	0,25	0,08	-0,06	-0,22	-0,45	-2,84	-0,73	-1,22	-1,14	-1,61	-2,04	-1,50	-1,54	-2,02

1.3 170 cm^3 in full load (nozzle-deposit system)

Engine data																					
Number of cylinders, z	-	1																			
Volumen desplazado por cilindro, V_cil	cm^3	171,26																			
Thermodynamic cycle per revolution, i	-	0,5																			
Fuel data																					
number of C atoms, alfa	-	1																			
number of H atoms, beta	-	1,861																			
number of O atoms, gamma	-	0,016																			
number of N atoms, delta	-	0																			
molecular weight, Mf	kg$_f$/kmol$_f$	17,08																			
Air Fuel Ratio stoiciometric, AFR_stq	kg$_a$/kg$_f$	14,25																			
lower heating value, LHV	MJ/kg	43,0																			
Measured parameters																					
point	-	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
Throttle opening	grados	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100
Engine speed, torque, fuel consumption																					
Engine speed, n	rpm	2998	3497	3747	3996	4245	4495	4744	4994	5243	5493	6001	6500	6999	7498	7996	8495	8993	9242	9492	9741
Actual Torque, T_e	Nm	12,95	13,38	13,60	13,44	13,53	13,45	13,49	13,41	13,17	12,95	12,74	12,99	13,27	12,97	12,49	11,57	10,44	9,64	9,20	8,88
Fuel consumption, m_f	kg/h	1,3170	1,6810	1,3240	0,9834	1,6940	1,3670	1,3060	1,6959	1,9580	2,3660	2,1790	2,3913	2,459	2,7740	2,3582	2,5711	2,861	2,885	2,6976	3,4532
	kg/s	0,0004	0,0005	0,0004	0,0003	0,0005	0,0004	0,0004	0,0005	0,0005	0,0007	0,0006	0,0007	0,0007	0,0008	0,0007	0,0007	0,0008	0,0008	0,0007	0,0010
Environmental conditions of the trial cell																					
barometric pressure, p_cell	mbar	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1022	1022	1022	1022	1022	1022
Temperature, t_cell	°C	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28	28
Humidity relative, phi_rel	%	47	47	47	47	47	47	47	47	47	47	47	47	47	47	47	47	47	47	47	47
Data for obtatining the intake air mass flow																					
Nozzle discharge coefficient	-	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93
Differential pressure, Dp_atm-dep	mbar	0,42	0,60	0,67	0,75	0,78	0,86	0,93	1,00	1,08	1,15	1,29	1,57	1,91	2,16	2,31	2,54	2,70	2,42	2,32	2,34
Presión de admisión y contrapresión de escape		0,00	0,00	0,00	-1,07	-2,13	-3,20	-4,40	-5,60	-6,80	-7,46	-8,80	-10,84	-13,95	-15,80	-16,95	-16,43	-18,20	-18,72	-19,25	-20,50
Pressure adm, Dp_atm-adm	mbar	-16	-12,52	-10,52	-8	-5	-4	-4	-1	-9	-11	-6	-2	-2	-4	-0,94	-4	-1,85	-4,96	-5,48	-4,65
p_intake		1.022	1.022	1.022	1.023	1.024	1.025	1.026	1.028	1.029	1.029	1.031	1.033	1.036	1.038	1.039	1.038	1.040	1.041	1.041	1.043
Pressure escape, Dp_esc-atm	mbar	30,52	61,12	63,23	64,04	76,75	71,08	72,00	71,35	76,07	77,73	80,07	87,20	102,84	108,12	100,11	114,23	113,11	110,1	109,88	100,8
Temperature adm	°C	31,26	32,42	32,56	32,68	32,93	33,32	33,85	34,40	34,95	35,69	35,70	35,82	36,19	36,50	35,46	37	36,66	38,01	39,27	38,37
lambda MOTEC	-	0,95	0,95	0,97	1,03	0,97	0,97	0,95	0,92	0,94	0,93	0,94	0,93	0,91	0,9	0,93	0,91	0,92	0,91	0,91	0,91
Results of the exhaust gases analysis (HC < 100 ppm y NOx < 1000 ppm)																					
Exhuast gases temperature, t_esc	°C	396	427,23	441,17	451	465	473	480	483	494	502	516	531	541	550	555,39	571	566,92	573,46	577,89	573,41
Vol. Correction of O2 in the dry exhaust gases, O2_D	%	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
Vol. Correction of CO2 in the dry exhaust gases, CO2_D	%	14,08	14,11	14,42	14,53	14,38	14,22	13,85	13,31	13,55	13,56	13,59	13,40	13,16	12,91	13,63	13,20	13,43	13,29	13,34	13,31
Vol. Correction of CO in the dry exhaust gases, CO_D	%	1,67	1,72	1,26	0,37	1,21	1,56	2,10	2,91	2,49	2,50	2,42	2,73	3,04	3,36	2,45	3,03	2,70	2,93	2,82	2,84
Vol. Correction of HC in the dry exhaust gases, HC_D	ppm C6	798	329	274	239	222	203	188	186	180	171	173	286	301	269	230	164	143	140	136	194
Vol. relation of NOx in the dry exhaust gases, NOx_D	ppm NO+NO2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Dilution factor, DF		1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00
measured lambda HORIBA	-	0,98	0,98	0,99	1,05	0,99	0,99	0,97	0,94	0,96	0,96	0,96	0,95	0,94	0,93	0,95	0,93	0,95	0,94	0,94	0,93
Vol. relation of O2 in the dry exhaust gases, O2_D	%	0,96	0,45	0,5	1,12	0,52	0,46	0,37	0,33	0,33	0,34	0,35	0,34	0,30	0,29	0,42	0,30	0,62	0,29	0,28	0,3
Vol. relation of CO2 in the dry exhaust gases, CO2_D	%	14,08	14,11	14,42	14,53	14,38	14,22	13,85	13,31	13,55	13,56	13,59	13,40	13,16	12,91	13,63	13,20	13,43	13,29	13,34	13,31
Vol. relation of CO in the dry exhaust gases, CO_D	%	1,67	1,72	1,26	0,37	1,21	1,56	2,10	2,91	2,49	2,50	2,42	2,73	3,04	3,36	2,45	3,03	2,7	2,93	2,82	2,84
Vol. relation of HC in the dry exhaust gases, HC_D	ppm C6	798	329,44	274,03	239	222	203	188	186	180	171	173	286	301	269	229,7	163,66	142,74	140,24	135,63	194,09
Temperature before catalyst	°C	207,07	198,96	208,24	216,47	227,18	236,62	243,68	248,39	252,72	258,48	267,03	286,20	301,07	313,15	276,2	327,90	280,26	305,06	318,79	286,86
Temperature after catalyst	°C	205,43	170,98	181,41	191,53	202,28	211,46	216,40	219,25	223,60	227,42	235,95	252,18	264,99	280,34	221,85	281	216,9	258,7	274,87	216,91

Calculated parameters																					
point	-	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
engine speedr, n	rpm	2998	3497	3747	3996	4245	4495	4744	4994	5243	5493	6001	6500	6999	7498	7996	8495	8993	9242	9492	9741
actual Torque, actual Power, mean effective pressure, specific fuel consumption																					
actual Torque, M_e	Nm	12,95	13,38	13,60	13,44	13,53	13,45	13,49	13,41	13,17	12,95	12,74	12,99	13,27	12,97	12,49	11,57	10,44	9,64	9,20	8,88
actual Power, W_e	kW	4,07	4,90	5,34	5,62	6,01	6,33	6,70	7,01	7,23	7,45	8,01	8,84	9,72	10,18	10,45	10,30	9,83	9,33	9,14	9,05
mean effective pressure, pme	kPa	950,4	981,8	997,8	986,2	992,7	986,6	990,1	983,8	966,2	949,9	935,2	953,4	973,4	951,6	916,1	849,3	766,2	707,2	675,0	651,2
specific fuel consumption, g_ef	g/kWh	323,9	343,0	248,2	174,9	281,6	216,0	194,8	241,9	270,8	317,8	272,1	270,4	252,9	272,4	225,6	249,7	290,9	309,3	295,0	381,4
Power and Torque corrected by environmental conditions																					
Saturated steam pressure, p_s,ens (kPa)	mbar	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8
Dry air pressure, p_ens-phi*p_s,ens (kPa)	mbar	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2
Factor de corrección, alfa-a	-	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99
Power corrtd, W_e,corr	kW	4,02	4,85	5,28	5,56	5,95	6,26	6,63	6,93	7,15	7,36	7,92	8,75	9,61	10,07	10,34	10,18	9,73	9,23	9,04	8,95
Torque corrct, M_e,corr	Nm	12,81	13,23	13,45	13,29	13,38	13,30	13,34	13,26	13,02	12,80	12,60	12,85	13,12	12,83	12,35	11,45	10,33	9,53	9,10	8,78
Relative dosage, volumetric efficiency and emissions (starting from measurements of air flow rate and fue consumption)																					
AFR, lambda and relative dosage																					
number of C atoms, alfa	-	1,000	1,000	1,000	1,000	1,000	1,000	1,000	1,000	1,000	1,000	1,000	1,000	1,000	1,000	1,000	1,000	1,000	1,000	1,000	1,000
number of H atoms, beta	-	1,861	1,861	1,861	1,861	1,861	1,861	1,861	1,861	1,861	1,861	1,861	1,861	1,861	1,861	1,861	1,861	1,861	1,861	1,861	1,861
number of O atoms, gamma	-	0,016	0,016	0,016	0,016	0,016	0,016	0,016	0,016	0,016	0,016	0,016	0,016	0,016	0,016	0,016	0,016	0,016	0,016	0,016	0,016
number of N atoms, delta	-	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000
Air Fuel Ratio stoiciometric, AFR_stq	kg_a/kg_f	14,25	14,25	14,25	14,25	14,25	14,25	14,25	14,25	14,25	14,25	14,25	14,25	14,25	14,25	14,25	14,25	14,25	14,25	14,25	14,25
air density of the trial cell, rho_adm	kg/m^3	1,182	1,182	1,182	1,182	1,182	1,182	1,182	1,182	1,182	1,182	1,182	1,182	1,182	1,182	1,182	1,182	1,182	1,182	1,182	1,182
air mass flow rate, m_a	kg/s	0,0048	0,0057	0,0061	0,0064	0,0066	0,0069	0,0072	0,0075	0,0077	0,0080	0,0085	0,0093	0,0103	0,0109	0,0113	0,0118	0,0122	0,0116	0,0113	0,0114
air fuel ratio, AFR	kg_a/kg_f	13,13	12,31	16,58	23,60	13,98	18,18	19,82	15,84	14,25	12,15	14,00	14,03	15,05	14,19	17,26	16,59	15,39	14,46	15,11	11,87
air excess, lambda	-	0,92	0,86	1,16	1,66	0,98	1,28	1,39	1,11	1,00	0,85	0,98	0,98	1,06	1,00	1,21	1,16	1,08	1,01	1,06	0,83
relative dosage, 1/lambda	-	1,09	1,16	0,86	0,60	1,02	0,78	0,72	0,90	1,00	1,17	1,02	1,02	0,95	1,00	0,83	0,86	0,93	0,99	0,94	1,20
Volumetric Efficiency																					
Volumetric Efficiency, nhu_v,ens	-	0,95	0,97	0,96	0,96	0,92	0,91	0,90	0,89	0,88	0,86	0,84	0,85	0,87	0,86	0,84	0,83	0,81	0,74	0,71	0,69
air density in the intake manifold, rho_adm	kg/m^3	1,152	1,151	1,153	1,155	1,157	1,157	1,155	1,156	1,145	1,141	1,147	1,150	1,149	1,146	1,153	1,144	1,147	1,139	1,134	1,138
Volumetric Efficiency, nhu_v,adm	-	0,97	1,00	0,99	0,98	0,94	0,93	0,92	0,91	0,90	0,89	0,86	0,87	0,90	0,89	0,86	0,85	0,83	0,77	0,74	0,72
Emissions of polluting gases																					
exhuast gases mass flow rate, m_esc,w = m_f+m_a	kg/s	0,0052	0,0062	0,0065	0,0067	0,0071	0,0073	0,0076	0,0079	0,0083	0,0086	0,0091	0,0100	0,0110	0,0117	0,0120	0,0126	0,0130	0,0124	0,0121	0,0123
bs_CO	g/kWh	70,9	72,8	51,1	14,8	47,4	60,0	79,1	110,1	95,4	96,9	91,6	103,0	114,6	129,0	93,7	123,6	119,6	130,0	124,5	129,3
bs_HC1	g/kWh	10,3	5,3	4,2	3,6	3,3	3,0	2,7	2,7	2,6	2,5	2,5	4,1	4,3	3,9	3,3	2,5	2,4	2,4	2,3	3,4
bs_NOx	g/kWh	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0
bs_CO2	g/kWh	939,0	938,7	918,0	912,6	884,9	859,5	819,8	791,0	815,9	825,9	808,5	794,3	779,3	778,9	819,2	845,9	934,3	926,4	925,0	952,2
Relative dosage, volumetric efficiency and emissions (starting from the gases analysis and fuel consumption)																					
AFR, lambda and relative dosage																					
AFR with Brettschneider method																					
K balance constant of the reaction H2O gases	-	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5
carbon atoms in the selected HC molecule	-	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0
(beta/alpha)/4	-	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653
(gamma/alpha)/2	-	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040
1+((beta/alpha)/4)-((gamma/alpha)/2)	-	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613
Lambda Brettschneider	-	0,93	0,94	0,95	0,98	0,96	0,95	0,94	0,91	0,92	0,92	0,93	0,91	0,91	0,90	0,92	0,91	0,92	0,91	0,92	0,91
relative dosage Brettschneider	-	1,08	1,06	1,05	1,02	1,04	1,05	1,07	1,09	1,08	1,08	1,08	1,09	1,10	1,11	1,08	1,10	1,09	1,09	1,09	1,09
AFR Brettschneider	kg_a/kg_f	13,19	13,40	13,61	13,96	13,65	13,53	13,33	13,02	13,18	13,18	13,21	13,04	12,91	12,80	13,18	12,98	13,12	13,04	13,08	13,04
Fuel consumption, m_f	kg/h	1,31	1,54	1,61	1,66	1,74	1,84	1,94	2,06	2,12	2,18	2,31	2,57	2,87	3,07	3,09	3,29	3,36	3,20	3,12	3,14
	kg/s	0,00036	0,00043	0,00045	0,00046	0,00048	0,00051	0,00054	0,00057	0,00059	0,00061	0,00064	0,00071	0,00080	0,00085	0,00086	0,00091	0,00093	0,00089	0,00087	0,00087
specific fuel consumptio, g_ef	g/kWh	322,4	315,0	302,4	295,5	288,5	290,2	289,7	294,3	292,7	292,9	288,3	291,0	294,8	301,8	295,5	319,1	341,4	343,0	341,0	347,2
Rendimiento volumétrico																					
Air consumption from lambda, m_a	kg/s	0,0048	0,0063	0,0050	0,0038	0,0064	0,0051	0,0048	0,0061	0,0072	0,0087	0,0080	0,0087	0,0088	0,0099	0,0086	0,0093	0,0104	0,0104	0,0098	0,0125
	kg/h	17,3660	22,5300	18,0165	13,7325	23,1253	18,4928	17,4064	22,0826	25,8079	31,1887	28,7847	31,1806	31,7523	35,5195	31,0694	33,3841	37,5369	37,6066	35,2774	45,0308
Volumetric Efficiency, nhu_v,ens	-	0,95	1,06	0,79	0,57	0,90	0,68	0,60	0,73	0,81	0,93	0,79	0,79	0,75	0,78	0,64	0,65	0,69	0,67	0,61	0,76
Volumetric Efficiency, nhu_v,adm for air density in intake manifold	-	0,98	1,09	0,81	0,58	0,92	0,69	0,62	0,74	0,84	0,97	0,81	0,81	0,77	0,80	0,66	0,67	0,71	0,70	0,64	0,79

Composition of exhaust gases and emissions																					
[H2O] wet	%	12,4	12,5	12,5	12,1	12,4	12,5	12,5	12,4	12,4	12,4	12,4	12,4	12,4	12,4	12,5	12,4	12,4	12,4	12,4	12,4
[O2] wet	%	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0
[CO2] wet	%	12,3	12,4	12,6	12,8	12,6	12,4	12,1	11,7	11,9	11,9	11,9	11,7	11,5	11,3	11,9	11,6	11,8	11,6	11,7	11,7
[CO] wet	%	1,5	1,5	1,1	0,3	1,1	1,4	1,8	2,5	2,2	2,2	2,1	2,4	2,7	2,9	2,1	2,7	2,4	2,6	2,5	2,5
[HC1] wet	%	0,4	0,2	0,1	0,1	0,1	0,1	0,1	0,1	0,1	0,1	0,1	0,2	0,2	0,1	0,1	0,1	0,1	0,1	0,1	0,1
[NO2] wet	%	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0
[H2] wet	%	0,4	0,4	0,3	0,1	0,3	0,4	0,5	0,8	0,7	0,7	0,6	0,7	0,8	0,9	0,6	0,8	0,7	0,8	0,8	0,8
B1	-	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457
B2	-	7,035	7,127	7,210	7,562	7,261	7,184	7,111	6,993	7,071	7,065	7,086	7,005	6,968	6,943	7,044	6,992	7,042	7,004	7,030	7,018
B3	-	1,081	1,064	1,048	1,021	1,044	1,054	1,070	1,095	1,082	1,081	1,079	1,093	1,104	1,113	1,082	1,098	1,087	1,094	1,090	1,093
[N2] wet	%	72,3	72,5	72,8	71,2	72,5	72,6	72,3	71,8	71,9	72,0	71,9	71,8	71,5	71,1	72,1	71,6	71,9	71,8	71,7	71,7
Summation [i]	%	99,3	99,5	99,4	96,6	99,0	99,4	99,3	99,3	99,1	99,2	99,1	99,2	99,0	98,8	99,4	99,1	99,2	99,3	99,1	99,1
Molecular weight of the exhaust gases, M_esc,w	kg/kmol	28,69	28,58	28,63	27,95	28,49	28,55	28,44	28,30	28,30	28,33	28,30	28,33	28,22	28,09	28,42	28,22	28,28	28,26	28,24	28,24
mass flow rate of damp fumes m_esc,w	g/h	18683	24211	19340	14716	24819	19860	18712	23778	27766	33555	30964	33572	34211	38294	33428	35955	40398	40492	37975	48484
mass flow rate of damp fumes m_esc,w	kg/s	0,0052	0,0067	0,0054	0,0041	0,0069	0,0055	0,0052	0,0066	0,0077	0,0093	0,0086	0,0093	0,0095	0,0106	0,0093	0,0100	0,0112	0,0112	0,0105	0,0135
Mol in the exhaust gases, n_esc,w = m_esc,w/M_esc,w	mol/h	6,51E+02	8,47E+02	6,76E+02	5,26E+02	8,71E+02	6,96E+02	6,58E+02	8,40E+02	9,81E+02	1,18E+03	1,09E+03	1,19E+03	1,21E+03	1,36E+03	1,18E+03	1,27E+03	1,43E+03	1,43E+03	1,34E+03	1,72E+03
bs CO	g/kWh	65,6	72,9	39,1	8,5	43,0	42,0	50,5	85,5	82,9	97,6	81,1	89,7	93,0	110,4	67,6	92,0	96,2	110,4	101,7	132,1
bs HC1	g/kWh	57,0	25,4	15,5	10,0	14,3	10,0	8,2	10,0	10,9	12,2	10,6	17,1	16,7	16,1	11,5	9,0	9,2	9,6	8,9	16,4
bs NOx	g/kWh	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0
bs CO2	g/kWh	869,2	939,9	703,4	526,1	802,7	602,1	523,7	614,6	708,6	831,4	715,5	692,1	632,6	666,7	590,8	629,7	751,7	786,5	756,1	973,0
Composition and specific heat of exhaust gases																					
Composition of exhaust gases																					
y O2 wet	-	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000
y CO2 wet	-	0,123	0,124	0,126	0,128	0,126	0,124	0,121	0,117	0,119	0,119	0,119	0,117	0,115	0,113	0,119	0,116	0,118	0,116	0,117	0,117
y CO wet	-	0,015	0,015	0,011	0,003	0,011	0,014	0,018	0,025	0,022	0,022	0,021	0,024	0,027	0,029	0,021	0,027	0,024	0,026	0,025	0,025
y H2 wet	-	0,004	0,004	0,003	0,001	0,003	0,004	0,005	0,008	0,007	0,007	0,006	0,007	0,008	0,009	0,006	0,008	0,007	0,008	0,008	0,008
y H2O wet	-	0,124	0,125	0,125	0,121	0,124	0,125	0,125	0,124	0,124	0,124	0,124	0,124	0,124	0,124	0,125	0,124	0,124	0,124	0,124	0,124
y N2 wet	-	0,723	0,725	0,728	0,712	0,725	0,726	0,723	0,718	0,719	0,720	0,719	0,718	0,715	0,711	0,721	0,716	0,719	0,718	0,717	0,717
sum yi	-	0,989	0,993	0,993	0,965	0,989	0,993	0,992	0,992	0,990	0,991	0,990	0,991	0,989	0,986	0,993	0,991	0,991	0,992	0,991	0,990
Specific heat in the exhaust gases																					
Differential temperature gases atmosphere, tesc-ta	°C	368	399	413	423	437	445	452	455	466	474	488	503	513	522	527	543	539	545	550	545
Average temperature of gases and environment, (tesc+ta)/2	°C	212	228	235	240	247	251	254	256	261	265	272	280	285	289	292	300	297	301	303	301
cp_esc,w	kJ/kg K	1,2	1,2	1,2	1,2	1,2	1,2	1,2	1,2	1,2	1,2	1,2	1,2	1,2	1,2	1,2	1,2	1,2	1,2	1,2	1,2
Energy balance																					
Thermal Power of the combustion, m_f H_f	kW	15,66	18,44	19,27	19,85	20,73	21,94	23,19	24,65	25,28	26,05	27,58	30,74	34,24	36,71	36,90	39,24	40,10	38,21	37,24	37,54
	%	100,0%	100,0%	100,0%	100,0%	100,0%	100,0%	100,0%	100,0%	100,0%	100,0%	100,0%	100,0%	100,0%	100,0%	100,0%	100,0%	100,0%	100,0%	100,0%	100,0%
Actual engine Power, We	kW	4,07	4,90	5,34	5,62	6,01	6,33	6,70	7,01	7,23	7,45	8,01	8,84	9,72	10,18	10,45	10,30	9,83	9,33	9,14	9,05
	%	25,97%	26,58%	27,68%	28,33%	29,02%	28,85%	28,90%	28,45%	28,60%	28,58%	29,04%	28,77%	28,40%	27,74%	28,33%	26,24%	24,53%	24,41%	24,55%	24,11%
Sensible heat of exhaust gases, H_esc,w	kW	2,3	3,0	3,2	3,4	3,7	3,9	4,1	4,3	4,6	4,9	5,3	6,0	6,7	7,3	7,6	8,2	8,4	8,1	8,0	8,1
	%	14,6%	16,1%	16,6%	17,2%	17,8%	17,7%	17,7%	17,6%	18,3%	18,9%	19,3%	19,6%	19,7%	20,0%	20,5%	20,9%	21,0%	21,2%	21,4%	21,5%
Energy due to the unburned content, H_inq	kW	1,0	1,3	0,8	0,2	0,9	1,0	1,2	2,2	2,2	2,7	2,4	2,9	3,3	4,1	2,6	3,5	3,5	3,8	3,4	4,4
	%	6,2%	7,0%	3,9%	0,9%	4,5%	4,4%	5,3%	8,9%	8,7%	10,2%	8,6%	9,5%	9,7%	11,3%	7,0%	8,9%	8,6%	9,9%	9,2%	11,7%
Heat dissipated in cooling system, Q_ref+Q_lub	kW	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0
	%	0,0%	0,0%	0,0%	0,0%	0,0%	0,0%	0,0%	0,0%	0,0%	0,0%	0,0%	0,0%	0,0%	0,0%	0,0%	0,0%	0,0%	0,0%	0,0%	0,0%
Other energy flows, E_otras	kW	8,3	9,3	10,0	10,6	10,1	10,8	11,2	11,1	11,2	11,0	11,9	13,0	14,4	15,0	16,3	17,3	18,4	17,0	16,7	16,0
	%	53,3%	50,3%	51,8%	53,6%	48,6%	49,0%	48,1%	45,0%	44,4%	42,4%	43,1%	42,2%	42,2%	41,0%	44,2%	44,0%	45,8%	44,5%	44,9%	42,7%

1.4 170 cm^3 in partial load (conventional filter)

80 % of wide open throttle

Engine data														
Number of cylinders, z	-	1												
Volumen desplazado por cilindro, V_cil	cm^3	171,26												
Thermodynamic cycle per revolution, i	-	0,5												
Fuel consumption														
number of C atoms, alfa	-	1												
number of H atoms, beta	-	1,861												
number of O atoms, gamma	-	0,016												
number of N atoms, delta	-	0												
molecular weight, Mf	$kg_f/kmol_f$	17,08												
Air Fuel Ratio stoiciometric, AFR_stq	kg_a/kg_f	14,25												
lower heating value, LHV	MJ/kg	43,0												
Measured parameters														
point	-	1	2	3	4	5	6	7	8	9	10	11	12	13
Throttle opening	grados	80	80	80	80	80	80	80	80	80	80	80	80	80
Engine speed, torque, fuel consumption														
Engine speed, n	rpm	2.999	3.748	4.495	5.243	6.001	6.500	6.999	7.498	7.997	8.496	8.994	9.393	9742,383
Actual Torque, T_e	Nm	11,60	12,14	12,62	12,79	12,83	12,88	12,63	12,25	11,67	10,80	9,78	8,98	8,197494
Fuel consumption, m_f	kg/h	1,0100	1,2400	1,3600	1,6700	1,6680	2,1400	2,5100	1,7900	2,0446	2,5600	2,1999	3,3200	2,4700
Environmental conditions of the trial cell														
barometric pressure, p_cell	mbar	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1.022
Temperature, t_cell	ºC	28	28	28	28	28	28	28	28	28	28	28	28	28
Humidity relative, phi_rel	%	47	47	47	47	47	47	47	47	47	47	47	47	47
Data for obtatining the intake air mass flow														
Nozzle discharge coefficient	-	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93
Differential pressure, Dp_atm-dep	mbar													
Intake and exhaust pressure														
Intake manifold absolute pressure, Dp_atm-adm	mbar	-1	-1	-6	-8	-10	-12	-15	-18	-19	-19	-23	-26	-27,02
Exhaust manifold absolute pressure, Dp_esc-atm	mbar	3,20	12,89	37,69	46,03	58,52	67,14	76,13	86,07	89,29	94,00	96,38	94,84	101,68
Intake temperature	ºC	39,34	38,86	38,60	39,19	39,75	40,81	41,62	42,44	43,76	44,11	45,45	46,58	47,33
		312,49	312,01	311,75	312,34	312,90	313,96	314,77	315,59	316,91	317,26	318,60	319,73	320,48
Results of the exhaust gases analysis (HC < 100 ppm y NOx < 1000 ppm)														
Exhuast gases temperature, t_esc	ºC	412	432	461	492	515	536	552	561	554	564	581	584	589,47
Vol. Correction of O2 in the dry exhaust gases, O2_D	%	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
Vol. Correction of CO2 in the dry exhaust gases, CO2_D	%	14,05	14,01	14,05	14,18	14,05	13,98	14,12	13,94	13,75	13,84	14,22	14,42	14,55
Vol. Correction of CO in the dry exhaust gases, CO_D	%	0,93	1,17	1,23	1,07	1,31	1,53	1,34	1,64	1,86	1,85	1,34	1,02	0,72
Vol. Correction of HC in the dry exhaust gases, HC_D	ppm C6	151	154	135	111	101	123	163	208	213	165	111	90	76
Vol. relation of NOx in the dry exhaust gases, NOx_D	ppm NO+NO2	0	0	0	0	0	0							
Dilution factor, DF		1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00
measured lambda HORIBA	-	1,00	0,99	0,99	1,00	0,99	0,98	0,99	0,98	0,97	0,96	0,98	1,00	1,01
Vol. relation of O2 in the dry exhaust gases, O2_D	%	2,44	0,74	0,52	0,47	0,45	0,39	0,45	0,46	1,17	0,81	0,93	1,03	1,05
Vol. relation of CO2 in the dry exhaust gases, CO2_D	%	14,05	14,01	14,05	14,18	14,05	13,98	14,12	13,94	13,75	13,84	14,22	14,42	14,55
Vol. relation of CO in the dry exhaust gases, CO_D	%	0,93	1,17	1,23	1,07	1,31	1,53	1,34	1,64	1,86	1,85	1,34	1,02	0,72
Vol. relation of HC in the dry exhaust gases, HC_D	ppm C6	151	154	135	111	101	123	163	208	213	165	111	90	75,5
Temperature before catalyst	ºC	224,51	226,07	245,66	275,70	297,15	319,82	338,10	352,24	325,74	350,20	366,88	363,26	369,69
Temperature after catalyst	ºC	166,74	166,28	179,40	205,21	224,14	247,82	264,62	279,97	254,36	274,79	290,30	287,30	293,42
Calculated parameters														
point	-	1	2	3	4	5	6	7	8	9	10	11	12	13
engine speedr, n	rpm	2999	3748	4495	5243	6001	6500	6999	7498	7997	8496	8994	9393	9742
actual Torque, actual Power, mean effective pressure, specific fuel consumption														
actual Torque, M_e	Nm	11,60	12,14	12,62	12,79	12,83	12,88	12,63	12,25	11,67	10,80	9,78	8,98	8,20
actual Power, W_e	kW	3,6	4,8	5,9	7,0	8,1	8,8	9,3	9,6	9,8	9,6	9,2	8,8	8,4
mean effective pressure, pme	kPa	851,3	891,1	926,1	938,6	941,1	945,3	926,9	899,1	856,2	792,4	717,8	658,6	601,5
specific fuel consumption, g_ef	g/kWh	277,2	260,2	228,9	237,8	207,0	244,0	271,1	186,1	209,3	266,5	238,8	376,0	295,3
Power and Torque corrected by environmental conditions														
Saturated steam pressure, p_s,ens (kPa)	mbar	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8
Dry air pressure, p_ens-phi*p_s,ens (kPa)	mbar	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2
Factor de corrección, alfa-a	-	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99
Power corrtd, W_e,corr	kW	3,6	4,7	5,9	6,9	8,0	8,7	9,2	9,5	9,7	9,5	9,1	8,7	8,3
Torque corrct, M_e,corr	Nm	11,5	12,0	12,5	12,6	12,7	12,7	12,5	12,1	11,5	10,7	9,7	8,9	8,1

Relative dosage, volumetric efficiency and emissions (starting from the gases analysis and fuel consumption)														
AFR, lambda and relative dosage														
AFR with Brettschneider method														
K balance constant of the reaction H2O gases	-	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5
carbon atoms in the selected HC molecule	-	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0
(beta/alpha)/4	-	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653
(gamma/alpha)/2	-	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040
1+((beta/alpha)/4)-((gamma/alpha)/2)	-	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613
Lambda Brettschneider	-	0,97	0,96	0,96	0,97	0,96	0,95	0,96	0,95	0,94	0,94	0,96	0,97	0,98
relative dosage Brettschneider	-	1,03	1,04	1,04	1,04	1,04	1,05	1,05	1,06	1,06	1,06	1,04	1,03	1,02
AFR Brettschneider	kg_a/kg_f	13,78	13,69	13,68	13,75	13,66	13,57	13,62	13,48	13,39	13,43	13,66	13,79	13,92
Fuel consumption, m_f	kg/h	-	-	-	-	-	-	-	-	-	-	-	-	-
specific fuel consumption, g_ef	g/kWh	-	-	-	-	-	-	-	-	-	-	-	-	-
Volumetric Efficiency														
air fuel rate from lambda, m_a	kg/s	0,0039	0,0047	0,0052	0,0064	0,0063	0,0081	0,0095	0,0067	0,0076	0,0095	0,0083	0,0127	0,0095
Volumetric Efficiency, nhu_v,ens	-	0,76	0,75	0,68	0,72	0,63	0,74	0,80	0,53	0,56	0,67	0,55	0,80	0,58
Volumetric Efficiency, nhu_v,adm for air density in intake manifold	-	0,79	0,77	0,71	0,75	0,66	0,78	0,85	0,57	0,60	0,72	0,60	0,87	0,63
Composition of exhaust gases and emissions														
[H2O] wet	%	12,0	12,1	12,2	12,2	12,2	12,3	12,3	12,3	12,3	12,3	12,4	12,3	12,3
[O2] wet	%	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0
[CO2] wet	%	12,4	12,3	12,3	12,5	12,3	12,3	12,4	12,2	12,1	12,1	12,5	12,6	12,8
[CO] wet	%	0,8	1,0	1,1	0,9	1,1	1,3	1,2	1,4	1,6	1,6	1,2	0,9	0,6
[HC1] wet	%	0,1	0,1	0,1	0,1	0,1	0,1	0,1	0,1	0,1	0,1	0,1	0,0	0,0
[NO2] wet	%	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0
[H2] wet	%	0,2	0,3	0,3	0,3	0,3	0,4	0,3	0,4	0,5	0,5	0,3	0,2	0,2
B1	-	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457
B2	-	7,543	7,451	7,413	7,436	7,388	7,315	7,328	7,261	7,243	7,224	7,302	7,363	7,444
B3	-	1,034	1,041	1,042	1,036	1,043	1,051	1,046	1,057	1,064	1,062	1,044	1,033	1,024
[N2] wet	%	70,5	70,9	71,2	71,4	71,3	71,6	71,7	71,6	71,3	71,7	72,1	72,3	72,1
Summation [i]	%	96,0	96,7	97,1	97,3	97,4	97,9	98,0	98,1	97,9	98,3	98,5	98,4	98,0
Molecular weight of the exhaust gases, M_esc,w	kg/kmol	27,65	27,81	27,93	27,98	27,99	28,09	28,16	28,17	28,06	28,18	28,30	28,32	28,25
mass flow rate of damp fumes m_esc,w	g/h	14930	18213	19960	24640	24461	31179	36706	25926	29428	36931	32240	49111	36848
mass flow rate of damp fumes m_esc,w	kg/s	0,0041	0,0051	0,0055	0,0068	0,0068	0,0087	0,0102	0,0072	0,0082	0,0103	0,0090	0,0136	0,0102
Mol in the exhaust gases, n_esc,w = m_esc,w/M_esc,w	mol/h	5,40E+02	6,55E+02	7,15E+02	8,81E+02	8,74E+02	1,11E+03	1,30E+03	9,20E+02	1,05E+03	1,31E+03	1,14E+03	1,73E+03	1,30E+03
bs CO	g/kWh	34,0	39,6	36,4	33,0	34,9	47,6	46,4	38,5	49,1	62,0	40,7	49,2	27,6
bs HC1	g/kWh	10,1	9,5	7,3	6,2	4,9	7,0	10,2	8,9	10,2	10,0	6,1	7,9	5,3
bs NOx	g/kWh	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0
bs CO2	g/kWh	806,0	744,4	653,2	687,0	588,5	683,1	767,4	514,7	569,8	728,4	678,1	1092,5	876,0

Composition and specific heat of exhaust gases														
Composition of exhaust gases														
y O2 wet	-	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000
y CO2 wet	-	0,124	0,123	0,123	0,125	0,123	0,123	0,124	0,122	0,121	0,121	0,125	0,126	0,128
y CO wet	-	0,008	0,010	0,011	0,009	0,011	0,013	0,012	0,014	0,016	0,016	0,012	0,009	0,006
y H2 wet	-	0,002	0,003	0,003	0,003	0,003	0,004	0,003	0,004	0,005	0,005	0,003	0,002	0,002
y H2O wet	-	0,120	0,121	0,122	0,122	0,122	0,123	0,123	0,123	0,123	0,123	0,124	0,123	0,123
y N2 wet	-	0,705	0,709	0,712	0,714	0,713	0,716	0,717	0,716	0,713	0,717	0,721	0,723	0,721
sum yi	-	0,959	0,966	0,971	0,972	0,974	0,978	0,979	0,980	0,978	0,982	0,985	0,984	0,980
Specific heat in the exhaust gases														
Differential temperature gases atmosphere, tesc-ta	ºC	384	404	433	464	487	508	524	533	526	536	553	556	561
Average temperature of gases and environment, (tesc+ta)/2	ºC	220	230	244	260	272	282	290	294	291	296	304	306	309
cp_esc,w	kJ/kg K	1,2	1,2	1,2	1,2	1,2	1,2	2,2	3,2	4,2	5,2	6,2	7,2	8,2

Energy balance														
Thermal Power of the combustion, m_f H_f	kW	12,06	14,81	16,24	19,95	19,92	25,56	29,98	21,38	24,42	30,58	26,28	39,66	29,50
	%	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00
Actual engine Power, We	kW	3,64	4,77	5,94	7,02	8,06	8,77	9,26	9,62	9,77	9,61	9,21	8,83	8,36
	%	0,30	0,32	0,37	0,35	0,40	0,34	0,31	0,45	0,40	0,31	0,35	0,22	0,28
Sensible heat of exhaust gases, H_esc,w	kW	1,91	2,45	2,88	3,81	3,97	5,28	11,74	12,28	18,05	28,61	30,68	54,58	47,13
	%	0,16	0,17	0,18	0,19	0,20	0,21	0,39	0,57	0,74	0,94	1,17	1,38	1,60
Energy due to the unburned content, H_inq	kW	0,44	0,68	0,78	0,83	1,02	1,51	1,55	1,34	1,74	2,16	1,35	1,56	0,83
	%	0,04	0,05	0,05	0,04	0,05	0,06	0,05	0,06	0,07	0,07	0,05	0,04	0,03
Heat dissipated in cooling system, Q_ref+Q_lub	kW	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
	%	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
Other energy flows, E_otras	kW	6,06	6,91	6,64	8,28	6,87	10,00	7,43	-1,86	-5,14	-9,80	-14,97	-25,31	-26,81
	%	0,50	0,47	0,41	0,42	0,34	0,39	0,25	-0,09	-0,21	-0,32	-0,57	-0,64	-0,91

60 % of wide open throttle

Engine data														
Number of cylinders, z	-	1												
Volumen desplazado por cilindro, V_cil	cm^3	171,26												
Thermodynamic cycle per revolution, i	-	0,5												
Fuel consumption														
number of C atoms, alfa	-	1												
number of H atoms, beta	-	1,861												
number of O atoms, gamma	-	0,016												
number of N atoms, delta	-	0												
molecular weight, Mf	$kg_f/kmol_f$	17,08												
Air Fuel Ratio stoiciometric, AFR_stq	kg_a/kg_f	14,25												
lower heating value, LHV	MJ/kg	43,0												
Measured parameters														
point	-	1	2	3	4	5	6	7	8	9	10	11	12	13
Throttle opening	grados	60	60	60	60	60	60	60	60	60	60	60	60	60
Engine speed, torque, fuel consumption														
Engine speed, n	rpm	3.001	3.749	4.496	5.244	6.003	6.501	6.999	7.498	7.997	8.496	8.996	9.395	9743,879
Actual Torque, T_e	Nm	10,54	11,87	12,04	11,93	11,57	11,58	11,24	10,60	9,93	9,17	8,43	7,78	7,27
Fuel consumption, m_f	kg/h	1,4700	1,4720	1,3400	1,5650	1,7700	2,0900	2,1300	0,5933	2,1700	2,1800	1,7700	3,3600	2,43
Environmental conditions of the trial cell														
barometric pressure, p_cell	mbar	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1.022
Temperature, t_cell	ºC	28	28	28	28	28	28	28	28	28	28	28	28	28
Humidity relative, phi_rel	%	47	47	47	47	47	47	47	47	47	47	47	47	47
Data for obtatining the intake air mass flow														
Nozzle discharge coefficient	-	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93
Differential pressure, Dp_atm-dep	mbar													
Intake and exhaust pressure														
Intake manifold absolute pressure, Dp_atm-adm	mbar	-5	-7	-10	-14	-16	-19	-23	-26	-27	-29	-32	-35	-37,4
Exhaust manifold absolute pressure, Dp_esc-atm	mbar	24,59	40,45	46,14	44,35	50,72	28,38	61,30	66,73	74,80	76,86	75,73	81,00	78,8
Intake temperature	ºC	34,13	35,35	36,54	38,50	39,67	41,04	41,17	43,21	44,42	46,18	46,30	46,52	47,75
		307,28	308,50	309,69	311,65	312,82	314,19	314,32	316,36	317,57	319,33	319,45	319,67	320,90
Resultados del análisis de gases de escape (HC < 100 ppm y NOx < 1000 ppm)														
Exhuast gases temperature, t_esc	ºC	433	450	472	504	527	510	549	565	573	575	576	576	586,82
Vol. Correction of O2 in the dry exhaust gases, O2_D	%	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
Vol. Correction of CO2 in the dry exhaust gases, CO2_D	%	14,39	14,41	14,45	14,45	14,34	14,19	14,15	14,15	14,16	14,16	14,06	14,04	14,05
Vol. Correction of CO in the dry exhaust gases, CO_D	%	0,69	0,76	0,58	0,43	0,87	0,98	1,33	1,27	1,29	1,35	1,56	1,56	1,60
Vol. Correction of HC in the dry exhaust gases, HC_D	ppm C6	159	150	129	96	93	207	248	290	280	211	136	118	108
Vol. relation of NOx in the dry exhaust gases, NOx_D	ppm NO+NO2	0	0	0	0	0	0	0	0	0	0	0	0	0
Dilution factor, DF		1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00
measured lambda HORIBA	-	1,01	1,01	1,02	1,04	1,01	1,00	0,99	0,99	0,99	0,98	0,98	0,97	0,98
Vol. relation of O2 in the dry exhaust gases, O2_D	%	0,86	0,81	0,92	1,26	1,09	4,20	0,88	0,73	0,69	0,67	0,60	0,60	0,53
Vol. relation of CO2 in the dry exhaust gases, CO2_D	%	14,39	14,41	14,45	14,45	14,34	14,19	14,15	14,15	14,16	14,16	14,06	14,04	14,052
Vol. relation of CO in the dry exhaust gases, CO_D	%	0,69	0,76	0,58	0,43	0,87	0,98	1,33	1,27	1,29	1,35	1,56	1,56	1,6
Vol. relation of HC in the dry exhaust gases, HC_D	ppm C6	159	150	129	96	93	207	248	290	280	211	136	118	108,47
Temperature before catalyst	ºC	201,47	207,30	228,69	251,70	278,01	231,13	291,49	323,93	328,52	326,06	317,39	298,46	302,73
Temperature after catalyst	ºC	151,65	159,37	173,18	197,36	214,62	182,62	232,96	258,95	270,09	270,09	274,55	283,19	280,33
Calculated parameters														
point	-	1	2	3	4	5	6	7	8	9	10	11	12	13
engine speedr, n	rpm	3001	3749	4496	5244	6003	6501	6999	7498	7997	8496	8996	9395	9744
actual Torque, actual Power, mean effective pressure, specific fuel consumption														
actual Torque, M_e	Nm	10,54	11,87	12,04	11,93	11,57	11,58	11,24	10,60	9,93	9,17	8,43	7,78	7,27
actual Power, W_e	kW	3,3	4,7	5,7	6,5	7,3	7,9	8,2	8,3	8,3	8,2	7,9	7,7	7,4
mean effective pressure, pme	kPa	773,2	870,9	883,6	875,2	848,8	849,6	824,4	777,8	728,4	672,9	618,8	571,1	533,6
specific fuel consumption, g_ef	g/kWh	443,9	316,0	236,3	238,9	243,4	265,2	258,7	71,3	261,0	267,2	222,8	438,9	327,5
Power and Torque corrected by environmental conditions														
Saturated steam pressure, p_s,ens (kPa)	mbar	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8	
Dry air pressure, p_ens-phi*p_s,ens (kPa)	mbar	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1022,0
Factor de corrección, alfa-a	-	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,97
Power corrtd, W_e,corr	kW	3,3	4,6	5,6	6,5	7,2	7,8	8,1	8,2	8,2	8,1	7,9	7,6	7,2
Torque corrct, M_e,corr	Nm	10,4	11,7	11,9	11,8	11,4	11,5	11,1	10,5	9,8	9,1	8,3	7,7	7,0

Relative dosage, volumetric efficiency and emissions (starting from the gases analysis and fuel consumption)														
AFR, lambda and relative dosage														
AFR with Brettschneider method														
K balance constant of the reaction H2O gases	-	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5
carbon atoms in the selected HC molecule	-	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0
(beta/alpha)/4	-	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653
(gamma/alpha)/2	-	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040
1+((beta/alpha)/4)-((gamma/alpha)/2)	-	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613
Lambda Brettschneider	-	0,97	0,97	0,98	0,98	0,97	0,96	0,95	0,95	0,95	0,95	0,95	0,95	0,95
relative dosage Brettschneider	-	1,03	1,03	1,02	1,02	1,03	1,04	1,05	1,05	1,05	1,05	1,05	1,05	1,05
AFR Brettschneider	kg_a/kg_f	13,88	13,86	13,94	14,02	13,85	13,74	13,58	13,58	13,58	13,60	13,56	13,56	13,55
Fuel consumption, m_f	kg/h	-	-	-	-	-	-	-	-	-	-	-	-	-
specific fuel consumption, g_ef	g/kWh	-	-	-	-	-	-	-	-	-	-	-	-	-
Volumetric Efficiency														
air fuel rate from lambda, m_a	kg/s	0,0057	0,0057	0,0052	0,0061	0,0068	0,0080	0,0080	0,0022	0,0082	0,0082	0,0067	0,0127	0,0091
Volumetric Efficiency, nhu_v,ens	-	1,12	0,90	0,68	0,69	0,67	0,73	0,68	0,18	0,61	0,57	0,44	0,80	0,56
Volumetric Efficiency, nhu_v,adm for air density in intake manifold	-	1,15	0,92	0,71	0,72	0,71	0,77	0,73	0,19	0,66	0,63	0,48	0,88	0,62
Composition of exhaust gases and emissions														
[H2O] wet	%	12,2	12,2	12,1	12,1	12,2	12,2	12,3	12,3	12,3	12,3	12,3	12,3	12,4
[O2] wet	%	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0
[CO2] wet	%	12,6	12,7	12,7	12,7	12,6	12,5	12,4	12,4	12,4	12,4	12,3	12,3	12,3
[CO] wet	%	0,6	0,7	0,5	0,4	0,8	0,9	1,2	1,1	1,1	1,2	1,4	1,4	1,4
[HC1] wet	%	0,1	0,1	0,1	0,1	0,0	0,1	0,1	0,2	0,1	0,1	0,1	0,1	0,1
[NO2] wet	%	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0
[H2] wet	%	0,2	0,2	0,1	0,1	0,2	0,2	0,3	0,3	0,3	0,3	0,4	0,4	0,4
B1	-	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457
B2	-	7,502	7,464	7,535	7,614	7,462	7,443	7,296	7,310	7,300	7,294	7,266	7,279	7,260
B3	-	1,027	1,029	1,022	1,017	1,029	1,038	1,049	1,049	1,049	1,048	1,052	1,051	1,052
[N2] wet	%	71,4	71,6	71,4	71,0	71,6	71,2	71,8	71,7	71,8	71,9	72,0	71,9	72,0
Summation [i]	%	97,0	97,4	96,9	96,3	97,4	97,0	98,2	97,9	98,1	98,3	98,5	98,3	98,6
Molecular weight of the exhaust gases, M_esc,w	kg/kmol	27,99	28,09	27,97	27,82	28,05	27,96	28,24	28,20	28,23	28,26	28,26	28,22	28,27
mass flow rate of damp fumes m_esc,w	g/h	21874	21872	20022	23505	26277	30801	31065	8653	31645	31821	25763	48935	35368
mass flow rate of damp fumes m_esc,w	kg/s	0,0061	0,0061	0,0056	0,0065	0,0073	0,0086	0,0086	0,0024	0,0088	0,0088	0,0072	0,0136	0,0098
Mol in the exhaust gases, n_esc,w = m_esc,w/M_esc,w	mol/h	7,81E+02	7,79E+02	7,16E+02	8,45E+02	9,37E+02	1,10E+03	1,10E+03	3,07E+02	1,12E+03	1,13E+03	9,12E+02	1,73E+03	1,25E+03
bs CO	g/kWh	40,1	31,2	18,0	13,7	27,6	33,7	43,6	11,5	42,7	45,8	44,0	86,8	66,2
bs HC1	g/kWh	16,8	11,2	7,3	5,5	5,3	13,0	14,8	4,8	16,8	13,0	6,9	11,9	8,2
bs NOx	g/kWh	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0
bs CO2	g/kWh	1312,8	930,6	705,3	721,4	713,8	766,8	729,6	201,4	737,0	754,2	622,5	1227,0	913,8

Composition and specific heat of exhaust gases														
Composition of exhaust gases														
y O2 wet	-	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000
y CO2 wet	-	0,126	0,127	0,127	0,127	0,126	0,125	0,124	0,124	0,124	0,124	0,123	0,123	0,123
y CO wet	-	0,006	0,007	0,005	0,004	0,008	0,009	0,012	0,011	0,011	0,012	0,014	0,014	0,014
y H2 wet	-	0,002	0,002	0,001	0,001	0,002	0,002	0,003	0,003	0,003	0,003	0,004	0,004	0,004
y H2O wet	-	0,122	0,122	0,121	0,121	0,122	0,122	0,123	0,123	0,123	0,123	0,123	0,123	0,124
y N2 wet	-	0,714	0,716	0,714	0,710	0,716	0,712	0,718	0,717	0,718	0,719	0,720	0,719	0,720
sum yi	-	0,969	0,973	0,969	0,963	0,973	0,969	0,980	0,978	0,979	0,982	0,984	0,983	0,985
Specific heat in the exhaust gases														
Differential temperature gases atmosphere, tesc-ta	ºC	405	422	444	476	499	482	521	537	545	547	548	548	559
Average temperature of gases and environment, (tesc+ta)/2	ºC	231	239	250	266	278	269	289	296	300	302	302	302	307
cp_esc,w	kJ/kg K	1,2	1,2	1,2	1,2	1,2	1,2	2,2	3,2	4,2	5,2	6,2	7,2	8,2

Energy balance														
Thermal Power of the combustion, m_f H_f	kW	17,6	17,6	16,0	18,7	21,1	25,0	25,4	7,1	25,9	26,0	21,1	40,1	29,0
	%	1,0	1,0	1,0	1,0	1,0	1,0	1,0	1,0	1,0	1,0	1,0	1,0	1,0
Actual engine Power, We	kW	3,3	4,7	5,7	6,5	7,3	7,9	8,2	8,3	8,3	8,2	7,9	7,7	7,4
	%	0,2	0,3	0,4	0,4	0,3	0,3	0,3	1,2	0,3	0,3	0,4	0,2	0,3
Sensible heat of exhaust gases, H_esc,w	kW	3,0	3,1	3,0	3,7	4,4	4,9	9,9	4,1	20,1	25,2	24,3	53,6	45,0
	%	0,2	0,2	0,2	0,2	0,2	0,2	0,4	0,6	0,8	1,0	1,2	1,3	1,6
Energy due to the unburned content, H_inq	kW	0,5	0,5	0,4	0,3	0,7	1,0	1,3	0,3	1,3	1,3	1,3	2,4	1,8
	%	0,0	0,0	0,0	0,0	0,0	0,0	0,1	0,0	0,0	0,1	0,1	0,1	0,1
Heat dissipated in cooling system, Q_ref+Q_lub	kW	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0
	%	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0
Other energy flows, E_otras	kW	10,8	9,3	7,0	8,1	8,8	11,2	6,0	-5,7	-3,8	-8,6	-12,4	-23,6	-25,2
	%	0,6	0,5	0,4	0,4	0,4	0,4	0,2	-0,8	-0,1	-0,3	-0,6	-0,6	-0,9

50 % of wide open throttle

Engine data		
Number of cylinders, z	-	1
Volumen desplazado por cilindro, V_cil	cm^3	171,26
Thermodynamic cycle per revolution, i	-	0,5
Fuel consumption		
number of C atoms, alfa	-	1
number of H atoms, beta	-	1,861
number of O atoms, gamma	-	0,016
number of N atoms, delta	-	0
molecular weight, Mf	$kg_f/kmol_f$	17,08
Air Fuel Ratio stoiciometric, AFR_stq	kg_a/kg_f	14,25
lower heating value, LHV	MJ/kg	43,0

Measured parameters														
point	-	1	2	3	4	5	6	7	8	9	10	11	12	13
Throttle opening	grados	50	50	50	50	50	50	50	50	50	50	50	50	50
Engine speed, torque, fuel consumption														
Engine speed, n	rpm	3.000	3.746	4.494	5.247	6.000	6.499	7.000	7.499	7.997	8.495	8.996	9.393	9741,286
Actual Torque, T_e	Nm	11,35	11,22	11,09	10,65	9,47	9,22	8,85	8,45	7,76	6,95	6,19	5,68	5,06
Fuel consumption, m_f	kg/h	0,7500	1,2400	1,4700	1,5300	1,6600	1,5900	1,7500	1,9700	1,5500	1,2300	1,2300	1,6700	1,54
Environmental conditions of the trial cell														
barometric pressure, p_cell	mbar	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1.022
Temperature, t_cell	ºC	28	28	28	28	28	28	28	28	28	28	28	28	28
Humidity relative, phi_rel	%	47	47	47	47	47	47	47	47	47	47	47	47	47
Data for obtaining the intake air mass flow														
Nozzle discharge coefficient	-	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93
Differential pressure, Dp_atm-dep	mbar	0,109	0,113	0,110	0,102	0,101	0,104	0,097	0,095	0,092	0,094	0,096	0,091	0,0957
Intake and exhaust pressure														
Intake manifold absolute pressure, Dp_atm-adm	mbar	-9	-12	-17	-20	-20	-24	-27	-31	-34	-37	-42	-45	-47,1
Exhaust manifold absolute pressure, Dp_esc-atm	mbar	9,07	27,94	33,95	33,89	38,55	44,52	50,27	52,76	58,60	53,99	43,16	49,75	48,31
Intake temperature	ºC	40,09	34,33	35,63	37,78	40,16	41,43	42,34	44,06	45,80	43,96	46,00	43,48	45,49

Exhaust gas analysis results (HC < 100 ppm and NOx < 1000 ppm)														
Exhuast gases temperature, t_esc	ºC	405	439	473	491	510	521	531	542	554	527	549	529	536,94
Vol. Correction of O2 in the dry exhaust gases, O2_D	%	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
Vol. Correction of CO2 in the dry exhaust gases, CO2_D	%	13,88	13,82	13,97	13,85	13,96	13,86	13,91	13,77	13,79	13,81	13,85	13,71	13,66
Vol. Correction of CO in the dry exhaust gases, CO_D	%	0,28	0,22	0,47	0,93	0,64	0,80	0,56	0,76	0,63	0,47	0,42	0,36	0,29
Vol. Correction of HC in the dry exhaust gases, HC_D	ppm C6	109	97	79	77	79	151	256	353	310	137	66	71	41
Vol. relation of NOx in the dry exhaust gases, NOx_D	ppm NO+NO2	0	0	0	0	0	0	0	0	0	0	0	0	0
Dilution factor, DF		1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00
measured lambda HORIBA	-	1,04	1,05	1,02	1,00	1,02	1,01	1,03	1,01	1,03	1,05	1,05	1,06	1,07
Vol. relation of O2 in the dry exhaust gases, O2_D	%	1,39	1,28	0,86	0,58	0,69	0,72	0,87	0,89	1,00	1,59	1,26	2,45	2,63
Vol. relation of CO2 in the dry exhaust gases, CO2_D	%	13,88	13,82	13,97	13,85	13,96	13,86	13,91	13,77	13,79	13,81	13,85	13,71	13,66
Vol. relation of CO in the dry exhaust gases, CO_D	%	0,28	0,22	0,47	0,93	0,64	0,80	0,56	0,76	0,63	0,47	0,42	0,36	0,29
Vol. relation of HC in the dry exhaust gases, HC_D	ppm C6	109	97	79	77	79	151	256	353	310	137	66	71	40,93
Temperature before catalyst	ºC	172,71	204,13	227,18	241,34	256,62	268,89	279,77	290,97	304,04	233,55	256,67	201,72	226,69
Temperature after catalyst	ºC	142,07	161,77	202,28	185,93	203,46	218,27	229,96	242,50	253,82	169,09	193,52	133,07	169,35

Calculated parameters														
point	-	1	2	3	4	5	6	7	8	9	10	11	12	13
engine speedr, n	rpm	3000	3746	4494	5247	6000	6499	7000	7499	7997	8495	8996	9393	9741
actual Torque, actual Power, mean effective pressure, specific fuel consumption														
actual Torque, M_e	Nm	11,35	11,22	11,09	10,65	9,47	9,22	8,85	8,45	7,76	6,95	6,19	5,68	5,06
actual Power, W_e	kW	3,6	4,4	5,2	5,8	5,9	6,3	6,5	6,6	6,5	6,2	5,8	5,6	5,2
mean effective pressure, pme	kPa	832,5	822,9	814,1	781,1	694,7	676,2	649,3	620,0	569,7	510,1	454,1	416,4	371,3
specific fuel consumption, g_ef	g/kWh	210,4	281,9	281,6	261,6	279,1	253,5	269,8	296,9	238,4	198,9	211,0	299,1	298,3
Power and Torque corrected by environmental conditions														
Saturated steam pressure, p_s,ens (kPa)	mbar	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8
Dry air pressure, p_ens-phi*p_s,ens (kPa)	mbar	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2
Factor de corrección, alfa-a	-	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99
Power corrtd, W_e,corr	kW	3,5	4,4	5,2	5,8	5,9	6,2	6,4	6,6	6,4	6,1	5,8	5,5	5,1
Torque corrct, M_e,corr	Nm	11,2	11,1	11,0	10,5	9,4	9,1	8,8	8,4	7,7	6,9	6,1	5,6	5,0

Relative dosage, volumetric efficiency and emissions (starting from the gases analysis and fuel consumption)														
AFR, lambda and relative dosage														
AFR with Brettschneider method														
K balance constant of the reaction H2O gases	-	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5
carbon atoms in the selected HC molecule	-	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0
(beta/alpha)/4	-	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653
(gamma/alpha)/2	-	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040
1+((beta/alpha)/4)-((gamma/alpha)/2)	-	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613
Lambda Brettschneider	-	0,99	0,99	0,98	0,97	0,98	0,97	0,97	0,96	0,97	0,98	0,98	0,99	0,99
relative dosage Brettschneider	-	1,01	1,01	1,02	1,03	1,02	1,03	1,03	1,04	1,03	1,02	1,02	1,01	1,01
AFR Brettschneider	kg_a/kg_f	14,07	14,10	14,00	13,82	13,94	13,83	13,86	13,73	13,80	13,97	14,03	14,05	14,10
Fuel consumption, m_f	kg/h	-	-	-	-	-	-	-	-	-	-	-	-	-
specific fuel consumption, g_ef	g/kWh	-	-	-	-	-	-	-	-	-	-	-	-	-
Volumetric Efficiency														
air fuel rate from lambda, m_a	kg/s	0,0029	0,0049	0,0057	0,0059	0,0064	0,0061	0,0067	0,0075	0,0059	0,0048	0,0048	0,0065	0,0060
Volumetric Efficiency, nhu_v,ens	-	0,58	0,77	0,75	0,66	0,63	0,56	0,57	0,59	0,44	0,33	0,32	0,41	0,37
Volumetric Efficiency, nhu_v,adm for air density in intake manifold	-	0,61	0,79	0,79	0,70	0,67	0,60	0,61	0,64	0,48	0,36	0,35	0,45	0,41
Composition of exhaust gases and emissions														
[H2O] wet	%	11,6	11,5	11,7	11,9	11,8	11,8	11,7	11,7	11,7	11,6	11,6	11,5	11,4
[O2] wet	%	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0
[CO2] wet	%	12,3	12,2	12,3	12,2	12,3	12,2	12,3	12,2	12,2	12,2	12,2	12,1	12,1
[CO] wet	%	0,2	0,2	0,4	0,8	0,6	0,7	0,5	0,7	0,6	0,4	0,4	0,3	0,3
[HC1] wet	%	0,1	0,1	0,0	0,0	0,0	0,1	0,1	0,2	0,2	0,1	0,0	0,0	0,0
[NO2] wet	%	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0
[H2] wet	%	0,1	0,1	0,1	0,2	0,2	0,2	0,1	0,2	0,2	0,1	0,1	0,1	0,1
B1	-	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457
B2	-	7,950	8,016	7,821	7,655	7,743	7,689	7,748	7,686	7,753	7,879	7,908	8,007	8,079
B3	-	1,013	1,011	1,018	1,032	1,023	1,031	1,028	1,038	1,033	1,020	1,016	1,014	1,011
[N2] wet	%	68,2	67,8	69,1	69,6	69,4	69,4	69,0	68,9	68,7	68,4	68,4	67,7	67,3
Summation [i]	%	92,5	91,9	93,7	94,8	94,3	94,4	93,8	93,8	93,4	92,8	92,8	91,8	91,2
Molecular weight of the exhaust gases, M_esc,w	kg/kmol	26,72	26,56	27,05	27,29	27,19	27,21	27,11	27,11	27,00	26,80	26,79	26,50	26,33
mass flow rate of damp fumes m_esc,w	g/h	11299	18721	22057	22670	24792	23576	26014	29012	22944	18411	18488	25136	23253
mass flow rate of damp fumes m_esc,w	kg/s	0,0031	0,0052	0,0061	0,0063	0,0069	0,0065	0,0072	0,0081	0,0064	0,0051	0,0051	0,0070	0,0065
Mol in the exhaust gases, n_esc,w = m_esc,w/M_esc,w	mol/h	4,23E+02	7,05E+02	8,16E+02	8,31E+02	9,12E+02	8,66E+02	9,60E+02	1,07E+03	8,50E+02	6,87E+02	6,90E+02	9,49E+02	8,83E+02
bs CO	g/kWh	8,2	8,7	18,1	32,6	24,2	27,3	20,5	30,3	20,4	12,9	12,3	15,2	12,3
bs HC1	g/kWh	5,8	7,0	5,5	4,9	5,4	9,4	17,0	25,6	18,2	6,9	3,5	5,4	3,2
bs NOx	g/kWh	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0
bs CO2	g/kWh	640,7	862,3	847,6	762,9	830,3	743,0	799,1	862,6	700,4	596,6	637,6	907,3	910,7

Composition and specific heat of exhaust gases														
Composition of exhaust gases														
y O2 wet	-	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000
y CO2 wet	-	0,123	0,122	0,123	0,122	0,123	0,122	0,123	0,122	0,122	0,122	0,122	0,121	0,121
y CO wet	-	0,002	0,002	0,004	0,008	0,006	0,007	0,005	0,007	0,006	0,004	0,004	0,003	0,003
y H2 wet	-	0,001	0,001	0,001	0,002	0,002	0,002	0,001	0,002	0,002	0,001	0,001	0,001	0,001
y H2O wet	-	0,116	0,115	0,117	0,119	0,118	0,118	0,117	0,117	0,117	0,116	0,116	0,115	0,114
y N2 wet	-	0,682	0,678	0,691	0,696	0,694	0,694	0,690	0,689	0,687	0,684	0,684	0,677	0,673
sum yi	-	0,924	0,918	0,937	0,948	0,943	0,943	0,937	0,936	0,933	0,927	0,928	0,917	0,912
Specific heat in the exhaust gases														
Differential temperature gases atmosphere, tesc-ta	ºC	377	411	445	463	482	493	503	514	526	499	521	501	509
Average temperature of gases and environment, (tesc+ta)/2	ºC	216	233	250	260	269	275	280	285	291	278	289	278	282
cp_esc,w	kJ/kg K	1,2	1,2	1,2	1,2	1,2	1,2	2,2	3,2	4,2	5,2	6,2	7,2	8,2

Energy balance														
Thermal Power of the combustion, m_f H_f	kW	8,96	14,81	17,56	18,28	19,83	18,99	20,90	23,53	18,51	14,69	14,69	19,95	18,39
	%	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00
Actual engine Power, We	kW	3,56	4,40	5,22	5,85	5,95	6,27	6,49	6,63	6,50	6,18	5,83	5,58	5,16
	%	0,40	0,30	0,30	0,32	0,30	0,33	0,31	0,28	0,35	0,42	0,40	0,28	0,28
Sensible heat of exhaust gases, H_esc,w	kW	1,42	2,56	3,27	3,50	3,98	3,88	8,00	13,25	14,08	13,27	16,59	25,18	26,96
	%	0,16	0,17	0,19	0,19	0,20	0,20	0,38	0,56	0,76	0,90	1,13	1,26	1,47
Energy due to the unburned content, H_inq	kW	0,10	0,14	0,34	0,69	0,52	0,61	0,48	0,72	0,47	0,29	0,26	0,30	0,23
	%	0,01	0,01	0,02	0,04	0,03	0,03	0,02	0,03	0,03	0,02	0,02	0,02	0,01
Heat dissipated in cooling system, Q_ref+Q_lub	kW	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
	%	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
Other energy flows, E_otras	kW	3,87	7,71	8,73	8,24	9,38	8,23	5,94	2,93	-2,54	-5,05	-7,99	-11,12	-13,95
	%	0,43	0,52	0,50	0,45	0,47	0,43	0,28	0,12	-0,14	-0,34	-0,54	-0,56	-0,76

40 % of wide open throttle

Engine data														
Number of cylinders, z	-	1												
Volumen desplazado por cilindro, V_cil	cm^3	171,26												
Thermodynamic cycle per revolution, i	-	0,5												
Fuel consumption														
number of C atoms, alfa	-	1												
number of H atoms, beta	-	1,861												
number of O atoms, gamma	-	0,016												
number of N atoms, delta	-	0												
molecular weight, Mf	$kg_f/kmol_f$	17,08												
Air Fuel Ratio stoiciometric, AFR_stq	kg_a/kg_f	14,25												
lower heating value, LHV	MJ/kg	43,0												
Measured parameters														
point	-	1	2	3	4	5	6	7	8	9	10	11	12	13
Throttle opening	grados	40	40	40	40	40	40	40	40	40	40	40	40	40
Engine speed, torque, fuel consumption														
Engine speed, n	rpm	2.998	3.747	4.498	5.243	6.003	6.500	6.999	7.498	7.998	8.497	8.996	9.395	9.745
Actual Torque, T_e	Nm	10,85	10,24	9,63	8,69	8,06	7,72	7,29	6,68	5,93	5,48	4,89	4,32	3,94
Fuel consumption, m_f	kg/h	0,8183	0,8390	1,3075	1,2600	1,2729	1,4700	1,6700	0,2600	0,9700	1,1986	1,7050	1,9401	1,6550
Environmental conditions of the trial cell														
barometric pressure, p_cell	mbar	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1.022
Temperature, t_cell	ºC	28	28	28	28	28	28	28	28	28	28	28	28	28
Humidity relative, phi_rel	%	47	47	47	47	47	47	47	47	47	47	47	47	47
Data for obtatining the intake air mass flow														
Nozzle discharge coefficient	-	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93
Differential pressure, Dp_atm-dep	mbar													
Intake and exhaust pressure														
Intake manifold absolute pressure, Dp_atm-adm	mbar	-13	-16	-23	-26	-26	-31	-35	-39	-43	-46	-51	-54	-57
Exhaust manifold absolute pressure, Dp_esc-atm	mbar	19,84	37,32	41,45	37,48	36,78	39,92	43,82	49,40	49,84	55,44	54,50	56,36	54,86
Intake temperature	ºC	40,49	39,16	40,82	43,03	49,12	46,50	47,92	49,35	50,74	52,02	51,87	52,39	52,86
Exhaust gas analysis results (HC < 100 ppm and NOx < 1000 ppm)														
Exhuast gases temperature, t_esc	ºC	301	449	482	492	477	526	538	550	563	574	573	569	574
Vol. Correction of O2 in the dry exhaust gases, O2_D	%	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
Vol. Correction of CO2 in the dry exhaust gases, CO2_D	%	14,60	14,87	14,84	14,80	14,71	14,59	14,53	14,51	14,62	14,79	14,88	14,90	14,92
Vol. Correction of CO in the dry exhaust gases, CO_D	%	0,60	0,60	0,64	0,78	0,88	1,14	0,98	0,83	0,56	0,68	0,54	0,56	0,52
Vol. Correction of HC in the dry exhaust gases, HC_D	ppm C6	170	170	174	174	182	280	335	355	271	189	97	97	93
Vol. relation of NOx in the dry exhaust gases, NOx_D	ppm NO+NO2	0	0	0	0	0	0							
Dilution factor, DF		1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00
measured lambda HORIBA	-	1,01	1,02	1,02	1,01	1,00	1,00	1,00	1,02	1,04	1,02	1,03	1,03	1,04
Vol. relation of O2 in the dry exhaust gases, O2_D	%	3,76	0,85	0,81	0,74	2,95	0,66	0,88	1,11	1,28	1,04	1,15	1,12	1,23
Vol. relation of CO2 in the dry exhaust gases, CO2_D	%	14,60	14,87	14,84	14,80	14,71	14,59	14,53	14,51	14,62	14,79	14,88	14,90	14,92
Vol. relation of CO in the dry exhaust gases, CO_D	%	0,60	0,60	0,64	0,78	0,88	1,14	0,98	0,83	0,56	0,68	0,54	0,56	0,52
Vol. relation of HC in the dry exhaust gases, HC_D	ppm C6	170	170	174	174	182	280	335	355	271	189	97	97	92,87
Temperature before catalyst	ºC	151,28	184,89	220,11	240,04	249,43	265,88	279,67	293,17	305,22	314,55	325,18	322,52	318,07
Temperature after catalyst	ºC	109,65	134,39	164,15	186,71	249,43	211,33	226,38	240,46	250,53	256,79	257,60	259,06	256,09
Calculated parameters														
point	-	1	2	3	4	5	6	7	8	9	10	11	12	13
engine speedr, n	rpm	2998	3747	4498	5243	6003	6500	6999	7498	7998	8497	8996	9395	9745
actual Torque, actual Power, mean effective pressure, specific fuel consumption														
actual Torque, M_e	Nm	10,85	10,24	9,63	8,69	8,06	7,72	7,29	6,68	5,93	5,48	4,89	4,32	3,94
actual Power, W_e	kW	3,4	4,0	4,5	4,8	5,1	5,3	5,3	5,2	5,0	4,9	4,6	4,3	4,0
mean effective pressure, pme	kPa	796,2	751,6	706,6	637,7	591,4	566,6	535,2	490,4	435,4	401,9	359,1	317,2	289,5
specific fuel consumption, g_ef	g/kWh	240,2	208,7	288,2	264,0	251,2	279,6	312,4	49,5	195,2	245,9	369,8	456,2	411,1
Potencia y par corregidas, y consumo específico corregido														
Saturated steam pressure, p_s,ens (kPa)	mbar	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8
Dry air pressure, p_ens-phi*p_s,ens (kPa)	mbar	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2
Factor de corrección, alfa-a	-	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99
Power corrtd, W_e,corr	kW	3,4	4,0	4,5	4,7	5,0	5,2	5,3	5,2	4,9	4,8	4,6	4,2	4,0
Torque corrct, M_e,corr	Nm	10,7	10,1	9,5	8,6	8,0	7,6	7,2	6,6	5,9	5,4	4,8	4,3	3,9

Relative dosage, volumetric efficiency and emissions (starting from the gases analysis and fuel consumption)														
AFR, lambda and relative dosage														
AFR with Brettschneider method														
K balance constant of the reaction H2O gases	-	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5
carbon atoms in the selected HC molecule	-	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0
(beta/alpha)/4	-	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653
(gamma/alpha)/2	-	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040
1+((beta/alpha)/4)-((gamma/alpha)/2)	-	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613
Lambda Brettschneider	-	0,98	0,98	0,98	0,97	0,97	0,96	0,96	0,96	0,97	0,97	0,98	0,98	0,98
relative dosage Brettschneider	-	1,02	1,02	1,03	1,03	1,03	1,04	1,04	1,04	1,03	1,03	1,02	1,02	1,02
AFR Brettschneider	kg_a/kg_f	13,91	13,92	13,90	13,85	13,80	13,65	13,68	13,73	13,88	13,88	13,98	13,98	13,99
Fuel consumption, m_f	kg/h	-	-	-	-	-	-	-	-	-	-	-	-	-
specific fuel consumption, g_ef	g/kWh	-	-	-	-	-	-	-	-	-	-	-	-	-
Volumetric Efficiency														
air fuel rate from lambda, m_a	kg/s	0,0032	0,0032	0,0050	0,0048	0,0049	0,0056	0,0063	0,0010	0,0037	0,0046	0,0066	0,0075	0,0064
Volumetric Efficiency, nhu_v,ens	-	0,63	0,51	0,67	0,55	0,48	0,51	0,54	0,08	0,28	0,32	0,44	0,48	0,39
Volumetric Efficiency, nhu_v,adm for air density in intake manifold	-	0,66	0,54	0,71	0,59	0,53	0,56	0,59	0,09	0,31	0,36	0,50	0,54	0,45
Composition of exhaust gases and emissions														
[H2O] wet	%	12,3	12,5	12,5	12,5	12,5	12,5	12,4	12,3	12,3	12,4	12,4	12,5	12,5
[O2] wet	%	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0
[CO2] wet	%	12,8	13,0	13,0	13,0	12,9	12,8	12,7	12,7	12,8	13,0	13,0	13,0	13,1
[CO] wet	%	0,5	0,5	0,6	0,7	0,8	1,0	0,9	0,7	0,5	0,6	0,5	0,5	0,5
[HC1] wet	%	0,1	0,1	0,1	0,1	0,1	0,1	0,2	0,2	0,1	0,1	0,1	0,1	0,0
[NO2] wet	%	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0
[H2] wet	%	0,1	0,1	0,2	0,2	0,2	0,3	0,2	0,2	0,1	0,2	0,1	0,1	0,1
B1	-	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457
B2	-	7,449	7,336	7,330	7,286	7,278	7,191	7,266	7,333	7,429	7,329	7,378	7,361	7,371
B3	-	1,024	1,024	1,025	1,029	1,033	1,044	1,042	1,039	1,027	1,027	1,019	1,020	1,019
[N2] wet	%	72,1	73,2	73,2	73,3	73,2	73,2	72,6	72,2	72,0	73,0	73,1	73,2	73,2
Summation [i]	%	97,9	99,4	99,4	99,7	99,6	100,0	99,0	98,4	97,9	99,3	99,2	99,4	99,4
Molecular weight of the exhaust gases, M_esc,w	kg/kmol	28,26	28,70	28,69	28,76	28,71	28,80	28,58	28,41	28,30	28,66	28,63	28,68	28,68
mass flow rate of damp fumes m_esc,w	g/h	12204	12518	19484	18709	18845	21542	24518	3829	14429	17832	25545	29054	24814
mass flow rate of damp fumes m_esc,w	kg/s	0,0034	0,0035	0,0054	0,0052	0,0052	0,0060	0,0068	0,0011	0,0040	0,0050	0,0071	0,0081	0,0069
Mol in the exhaust gases, n_esc,w = m_esc,w/M_esc,w	mol/h	4,32E+02	4,36E+02	6,79E+02	6,51E+02	6,56E+02	7,48E+02	8,58E+02	1,35E+02	5,10E+02	6,22E+02	8,92E+02	1,01E+03	8,65E+02
bs CO	g/kWh	18,7	16,0	23,5	26,1	27,9	39,7	38,6	5,2	14,1	21,3	25,6	32,7	27,4
bs HC1	g/kWh	9,6	8,2	11,6	10,6	10,5	17,8	23,9	4,1	12,4	10,8	8,4	10,3	8,9
bs NOx	g/kWh	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0
bs CO2	g/kWh	714,6	621,6	855,9	776,9	733,9	799,2	899,1	143,8	579,2	727,7	1109,6	1367,1	1235,6

Composition and specific heat of exhaust gases														
Composition of exhaust gases														
y O2 wet	-	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000
y CO2 wet	-	0,128	0,130	0,130	0,130	0,129	0,128	0,127	0,127	0,128	0,130	0,130	0,130	0,131
y CO wet	-	0,005	0,005	0,006	0,007	0,008	0,010	0,009	0,007	0,005	0,006	0,005	0,005	0,005
y H2 wet	-	0,001	0,001	0,002	0,002	0,002	0,003	0,002	0,002	0,001	0,002	0,001	0,001	0,001
y H2O wet	-	0,123	0,125	0,125	0,125	0,125	0,125	0,124	0,123	0,123	0,124	0,124	0,125	0,125
y N2 wet	-	0,721	0,732	0,732	0,733	0,732	0,732	0,726	0,722	0,720	0,730	0,731	0,732	0,732
sum yi	-	0,978	0,993	0,993	0,996	0,995	0,998	0,989	0,982	0,978	0,992	0,992	0,994	0,993
Specific heat in the exhaust gases														
Differential temperature gases atmosphere, tesc-ta	ºC	273	421	454	464	449	498	510	522	535	546	545	541	546
Average temperature of gases and environment, (tesc+ta)/2	ºC	164	239	255	260	252	277	283	289	296	301	301	299	301
cp_esc,w	kJ/kg K	1,2	1,2	1,2	1,2	1,2	1,2	2,2	3,2	4,2	5,2	6,2	7,2	8,2

Energy balance														
Thermal Power of the combustion, m_f H_f	kW	9,77	10,02	15,62	15,05	15,20	17,56	19,95	3,11	11,59	14,32	20,37	23,17	19,77
	%	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00
Actual engine Power, We	kW	3,41	4,02	4,54	4,77	5,07	5,26	5,35	5,25	4,97	4,87	4,61	4,25	4,03
	%	0,35	0,40	0,29	0,32	0,33	0,30	0,27	1,69	0,43	0,34	0,23	0,18	0,20
Sensible heat of exhaust gases, H_esc,w	kW	1,11	1,76	2,95	2,89	2,82	3,58	7,64	1,78	9,01	14,07	24,00	31,45	30,88
	%	0,11	0,18	0,19	0,19	0,19	0,20	0,38	0,57	0,78	0,98	1,18	1,36	1,56
Energy due to the unburned content, H_inq	kW	0,23	0,23	0,38	0,45	0,51	0,75	0,74	0,10	0,25	0,37	0,42	0,50	0,39
	%	0,02	0,02	0,02	0,03	0,03	0,04	0,04	0,03	0,02	0,03	0,02	0,02	0,02
Heat dissipated in cooling system, Q_ref+Q_lub	kW	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
	%	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
Other energy flows, E_otras	kW	5,03	4,01	7,75	6,94	6,81	7,97	6,22	-4,02	-2,65	-5,00	-8,66	-13,03	-15,53
	%	0,51	0,40	0,50	0,46	0,45	0,45	0,31	-1,29	-0,23	-0,35	-0,43	-0,56	-0,79

20 % of wide open throttle

Engine data		
Number of cylinders, z	-	1
Volumen desplazado por cilindro, V_cil	cm^3	171,26
Thermodynamic cycle per revolution, i	-	0,5
Fuel consumption		
number of C atoms, alfa	-	1
number of H atoms, beta	-	1,861
number of O atoms, gamma	-	0,016
number of N atoms, delta	-	0
molecular weight, Mf	$kg_f/kmol_f$	17,08
Air Fuel Ratio stoiciometric, AFR_stq	kg_a/kg_f	14,25
lower heating value, LHV	MJ/kg	43,0

Measured parameters													
point	-	1	2	3	4	5	6	7	8	9	10	11	12
Throttle opening	grados	20	20	20	20	20	20	20	20	20	20	20	20
Engine speed, torque, fuel consumption													
Engine speed, n	rpm	3.001	3.749	4.497	5.245	6.003	6.502	7.001	7.499	7.998	8.495	8.995	9.393
Actual Torque, T_e	Nm	7,13	6,05	5,66	5,21	4,57	4,29	3,74	3,38	2,85	2,45	1,75	0,75
Fuel consumption, m_f	kg/h	0,6920	0,9190	0,9500	0,8700	0,9400	0,8400	0,9300	0,4650	1,3800	1,2900	0,9900	1,2900
Environmental conditions of the trial cell													
barometric pressure, p_cell	mbar	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1.022	1.022
Temperature, t_cell	ºC	28	28	28	28	28	28	28	28	28	28	28	28
Humidity relative, phi_rel	%	47	47	47	47	47	47	47	47	47	47	47	47
Data for obtaining the intake air mass flow													
Nozzle discharge coefficient	-	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93	0,93
Differential pressure, Dp_atm-dep	mbar												
Intake and exhaust pressure													
Intake manifold absolute pressure, Dp_atm-adm	mbar	-81	-107	-144	-168	-197	-207	-231	-252	-274	-291	-283	-293
Exhaust manifold absolute pressure, Dp_esc-atm	mbar	7,98	6,43	5,30	5,62	4,15	3,97	2,78	2,75	0,11	0,34	0,41	2,61
Intake temperature	ºC	40,43	41,45	43,34	44,23	45,07	45,88	46,75	47,60	48,26	49,08	48,62	49,75
		313,58	314,60	316,49	317,38	318,22	319,03	319,90	320,75	321,41	322,23	321,77	322,90

Exhaust gas analysis results (HC < 100 ppm and NOx < 1000 ppm)													
Exhuast gases temperature, t_esc	ºC	385	429	462	475	503	524	552	568	591	612	594	649
Vol. Correction of O2 in the dry exhaust gases, O2_D	%	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
Vol. Correction of CO2 in the dry exhaust gases, CO2_D	%	13,94	13,93	14,05	14,26	14,14	14,13	14,35	14,29	14,44	14,47	14,38	14,76
Vol. Correction of CO in the dry exhaust gases, CO_D	%	1,02	0,50	0,32	0,65	0,87	0,79	0,59	0,77	0,67	0,77	0,93	0,30
Vol. Correction of HC in the dry exhaust gases, HC_D	ppm C6	143	291	220	126	97	94	69	86	65	43	30	7
Vol. relation of NOx in the dry exhaust gases, NOx_D	ppm NO+NO2	0	0	0	0	0	0						
Dilution factor, DF		1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00
measured lambda HORIBA	-	0,99	1,02	1,05	1,01	1,00	1,01	1,02	1,01	1,01	1,01	0,99	1,03
Vol. relation of O2 in the dry exhaust gases, O2_D	%	1,44	3,89	1,55	1,35	1,31	1,45	1,46	1,36	1,18	0,96	3,23	0,93
Vol. relation of CO2 in the dry exhaust gases, CO2_D	%	13,94	13,93	14,05	14,26	14,14	14,13	14,35	14,29	14,44	14,47	14,38	14,76
Vol. relation of CO in the dry exhaust gases, CO_D	%	1,02	0,50	0,32	0,65	0,87	0,79	0,59	0,77	0,67	0,77	0,93	0,30
Vol. relation of HC in the dry exhaust gases, HC_D	ppm C6	143	291	220	126	97	94	69	86	65	43	30	7
Temperature before catalyst	ºC	104,13	131,58	169,40	187,60	206,85	223,83	246,10	258,60	275,67	297,32	230,90	292,50
Temperature after catalyst	ºC	71,16	89,38	122,56	140,86	155,30	171,57	185,45	197,84	210,57	222,35	168,77	211,70

Calculated parameters													
point	-	1	2	3	4	5	6	7	8	9	10	11	12
engine speedr, n	rpm	3001	3749	4497	5245	6003	6502	7001	7499	7998	8495	8995	9393
actual Torque, actual Power, mean effective pressure, specific fuel consumption													
actual Torque, M_e	Nm	7,13	6,05	5,66	5,21	4,57	4,29	3,74	3,38	2,85	2,45	1,75	0,75
actual Power, W_e	kW	2,2	2,4	2,7	2,9	2,9	2,9	2,7	2,7	2,4	2,2	1,6	0,7
mean effective pressure, pme	kPa	523,3	443,6	415,5	382,2	335,0	314,5	274,5	247,9	208,9	180,0	128,1	55,1
specific fuel consumption, g_ef	g/kWh	308,8	387,2	356,3	304,1	327,5	287,9	339,1	175,3	578,7	591,1	601,8	1746,6
Power and Torque corrected													
Saturated steam pressure, p_s,ens (kPa)	mbar	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8	37,8
Dry air pressure, p_ens-phi*p_s,ens (kPa)	mbar	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2	1004,2
Factor de corrección, alfa-a	-	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99	0,99
Power corrtd, W_e,corr	kW	2,2	2,3	2,6	2,8	2,8	2,9	2,7	2,6	2,4	2,2	1,6	0,7
Torque corrct, M_e,corr	Nm	7,1	6,0	5,6	5,2	4,5	4,2	3,7	3,3	2,8	2,4	1,7	0,7

Relative dosage, volumetric efficiency and emissions (starting from the gases analysis and fuel consumption)													
AFR, lambda and relative dosage													
AFR with Brettschneider method													
K balance constant of the reaction H2O gases	-	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5	3,5
carbon atoms in the selected HC molecule	-	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0	6,0
(beta/alpha)/4	-	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653	0,4653
(gamma/alpha)/2	-	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040	0,0040
1+((beta/alpha)/4)-((gamma/alpha)/2)	-	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613	1,4613
Lambda Brettschneider	-	0,96	0,97	0,98	0,98	0,97	0,97	0,98	0,97	0,98	0,98	0,97	0,99
relative dosage Brettschneider	-	1,04	1,03	1,02	1,02	1,03	1,03	1,02	1,03	1,02	1,02	1,03	1,01
AFR Brettschneider	kg_a/kg_f	13,75	13,87	13,99	13,91	13,84	13,87	13,97	13,89	13,94	13,91	13,86	14,13
Fuel consumption, m_f	kg/h	-	-	-	-	-	-	-	-	-	-	-	-
specific fuel consumption, g_ef	g/kWh	-	-	-	-	-	-	-	-	-	-	-	-
Volumetric Efficiency													
air fuel rate from lambda, m_a	kg/s	0,0026	0,0035	0,0037	0,0034	0,0036	0,0032	0,0036	0,0018	0,0053	0,0050	0,0038	0,0051
Volumetric Efficiency, nhu_v,ens	-	0,522	0,560	0,486	0,380	0,357	0,295	0,305	0,142	0,396	0,348	0,251	0,319
Volumetric Efficiency, nhu_v,adm for air density in intake manifold	-	0,590	0,653	0,595	0,479	0,467	0,392	0,419	0,200	0,577	0,520	0,371	0,480
Composition of exhaust gases and emissions													
[H2O] wet	%	12,0	11,7	11,7	12,0	12,1	12,0	12,1	12,1	12,2	12,3	12,3	12,2
[O2] wet	%	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0
[CO2] wet	%	12,3	12,3	12,4	12,5	12,4	12,4	12,6	12,6	12,7	12,7	12,6	13,0
[CO] wet	%	0,9	0,4	0,3	0,6	0,8	0,7	0,5	0,7	0,6	0,7	0,8	0,3
[HC1] wet	%	0,1	0,2	0,1	0,1	0,1	0,0	0,0	0,0	0,0	0,0	0,0	0,0
[NO2] wet	%	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0	0,0
[H2] wet	%	0,3	0,1	0,1	0,2	0,2	0,2	0,1	0,2	0,2	0,2	0,2	0,1
B1	-	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457	1,457
B2	-	7,553	7,757	7,812	7,587	7,547	7,590	7,592	7,531	7,517	7,465	7,436	7,563
B3	-	1,037	1,028	1,019	1,025	1,030	1,028	1,020	1,026	1,022	1,024	1,029	1,009
[N2] wet	%	70,2	69,0	69,1	70,7	70,7	70,5	71,0	71,1	71,5	71,9	71,9	72,0
Summation [i]	%	95,7	93,7	93,7	96,1	96,3	95,9	96,4	96,7	97,2	97,7	97,8	97,6
Molecular weight of the exhaust gases, M_esc,w	kg/kmol	27,55	27,10	27,10	27,72	27,72	27,63	27,79	27,87	28,01	28,15	28,15	28,17
mass flow rate of damp fumes m_esc,w	g/h	10206	13666	14237	12973	13948	12491	13921	6923	20618	19240	14709	19512

Composition and specific heat of exhaust gases													
Composition of exhaust gases													
y O2 wet	-	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000	0,000
y CO2 wet	-	0,123	0,123	0,124	0,125	0,124	0,124	0,126	0,126	0,127	0,127	0,126	0,130
y CO wet	-	0,009	0,004	0,003	0,006	0,008	0,007	0,005	0,007	0,006	0,007	0,008	0,003
y H2 wet	-	0,003	0,001	0,001	0,002	0,002	0,002	0,001	0,002	0,002	0,002	0,002	0,001
y H2O wet	-	0,120	0,117	0,117	0,120	0,121	0,120	0,121	0,121	0,122	0,123	0,123	0,122
y N2 wet	-	0,702	0,690	0,691	0,707	0,707	0,705	0,710	0,711	0,715	0,719	0,719	0,720
sum yi	-	0,956	0,936	0,936	0,960	0,962	0,958	0,963	0,967	0,972	0,977	0,978	0,976
Specific heat in the exhaust gases													
Differential temperature gases atmosphere, tesc-ta	ºC	357	401	434	447	475	496	524	540	563	584	566	621
Average temperature of gases and environment, (tesc+ta)/2	ºC	207	228	245	252	266	276	290	298	309	320	311	339
cp_esc,w	kJ/kg K	1,2	1,2	1,2	1,2	1,2	1,2	2,2	3,2	4,2	5,2	6,2	7,2

Energy balance													
Thermal Power of the combustion, m_f H_f	kW	8,27	10,98	11,35	10,39	11,23	10,03	11,11	5,55	16,48	15,41	11,83	15,41
	%	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,00
Actual engine Power, We	kW	2,24	2,37	2,67	2,86	2,87	2,92	2,74	2,65	2,38	2,18	1,65	0,74
	%	0,27	0,22	0,23	0,28	0,26	0,29	0,25	0,48	0,14	0,14	0,14	0,05
Sensible heat of exhaust gases, H_esc,w	kW	1,22	1,83	2,06	1,93	2,21	2,07	4,46	3,32	13,54	16,22	14,33	24,25
	%	0,15	0,17	0,18	0,19	0,20	0,21	0,40	0,60	0,82	1,05	1,21	1,57
Energy due to the unburned content, H_inq	kW	0,34	0,22	0,15	0,27	0,39	0,32	0,26	0,17	0,43	0,46	0,43	0,18
	%	0,04	0,02	0,01	0,03	0,03	0,03	0,02	0,03	0,03	0,03	0,04	0,01
Heat dissipated in cooling system, Q_ref+Q_lub	kW	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
	%	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00	0,00
Other energy flows, E_otras	kW	4,47	6,55	6,47	5,33	5,76	4,73	3,65	-0,59	0,12	-3,46	-4,58	-9,76
	%	0,54	0,60	0,57	0,51	0,51	0,47	0,33	-0,11	0,01	-0,22	-0,39	-0,63

1.5 Different type of trial for 125 cm^3 in WOT

test bench with conventional filter							
rpm	3500	4000	4250	4500	4750	5000	5250
power [kW]	3,15	3,71	3,97	4,18	4,43	4,68	4,91
torque on the wheel [Nm]	85,6	88,4	88,9	88,6	88,8	89,2	89,0
engine torque [Nm]	8,58	8,86	8,91	8,88	8,90	8,94	8,92
rpm	5500	6000	6500	7000	7500	8000	9000
power [kW]	5,12	5,70	6,43	7,11	7,59	7,81	7,61
torque on the wheel [Nm]	88,7	90,5	94,1	96,7	96,3	93,0	80,6
engine torque [Nm]	8,89	9,07	9,43	9,69	9,65	9,32	8,08

test bench with conventional filter							
rpm	3500	4000	4250	4500	4750	5000	5250
power [kW]	3,25	3,78	4,01	4,15	4,31	4,49	4,74
torque on the wheel [Nm]	88,6	90,1	90,2	87,8	85,6	85,5	86,1
engine torque [Nm]	8,88	9,03	9,04	8,80	8,58	8,57	8,63
rpm	5500	6000	6500	7000	7500	8000	9000
power [kW]	5,00	5,53	6,12	6,59	6,79	6,58	6,08
torque on the wheel [Nm]	86,7	87,8	89,6	89,8	86,1	78,5	64,4
engine torque [Nm]	8,69	8,80	8,98	9,00	8,63	7,87	6,46

mechanical losses							
rpm	3500	4000	4250	4500	4750	5000	5250
power [kW]	0,90	1,05	1,13	1,22	1,35	1,47	1,61
torque on the wheel [Nm]	24,5	25,1	25,4	26,9	27,0	28,0	29,2
engine torque [Nm]	2,45	2,51	2,54	2,69	2,71	2,81	2,93
rpm	5500	6000	6500	7000	7500	8000	-
power [kW]	1,76	2,00	2,16	2,39	2,68	3,10	-
torque on the wheel [Nm]	30,4	31,7	31,7	32,5	34,0	35,0	-
engine torque [Nm]	3,05	3,18	3,17	3,26	3,41	3,51	-

1.6 Different type of trial for 170 cm^3 in WOT

test bench with the nozzle-deposit system in intake										
rpm	2998	3497	3747	3996	4245	4495	4744	4994	5243	5493
power [kW]	4,07	4,90	5,34	5,62	6,01	6,33	6,70	7,01	7,23	7,45
torque on the wheel [Nm]	129,20	133,47	135,64	134,06	134,95	134,12	134,60	133,74	131,35	129,13
engine torque [Nm]	12,95	13,38	13,60	13,44	13,53	13,45	13,49	13,41	13,17	12,95
rpm	6001	6500	6999	7498	7996	8495	8993	9242	9492	9741
power [kW]	8,01	8,84	9,72	10,18	10,45	10,30	9,83	9,33	9,14	9,05
torque on the wheel [Nm]	127,13	129,60	132,32	129,36	124,54	115,45	104,16	96,14	91,76	88,53
engine torque [Nm]	12,74	12,99	13,27	12,97	12,49	11,57	10,44	9,64	9,20	8,88

testh bench with conventional filter							
rpm	2997	3745	4493	5241	5999	6498	6997
power [kW]	3,64	4,83	6,02	7,25	8,46	9,33	9,93
torque on the wheel [Nm]	115,63	122,77	127,73	131,69	134,41	136,71	135,19
engine torque [Nm]	11,59	12,31	12,81	13,20	13,47	13,71	13,55
rpm	7496	7994	8494	8993	9392	9741	-
power [kW]	10,25	10,46	10,36	10,09	9,50	8,94	-
torque on the wheel [Nm]	130,27	124,68	116,14	106,87	96,38	87,43	-
engine torque [Nm]	13,06	12,50	11,64	10,71	9,66	8,76	-

APPENDIX 2

2.1 MAP mapping for 170 cm^3

	20	40	50	60	80	100
160.535	1	1	1	1	1	1.0232
314.159	0.9411	1.0091	1.013	1.0168	1.0214	1.022
340.339	1	1	1	1	1	1.022
366.519	1	1	1	1	1	1.022
392.699	0.9155	1.0058	1.0102	1.0149	1.0213	1.022
418.879	1	1	1	1	1	1.0209
445.059	1	1	1	1	1	1.0199
471.239	0.8785	0.9992	1.0052	1.0116	1.0164	1.0188
497.419	1	1	1	1	1	1.0176
523.599	1	1	1	1	1	1.0164
549.779	0.8544	0.9958	1.0018	1.0082	1.0144	1.0152
575.959	1	1	1	1	1	1.0145
628.319	0.8246	0.9956	1.0024	1.0064	1.012	1.0132
680.678	0.8147	0.9911	0.998	1.0032	1.01	1.0112
733.038	0.7915	0.9866	0.9947	0.999	1.0067	1.0081
785.398	0.7697	0.9829	0.9909	0.9956	1.004	1.0062
837.758	0.7481	0.9794	0.9877	0.9947	1.0029	1.0051
890.118	0.7308	0.9759	0.9845	0.9934	1.0026	1.0056
942.478	0.739	0.9714	0.9803	0.9897	0.9988	1.0038
968.658	1	1	1	1	1	1.0033
994.838	0.729	0.9675	0.977	0.9868	0.9964	1.0028
1021.018	0.7203	0.965	0.9749	0.9846	0.995	1.0015

2.2 MATLAB script for 125 cm^3

```
Vd=124.7*1e-6;
Cpump=(10^8)/(287*4*pi);
InjFlowRate=1.1812*1e-6;
InjTurnON=1500;
BattComp=0.8;
MAPVect=[1 1 1 1 1 1 1 1 1 1 1 1 1 1 1 1 1.015;0.9563 1
0.9832 1 1 1 1 1 1 1 0.9964 1 1.0046 1 1 1 1.015;1 1 1 1
1 1 1 1 1 1 1 1 1 1 1 1 1.016;1 0.9585 1 1 1 0.9895 1 1
1.0007 1 1 1 1 1.0087 1 1 1.02;1 1 1 1 1 1 1 1 1 1 1 1
1 1 1 1.02;1 1 1 1 1 1 1 1 1 1 1 1 1 1 1 1 1.016;1 1 1
0.9585 1 1 1 0.9895 1 1 1 0.9997 1 1 1 1.009 1.013;1 1 1
1 1 1 1 1 1 1 1 1 1 1 1 1.01;1 1 1 1 1 1 1 1 1 1 1 1
1 1 1 1.009;1 1 1 1 0.9491 1 0.9819 1 1 0.997 1 1 1 1
1.0037 1 1.008;1 1 1 1 1 1 1 1 1 1 1 1 1 1 1 1 1.005;1 1
1 1 1 1 1 1 1 1 1 1 1 1 1 1 1.008];
PressVect1=[0.9491 0.9563 0.9585 0.9819 0.9832 0.9895
0.9964 0.997 0.9997 1.0007 1.0037 1.0046 1.005 1.008
1.0087 1.009 1.01 1.013 1.015 1.016 1.02];
SpeedVect1=[366.519 418.879 445.059 471.239 497.419 523.6
549.779 575.959 628.319 680.678 733.038 785.398];
```

```
ThrotVect1=[21.2 25.2 27.9 29.8 31.4 33 37.5 37.9 42.4
42.5 43.3 48.1 54.4 57.4 59.6 61.6 100];
EstMAPVect=[0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0;0 0 0 0 0 0
0 0 0 0 0 0 0 0 0 0 0;0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0;0
0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0;0 0 0 0 0 0 0 0 0 0 0 0 0
0 0 0 0;0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0;0 0 0 0 0 0 0 0
0 0 0 0 0 0 0 0 0;0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0;0 0 0
0 0 0 0 0 0 0 0 0 0 0 0 0 0;0 0 0 0 0 0 0 0 0 0 0 0 0 0 0
0 0;0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0;0 0 0 0 0 0 0 0 0 0
0 0 0 0 0 0 0];
EstSpeedVect=[0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0;0,0,0,0,0
,0,0,0,0,0,0,0,0,0,0,0,0;0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,
0;0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0;0,0,0,0,0,0,0,0,0,0,0
,0,0,0,0,0,0;0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0;0,0,0,0,0,
0,0,0,0,0,0,0,0,0,0,0,0;0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0
;0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0;0,0,0,0,0,0,0,0,0,0,0,
0,0,0,0,0,0;0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0;0,0,0,0,0,0
,0,0,0,0,0,0,0,0,0,0,0;0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0;
0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0;0,0,0,0,0,0,0,0,0,0,0,0
,0,0,0,0,0;0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0;0,0,0,0,0,0,
0,0,0,0,0,0,0,0,0,0,0;0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0;0
,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0;0,0,0,0,0,0,0,0,0,0,0,0,
0,0,0,0,0;0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0];
EstThrotVect=[0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0;0
0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0;0 0 0 0 0 0 0 0 0
0 0 0 0 0 0 0 0 0 0 0 0;0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0
0 0 0 0;0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0;0 0 0 0
0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0;0 0 0 0 0 0 0 0 0 0 0 0
0 0 0 0 0 0 0 0 0;0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0
0;0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0;0 0 0 0 0 0 0
0 0 0 0 0 0 0 0 0 0 0 0 0 0;0 0 0 0 0 0 0 0 0 0 0 0 0 0 0
0 0 0 0 0 0;0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0];
VE=[0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0.57893 0 0;0
0.40489 0 0 0.50636 0 0.58379 0 0 0 0 0.53853 0 0 0 0 0 0
0.45899 0 0;0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0.66555
0;0 0 0.39744 0 0 0.45706 0 0 0 0.56493 0 0 0 0 0.62239 0
0 0 0 0 0.63749;0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0
0.76863;0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0.83581 0;0
0 0.37878 0 0 0.53669 0 0 0.63129 0 0 0 0 0 0 0.67276 0
0.71907 0 0 0;0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0.70853 0 0
0 0;0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0.76588 0 0 0 0
0;0.42044 0 0 0.64809 0 0 0 0.82301 0 0 0.87664 0 0
0.75035 0 0 0 0 0 0 0;0 0 0 0 0 0 0 0 0 0 0 0 0.73514 0 0
0 0 0 0 0 0;0 0 0 0 0 0 0 0 0 0 0 0 0 0.79251 0 0 0 0 0 0
0];
Lambda=[0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0.90919 0 0;0
0.92281 0 0 0.85395 0 0.83911 0 0 0 0 0.9642 0 0 0 0 0 0
0.91056 0 0;0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0.90384
0;0 0 0.98488 0 0 0.84186 0 0 0 0.83906 0 0 0 0 0.89251 0
0 0 0 0 0.90732;0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0
0.89726;0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0.90336 0;0
```

```
0 0.87808 0 0 0.84654 0 0 0.84638 0 0 0 0 0 0 0.8918 0
0.90386 0 0 0;0 0 0 0 0 0 0 0 0 0 0 0 0 0 0.8973 0 0
0 0;0 0 0 0 0 0 0 0 0 0 0 0 0 0 0.88839 0 0 0 0
0;0.92866 0 0 0.87378 0 0 0 0.85116 0 0 0.85161 0 0
0.88253 0 0 0 0 0 0 0;0 0 0 0 0 0 0 0 0 0 0.88012 0 0
0 0 0 0 0 0;0 0 0 0 0 0 0 0 0 0 0 0.84946 0 0 0 0 0 0
0];
Tadmission=[0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 312.69 0
0;0 324.58 0 0 322.45 0 320 0 0 0 0 317.02 0 0 0 0 0 0
312.36 0 0;0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 312.65
0;0 0 324.52 0 0 322.16 0 0 0 319.56 0 0 0 0 316.2 0 0 0
0 0 313.24;0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0
313.52;0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 313.15 0;0 0
321.88 0 0 319.6 0 0 317.41 0 0 0 0 0 0 314.97 0 313.64 0
0 0;0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 313.56 0 0 0 0;0 0 0
0 0 0 0 0 0 0 0 0 0 0 314.09 0 0 0 0 0;322.58 0 0
321.13 0 0 0 322.09 0 0 319.48 0 0 314.58 0 0 0 0 0 0 0;0
0 0 0 0 0 0 0 0 0 0 0 315.62 0 0 0 0 0 0 0 0;0 0 0 0 0 0
0 0 0 0 0 0 0 316.5 0 0 0 0 0 0 0];
InjFlowRateTAB=[0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0.986
515501000000,0,0;0,1.07960650100000,0,0,1.15837803700000,
0,1.12795115500000,0,0,0,0,0.868218833000000,0,0,0,0,0,0,
0.750655222000000,0,0;0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0
,0,1.11087962500000,0;0,0,0.987487904000000,0,0,1.0439809
3900000,0,0,0,1.11214998300000,0,0,0,0,1.09137697800000,0
,0,0,0,0,1.05235428800000;0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0
,0,0,0,0,1.28963527400000;0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0
,0,0,0,1.39061933400000,0;0,0,1.02230633200000,0,0,1.1942
9762000000,0,0,1.23749059300000,0,0,0,0,0,0,1.17898630700
000,0,1.22069264200000,0,0,0;0,0,0,0,0,0,0,0,0,0,0,0,0,0,
0,0,1.20615014700000,0,0,0,0;0,0,0,0,0,0,0,0,0,0,0,0,0,0,
0,1.29408097300000,0,0,0,0,0;1.22678257900000,0,0,1.48550
762000000,0,0,0,1.59398075000000,0,0,0.955104055000000,0,
0,1.25044928400000,0,0,0,0,0,0,0;0,0,0,0,0,0,0,0,0,0,0,0,
1.16737191400000,0,0,0,0,0,0,0,0;0,0,0,0,0,0,0,0,0,0,0,0,
0,1.28747166800000,0,0,0,0,0,0,0];
RampRateKiX=[100 200 300 400 500 600];
RampRateKiY=[0 0.2 0.4 0.6 0.8 1];
RampRateKiZ=[0.012 0.024 0.036 0.048 0.06 0.072;0.024
0.048 0.072 0.096 0.12 0.144;0.036 0.072 0.108 0.144 0.18
0.216;0.048 0.096 0.144 0.192 0.24 0.288;0.06 0.12 0.18
0.24 0.3 0.36;0.072 0.144 0.216 0.288 0.36 0.432];
ego_sw=1;
enginespeed=785.398;
throttle_time=[0 10 20];
throttle_opening=[100 100 100];
hys=25;
map_sw=1;
max_ego=1.2;
max_press1=10;
max_speed1=1000;
```

```
max_throt1=100;
min_press=0.05;
min_speed=50;
min_throt=3;
speed_sw=1;
st_range=0.0001;
throttle_sw=1;
zero_thresh=250;
```

2.2 MATLAB script for 170 cm^3

```
Vd=(((pi*61^2)/4)*58.6)*1e-9;
Cpump=(10^8)/(287*4*pi);
InjFlowRate=1.386851206*1e-6;
InjTurnON=1500;
BattComp=1.2807;
MAPVect=[1,1,1,1,1,1.02320000000000;0.941100000000000,1.0
0910000000000,1.01300000000000,1.01680000000000,1.0214000
0000000,1.02200000000000;1,1,1,1,1,1.02200000000000;1,1,1
,1,1,1.02200000000000;0.915500000000000,1.00580000000000,
1.01020000000000,1.01490000000000,1.02130000000000,1.0220
0000000000;1,1,1,1,1,1.02090000000000;1,1,1,1,1,1.0199000
0000000;0.878500000000000,0.999200000000000,1.00520000000
000,1.01160000000000,1.01640000000000,1.01880000000000;1,
1,1,1,1,1.01760000000000;1,1,1,1,1,1.01640000000000;0.854
400000000000,0.995800000000000,1.00180000000000,1.0082000
0000000,1.01440000000000,1.01520000000000;1,1,1,1,1,1.014
50000000000;0.824600000000000,0.995600000000000,1.0024000
0000000,1.00640000000000,1.01200000000000,1.0132000000000
0;0.814700000000000,0.991100000000000,0.998000000000000,1
.00320000000000,1.01000000000000,1.01120000000000;0.79150
0000000000,0.986600000000000,0.994700000000000,0.99900000
0000000,1.00670000000000,1.00810000000000;0.7697000000000
00,0.982900000000000,0.990900000000000,0.995600000000000,
1.00400000000000,1.00620000000000;0.748100000000000,0.979
400000000000,0.987700000000000,0.994700000000000,1.002900
00000000,1.00510000000000;0.730800000000000,0.97590000000
0000,0.984500000000000,0.993400000000000,1.00260000000000
,1.00560000000000;0.739000000000000,0.971400000000000,0.9
80300000000000,0.989700000000000,0.998800000000000,1.0038
0000000000;1,1,1,1,1,1.00330000000000;0.729000000000000,0
.967500000000000,0.977000000000000,0.986800000000000,0.99
6400000000000,1.00280000000000;0.720300000000000,0.965000
000000000,0.974900000000000,0.984600000000000,0.995000000
000000,1.00150000000000];
PressVect1=[0.720300000000000,0.729000000000000,0.7308000
00000000,0.739000000000000,0.748100000000000,0.7697000000
00000,0.791500000000000,0.814700000000000,0.8246000000000
00,0.854400000000000,0.878500000000000,0.915500000000000,
0.941100000000000,0.965000000000000,0.967500000000000,0.9
```

```
714000000000000,0.974900000000000,0.975900000000000,0.9770
00000000000,0.979400000000000,0.980300000000000,0.9829000
00000000,0.984500000000000,0.984600000000000,0.9866000000
00000,0.986800000000000,0.987700000000000,0.9897000000000
00,0.990900000000000,0.991100000000000,0.993400000000000,
0.994700000000000,0.995000000000000,0.995600000000000,0.9
95800000000000,0.996400000000000,0.998000000000000,0.9988
00000000000,0.999000000000000,0.999200000000000,1.0015000
0000000,1.00180000000000,1.00240000000000,1.0026000000000
0,1.00280000000000,1.00290000000000,1.00320000000000,1.00
330000000000,1.00380000000000,1.00400000000000,1.00510000
000000,1.00520000000000,1.00560000000000,1.00580000000000
,1.00620000000000,1.00640000000000,1.00670000000000,1.008
10000000000,1.00820000000000,1.00910000000000,1.010000000
00000,1.01020000000000,1.01120000000000,1.01160000000000,
1.01200000000000,1.01300000000000,1.01320000000000,1.0144
0000000000,1.01450000000000,1.01490000000000,1.0152000000
0000,1.01640000000000,1.01680000000000,1.01760000000000,1
.01880000000000,1.01990000000000,1.02090000000000,1.02130
000000000,1.02140000000000,1.02200000000000,1.02320000000
000];
SpeedVect1=[160.535 314.159 340.339 366.519 392.699
418.879 445.059 471.239 497.419 523.599 549.779 575.959
628.319 680.678 733.038 785.398 837.758 890.118 942.478
968.658 994.838 1021.018];
ThrotVect1=[20 40 50 60 80 100];
EstMAPVect=[0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0
,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,
0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0
,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;
0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0];
EstSpeedVect=[0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0
,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,
0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0
,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,
0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0
,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,
0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0
,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,
0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0
,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,
0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0
,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,
0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0
,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,
0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0
,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,
0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0,0,0;0,0,0,0
,0,0;0,0,0,0,0,0];
EstThrotVect=[0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0
,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,
```

0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0
,0,0;0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,
0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0
,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0;0,0,
0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0
,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,
0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0;0,0,0,0,0,0,0
,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,
0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0
,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0;0,0,0,0,0,0,0,0,0,0,0,
0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0
,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,
0,0,0,0,0,0,0,0,0,0,0,0,0;0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0
,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,
0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0
,0,0,0,0,0,0,0,0;0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,
0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0
,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,
0,0,0,0;0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0
,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,
0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0;0
,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,
0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0
,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0;0,0,0,0,0,
0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0
,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,
0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0;0,0,0,0,0,0,0,0,0,0
,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,
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0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,1.35134170000000,0,
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0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,1.33317390000000,0,0,
0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0,0
,0,0,0,0,0,0,0,0,0];
RampRateKiX=[100 200 300 400 500 600];
RampRateKiY=[0 0.2 0.4 0.6 0.8 1];
RampRateKiZ=[0.012 0.024 0.036 0.048 0.06 0.072;0.024
0.048 0.072 0.096 0.12 0.144;0.036 0.072 0.108 0.144 0.18
0.216;0.048 0.096 0.144 0.192 0.24 0.288;0.06 0.12 0.18
0.24 0.3 0.36;0.072 0.144 0.216 0.288 0.36 0.432];
ego_sw=1;
enginespeed=994.838;
throttle_time=[0 10 20];
throttle_opening=[100 100 100];
hys=25;
map_sw=1;
max_ego=1.2;
max_press1=10;
max_speed1=1500;
max_throt1=100;
```

```
min_press=0.05;
min_speed=50;
min_throt=3;
speed_sw=1;
st_range=0.0001;
throttle_sw=1;
zero_thresh=250;
```

2.3 Injection timing with IFR not constant for 125 cm^3 in full load

Throttle %	rpm	experimental analysis			model results			Error %		
		PW	Fuel rate	Air flow R	PW	Fuel rate	Air flow R			
		ms	g/s	g/s	ms	g/s	g/s	PW	Fuel rate	Air flow R
100	3500	7,683	0,192	2,490	7,394	0,184	2,382	3,77	4,34	4,55
100	4000	7,953	0,174	2,256	7,659	0,167	2,160	3,70	4,22	4,44
100	4250	7,863	0,270	3,476	7,573	0,258	3,328	3,69	4,23	4,43
100	4500	7,913	0,273	3,525	7,634	0,262	3,382	3,52	4,03	4,22
100	4750	7,878	0,351	4,486	7,595	0,336	4,301	3,59	4,11	4,31
100	5000	7,889	0,399	5,135	7,587	0,381	4,909	3,82	4,37	4,60
100	5250	7,748	0,360	4,639	7,424	0,343	4,415	4,19	4,80	5,07
100	5500	7,778	0,374	4,788	7,435	0,355	4,545	4,41	5,08	5,36
100	6000	7,898	0,446	5,647	7,531	0,422	5,345	4,64	5,32	5,64
100	6500	8,018	0,475	5,993	7,649	0,450	5,659	4,61	5,26	5,91
100	7000	8,408	0,504	6,323	7,952	0,473	5,933	5,42	6,15	6,58
100	7500	8,502	0,603	7,304	8,042	0,566	6,854	5,41	6,14	6,56

2.4 Injection timing with IFR not constant for 170 cm^3 in full load

Throttle %	rpm	experimental analysis			model results			Error %		
		PW	Fuel rate	Air flow R	PW	Fuel rate	Air flow R			
		ms	g/s	g/s	ms	g/s	g/s	PW	Fuel rate	Air flow R
100	3000	10,428	0,364	4,802	10,330	0,361	4,753	0,94	1,00	1,01
100	3250	10,486	0,396	5,274	10,371	0,392	5,206	1,10	1,22	1,29
100	3500	10,640	0,429	5,746	10,510	0,423	5,668	1,22	1,35	1,36
100	3750	10,568	0,448	6,099	10,436	0,442	6,013	1,25	1,38	1,40
100	4000	10,478	0,462	6,447	10,322	0,455	6,347	1,48	1,53	1,55
100	4250	10,387	0,482	6,580	10,229	0,474	6,468	1,52	1,68	1,70
100	4500	10,493	0,510	6,902	10,311	0,500	6,768	1,73	1,91	1,93
100	4750	10,588	0,539	7,189	10,379	0,528	7,031	1,98	2,18	2,20
100	5000	10,823	0,573	7,464	10,580	0,559	7,278	2,25	2,48	2,49
100	5250	10,567	0,588	7,748	10,303	0,572	7,533	2,50	2,76	2,78
100	5500	10,395	0,606	7,986	10,108	0,587	7,741	2,76	3,06	3,07
100	6000	10,274	0,641	8,472	9,965	0,620	8,187	3,01	3,33	3,35
100	6500	10,753	0,715	9,321	10,407	0,690	8,989	3,22	3,55	3,56
100	7000	11,209	0,796	10,281	10,807	0,765	9,875	3,58	3,94	3,95
100	7500	11,279	0,854	10,931	10,847	0,818	10,471	3,83	4,20	4,22
100	8000	10,823	0,858	11,307	10,435	0,824	10,857	3,59	3,96	3,98
100	8500	10,519	0,913	11,848	10,106	0,873	11,331	3,93	4,34	4,36
100	9000	10,030	0,932	12,234	9,630	0,891	11,689	3,99	4,44	4,46
100	9250	9,537	0,889	11,584	9,119	0,845	11,015	4,38	4,90	4,91
100	9500	9,087	0,866	11,325	8,657	0,820	10,720	4,73	5,32	5,34
100	9750	9,066	0,873	11,386	8,650	0,828	10,795	4,59	5,16	5,18

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