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STUDIUL EFECTELOR INJECȚIEI DE APĂ ASUPRA PERFORMANȚELOR MOTOARELOR CU ARDERE INTERNĂ

ASSESSMENT OF WATER INJECTION ON INTERNAL COMBUSTION ENGINES PERFORMANCES

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Contents

Abstract

The research activity included in the PhD thesis entitled “ASSESSMENT OF WATER INJECTION ON INTERNAL COMBUSTION ENGINES PERFORMANCES” was carried out over three years as follows: In the first year I conducted the study of scientific works in the field of water injection in internal combustion engines. The study was carried out at the Department of Thermal Systems and Road Vehicles at the University of “Dunărea de Jos” in Galati under the guidance of the entire coordination team. The second year was carried out at Minho University in Guimaraes, Portugal, where a water injection stand was developed, and experimental tests were performed. The numerical model was developed in the third year of the PhD at the University of “Dunărea de Jos” in Galati.

The current trend in engine development is to reduce fuel consumption and pollutant emissions, including GHG (greenhouse gases). Hybrid systems installed on vehicles equipped with spark ignition engines are becoming more and more used for urban traffic. Most hybrid vehicles are equipped with thermal engines with small engine capacity, enabling the lowering of the fuel consumption and GHG production.

The water injection in internal combustion engines has a primary role in improving the internal cooling of the engine. The need for internal engine cooling occurs when the engine is working at a high load, with a high spark ignition advance, which may lead to the appearance of knock. In addition to cases in where spark can be delayed or it is necessary to use a high-octane fuel, the water injection can also be used to reduce the maximum temperature at the end of the compression process.

To supress or reduce knock intensity it is necessary to reduce the compression ratio of the engine, reduce the spark ignition advance or it is necessary to use a fuel with a higher-octane number. Also, water injection in internal combustion engines has a primary role in internal cooling of the engine. As it reduces the maximum temperature at the end of the compression process.

There is plenty of water around us, it does not require special condition to store it and is not harmful to the environment. Water can also be collected from the engine exhaust or from the vehicle's HVAC (Heating, ventilation, and air conditioning) system. Current fuel injection systems (hardware and software) may be used to perform water injection.

But like any other technology, water injection has also disadvantaged. Depending on the geographical area in which the car is used, there is a risk that water will freeze, block, or even damage the water injection system. If the homogeneity of the air-fuel-water mixture is not well implemented, abnormalities may occur during the combustion process in the cylinder.

Most research on water injection aims at suppressing knock, to increase the brake power, and possibly to reduce brake specific fuel consumption (BSFC) but is usually performed only at very high loads (wide open throttle – WOT). However, there is no research on the negative effects of water and the impact of water injection at low engine speeds and loads on internal combustion engines.

The main objective of the PhD thesis is to continue research in the field of water injection in areas where there are still gaps in the literature. Thus, experimental research is needed to evaluate the impact of water injection in combustion engines, especially at medium and low loads. To cover as wide a range of tests as possible, experimental tests with water injection into the intake manifold and direct water injection into the cylinder will be performed. The experimental data should be taken as accurately as possible from the experimental stand, so that a mathematical model can be created, to predict the evolution of the combustion chamber pressure.

The objectives pursued in the thesis are:

- The analysis of the current state of water injection systems in internal combustion engines
- The development of an experimental stand for performing manifold water injection and in cylinder direct water injection tests
- Real time control and data acquisition
- Water injection impact on at medium and low loads on combustion and engine cycle
- Water injection impact on pollutant emissions
- 3D modelling of the experimental stand
- CFD analysis and in cylinder pressure prediction.

The development of the experimental stand requires first a study of the experimental works, books, even PhD theses in this field. Hence, the study of literature has divided them into two main categories: direct water injection in the engine cylinder and water injection in the intake manifold. In addition to the published experimental results, it is also followed the engines on which the tests are performed, their cylinder capacity and the way in which the determinations are made (load, speed, water/fuel ratio, ignition/injection advance, etc.)

The thesis is structured in four main chapters, as follows:

The first chapter is titled *Literature Review*, it presents the current state of the art in the field of water injection. It is structured according to the type of water injection (direct injection or injection in the intake manifold) with emphasis on the results obtained along with the way the determinations were made and the type of engine used.

The second chapter is entitled *Experimental part*. This is an extensive chapter describing how the experimental stand was developed, along with the equipment used for the adjustment and real-time data acquisition. Further on, the methods of working and analysing the raw data taken from the stand are presented. The chapter continues with the interpretation of the data taken from the stand for both direct water injection and water injection into the intake manifold. The chapter ends with the main conclusions of the experimental results.

The third chapter is entitled *Numerical Model*. In this chapter the numerical model for the prediction of cylinder pressure is presented. The chapter starts with the 3D modelling of the stand and continues with a series of comparative studies between different numerical models that are the basis for the creation of the numerical model for the prediction of the cylinder pressure evolution for the analysed engine. Finally, experimental data are compared with the data obtained from the simulation.

The fourth chapter is entitled *Conclusions, contributions, and recommendations*. This chapter presents the final conclusions of the PhD thesis, personal contributions to achieving the thesis objectives and future research guidelines in the field of water injection in internal combustion engines.

The uniqueness of the PhD thesis is primarily due to the type of engine chosen. At the time of the PhD thesis, no articles, or papers on water injection in side-valve spark ignition engines were found. Also, the uniqueness of the thesis is given by the test parameters, both for direct water injection and for the injection in the intake manifold.

Nomenclatures and Abbreviations

ε	Turbulent Energy Dissipation
μ_τ	Turbulent Viscosity
Gb	Floatability
k	Turbulence Kinetic Energy
p	Pressure
P_r	Prandtl Number
t	Time
V	Volume
YM	Compressibility
β	Thermal Expansion Coefficient
ε	Compress Ratio
λ	Stoichiometry
ρ	Density
ω	Specific Dissipation Rate Of Turbulent Frequency

0D	Zero Spatial Dimensions
3D	Three Spatial Dimensions
ATDC	After Top Dead Center
AVX	Advanced Vector Extensions
BSFC	Brake-Specific Fuel Consumption
BTDC	Before Top Dead Center
CAD	Crank Angle Degrease
CFD	Computational Fluid Dynamic
CLT	Coolant Liquid Temperature
CO	Carbon monoxide
CO2	Carbon dioxide
COV	Coefficient Of Variation
CPU	Central Processing Unit
DDES	Delayed Detached Eddy Simulation
DES	Detached Eddy Simulation
DNS	Direct Numerical Simulation
DOHC	Dual overhead cam
DWI	Direct Water Injection
ECU	Electronic Computer Unit
EEA	European Economic Area
EGR	Exhaust Gas Recirculation
EGT	Exhaust Gas Temperature
EOC	End of Combustion
FEM	Finite Element Method
HC	Hydrocarbon
HP	High Pressure
HVAC	Heating Ventilation and Air Conditioning
IAT	Intake Air Temperature

IMEP	Indicated Mean Effective Pressure
LES	Large Eddy Simulation
LLNL	Lawrence Livermore National Laboratory
LP	Low Pressure
MON	Motor Octane Number
NI	National Instruments
NOx	Nitrogen Oxides
OHC	Overhead Cam
OHV	Overhead Valve
PFI	Port Fuel Injection
PM	Particulate Matter
PV	Pressure Volume
PWI	Port Water Injection
RAM	Random Access Memory
RANS	Reynolds Averaged Navier Stokes
RMS	Root Mean Square
RNG	Re Normalisation Group
SI	Spark Ignition
SGS	Sub Grid Scale
SOC	Start Of Combustion
SST	Shear Stress Transport
SV	Side valve
TDC	Top Dead Center
TDP	Thermal Dissipation Power
TKE	Turbulent Kinetic Energy
VOCs	Volatile Organic Compounds
W/F	Water Fuel
WI	Water Injection
WOT	Wide Open Throttle

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Chapter 1 Literature Review

1.2. Approaches on water injection

The concept of water injection is not a new one. Analysing its evolution over time, it is noted that this concept was used when it was intended to increase engine performance. The research in this field in the public domain is rather limited. Among the first research carried out as I mentioned, are those on aircraft engines. In 1938, Kuhring [1] managed to achieve a 90 hp power increase for a supercharged aircraft engine, with a W/F ratio of 0.83. He noted a knock suppression and an improvement in engine performance starting with a W/F ratio of 0.3.

The experimental study published by James in 1950 [9] aimed to improve the performance of spark ignition engines by increasing the compression ratio, and achieving the injection of water-alcohol mixture to suppress knock. He is also the first researcher who makes a comparison of fuel with different octane numbers and water injection, further emphasizing the mitigation knock feature of water injection.

At the same time, the army was financing projects [10] to study the concept of total cooling of direct water injection engines. The engines were operated commonly under heavy conditions on military equipment. The amount of water required for internal cooling was 5.5 higher than the amount of fuel. For this reason, it was necessary to recover as much water as possible from the engine exhaust. These studies were carried out by Weatherford in 1970 [11], and continued by Melton in 1975 [11].

Polluting emissions, today a delicate subject, were ignored in the past. The first reports of the links between pollutant emissions and water injection were made by Nicholls [12] and Melton [11]. After them, in 2003 Brusca [13] did more detailed research on pollutant emissions and water injection.

At the time of writing the thesis, a special attention was paid by researchers to the injection of water into engines, so in the last two years several articles started to be published on these topics.

1.3. Water injection system

The first water injection systems on (OldsMobile) automobile and airplane internal combustion piston engines used modified carburettors [1] or a continuous water spray nozzle at certain power regimes [4] [7]. The limitations were given by the level of technology present on internal combustion engines. With the advent of transistors and the development of car computers, it was possible to adapt fuel injection systems.

Direct water injection was initially performed by modifying the mechanical diesel injection pumps and injector nozzles [11]. For mechanical injection pumps, the mechanism is lubricated with diesel fuel, more precisely with the help of the sulphur or other additives from diesel. The problems encountered at that time, were due to the lack of lubrication of the pump mechanism. Recent studies have shown that special high-pressure pumps that are used for water injection is a common rail type. The injectors are specially developed for direct water injection or gasoline direct injectors (GDI).

Due to the complexity of the direct injection system, there have been cases in which researchers have preferred to inject water mixed with the fuel, using the same injection system. More specifically, these researchers focused on developing additives in order to create stable emulsions of fuel (diesel) and water. The issues raised by these emulsions are the stability over time and the inability to vary the fuel-water ratio during operation.

1.4. Direct Water Injection

The direct water injection into the cylinder is aimed at achieving total internal cooling of the internal combustion engine. However, Melton [11] has prepared a detailed report on the direct water injection impact on the compression ignition engine. Figure 1.1 shows the evolution of power produced depending on the time of direct water injection. It can be seen that the highest power output values are recorded for the water injection performed at the end of the compression process.

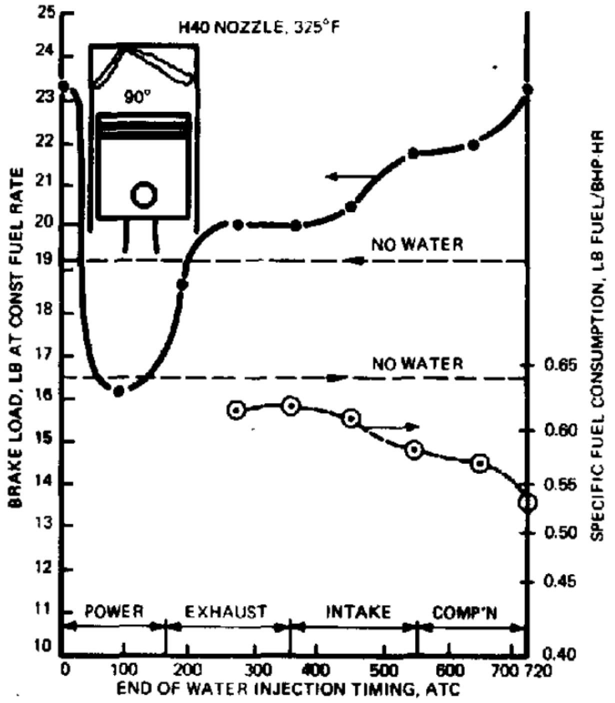


Figure 1.1. Direct Water injection performance [11]

When water injection is performed during combustion, a considerable decrease in power is produced. A delay in combustion has also been observed, due to water vapor. Therefore, to compensate for this shortcoming, it was necessary to advance the fuel injection. As expected in Figure 1.1, on the secondary axis, it can be seen that the lowest specific fuel consumption is almost 14% down from the “no water” value. In addition to these advantages, it is noted that the risk of oil contamination with water is higher when the water injection is performed during the process of compressing the air in the cylinder, the greatest contamination being achieved when the water injection is performed at the end of the compression process.

The experimental study conducted by Cordier [15] shown that water injection decreases the specific fuel consumption. Figure 1.2 shows that the evolution of specific consumption for both injection systems is similar. The high performance of the direct water injection system is observed from a low W/F ratio.

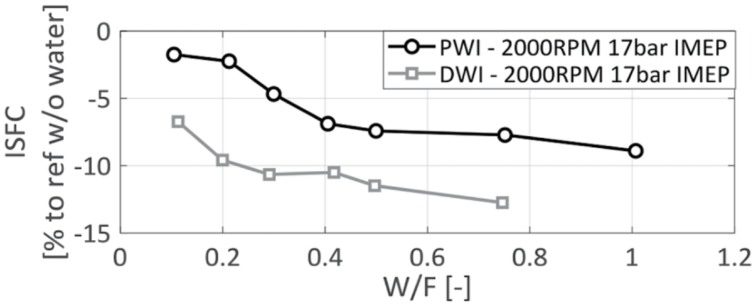


Figure 1.2. ISFC PWI and DWI [15]

In the experimental study [16] performed on a spark ignition engine, and direct injection of fuel and water, it is noticed that with the increase of the injected water ratio, the maximum pressure in the cylinder decreases and the energy release curve in the cylinder changes. Due to water vapor, combustion is delayed and the total combustion time increases, as can be observed in Figure 1.3

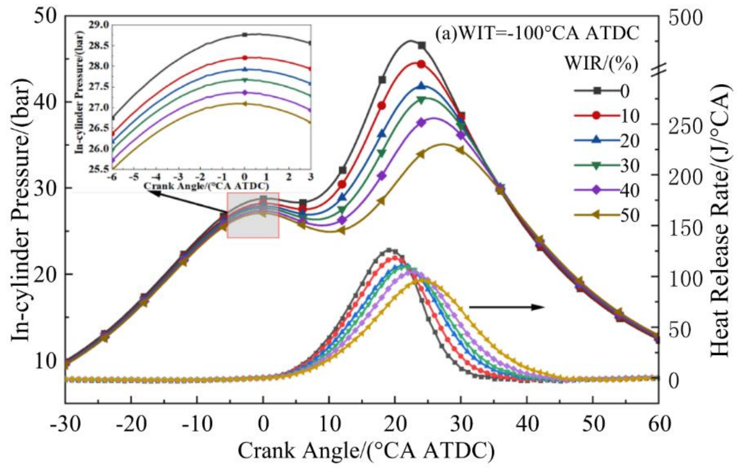


Figure 1.3. In-cylinder pressure and Heat Release Rate [17]

1.5. Intake manifold water injection

Knock inhibition was the most visible feature of water injection in internal combustion engines. In 2003, Brusca [13] performed experimental tests to be able to quantify the increase in knock resistance when water injection is performed. He used the MON (Motor Octane Number).

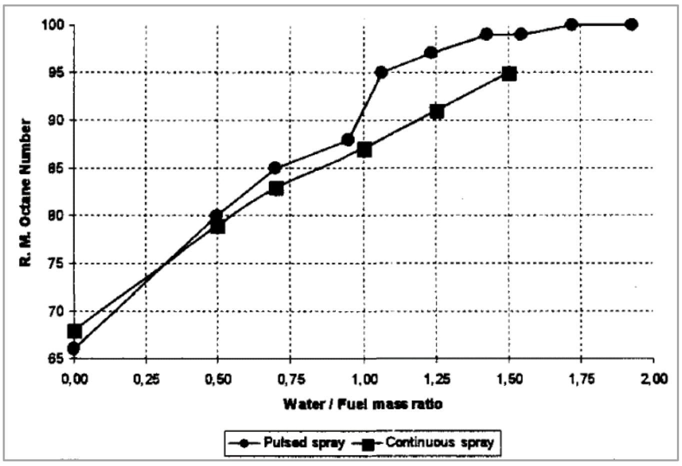


Figure 1.4. Octane number and water injection [13]

A fuel with an octane number of 67 and two water injection systems were used to perform the tests. The pulsed spray was a system with sequential water injection, such as fuel injection system, and the continuous spray was a system with a nozzle that continuously sprayed water, being the simplest water injection system that can be implemented. As can be seen in Figure 1.4, the self-ignition resistance of the mixture in the cylinder increases with the water injection. Thus, for a fuel water ratio of 1, knock resistance increased from 67 by almost 19 units reaching the value of 86. It is also noted that a sequential water injection system is more efficient, recording higher values of knock resistance.

Golzari [20] performed tests on an SI engine with direct fuel injection and port water injection (PWI). The engine used has a single cylinder, with a compression ratio of ε=11.43. Therefore, the engine has a stable operation, with an advance of only 3 Crank Angle Degrees Before Top Dead Center (CAD BTDC). The maximum spark ignition advance is dependent on the amount of water injected. As the spark ignition advance increases, the maximum pressure in the cylinder also increases.

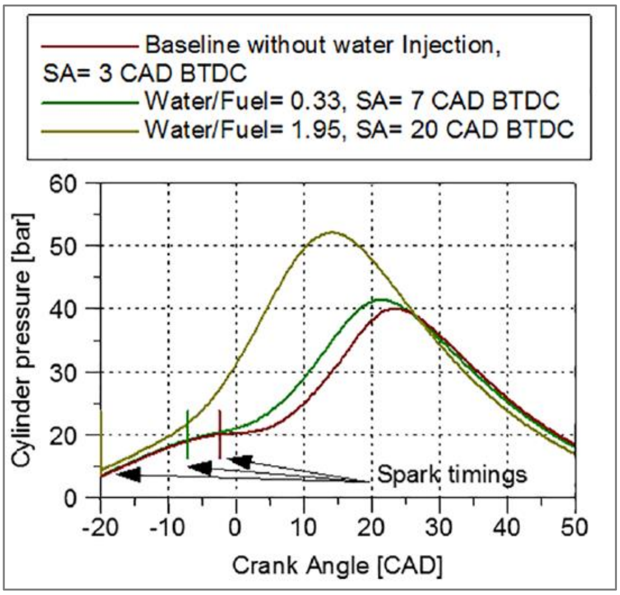


Figure 1.5. Cylinder pressure and water injection [20]

Reza [20] has also performed tests on a single-cylinder spark ignition engine with direct fuel injection and water injection into the intake manifold. The indicated efficiency increases until a water/fuel ratio of 0.75 is reached.

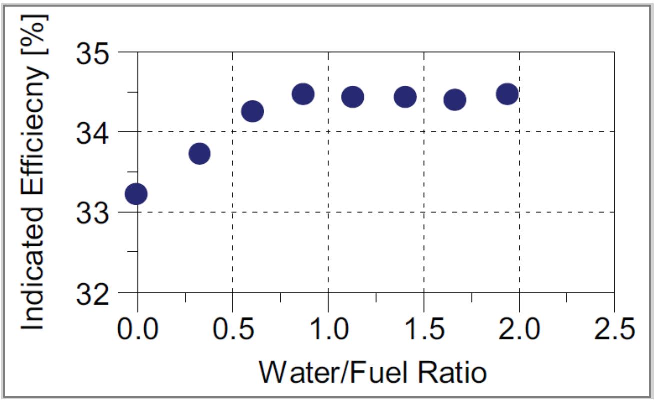


Figure 1.6. Indicated efficiency depending W/F Ratio [20]

Chapter 2. Experimental part

2.1. Experimental Stand

An internal combustion engine with spark ignition was used on the experimental stand. The engine is a single-cylinder one, and its cooling is done with air (Figure 2.1). Forced temperature with a variable flow fan is used to maintain a constant temperature. The engine speed is adjusted with an electronic throttle, and the load is adjusted with an electric brake.

Figure 2.1. Experimental stand

The engine has a fuel injection into the intake manifold and electronic ignition controlled by an EMU stand-alone computer from the ECU Master. The oxygen in the exhaust gas is measured using a Bosch wide-band sensor. In addition to the fuel injection system, two more individual water injection systems were added.

The first water injection system sprays water into the intake manifold. This system consists of the water tank, pump, filters, pressure regulator and injector as shown in the Figure 2.4. The components of this system are identical to those of the fuel injection system.

The second system injects water directly into the combustion chamber, as shown in the Figure 2.4. This system consists of water tank, high pressure pump, filters, pressure regulator, injection ramp, injector and liquid cooler. Given the complexity of the direct injection system, several phenomena will be further explained in this thesis.

Regarding the type of the engine, a flathead engine was chosen for the multiple possibilities of positioning the water injector. In addition, changing the compression ratio is not a difficult operation, as the shape of the combustion chamber in the cylinder head is a simple one.

Due to the simple design, several cylinder heads with compression ratio from 7.5 to 14.2 were made. The variation of the compression ratio was only carried out by changing the volume of the combustion chamber in the cylinder head and by changing the thickness of the cylinder head gasket.

Due to the engines design with cylinder head valves of the OHV, OHC and DOHC type, the water injection can be carried out so that the water jet is sprayed towards the piston head. This placement is also performed on the experimental stand, being considered as the water injection area 1, as shown in Figure 2.2. As mentioned before, the side valve (S.V.) engine allows an easy installation of the water injector and for the other areas 2 and 3 according to Figure 2.2. Thus, the water jet can be directed towards the exhaust gas valve and to the inlet valve. Since both valves are in the vicinity of the spark plug, there is a risk that the water jet will affect the formation of the flame core.

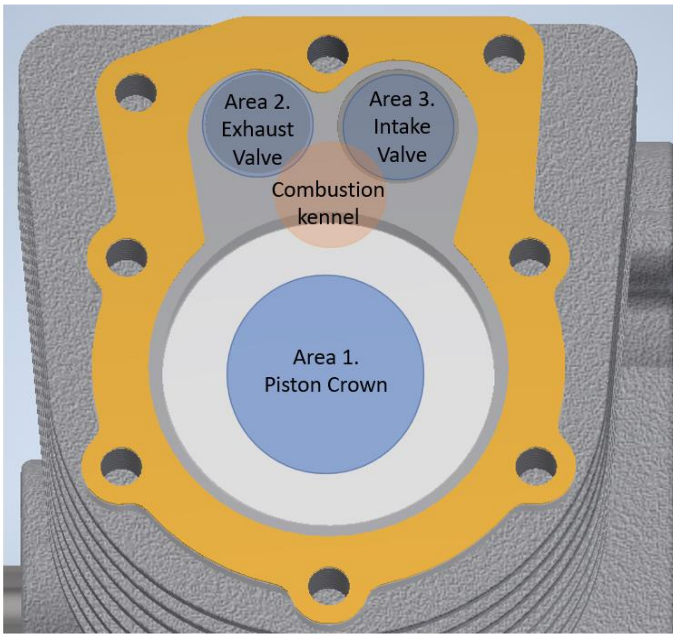


Figure 2.2. Possibilities for fitting the water injector in the combustion chamber

At the same time, according to the specialised literature, the exhaust gas valve is hotter, which may encourage the appearance of pre-ignition. Because of this, it was decided to install the injector in order to be able to spray water in area 2. On the other hand, the intake valve (area 3) is cooled during the intake process of the air-fuel mixture. The installation of the injector to ensure the spraying of water in area 2 aims at cooling the exhaust valve. Thus, the fine water droplets will absorb a larger amount of heat from the metal surface of the exhaust valve, than if the water jet is directed to the intake valve (area 3). For this reason, it was decided to perform direct injection only in area 1 and area 2.

To cool the piston, the injector must spray water towards the middle of the piston head. In this way, the amount of water in the immediate vicinity of the spark plug electrode is reduced. The second case involves mounting the injector above the exhaust gas valve, the water jet being directed towards the middle of the valve. The exhaust valve has been mechanically manufactured to have flat surfaces, to avoid liquid accumulation and uneven water dissipation.

The sequential control of the fuel injection allowed separate control of the injectors. Thus, it was possible to separate the water injection outlets from the fuel injectors and to adjust the percentage injection time according to the main fuel injection. In addition, the calculation unit also allows the adjustment of the injection time, which is necessary for performing tests with direct water injection. In addition, the standalone calculation unit can record and store data for a defined period, and can export it to editable files, providing information necessary for further analysis.

Two boards with a 12-bit resolution and a sample rate of 20kS/s, from National Instruments were used to monitor other parameters with small variations in time, and to acquire signals such as temperatures at various points on the engine, air flows, temperature, and atmospheric air pressure.

A data acquisition board from National Instruments [45] was used to acquire signals that have a large variation over time. This board is distinguished by a sample rate of 250kS/s and 16-bit resolution. The acquired signals are shown in Figure 2.3.

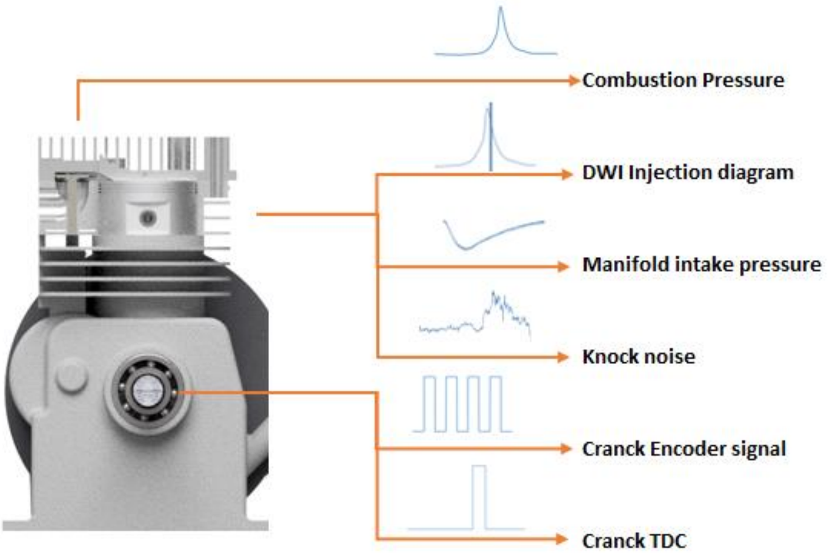


Figure 2.3. Fast data acquisition signals

To perform these tests, the injector was located so that the jet would spray onto the piston head [46] [47] [16], but, inside the combustion chamber, different locations could be selected. Figure 2.4 presents the water injection system of the engine diagram. An indirect water injection was located in front of the inlet valve, as well.

The fuel injection system in the intake manifold is one of the most used types of injection, due to its simplicity and low manufacturing cost. The working pressure is 3-4 bar. As this injection system is simple and reliable, many manufacturers continue to use it on today's engines in order to have a competitive price.

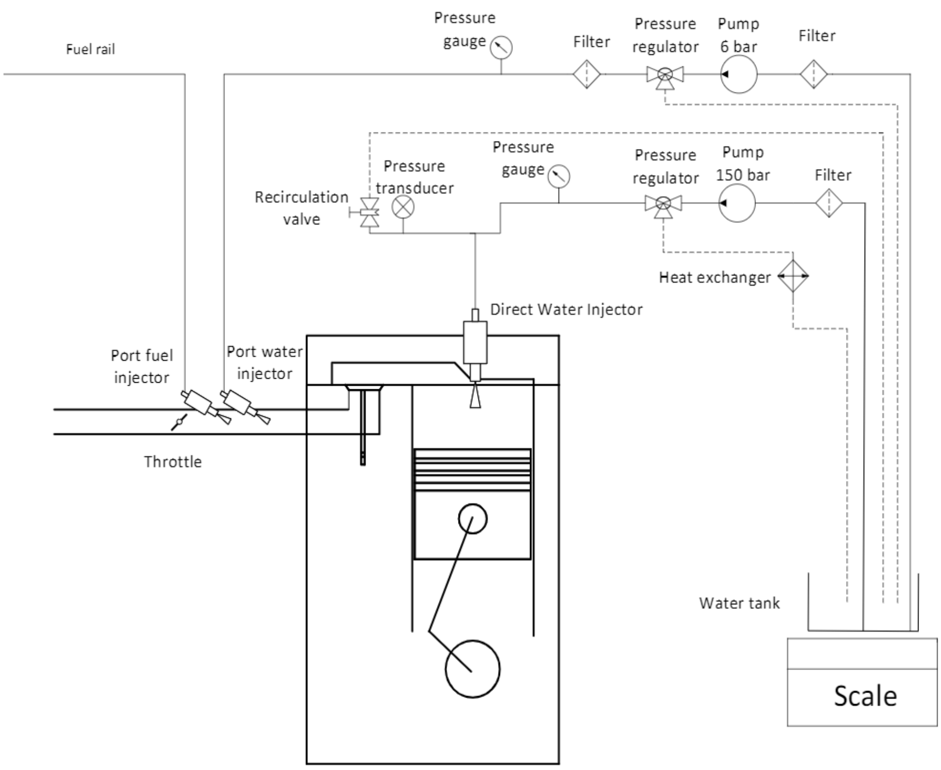


Figure 2.4. Schematic of water injection system

Even if the most important signal is the pressure in the combustion chamber, the accuracy of the measurements and test results is dependent on the other signals for real time calibration. The data acquisition software was created and developed in LabView [48]. Figure 2.5 shows the graphical interface of the data acquisition, monitoring and control system for the experimental stand. The main components of the data acquisition system are further described in this chapter.

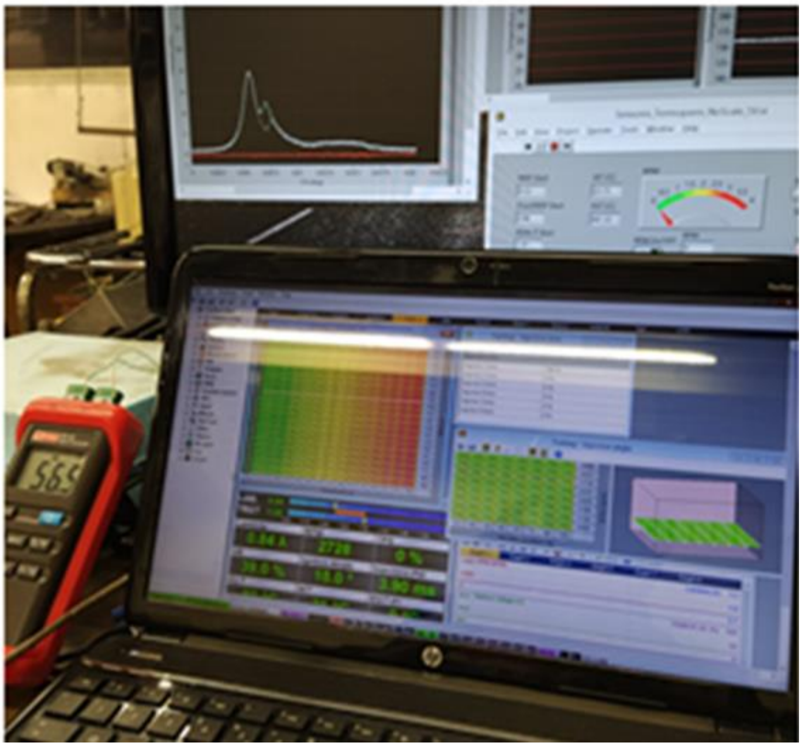


Figure 2.5. Data Acquisition software

To perform DWI, a Bosch direct injector [50] from a direct fuel injection stock engine has been utilised. Due to the high mass flowrate of liquid sprayed by this injector, it was decided to weld five spray holes out of a total of six. A high-pressure water circuit was fitted to the experimental stand.

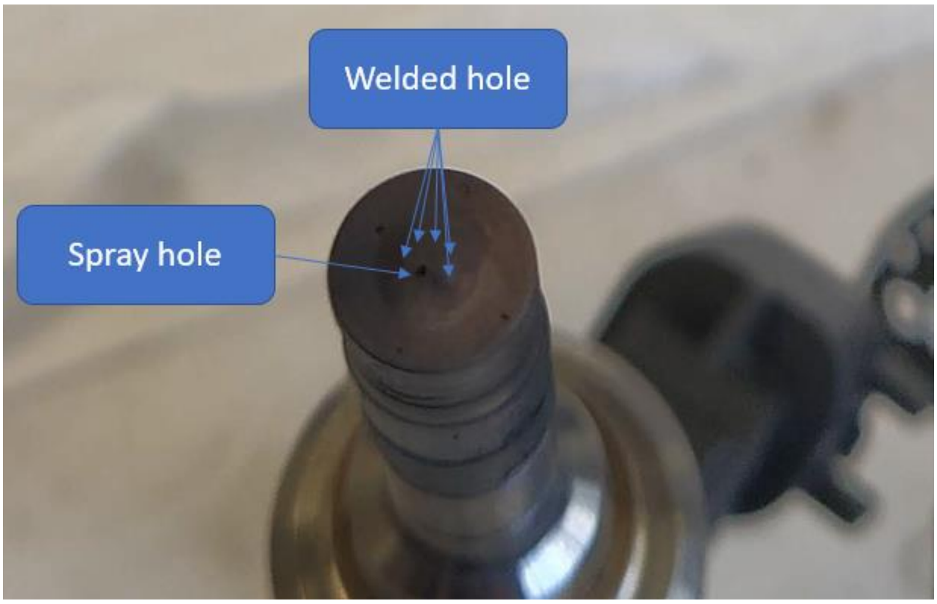


Figure 2.6. Direct injector with welded hole spray

2.2. Working methods

The experimental studies on internal combustion engines aimed at improving performances, improving combustion, and reducing pollutant emissions. Thus, recording engine operating parameters can provide information on the effects of the deployment of new technologies on internal combustion engines and the possibility of continuous improvement.

Thus, the signal received from the inductive sensor on the engine flywheel is not sufficient to encode the position of the crankshaft. If for a normal performance of an internal combustion engine, a deviation of a few degrees from the TDC does not have a noticeable impact, for research this may introduce high errors since the beginning of the data acquisition.

The first measurements of the pressure in the combustion chamber were made with a mechanical indicator [51]. Over time, with the technological advances, the sensors for measuring the pressure in the combustion chamber have become more compact, stable, with a very short response time, able to measure the pressure with high accuracy and withstand the transient conditions in the combustion chamber. This enables most of the combustion parameters in the engine to be determined, contributing to the permanent development of internal combustion engines.

Piezoelectric transducers only respond to pressure variations, and therefore it is necessary to determine a reference for each motor cycle. The accuracy of pressure measurement with piezoelectric sensors decreases when the sensor is exposed to a variable heat flow such as during combustion. Temperature variations influence internal voltages in the diaphragm and housing, thus changing the sensitivity of the piezoelectric element. This phenomenon causes a distortion of the output signal of the transducer, called drifting signal caused by temperature (temperature drift).

Also, the signal from the piezoelectric crystal to the transducer, respectively the acquisition board transducer must be well shielded, because in spark ignition engines there is a higher risk that the signal may be noisy, due to the high voltages that provide the engine ignition.

Hence, to monitor the pressure in the combustion chamber, an optical pressure sensor from Optrand [56] was used. Figure 2.7 shows the working principle. The light source achieved in the electronic conversion module is passed on, via the optical fibre onto the surface of the diaphragm. This diaphragm reflects the light and is passed on another channel back to the electronic conversion module, where a photodiode converts the light into an electric signal. The cylinder gas pressure deforms the diaphragm, changing the reflecting surface, hence changing the reflected light flux.

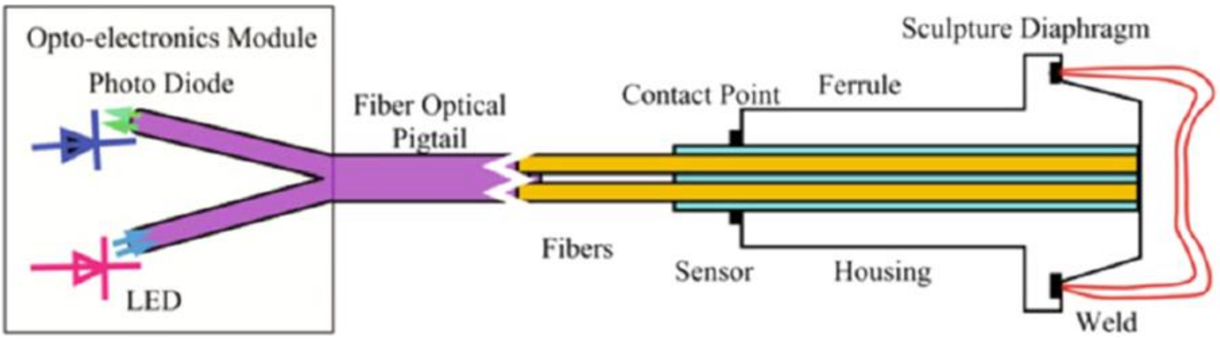


Figure 2.7. Operating principle of the optical sensor [53] [57]

To determine the position of the crankshaft with better accuracy, it has been chosen an optical incremental encoder. The output signal is represented by a series of square pulses.

The basic components are light-emitting diodes, light detectors, and a coded (perforated) disc, which allows the light produced by the led to pass only through certain slots. The disc is installed in line with the crankshaft, and the remaining components are fixed on the engine housing. Thus, depending on the position of the crankshaft, the coded disk interrupts the flow of incident light on the receiving diode. These pulse variations are interpreted by the acquisition software as 1 and 0. This means that the pressure can be read as the signal increases, decreases, or changes.

As mentioned above, due to the internal expansions and the variation of the heat flow received by the pressure sensor in the combustion chamber, there is a risk that the signal will drift. For this reason, it is necessary to use additional pressure sensors in the intake and exhaust manifold in order to be able to calibrate the indicated diagram. This eliminates possible calculation errors. It is recommended to use piezoelectric sensors to measure the absolute pressure in the intake manifold with an accuracy of + -10mbar.

In the figure below, it can be observed that the pressure measured by the pressure sensor in the combustion chamber can slide to a higher value (such as in Figure 2.8) or to a lower value, introducing errors in further calculations. Hence, a pressure sensor installed in the intake manifold can solve this problem.

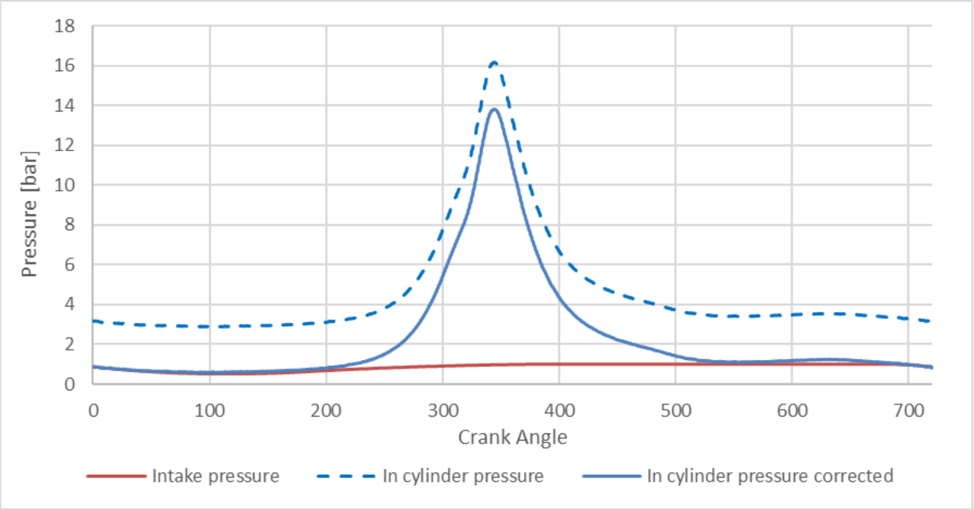


Figure 2.8. Motoring test for calibration

As it can be seen, it is necessary for the fast data acquisition board to, simultaneously, at least, take data from the pressure sensor in the combustion chamber, crankshaft position encoders, and intake manifold pressure sensor. In addition to these signals, other actuator signals, combustion noise are added, depending on the purpose of the experimental observations [60] [61]. The accuracy of the data taken from the experimental stand and its analysis must be high, especially when the differences in measured performance are small. This requires permanent sensor calibration and use of additional control sensors. In addition, the variation of engine cycles occurs.

Marvin's method.

This specific method requires tracking the volume-pressure diagram on a logarithmic scale. This allows the calculation of the value for the polytropic coefficient throughout compression and expansion, which shows data about the beginning and end of combustion, as it is further explained.

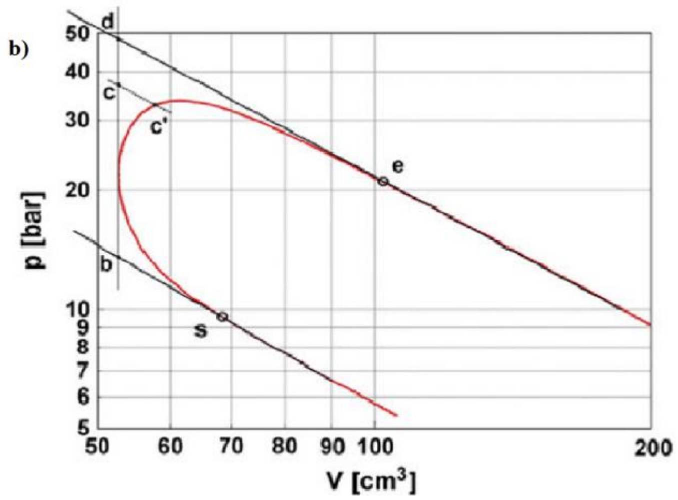


Figure 2.9. Marvin's graphical method [57]

The percentage of burned mass, the start, and the end of the combustion process can be graphically assessed, by using this method. The compression process of the fresh mixture and the expansion of the burned gas shall be deemed to have a constant polytropic evolution, pV^n. Figure 2.9 displays the pressure line deviation (red line) from the straight line (theoretical compression or expansion processes – black lines) point to the start of combustion (SOC – point *s*). Correspondingly, the end of combustion (EOC – point *e*) point can also be determine.

2.3. Pollutant emission measurement system

Pollutant emission measurement were performed using an AVL DiGas 4000 device. Considering the vulnerability of this device to humidity, it was chosen to introduce a gas cooler (air-water) with a condensation divider. Hence, the gas test is cooled, the condensation is collected in a hermetically vessel container and the cooled gases follow their path to the analyser.

Hydrocarbon (HC) emissions significantly contribute to the formation of smog, commonly in the presence of NOx. Incomplete combustion of fuel in internal combustion engines helps the appearance of hydrocarbon emissions..

Nitrogen oxides (NOx) causes acid rain and smog in cities. The acid rain may greatly affect agricultural crops that are close to urban areas [69], phenomenon determined by the evolution of air currents. The maximum temperature in the cylinder influences the quantity of nitrogen emissions [15]. Hence, if high temperatures are produced inside the combustion chamber, due to complete and rapid combustion and high compression ratio, the level of NOx emission will be extremely high. The exhaust gas recirculation (EGR) system has the key role of reducing the temperature in the combustion chamber, thus reducing the NOx level produced [70] by the spark ignition engines. However, if the temperature inside the combustion chamber decreases, combustion has a tendency to get worse and HC and CO emissions increase.

The need for a closed-loop control system, for internal combustion engines, arose from the use of the three-way catalyst, which significantly reduces polluting gases, but required a perfectly stoichiometric mixture. Thus, the spark ignition engines (SI) were originally set out with a lambda sensor and catalytic converter, where operation with a stoichiometric mixture $\lambda=1\pm1\%$ was required.

2.4. Direct water injection

2.4.1. Impact on engine cycle

Direct water injection in internal combustion engines involves high costs and requires more expensive and more efficient equipment than water injection in the intake manifold. If only the water-fuel ratio (W/F ratio) matters for the water injection in the intake manifold, in addition to the W/F ratio, the timing for the water injection is another important parameter for the direct water injection. For the first tests, the direct water injector is placed so that the water jet is directed towards the piston head (in area 1 - Figure 2.2). This injector location is specific to OHV, OHC and DOHC engines.

Direct water injection (DWI), and therefore the resulting water vapour, can change the ignition start time (ignition delay), if performed before the start of combustion. Also, it can also change the combustion time of the air-fuel mixture. These effects were measured in the side valve engine, adjusted at a compression ratio of ε=7.5. The above method was used to calculate the combustion time.

The first test was performed without water injection, therefore becoming a reference for the other tests. A constant speed of 3000 rpm was used for all the tests, along with similar fuel injection (6.5ms) and ignition advance (9 Crank Angle Degrees Before Top Dead Centre – CAD BTDC). The events are illustrated in Figure 2.10 (a). The time necessary to ignite the air-fuel mixture (ignition delay) is approximately 0.5 crank angle degrees (CAD) and the combustion occurs for 69.5 CAD.

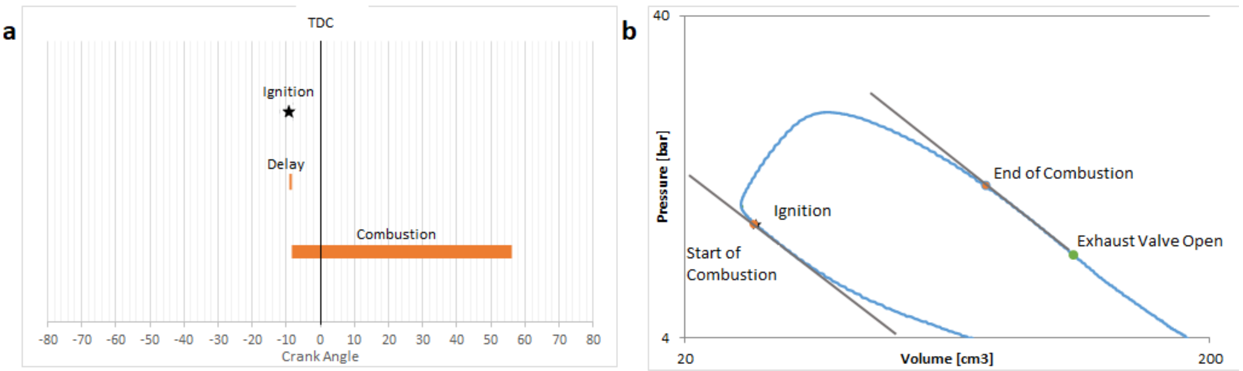


Figure 2.10. a) Timing map – No water injection b) p-V diagram

The following test used water injected into the piston crown with a water/fuel ratio of 1.3:1. DWI was performed at the start of the compression stroke (at 80 CAD BTDC). Figure 2.11 shows that when the DWI is performed into the cylinder at the beginning of the compression process, the ignition delay of the mixture increases from 0.5 to 2.3 CAD, and the combustion time is reduced from 69.5 to 62 CAD.

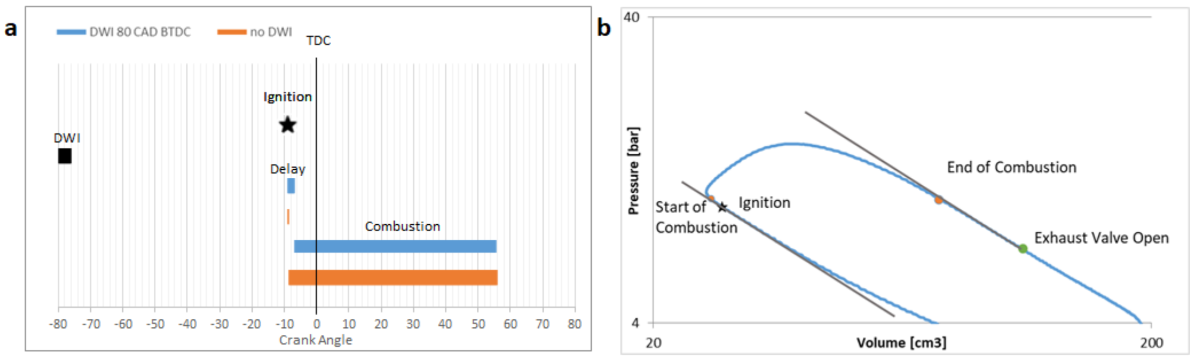


Figure 2.11. a) Timing map – DWI at 80 CAD BTDC b) p-V diagram

On another note, when the direct water injection (DWI) is performed later on the compression process (at 30 CAD BTDC), with a small advance compared to the spark discharge (at 9 CAD BTDC), the combustion start delay is reduced slightly to 2 CAD, but the total combustion time has been extended to 75 CAD, as shown in Figure 2.12.

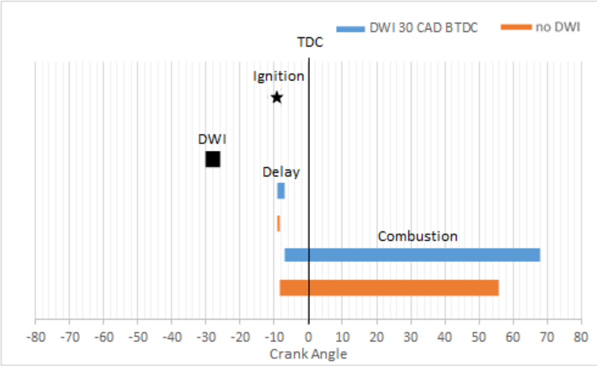


Figure 2.12. Timing map – DWI at 30 CAD BTDC

When the water injection is started when the piston is at TDC, during the combustion period, it can be observed that the total combustion time is reduced to 58 CAD, which is the shortest combustion time so far. As illustrated in Figure 2.13(a), the water was sprayed after the initial development of the kernel flame, so the combustion delay remains unchanged.

When the water injection is performed at 30 Crank Angle Degrees After Top Dead Centre (CAD ATDC), the combustion ends shortly after the water injection. In such a case, the combustion time is very short, lasting only 46.5 CAD, as it can be seen in Figure 2.13 (b).

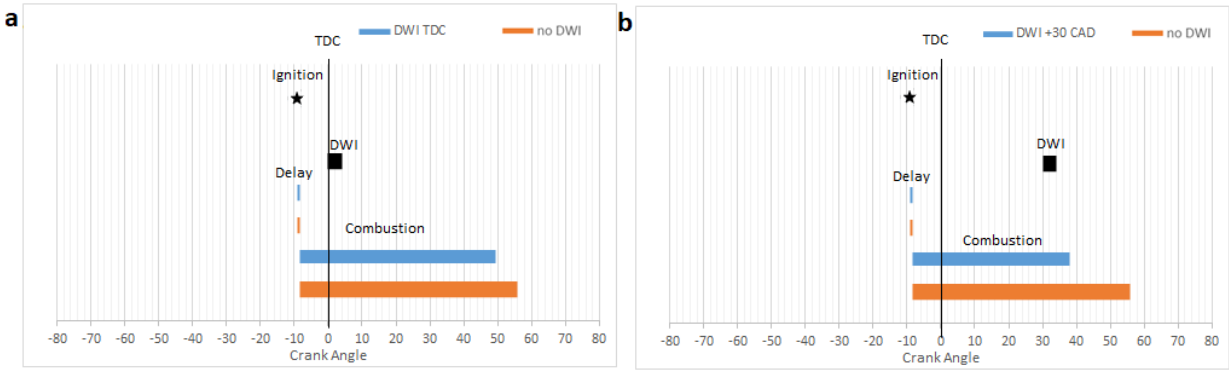


Figure 2.13. Timing map a) DWI at TDC b) DWI 30 ATDC

According to the analysed data, the DWI performed so late in the combustion period may "freeze" the combustion slightly after it occurs. Probably a similar effect was already happening when the injection took place at TDC. In both cases a slight reduction of torque and an increase in HC emission was identified, being more important for the late injection [71]. In the calculations already presented above, the same advance was used to produce the spark, in order to determine the effects of water injection on the combustion time.

2.4.3. Direct water injection on the exhaust valve, at the end of compression process

Considering that the internal combustion side-valve engine has a special design of the combustion chamber, it was decided to install the injector to spray water on the exhaust valve plate (area 2 - Figure 2.2). Due to the high temperature of the exhaust valve, as the compression ratio increases, this can become a hot spot leading to abnormal combustion in the cylinder. For the following tests it was decided to install a cylinder head with a compression ratio of $\varepsilon = 11.7$.

The water injector was placed to spray the water onto the exhaust valve, and 12 CAD BTDC was chosen as the injection moment. Thus, the water injection was be performed before the spark had occurred, to avoid any spray of water particles on the flame core or the spark plug electrodes.

As expected, the water injection performed before the spark delayed the formation of the mixture. Figure 2.14 shows the diagram of the pressure in the combustion chamber as a function of CA.

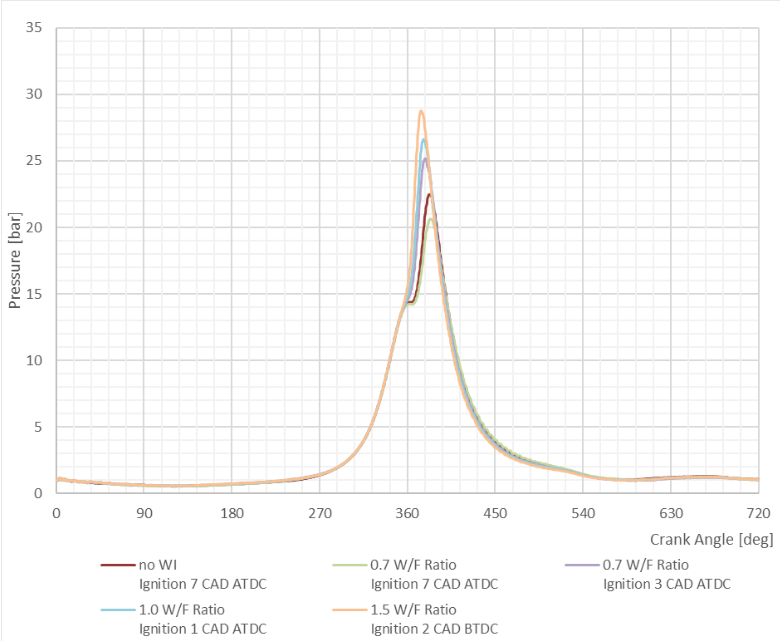


Figure 2.14. In cylinder pressure - DWI

Figure 2.15 shows the indicated work consumption for the intake stroke with fresh mixture in the cylinder, related to the energy of the injected fuel. There are small variations of this ratio compared to the reference (no WI), due to a change in the composition of the residual gas in the cylinder.

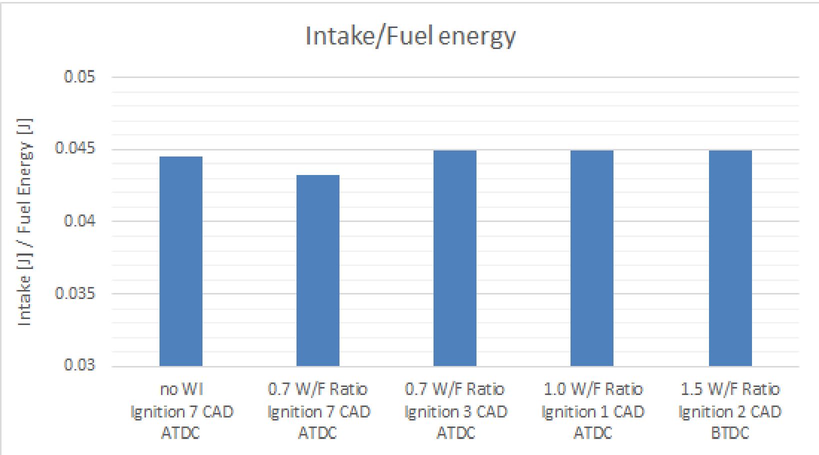


Figure 2.15. Intake / Fuel energy

As shown in

Figure 2.16, due to the direct water injection, the indicated work required for compression, increases. Even if a linear increase with the W / F ratio is expected, the effect of water injections is different at the same W / F ratio, but with different spark ignition advance. This was also observed by Melton [10]. They observed transient tendencies of the engine volumetric efficiency with variations of 2-3%. Under certain conditions, the water injection cooled the engine components, ensuring a volumetric efficiency increase, and in other situations, the steam produced by the water evaporation worsened the gas exchange in the engine.

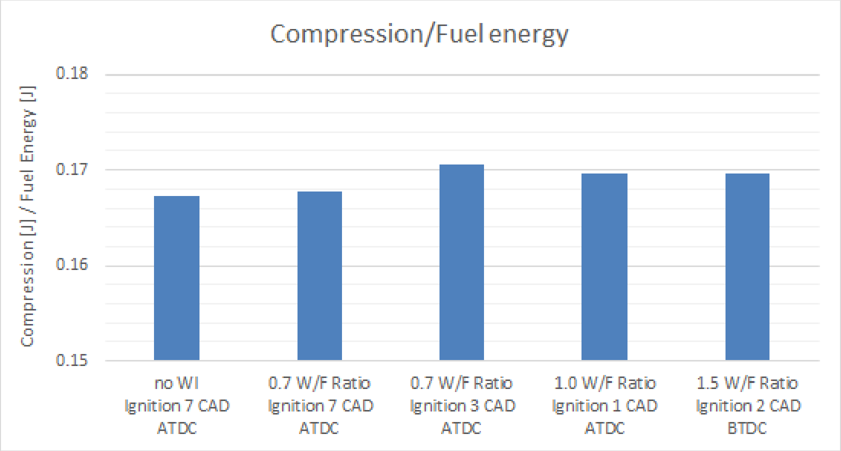


Figure 2.16. Compression / Fuel energy

If it was initially observed that for a W/F ratio of 1.5 the maximum pressure in the cylinder had a higher value (Figure 2.14), analysing the work during an engine cycle (Figure 2.17), it is noted that results are slightly different. Thus, the ratio between the energy released by the gas during the power stroke, and the chemical energy of the fuel indicates a decrease for a ratio of W / F 1.0 and 1.5 with 1%.

This phenomenon may be due to the possibility that the fine water droplets absorb energy from the burned gases, not only from the metal surface of the combustion chamber, or there is a risk that these water droplets will stop the combustion process more quickly. These results will be analysed in other data sets.

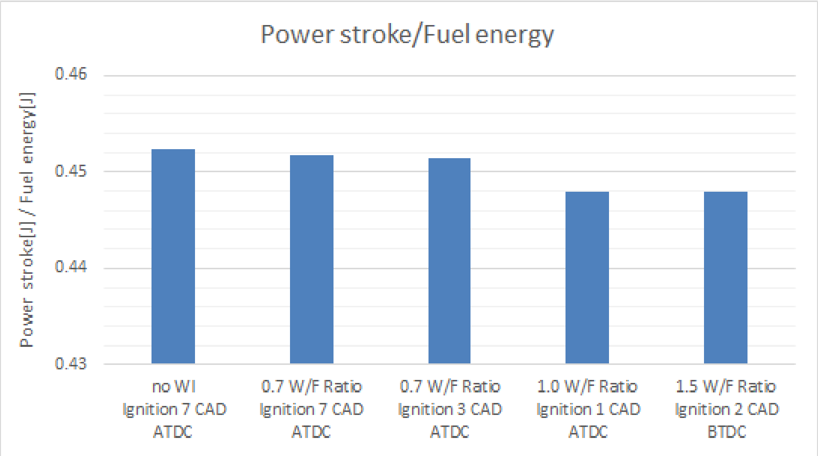


Figure 2.17. Power Stroke / Fuel Energy

Surprisingly, the mechanical work used to exhaust the gases from the cylinder is slightly lower for the water injection cases compared to the no WI reference (Figure 2.18). However, it was expected that the steam produced by the vaporizing water increased the work required for pumping.

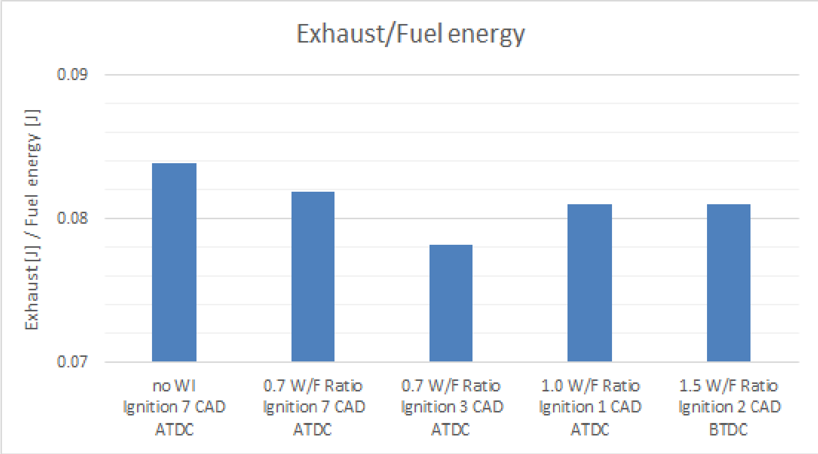


Figure 2.18. Exhaust / Fuel Energy

Finally, the net mechanical work at the end of the engine cycle was calculated in relation to the chemical energy of the fuel (Figure 2.19). There is an increase of 1.3 % in net work for using a W/F ratio of 0.7, as compared to the no Wi case, even if the pressure peak in the combustion chamber has decreased.

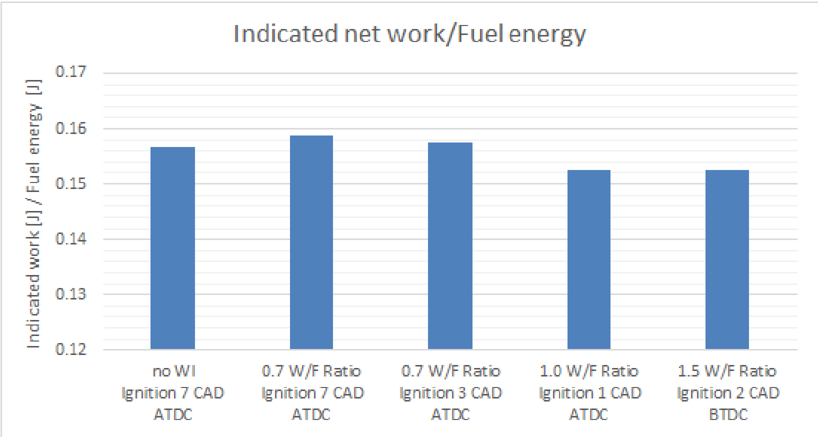


Figure 2.19. Indicated net work / Fuel energy

2.4.4. Water injection performed before and during combustion

The experimental investigations was continued with the same positioning of the water injector (onto the exhaust valve). The effect of the direct water injection was followed by a variation in the injection timing. A fuel injection time of 6.26ms and a W/F ratio of 1.5 was used. The evolution of the pressure in the cylinder can be followed in Figure 2.20

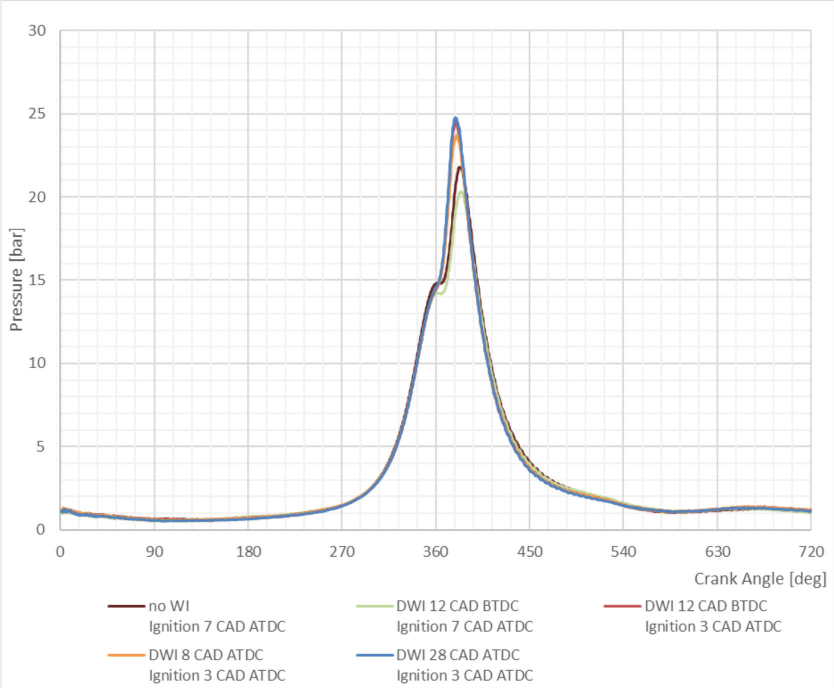


Figure 2.20. In cylinder pressure - W/F ratio 1.5

As expected, the water injection performed before the flame core production causes the mixture to delay ignition and changes the combustion time. In Figure 2.21 it can be observed that the time changing of the direct water injection and the advance in spark production has a high impact in the gas expansion process.

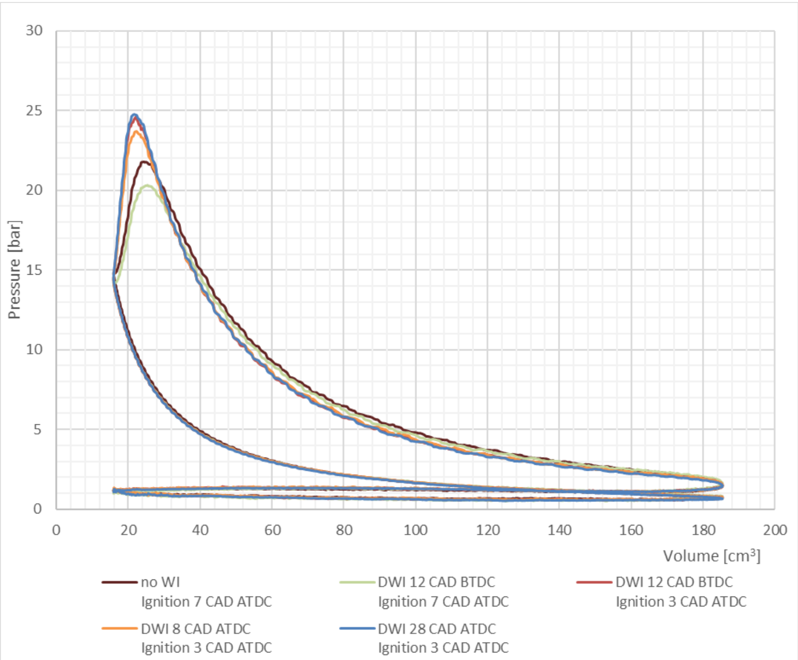


Figure 2.21. Pressure - volume diagram - W/F ratio 1.5

Moreover in Figure 2.22, the energy released during the power stroke is analysed, in relation to the chemical energy of the fuel. As seen in the previous graphs, the large amount of water decreases the indicated work produced during the power stroke.

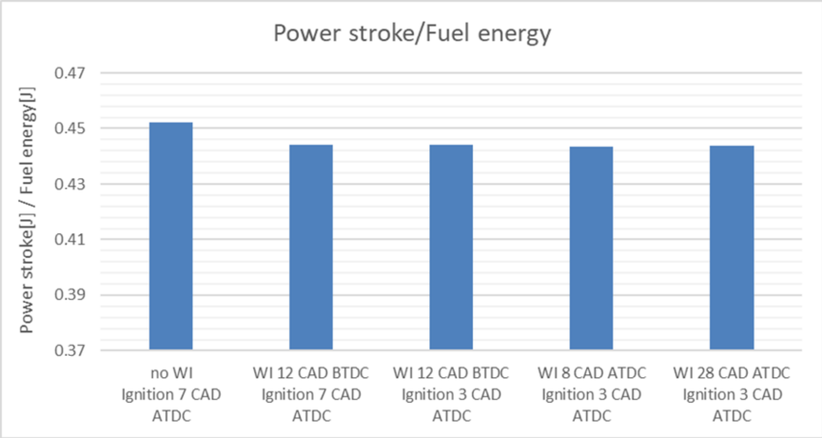


Figure 2.22. Power stroke / Fuel energy – W/F ratio 1.5

As expected, if the work produced is being analysed, a decrease of 11% is observed when the water injection is performed at 8 ATDC CAD, just a few degrees later from the production of the flame core, according Figure 2.23.

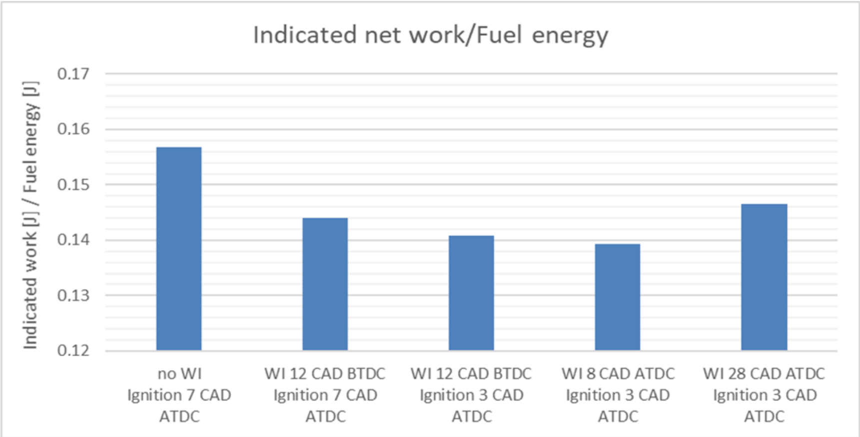


Figure 2.23. Indicated net work / Fuel energy – W/F ratio 1.5

The tests performed show that injecting a large quantity of water, equivalent to a fuel water ratio of 1 or more, worsens the engine performance. At the same time, if direct water injection is performed shortly after the spark is produced, the performance decreases even more.

2.4.5. DWI performed at the end of combustion and after the combustion

Experimental tests at medium load continued using a fuel water ratio of 0.7. During the tests, it is noted that the engine has performance variations (fluctuations) due to water injection. Due to these fluctuations in engine performance when direct water injection is performed or stopped, the data are taken after a period of stabilization of the operating mode, enabling the engine to stabilize. To reduce the errors caused by the pressure variations in the combustion chamber, after stabilizing the operating mode, the data were acquired for at least 50 cycles.

As presented in Figure 2.24, for a water/fuel ratio of 0.7, the maximum pressure in the combustion chamber increased by almost one bar (colour blue), without influencing the evolution of the burned gas expansion. Overall, the indicated mechanical work increased by 2.3%.

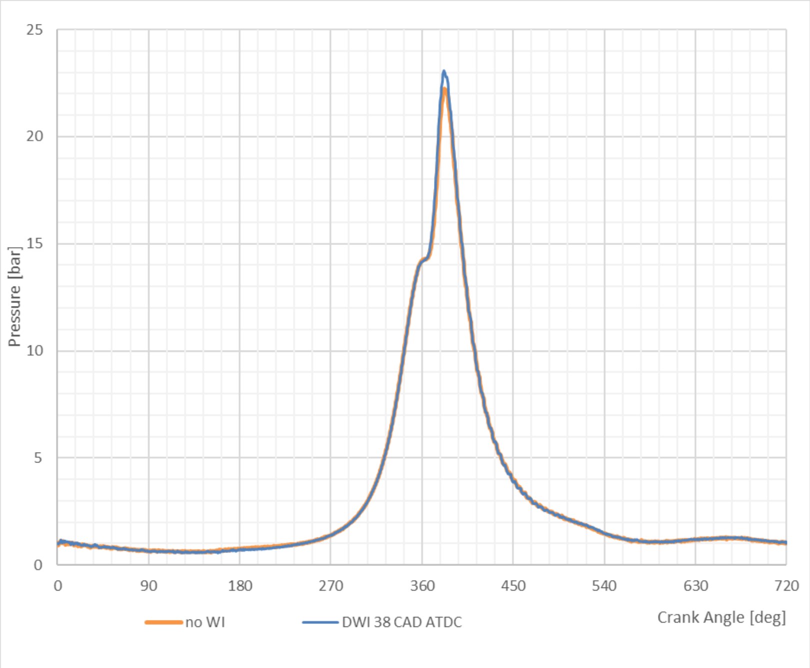


Figure 2.24. In cylinder pressure

Further tests were performed to see if the direct water injection carried out late - after the mixture has been burned in the cylinder - could produce steam by absorbing heat from the metal surfaces of the combustion chamber. The injector was still located above the exhaust valve.

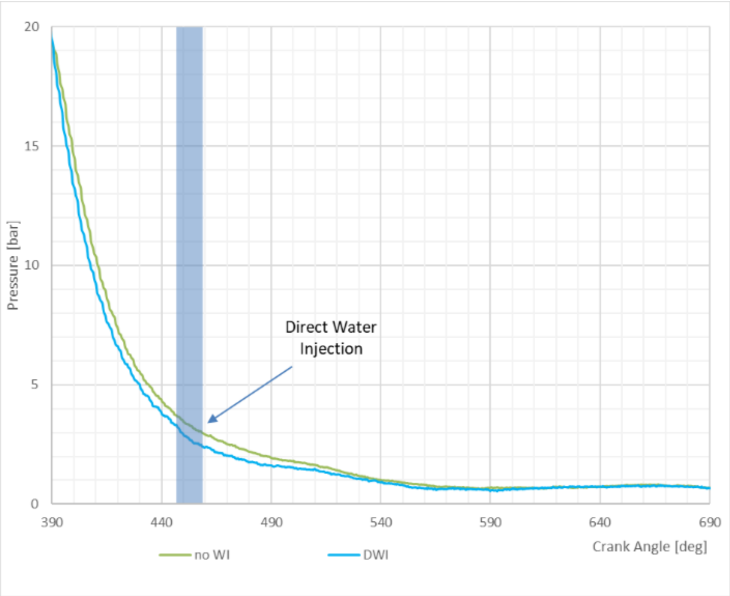


Figure 2.25. In cylinder pressure - DWI

Instead, it was observed that, the injected water does not only absorb heat from the metal surface, but also from the hot burned gases. Therefore, as it can be seen in Figure 2.25 (the evolution of pressure depending on the position of the crankshaft) the pressure in the combustion chamber decreases when the water injection is carried out.

2.5. Port Water Injection

2.5.1. Low compress ratio – Performance analysis

After performing several tests with and without water injection, it was observed that the engine volumetric efficiency (mass of intake air related to the engine swept volume for one cycle) decreased when water injection into the intake manifold was performed. These tests were performed at a fixed throttle position. Therefore, it was necessary to decrease the amount of injected fuel to maintain a stoichiometric mixture.

Figure 2.26 illustrates the evolution of the pressure inside the cylinder throughout the volume variation during one engine cycle. The tests were carried out without water injection for various quantities of injected water. The water/fuel ratio (W/F ratio, in mass) ranged from 0.21 to 1.0. The ignition advance has been constantly adjusted to the knock limit. As it can be observed, the maximum pressure occurred for the case of no water injection (no WI), and the maximum pressure decreased as the W/F ratio increased from 0.25 to 1.0.

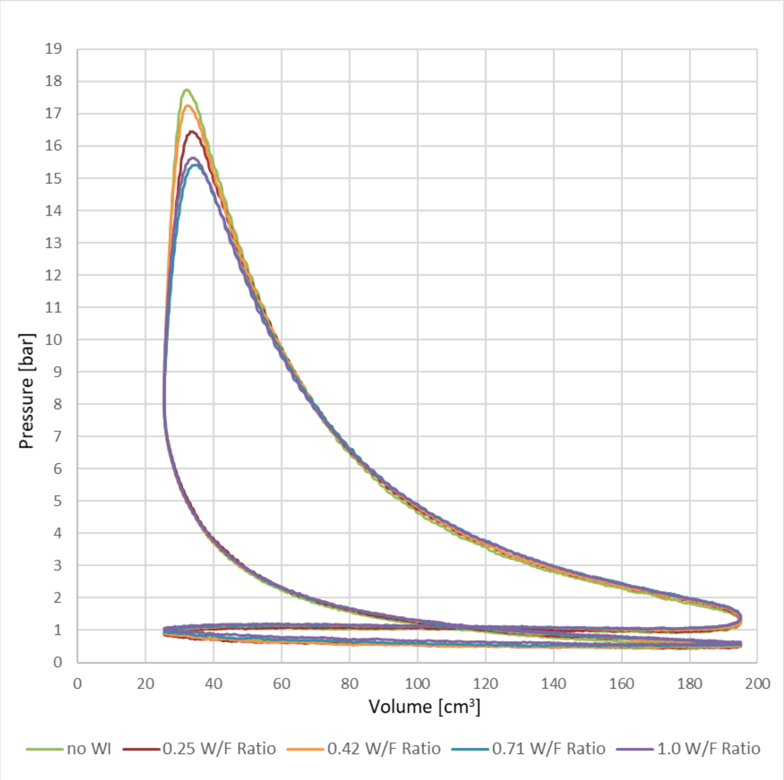


Figure 2.26. Pressure Volume diagram

The indicated work for one cycle was calculated with equation 2.7 [73].

$$W = \int pdV \tag{2.7}$$

For each condition (no WI and the four W/F) five calculations were performed for:
- intake;
- compression;
- expansion (power);
- exhaust;
- indicated work.

By comparing the work produced throughout the indicated cycle relate to the energy of the injected fuel (Figure 2.27), it can be noticed that the highest ratio is that of the cycle without water injection. When using a W/F Ratio of 0.25 or 0.42, the performance declines by 1.5%. When the W/F=1.0 was applied, the decrease was amplified up to 14%, a very high value. Nonetheless, this could be reduced by the optimization of the valve timing, but was not performed in this research.

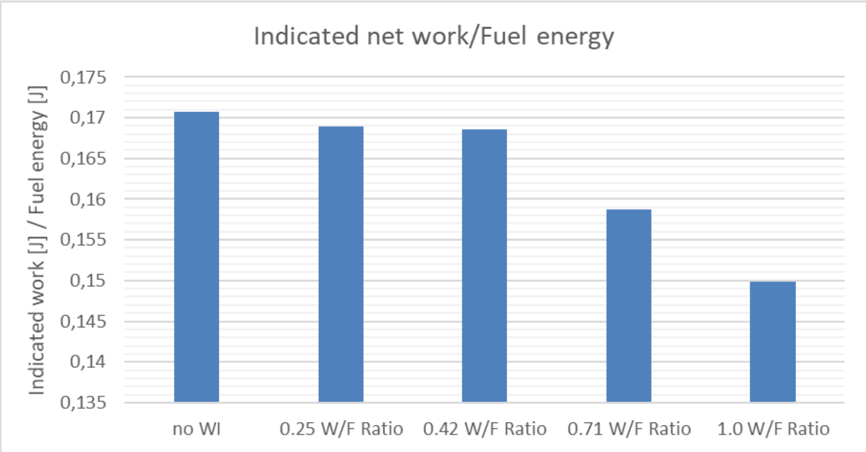


Figure 2.27. Indicated work [J] / Fuel energy [J]

2.5.2. Low compress ratio – Pollutant emissions

The nitrogen oxides NO and NO_2 are also known as NO_x. In general, they are formed due to the interaction between N_2 and O_2 at high temperatures, as per equation 2.8 [74] [75].

$$N_2 + O_2 \rightarrow NO + NO_2 \quad (2.8)$$

For the W/F ratio of 0.42, NOx emissions were lowered by 42% when the ignition was advanced from 5 crank angle degrees before top dead centre (CAD BTDC) to 15 CAD BTDC. When the water/fuel injection ratio was modified to 1.00, nitrogen oxide emissions were lowered by up to 80% when the ignition was advanced from 10 to 25 CAD BTDC. Furthermore, it was noticed that with the increase of the spark ignition advance, the pressure peak in the combustion chamber increased. At the same time, there was a trend for the engine to overheat.

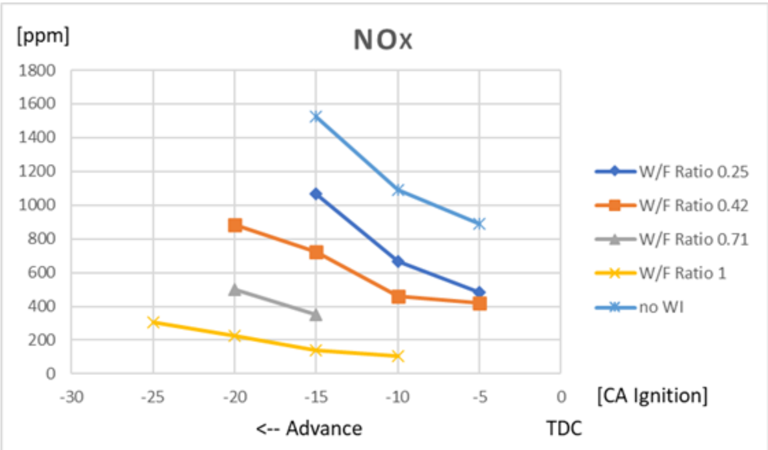


Figure 2.28. NOx emissions – compress ratio 7.5

Figure 2.29 (a), illustrates that the CO level lowered when the water injection was used. For a fuel water ratio of 1.00, the CO level lowered by 43% and the CO_2 emissions are on an average of 4% higher (**Error! Reference source not found.**b)

Figure 2.29. (a) Carbon monoxide (CO) emissions (b) Carbon dioxide (CO2) emissions

As it can be observed in Figure 2.55, as the amount of water injection (WI) increases (W/F ratio increases), the level of hydrocarbons increases.

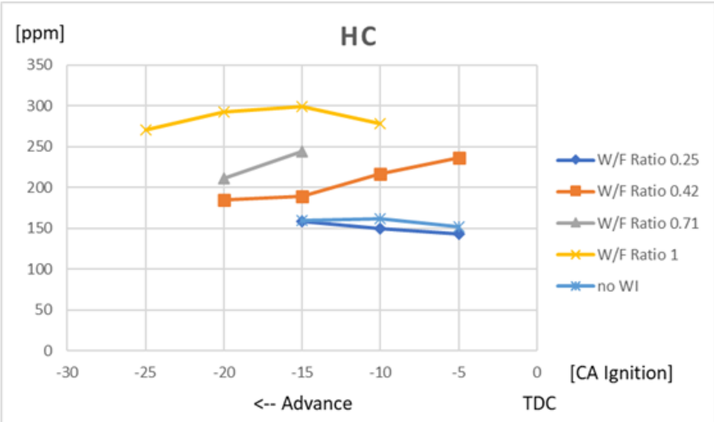


Figure 2.30. Hydrocarbon (HC) emissions – compress ratio 7.5

2.5.7. Comparative analyses

Experimental tests were performed for compression ratios of 7.5, 11.7 and 13. To be able to compare the results obtained with the 3 different cylinder heads, a comparative analysis was carried out. This was aimed at assessing the impact of the water injection into the engine with different water-fuel percentages.

As shown in Figure 2.31, the energy released during the power stroke engine cycle, records a maximum value when the water injection is performed with a certain ratio. Specifically, for a compression ratio of ε = 7.5, the best performance was recorded for a W/F ratio of 1. Increasing the compression ratio to ε = 11.7, the maximum performance is recorded when using a W/F ratio. of 0.71. Finally, for a compression ratio of ε = 13, the maximum recorded performance is when using a W/F ratio of 0.42.

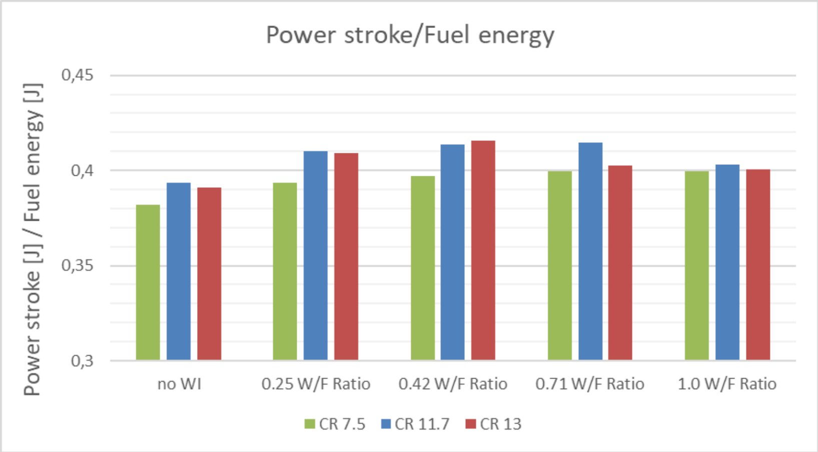


Figure 2.31. Compress ratio comparation

Theoretically, if the compression ratio increases, the engine's thermal efficiency should also increase. However, in practice, the compression ratio may increase to a certain maximum value, depending on the octane number of the fuel used. Exceeding this value, engine's performance decreases. In relation to the results of these experimental tests, it was noticed that the highest amount of converted energy from the fuel is when using a compression ratio of ε = 11.7, with a W/F ratio of 0.71 and compression ratio of ε = 13, with a ratio W/F of 0.42.

Given that only one engine stroke produces work, and the other three are work consumers (in the case of aspirated four stroke engines), it was noted that, despite good energy conversion rate performance in the injected fuel, the indicated mechanical work is much lower for the cases with compression ratio of 11.7 and 13.

Due to a lower consumption of mechanical work for the other engine strokes (intake, compression and exhaust), the case without water injection records the best performance when at low compression ratio. As expected, the highest consumption of work is for the compression process. Thus, with the increase of the compression ratio, according to **Error! Reference source not found.**, the mechanical work consumed for gas compression is much higher than the performance gain on the power stroke cycle (Figure 2.32).

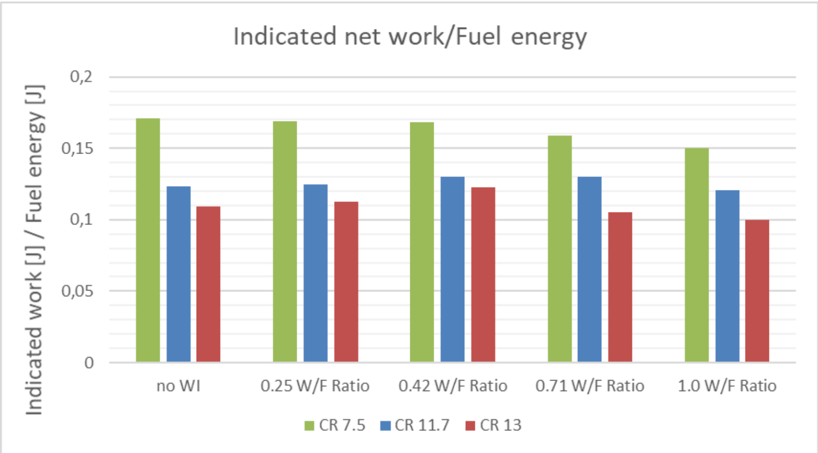


Figure 2.32. Indicated net work – compress ratio comparation

2.5.8. Partial conclusions

To conclude the chapter in which the experimental data were presented, I would like to bring up again that the engine used is an air-cooled side valve spark ignition engine. For this type of engine, the compression ratio is between 6 and 7.5. This is a low compression ratio for OHC or DOHC engines. Operating with a compression ratio of 11.7 is also a lot for the current OHC and DOHC engines. The purpose of the tests was to study the result of water injection under difficult combustion conditions. For this reason, tests were also performed for a compression ratio of 13 which is an extremely high compression ratio for a side-valve engine.

One of the objectives of the thesis is to study the influence of direct water injection on combustion in spark ignition engines. Thus, for the engine under analysis the following was found:

- Direct water injection performed before ignition events can delay ignition by up to 2.3 degrees. Without water injection, the delay is 0.5 degrees at 3000 rpm.
- Direct water injection performed before the spark increases the combustion time.
- Direct water injection after ignition shortens the total combustion time.
- Direct water injection at the mid-late burn 'kills' combustion in the cylinder.
- Direct water injection performed at the end of the combustion process can increase engine performance by 2.3%. At the same time, if the amount of water injected is higher, it will cause internal cooling of the engine and reduce the pressure in the cylinder (water absorbs heat from the combustion gases and not only from the metal surfaces of the engine).

Following the performance of the engine during water injection into the intake manifold, it is surprising to see that water improves combustion at medium loads. The optimum fuel-water ratio for a stoichiometric mixture depends on the compression ratio used. At the same time, it is found that water injection increases the mechanical work consumed for the intake and exhaust process. Due to water injection, the residual gas remaining in the cylinder (from the previous cycle) contains a higher amount of water/steam which decreases the volumetric efficiency of the next cycle.

According to the analysis of the diagrams taken from the experimental stand it was observed that the most energy released in power stroke is when water injection is performed for the three compression ratios analysed. On the other hand, if the whole engine cycle is analysed, it is noticeable that the maximum indicated work is for the no water injection case, because the pumping losses due to water/steam in the cylinder increase. These can be eliminated by adapting the camshaft profile to the new working conditions or by adding a turbo compressor. Internal engine cooling was possible with both types of water injection. Maximum spark advance depends on the water-fuel ratio used.

As regards of pollutant emissions, it was observed that nitrogen oxide emissions depend on the compression ratio and on the W/F ratio. Thus, as the water ratio increases, NO_x emissions decrease. At the same time, HC emissions increases as the water increases and/or in the presence of abnormal knock combustion (**Error! Reference source not found.** b). CO_2 and CO e missions have an inversely proportional trend, so if there is an increase in the power stroke/fuel energy ratio, CO_2 emissions increase and CO emissions decrease.

Chapter 3. Numerical Model

3.2. Pre-processing

The first step for CFD simulation is the pre-processing. In order to reduce the computation time of the CFD simulation, the inner surfaces that come into contact with the fluid flow are exported. For a more accurate simulation, the complete intake and exhaust path was introduced.

The import of the file was done in Converge Studio where the geometry is verified and corrected. CAD programs do not always export connectivity information for surface triangles leading to errors during surface import. After importing the surfaces and correcting the errors, information about the internal combustion engine geometry such as piston diameter, connecting rod length, crankshaft radius, valve opening profile (Annex 1), as well as information about the engine status such as speed, parts temperatures, pressures, and so on, are entered into Converge Studio [80] (Figure 3.1).

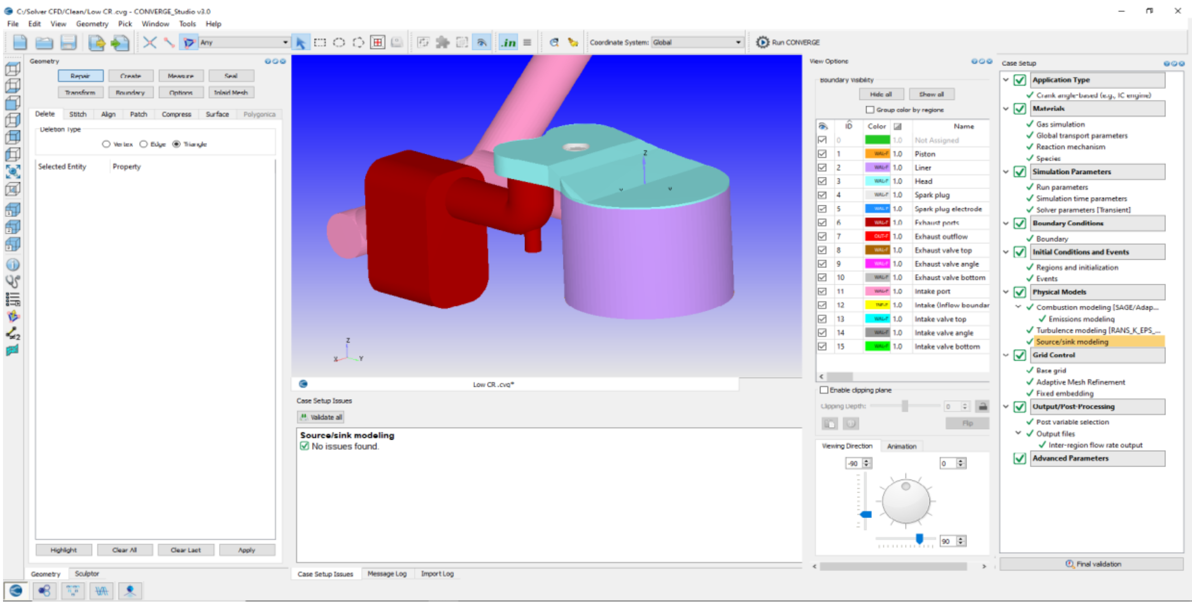


Figure 3.1. Converge Studio 3.0

3.3. Discretization and grid generation

Making a quality mesh is the first step in a CFD simulation that allows the results to converge to the right solution. At the same time, it must be taken into account that for the solution obtained, the load on the computer processing resources must be minimal [79]. Converge CFD program contains several tools for changing the grid size before and during the simulation. Grid scaling allows to modify the global grid to make it coarser or finer. If a coarser or finer grid is needed, after a certain time from the start of the simulation or at a certain event, the Grid scaling function or the fixed embedding function can be used.

3.4. Turbulence modelling

Turbulent flows are characterized by different eddies with a very wide range of time and length scales. The largest eddies are comparable in size to the characteristic length of statistically stationary flow, while the smallest scales are responsible for dissipating turbulent kinetic energy.

In internal combustion engines, the gas flow is turbulent, as in most engineering applications. Theoretically, the whole spectrum of turbulence scales can be solved by using a Direct Numerical Simulation (DNS) approach. The main advantage is that it does not require any modelling. At the same time, the resources needed for direct calculation are colossal, making solving the problem unfeasible.

On the other hand, the LES turbulence model uses a slightly different approach. Thus, the large eddies are solved directly, and the small size eddies are modelled numerically (Figure 3.2). As for the RANS turbulence models, they are fully modelled, greatly reducing the need for computer resources and simulation time.

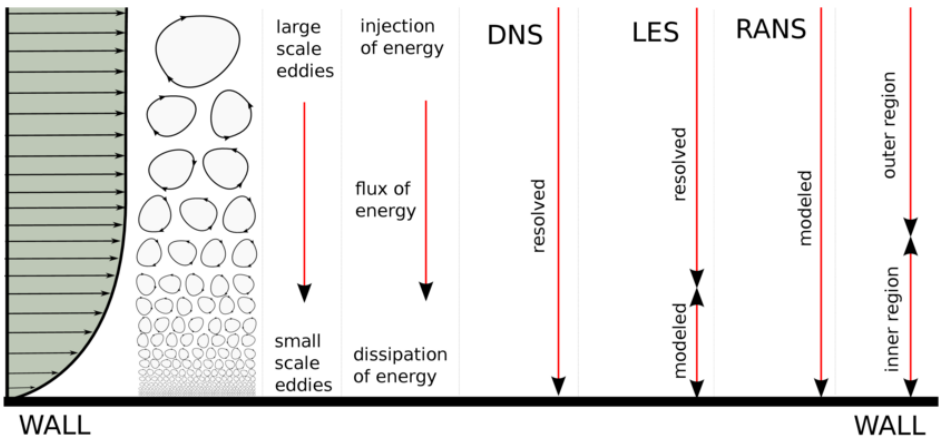


Figure 3.2. Turbulence modelling [83]

Of course, these models have both advantages and disadvantages. These are shown in the hierarchy, in Figure 3.3. DNS method best predicts physical phenomena but, it is not feasible for common engineering issues. Because in internal combustion engines we are dealing with large turbulence, high Reynolds numbers will result. This requires increased computing resources, and therefore costs. It is obvious that for large Reynolds numbers, costs become unjustified.

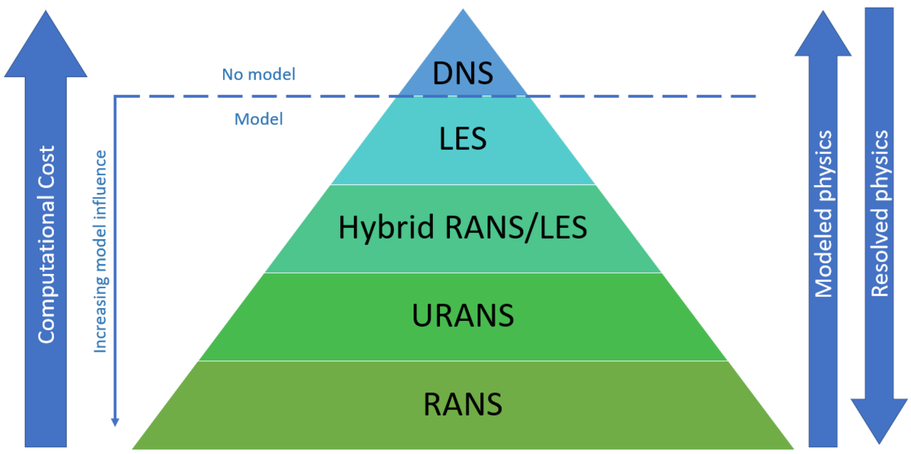


Figure 3.3. Hierarchy of turbulence models based on computational cost

3.7. CFD simulation and engine cylinder pressure predictability

Combustion in internal combustion engines is a very complex phenomenon. Simulating the processes inside the combustion chamber requires an understanding of thermo-chemical reactions and fluid dynamics. However, turbulence and heat transfer directly influence how chemical reactions take place. The study aims to visualise the differences between the different numerical heat transfer models presented above and their impact on chemical reactions in the combustion chamber, temperature evolution, pressure, and pollutant emissions.

Matching the data obtained from the CFD simulation with the data taken from the stand following experimental tests is a challenge because there are many unknowns and conditions that need to be introduced in the numerical model. The main way to validate the experimental data is to overlap the in-cylinder pressure curves as a function of crankshaft position. To achieve this, it is necessary to divide the diagram into several areas, as shown in Figure 3.4.

The first area is the air intake process into the cylinder. For the correct overlap of this area, it is necessary to know the correct profile of the intake camshaft, atmospheric pressure and air temperature. If determinations are performed on spark ignition engines with partial load (throttle plate partially open), then the 3D model imported into the program must contain the intake manifold and throttle body, with special attention to the fluid flow around the throttle plate profile. Additional experimental data on the air flow, i.e., the amount of air captured by the cylinder, is also required.

Ensuring good predictability of the mathematical model requires initial calibration of the intake (0-180 degrease) and exhaust (540-720 degrease) areas. After several CFD simulations are carried out and a good overlap is ensured, then we move to the more difficult part; calibration of the air-fuel mixture compression, ignition, combustion, and hot gas expansion areas.

The second issues are the end of the compression process and the initiation of combustion, more precisely, producing the spark. The diagrams overlap in this area is influenced by engine geometry (compression ratio deviations, pressure losses through valve or piston blow by), heat transfer.

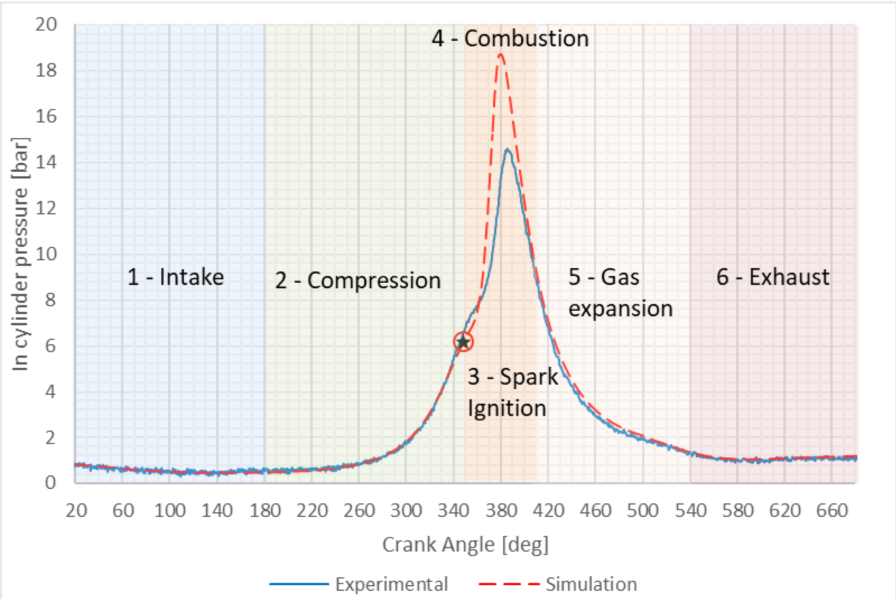


Figure 3.4. In cylinder pressure

The pressure evolution in area 3 - 4 is influenced by the type of surrogate fuel used, the reaction mechanism, the reaction speed of combustion, the heat transfer and temperature of the gas in the cylinder, the turbulence in the combustion chamber, and the quality of the mesh, as we have seen in comparative studies. Area 5, exhaust gas expansion, is influenced by the heat transfer in the combustion chamber, the end of cylinder reaction and the compression ratio.

In the previous sub-chapters, the influences of different types of chemical reaction mechanisms and wall heat transfer models on the predictability of cylinder pressure were analysed. Further, in Figure 3.5 the average cylinder pressure for several experimental and simulated cycles can be seen. A good overlap of theoretical and experimental model can be observed. The areas with differences between the two diagrams are the over-lap area of the inlet and exhaust valves, the combustion initiation, and the predicted peak pressure in the cylinder. The pressure differences between the exhaust and the inlet end area may be due to errors in determining the actual camshaft profile and errors caused by temperature difference.

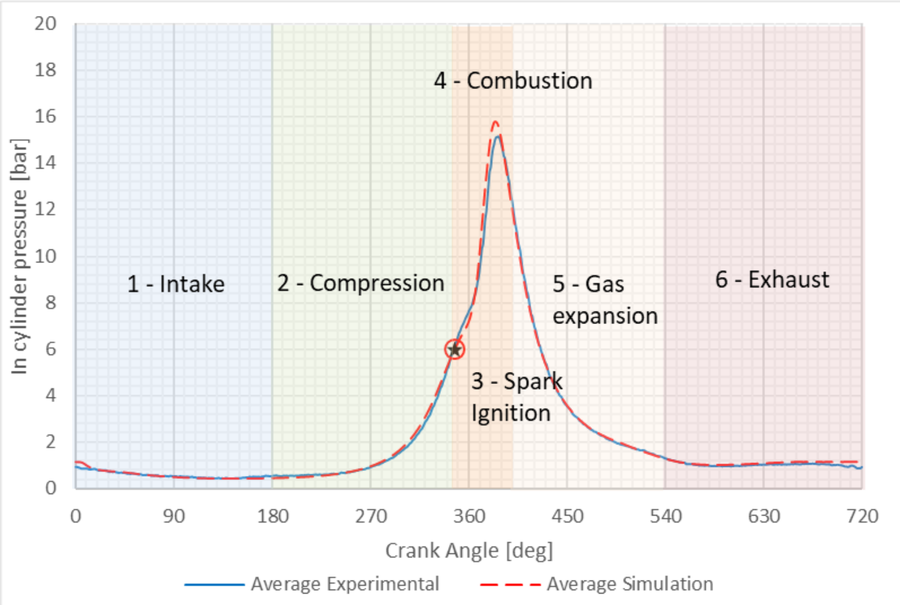


Figure 3.5. In cylinder pressure experimental and simulation

To be able to compare the results, the error between the real and simulated mean pressure will be further analysed. *Table 3.1* shows the absolute error and the relative error depending on the engine cycle stages. The highest absolute average error has a value of 0.29 bar and is recorded for the combustion process. On the other hand, the highest relative error has a value of 8.4% and is recorded for the exhaust gas process. The error is caused by a possible error when measuring the actual camshaft profile or lack of data simulating the path of the suction blower of the exhausted gas.

Table 3.1. Average error

	Intake	Compression	Combustion	Expansion	Exhaust
Average absolute error [bar]	-0.01	-0.06	-0.29	0.02	-0.09
Average relative error [%]	-0.97%	-0.88%	-1.71%	-0.34%	-8.40%

The negative value of the error is given by the tendency of the numerical model to overestimate the pressure in the cylinder. If we discuss the overall error per engine cycle, we find that we have an average absolute error of -0.05 bar, which is a relative error of -2.76%. In Figure 3.6 it can be seen the absolute error for the prediction of cylinder pressure as a function of crankshaft position. At the end of the compression process with the initiation of combustion there is a slight pressure deviation. Deviation encountered during simulations by researchers who created chemical reaction models [111] and by researchers who created wall heat transfer models [120] [117].

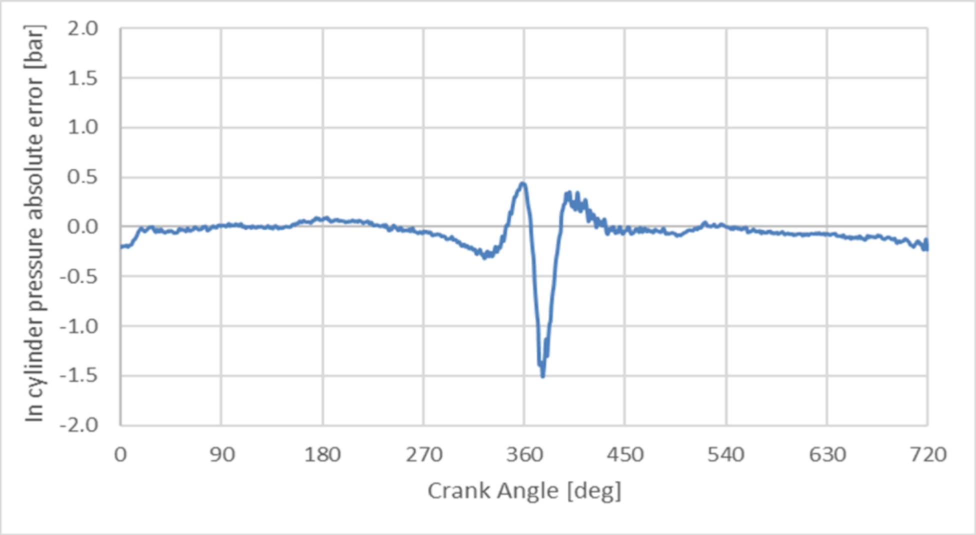


Figure 3.6. In cylinder pressure prediction absolute error

Figure 3.7 shows the evolution of relative errors depending on the position of the crankshaft. The highest absolute errors are in the overlap area of the exhaust and intake valves and for the intake valve closing process (near to 180 degrease). In addition, the vibration of the absolute error diagram in the intake and exhaust areas is given by noise during the experimental data acquisition.

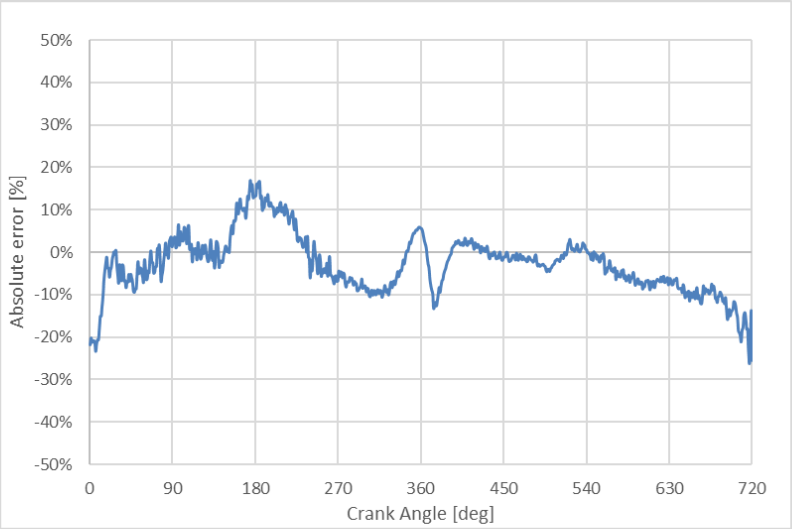


Figure 3.7. In cylinder pressure prediction relative error

These error calculations are valid for pressures averages of several experimental and simulated engine cycles. If errors are neglect due to valve overlap and pressure variation from cycle to cycle for both experimental and simulated data are considered, it will be noticeable that errors are smaller. This can be seen in Figure 3.8. It is also noticeable that the pressure peak predicted by the simulation lies between the real values taken from the stand. However, the error of the pressure prediction when the combustion is initiated is still present.

Given the good overlap of the graphics, these will process and analyse the 3D files. The post-processing is performed by the Converge CFD program and the visualization of the 3D CFD analysis results is carried out with the Tecplot program. Tecplot is composed by software tools for visualization, analysis software and post processing.

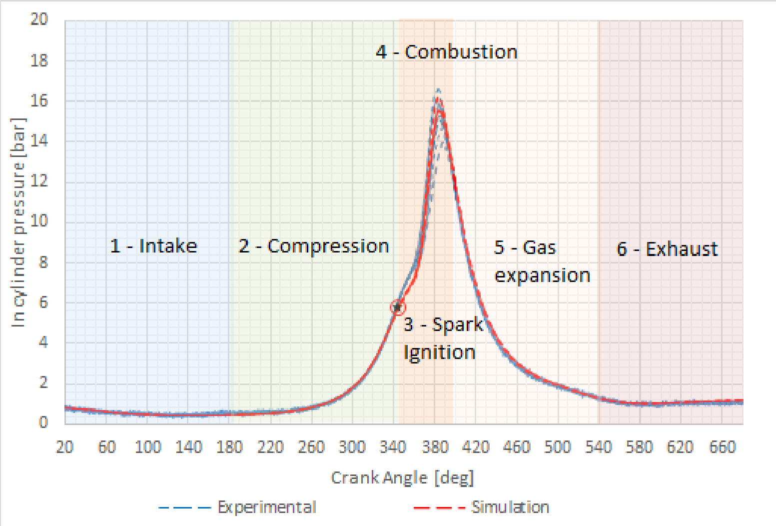
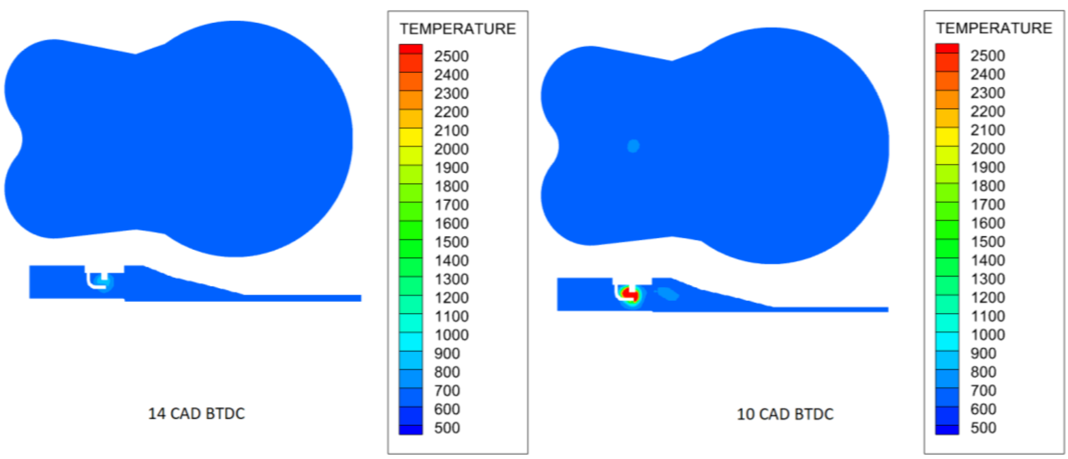


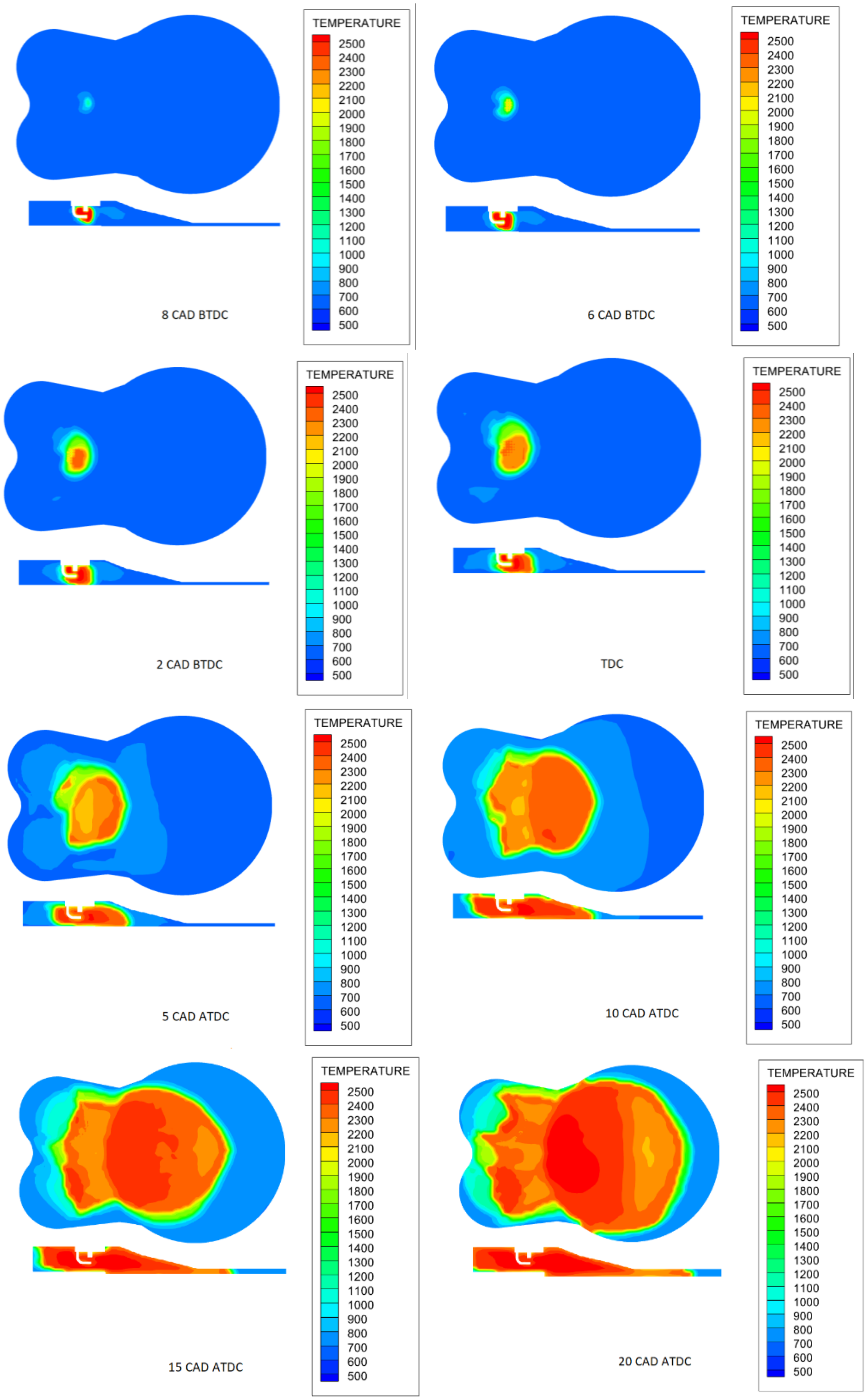
Figure 3.8. In cylinder pressure multiple cycle

The analysis of the temperature in the cylinder and the formation of the flame core are performed with Tecplot software. Figure 3.9 shows the combustion process for the analysed engine. The spark is produced with a 15-degree advance, as in the real case. For a better visualization two sections have been made in the 3D model. The first one is a top view representing a section through the cylinder head gasket. The second view is a side view with a section through the middle of the spark plug.

It can be seen in Figure 3.9, that the flame core formation has a duration of almost 5 CA: from 15 CAD BTDC (when the spark is produced) to 10 CAD BTDC. According to the experimental data, for the engine type analysed, the flame core formation is 0.5-1.0 degrease at 3000rpm. This could be one of the causes of the pressure differences between the experimental and simulated model in the range 15 CAD BTDC to TDC. Although a chemical reaction model with rapid combustion initiation was chosen, it seems that the spark advance should be increased more to compensate this delay.

Cold engine walls and large combustion chamber surface area produce freezing of chemical reactions in the vicinity of the walls, as can be observed at 30 - 45 CA ATDC. This was particularly observed in experimental tests with an absurd high compression ratio for this engine. Indeed, the cylinder head in the area furthest from the spark plug did not show any hydrocarbon deposits on the walls. This means that no combustion took place in that area. In view of this, the fuel combustion in the cylinder should be analysed.




TEMPERATURE
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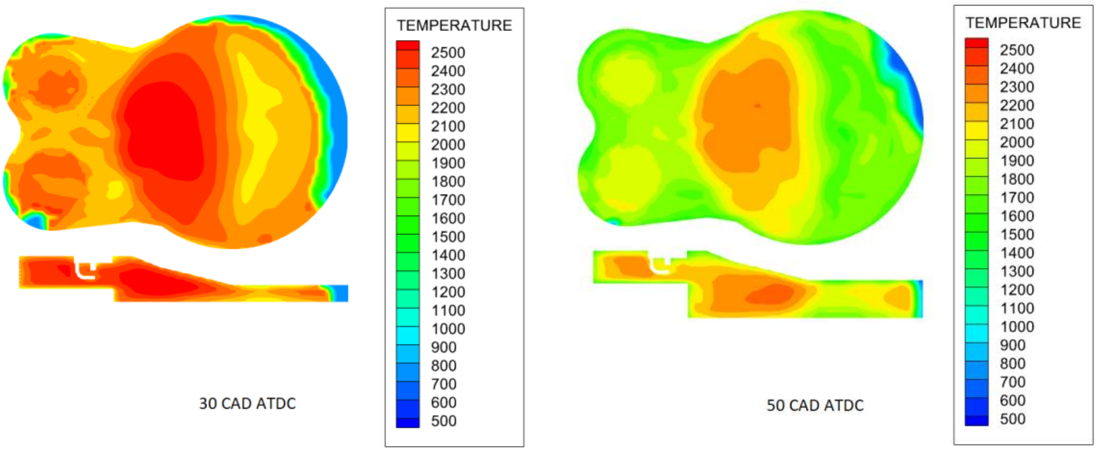
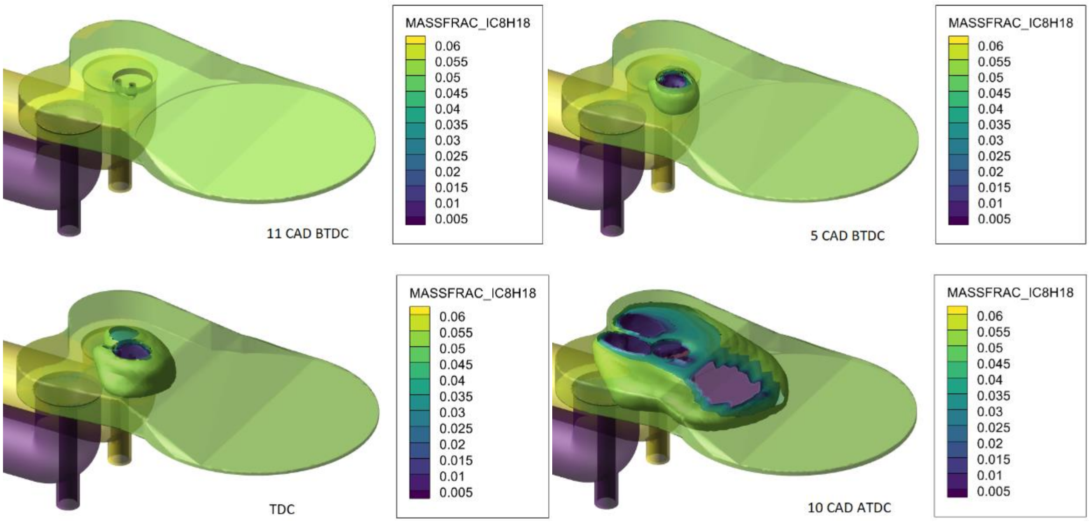


Figure 3.9. In cylinder temperature – CFD Simulation

Figure 3.10 shows the mass fraction for the used fuel (iso-octane iC_8H_{18}). For a better visualization of how the fuel is consumed in the cylinder, three iso-surfaces have been created for the following values: 0.01, 0.03 and 0.05. The iso-surface represents a surface created for constant value points. This is a three-dimensional analogue for isolines.

If previously was stated that for the simulation the combustion starts at 10 CAD BTDC, it can be seen in Figure 3.10, that we have a first iso-surface for a mass fraction of 0.05 at 11 CAD BTC. It indicates us that the combustion process starts 1CAD earlier. At 10 CAD BTDC it is clear how the flame core grows and consumes the fuel in its vicinity. Analysing the mass fraction at the end of combustion shows that there remains unoxidized fuel in the region furthest from the spark plug. This is also seen in Figure 3.9 where the temperature in that area is lower than in the rest of the combustion chamber.



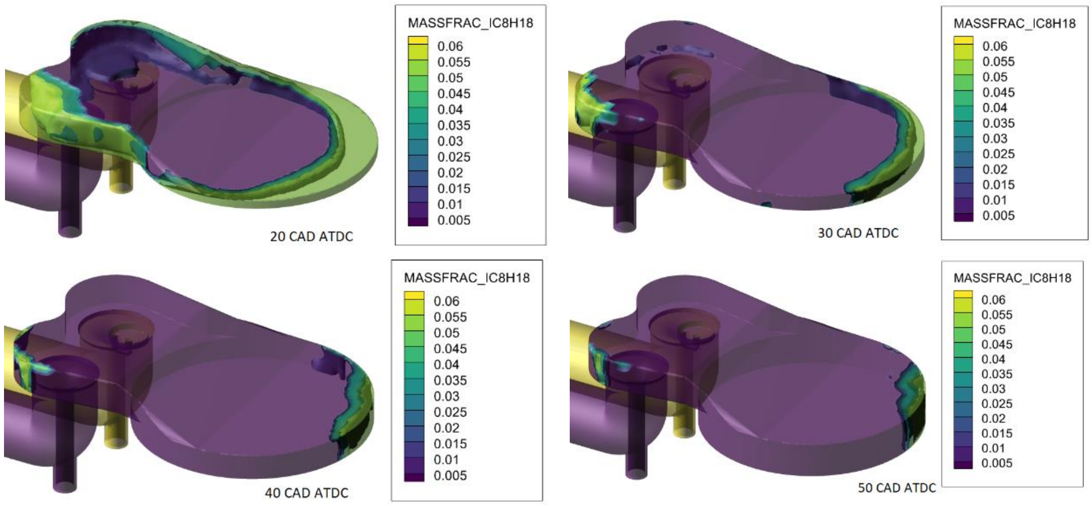


Figure 3.10. Fuel mass fraction

3.8. Partial conclusions

The predictability of cylinder pressure evolution depends on many variables introduced in the simulation. Thus, the first thing analysed is the engine geometry which can introduce errors from the beginning of the simulation. Engine wear and construction tolerances can also introduce errors. The quality of the mesh is closely related to the turbulence model used. If the fluid introduced into the cylinder is not pre-mixed (spray simulation is used) special attention should be paid to the eddies and how the mixing of the waste gas with fresh air and the sprayed fuel is achieved.

The combustion simulation in spark ignition engines is dependent on the chemical reaction mechanisms and the gasoline surrogate used. The predictability of in cylinder pressure evolution is closely related to these mechanisms. Complex chemical reaction models require high computational resources.

The heat transfer model has a high impact on the predictability of cylinder pressure as it influences the amount of fuel captured in the cylinder, the initiation of combustion and the release of heat into the cylinder.

Finally, the numerical model applied to predict cylinder pressure used a RANS - RNG turbulence model, a wall heat transfer model was the one proposed by Angelberger, and a simple chemical reaction mechanism using iso-octane as fuel was used to simulate combustion. Several simulations were run, and the predicted average cylinder pressure was compared with the experimentally obtained average cylinder pressure. The absolute error for the cylinder pressure prediction was -0.05 bar and the relative error was -2.76%. The negative sign indicates that the average pressure obtained from the simulation is higher than the average pressure obtained experimentally. However, if the pressure variation of the engine cycles (experimental and simulated) is con.sidered, it will be observed that the overlapping of the graphs is better.

Chapter 4. Conclusions, contributions and recommendations

4.1. Final conclusions

The studies of the current works show that water injection at full load improves the performance of internal combustion engines. The most representative results are increased maximum power and lower specific fuel consumption. Internal engine cooling allows the engine to run with a higher ignition advance (or injection advance for diesel engine). Most of the articles in the field does not publish all the information and adjustments on the engine for water injection and non-water injection cases. In these cases, repeatability of tests is impossible. It has also been noticed that water injection has a knock inhibition effect.

In terms of pollutant emissions, an increase in hydrocarbon (HC) emissions and a decrease in nitrogen oxide (NOx) emissions were noted. The decrease in NOx emissions is due to the decrease in combustion chamber temperature. A decrease in the exhaust gas temperature (EGT) has also been observed due to a change in the chemical composition of the engine flow.

Experimental tests have shown that the effects of direct water injection depend on the timing of water injection and the water/fuel ratio used. Thus, the following clarifications can be made:

- Direct water injection performed before ignition events can delay ignition by up to 2.3 degrees. Without water injection, the delay is 0.5 degrees at 3000 rpm.
- Direct water injection performed before the spark increases the combustion time.
- Direct water injection after ignition shortens the total combustion time.
- Direct water injection at the mid-late burn 'kills' combustion in the cylinder.
- Direct water injection performed at the end of the combustion process can increase engine performance by 2.3%. At the same time, if the amount of water injected is higher, it will cause internal cooling of the engine and reduce the pressure in the cylinder (water absorbs heat from the combustion gases and not only from the metal surfaces of the engine).

Intake manifold injection is the easiest to implement on series-produced vehicles and is also low cost. It has been observed that water injection improves combustion for the compression ratios analysed. The ignition timing for the experimental tests was adjusted to the knock limit in order to obtain the maximum performance for each determination, and the air-fuel mixture was always stoichiometric by changing the fuel injection time rather than by changing the intake air flow (changing the throttle position). While analysing the extracted diagrams during the determinations, it was found that the optimal water/fuel ratio depends on the compression ratio used. For a compression ratio of 7.5 it was found that the highest amount of energy released during the combustion process is when a water/fuel ratio between 0.71 and 1.00 is used.

As the compression ratio increases, it has been observed that the optimum water/fuel ratio decreases. According to the experimental data, for a compression ratio of 11.7 the highest amount of energy released is for a water/fuel ratio between 0.42 and 0.71, and for a compression ratio of 13, the best results recorded are for a water/fuel ratio of 0.42. Comparing these results, it can be noticed that water injection in the intake manifold improves combustion especially in difficult combustion regimes.

On the other hand, if the whole engine cycle is analysed, it is noticeable that the maximum indicated work is for the no water injection case, because the pumping losses due to water/steam in the cylinder increase. These can be eliminated by adapting the camshaft profile to the new working conditions or by adding a turbo compressor. Internal engine cooling was possible with both types of water injection. Maximum spark advance depends on the water-fuel ratio used.

In terms of pollutant emissions, it was observed that the maximum NOx level produced depends on the compression ratio, ignition advance, and water/fuel ratio used. Thus, the increase of ignition advance tends to overheat the engine, and the NOx level increases greatly. If the advance is too high and the combustion is with knock, the NOx level does not have the same aggressive increase. At the same time, there is an increase in CO and HC levels and a decrease in the level of CO_2 produced.

For a water/fuel ratio of 1.0 it was noticed that NOx emissions decrease by up to 80%. On the other hand, if an optimal water/fuel ratio is achieved depending on the compression ratio, it is observed that the CO level decreases, the CO2 level increases, and the HC emissions have a small variation.

The numerical model was performed in Converge CFD for combustion without water injection. Cold flow simulation shows that the turbulence models analysed predict turbulence of different intensity. The highest turbulence in the cylinder is predicted by the RANS turbulence model followed by LES and DES. At the same time if we analyse the evolution of the velocity field, we notice a different formation of eddies for the three models. These differences can have an influence if we analyse the mixing of fuel with air, or in the case of direct injection into the cylinder. Since a premixture is used to simulate engine combustion, the influences of these turbulence models are minimal.

The comparative study of the reaction mechanisms shows clear differences between the models used. If complex combustion models are used, more computational resources or more time is needed to solve the equations. On the other hand, simple combustion models do not recognise all the reactions of the real fuel, but the equations can be solved on workstations within a reasonable time.

The influence of heat transfer models on cylinder pressure predictability is significant. Firstly, the heat transfer model influences the pressure predictability during the hot gas expansion. Secondly, according to the heat transfer models simulations strongly influence the flame core formation and the initiation of reactions in the cylinder by modifying the way heat is released into the cylinder. The maximum pressure in the cylinder and the pressure during gas expansion will be strongly influenced by the change of the heat release curve. To reduce these differences, the combustion time is limited, according to experimental data.

The predictability of cylinder pressure evolution is dependent on all the models presented above. Finally, it was chosen to use iso-octane primary fuel along with a simple chemical kinect model for the final numerical model. The chosen turbulence model is RANS RNG, and for the wall heat transfer model it was chosen the model proposed by Angelberger. After calibrating the numerical model with the input data taken from the stand, several simulations were run. The absolute error between the experimental data and the numerical model is -0.05bar, and the relative error is -2.76%. The negative sign indicates that the average pressure obtained from the simulation is higher than the average pressure obtained experimentally. However, if the pressure variation of the engine cycles (experimental and simulated) is considered, it will be observed that the overlapping of the graphs is better.

4.2. Personal contributions

The literature and scientific works have shown that there is insufficient public information about the impact of water injection on combustion in spark ignition engines. There are also shortcomings in terms of pollutant emissions and engine performance at partial loads.

Water injection systems used in the literature and the limitations of those systems have been identified. Thus, a first picture of the experimental water injection stand that must work with the two types of injection individually was outlined. Another condition was that all the operating parameters of the engine and water injection systems could be controlled and monitored in real time.

The second year of my PhD began with the challenge of developing an experimental water injection stand at the University of Minho, in Portugal. There, I had to put into practice what I had studied and at the same time to study related areas in order to solve problems that arose during the completion of the experimental stand or during experimental tests.

Performing experimental tests requires a fully functional stand. So, first of all a complete overhaul of the internal combustion engine was done and, on this occasion, measurements of the parts were made in order to realize the 3D model. Technical problems were solved, reassembled on the stand along with the electric brake and the related sensors.

To modify the compression ratios, the cylinder head were designed and manufactured with a smaller combustion chamber (to increase the compression ratio). Due to the long waiting and production time on CNC machine tools, some of the cylinder head were produced on the machine tools available in the laboratory. Replacing the cylinder head also requires changing the cylinder gasket. Changing the shape of the combustion chamber required redesigning the shape of the cylinder gasket. Due to the long lead time (more than one month) for laser cutting, the cylinder gasket was handmade from metal (copper and aluminium) in different thicknesses (to be able to adjust the compression ratio).

During experimental tests with water injection in the internal combustion engine, technical problems started. Due to the cyclic thermal shocks amplified by direct water injection it was necessary to increase the tightening moment of the bolts, higher than the value recommended by the manufacturer. The engine block is made of an aluminium alloy with low mechanical resistance. For this reason, it was necessary to modify the bolts clamping system to work under the new conditions and the higher torque.

Exhaust gas analysis was possible due to additional safety measures. In order to reduce the amount of water that can get into the pollutant emission analyser, a gas cooler with condensate separation was developed. The purpose of this device is to protect the sensors in the pollutant emission measuring station during experimental tests with water injection, as the amount of water in the exhaust gas is much higher than under normal operating conditions.

There were performed changes of the electrical installations for engine control and data acquisition to add additional sensors according to the experimental tests. There were carried out as well air-fuel mixture and ignition maps for the studied engine according to the compression ratio and water/fuel ratio used.

The way of performing the experimental tests and data acquisition, along with the choice of direct water injection strategies proves the originality of the PhD thesis, as it addresses scientific topics that have not been studied yet.

3D design of the experimental stand, and generation of the engine geometry for finite element and CFD analysis have been performed. Presentation of the project to the Converge team and request of technical support and academic license for the numerical model of the thesis.

Performing numerical simulations and comparative studies to analyse the impact of chemical reaction models on heat release during the combustion process and cylinder pressure predictability. The study of the differences between thermal wall models (GRU-Unimore model, Angelberger model, O'Rourke model and Han and Reiz model) and their impact on combustion and cylinder pressure predictability have been caried out. Comparative study between LES, DES and RANS turbulence models and analysis of flow velocities in the cylinder have been performed as well.

4.3. Recommendations and perspectives

The thesis addresses a topic of current interest and opens new research directions in the field of internal combustion engines:

1. Internal combustion engines will continue to be used as the main source of mechanical energy and can be used as range extender for electric vehicles until the charging network infrastructure is developed. In terms of using ICEs as range extenders, attention should be paid to engines with small cylinder capacity.
2. Study of water injection in heavy duty internal combustion engines operating at low revs.
3. Study of direct water injection effects in engines using the Atkinson cycle or for engines using ceramic coating thermal barriers.
4. Study of direct water injection effects in engines using bio fuels or fast burning fuels.
5. Study of water injection effects in engines with variable compression ratio.
6. Study of water injection effects in internal combustion engines using OHC or DOHC timing systems.

The numerical model developed covers part of the experimental tests performed. The numerical study can be further analysed for the following scenarios:

1. Numerical model check on side-valve engines with high and very high compression ratios.
2. Study of turbulence when spraying fuel is performed.
3. Turbulence and combustion study when using water injection in the intake manifold.
4. Turbulence and combustion study when direct water injection is used.
5. Study of the chemical composition of the waste gases when water injection is used and the impact on the subsequent cycle.
6. Induction of additional turbulence in the cylinder and its impact on combustion.

List of Publications

Web of Science

1. **Frătița, Michael**; Popescu, Florin; Martins, Jorge; Brito, F. P.; Costa, Tiago, *Direct water injection and combustion time in SI engines,* Published in Energy Reports 2021, Tmrees, EURACA, 28 to 30 May 2021, Athens, Greece
DOI: 10.1016/j.egyr.2021.07.061

2. **Frătița, Michael**; Popescu, Florin; Martins, Jorge; Brito, F. P.; Costa, Tiago*, Water injection as a way for pollution control,* Published in Energy Reports 2021, Tmrees, EURACA, 28 to 30 May 2021, Athens, Greece
DOI: 10.1016/j.egyr.2021.07.099

3. **Frătița, Michael**; Popescu, Florin; Martins, Jorge; Brito, F. P.; Costa, Tiago; Ion, Ion*, Water injection in spark ignition engines-Impact on engine cycle* Published in Energy Reports 2021, Tmrees, EURACA, 28 to 30 May 2021, Athens, Greece
DOI: 10.1016/j.egyr.2021.07.113

4. Calin, Cristina; Ion, Ion V.; Rusu, Eugen; **Frătița, Michael***, Performance analysis of a RDF gasification and solar thermal energy based CCHP system,* Published in Energy Reports 2021, 6th International Conference on Advances on Clean Energy Research, ICACER 2021 April 15–17, 2021, Barcelona, Spain
DOI: 10.1016/j.egyr.2021.06.032

5. **Frătița, M**.; Uzuneanu, K.; Balanescu, D. T., *About I-beam versus H-beam connecting rod design using Inventor Autodesk 2018,* The 8th International Conference on Advanced Concepts in Mechanical Engineering, ACME2018 Iasi, Romania
DOI: 10.1088/1757-899X/444/7/072008

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1. **Frătița, M**., Uzuneanu, K., Popescu, F., Ion, I., *The analysis of the thermal barrier coating using Ansys software,* IOP Conference Series: Materials Science and Engineering, 2020, 997(1), 012143, ACME2020, Iasi, Romania
DOI:10.1088/1757-899X/997/1/012143

2. **Frătița, M**., Popescu, F., Uzuneanu, K., Ion, I.V., Angheluă, C.M*, About Structural and Thermal Analysis of Diesel Engine Piston Using Ansys Software,* IOP Conference Series: Materials Science and Engineering, 2019, NACOT2019, Galati, Romania
DOI: 10.1088/1757-899X/595/1/012041

3. **Frătița, M**., Popescu, F., Uzuneanu, K., Mereuă, V., Ion, I*, Fatigue analysis of the connecting rod in internal combustion engines,* IOP Conference Series: Materials Science and Engineering, 2019, 485(1), 012008, UGALMAT2019, Galati, Romania
DOI: 10.1088/1757-899X/485/1/012008

4. Chirica, I., Angheluta, C.M., Perijoc, S.D., Hobjilă, A.I., **Frătița, M.** *Mesh independence of a transient multiphase fluid-solid interaction,* Journal of Physics: Conference Series, 2019, 1297(1), International Scientific Conference SEA-CONF 2019
DOI: 10.1088/1742-6596/1297/1/012026

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WORK EXPERIENCE

2019 – Present Teaching assistant at „Dunărea de Jos", University of Galați
Department Thermal System and Road Vehicles

2018 – 2019 DA-SPACE Open Innovation to Raise Entrepreneurship Skills and Public Private Partnership in Danube Regio
Academic mentor in the project

2016 – 2018 Engineer at former ArcelorMittal Galați (LIBERTY Galati)
ArcelorMittal Galați – UTILITY Department
2017 – 2018 Process engineer at Turbogenerators and Turbo-blower
2016– 2017 Exploitation engineers at the Blowing Thermal Power Station

2015 Project assistant at ArcelorMittal Galati 2015 (2 months of Internship Program)
Project assistant on increasing the reliability of pumps supplying demineralised water to superheated steam boilers.

EDUCATION AND TRAINING

2018 – Present PhD at University "Dunărea de Jos" of Galați
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Training course on leadership skills teamwork, organizing and coordinating a team, time management, conflict management, goal setting, performance management

2016 - 2018 Master at University "Dunărea de Jos" of Galați
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2012 – 2016 Undergraduate studies
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Native Language: Romanian

Other language(s)	UNDERSTANDING		SPEAKING		WRITING
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English language	B1	B1	B1	B1	B1

SOCIAL SKILLS AND COMPETENCES

Digital skills CFD	Converge CFD Independent user	Ansys Independent user	Tecplot Independent user	
Digital skills FEM	Autodesk Inventor Expert user	Ansys Independent user	Autodesk Nastran Basic user	
Digital skills CAD/CAM	Autodesk Inventor Expert user	Solid Works Independent user	AutoCAD Independent user	CATIA V5 Independent user

Competences: The AMG and Unilever Engineers' League internships gave me the opportunity to develop my organizational and communication skills, meet deadlines and use analysis charts to see the real problem and find the root of it. During this time, I learned how important the team is, how to listen to my teammates and how to really lead a team.

Driving licence: ▪ B Category

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