PERFORMANCE AND NOX EMISSIONS OF A DIESEL ENGINE WITH WATER INJECTION

PRESTASI DAN PENCEMARAN NOx BAGI ENJIN DIESEL DENGAN SEMBURAN AIR

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Abstract

The emission like nitrogen oxide is an unwanted pollutant from diesel engines. One out of many methods to decrease the nitrogen oxides is the water injection. One measure to inject water into the engine is the injection into the intake manifold. The injection of the water is possible at three different locations at the investigated engine. One injection is with the airflow, the other is against the airflow and the third is towards the airflow. Water storage and the pressure supply for the injector are developed to support the injector with water under pressure. The amount of injected water is being varied from 0 to 100% water/fuel ratio in steps of 20%. Parallel to this work a simulation model of the engine is created to simulate the behaviour of the water injection. The simulation results show a decrease of the nitrogen oxide, carbon monoxide and carbon dioxide emissions as well as an increase of the performance and a decrease of the fuel consumption. These results are compared with the measurement results from the engine on the test bed. The results of the test runs show the same tendency as the simulation. The values of the simulated and measured parameters differ slightly due to estimation in the simulation model and tolerances of the measurement equipment and test bed. The temperature and the nitrogen oxide emission decrease drastically. This benefit has the drawback of an increase of the particle matter emission. The combustion process is slightly affected by the water injection. The performance parameters are very slightly affected with a little decrease of the efficiency and torque. The fuel consumption is constant with the water injection.

Abstrak

Pelepasan seperti nitrogen oksida adalah pencemar yang tidak diingini daripada enjin diesel. Salah satu daripada banyak kaedah untuk mengurangkan oksida nitrogen adalah suntikan air. Salah satu langkah untuk menyuntik air ke dalam enjin suntikan ke dalam pancarongga pengambilan. Suntikan air boleh didapati di tiga lokasi yang berbeza pada enjin disiasat. Satu suntikan adalah dengan aliran udara, yang lain adalah terhadap aliran udara dan yang ketiga ialah arah aliran udara. Simpanan bekalan air dan tekanan untuk penyuntik dibangunkan untuk menyokong penyuntik dengan air di bawah tekanan. Jumlah air yang disuntik sedang berubah dari nisbah air / bahan api 0 hingga 100% dalam langkah 20%. Selari dengan kerja-kerja ini model simulasi enjin dicipta untuk meniru tingkah laku suntikan air. Keputusan simulasi menunjukkan penurunan nitrogen oksida, karbon monoksida dan karbon dioksida pelepasan serta peningkatan prestasi dan pengurangan penggunaan bahan api. Keputusan ini berbanding dengan keputusan pengukuran dari enjin di atas katil ujian. Keputusan larian ujian menunjukkan kecenderungan yang sama seperti simulasi. Nilai-nilai parameter simulasi dan diukur berbeza sedikit kerana dianggarkan dengan model simulasi dan had terima peralatan pengukuran dan katil ujian. Suhu dan pelepasan oksida nitrogen berkurangan secara drastik. Manfaat ini mempunyai kelemahan daripada peningkatan pelepasan zarah perkara. Proses pembakaran sedikit terjejas oleh suntikan air. Parameter prestasi sangat sedikit terjejas dengan penurunan sedikit daripada kecekapan dan tork. Penggunaan bahan api adalah tetap dengan suntikan air.

Glossary

Symbol	Unit	Denotation
A	[-]	Peak cylinder pressure factor
B	[bar/(m/s)]	Mean speed piston factor
BMEP	[bar]	Brake mean effective pressure
BSFC	[g/kWh]	Brake specific fuel consumption
C	[bar/(m/s)2]	Mean piston speed squared factor
c_p	[J/kg K]	Specific heat capacity at constant pressure
$C_{p,m}$	[m/s]	Mean piston speed
$C_{u/cm}$	[-]	Swirl velocity
C_v	[J/kg K]	Specific heat capacity at constant volume
FC	[l/min]	Fuel consumption over time
FMEP	[bar]	Friction mean effective pressure
η	[-]	Efficiency
η_n	[-]	Brake efficiency
η_v	[-]	Volumetric efficiency
H_{vap}	[J/mol]	Enthalpy of vaporization
IMEP	[bar]	Indicated mean effective pressure
i	[-]	Number of cylinder
m_a	[kg]	Mass of air
m_{th}	[kg]	Theoretical mass
m_w	[kg]	Mass of water
m_v	[kg]	Mass of vapor
N, n	[rpm]	Crankshaft rotational speed
n_R	[-]	Number of crank revolutions per power
Q_HV	[MJ/kg]	Heating value of fuel
q_A	[J]	Dissipated heat
q_B	[J]	Combustion heat
q_{Bp}	[J]	Combustion heat at constant pressure
q_{BV}	[J]	Combustion heat at constant volume
ω	[%]	Ratio mass water vapor and mass dry air
Ω	[%]	Ratio water/fuel
P	[kW]	Power
P_e	[kW]	Brake power
p	[bar]	Pressure
$P_{Cyl.max}$	[bar]	Maximum cylinder pressure
p_v	[bar]	Partial pressure of vapor

φ	[%]	Relative humidity
R	[J/kg K]	Gas constant
R_a	[J/kg K]	Gas constant of dry air
R_v	[J/kg K]	Gas constant of vapor
T	[K]	Temperature
T	[Nm]	Torque
τ_{id}	[ms]	Ignition delay time
V	[m3]	Volume
V_d	[m3]	Displaced cylinder volume
W	[J]	Theoretical work
$W_{c,i}$	[J]	Indicated work per cycle

Abbreviations

ASEAN	Association of Southeast Asian Nations
BDC	Bottom Dead Center
BOI	Begin of injection
CA	Crank angle
CFD	Computational Fluid Dynamics
D8	Developing 8 Countries
DOC	Diesel oxidation catalytic converter
ECU	Electronic control unit
EGR	Exhaust Gas Recirculation
EVC	Exhaust valve closing
EVO	Exhaust valve opening
G15	Group of 15
IVC	Intake valve closing
IVO	Intake valve opening
PM	Particle matter
PPM	Parts per million
PWM	Pulse-width modulation
SCR	Selective Catalytic Reduction
TDC	Top Dead Center
TDCF	Top Dead Center Firing
WOT	Wide open throttle

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Motivation

The population of the world is increasing very fast the last 200 years. In July 2014 over 7 billion people are living on the earth and over one billion cars are used by these people [1]. Every day these number increase and the problems caused by this are not getting less. Fewer parking spaces in crowded metropolitan regions, increase of traffic jams and the consumption of fossil fuels are only a few problems caused by the cars. Another serious problem for the most cities and metropolitan regions are especially the pollutants of internal combustion engines. In cities with a lot of traffic the smog is a big issue and is not welcome from the inhabitants. Not only in cities are the smog and the other pollutants not welcome. At many landscapes the nature is suffering from the pollutants. To act against these problems the car manufacturers are required to limit the pollutants emitted by the internal combustion engines. This limitation is done by governments, like the Environment Protection Agency EPA in the USA or whole unions, like the European Union EU. Some countries in South East Asia like Indonesia, Singapore, Thailand, Vietnam and Malaysia orientate to these emission limitations [2]. Over the last years the tightening of emission laws and the increase of fuel prices are leading to a rethinking in the technology of internal combustion engines. Until the year 2020 the European Union will limit the CO_2 emission of new passenger cars about 40% as it is nowadays limitation. This corresponds to an emission limit of about 95 CO_2 g/km [3]. In the past there were a lot of improvements to reduce the emissions such as common rail systems, turbocharger, exhaust gas recirculation, particle filter or selective catalytic reduction. In some conditions this is not enough to reach the latest emission laws. So there needs to be another technology to decrease the amount of emissions. This technology involves the addition of water to the combustion process which leads to a reduction of the local temperature and as a consequence to a decrease of NO_x emission. There are several possibilities to implement the so called water injection. This can be carried out by using a separate injector in the intake, an on-board mixer, direct injection in the combustion chamber or with an emulsion of fuel and water. One of these methods will be further analysed in this project.

1 Fundamentals

Before starting the measurements there are some fundamentals, which are necessary to understand how a diesel engine works and how it is possible to reduce the emissions and fuel consumption by using of a water injection system

1.1 Diesel Engine

1.1.1 History

The diesel engine was invented back in 1894 by the German engineer Rudolf Diesel. His idea was to build an engine with a significantly higher efficiency than steam engines. These steam engines were very popular at that time but they had a low efficiency. To reach the aim of a high efficiency, Diesel's idea is based on the use of an isothermal cycle. This idea goes back to the theory of the French physicist Sadi Carnot. He invented the Carnot process which has a theoretical efficiency of up to 90 %. It took many years for the diesel engine to get accepted by the people. Nowadays, the diesel engine is the most used combustion engine in the world. They are used in fixed-installations for power sets, passenger cars, heavy goods vehicles, construction and agricultural machinery, railway locomotives and ships [4].

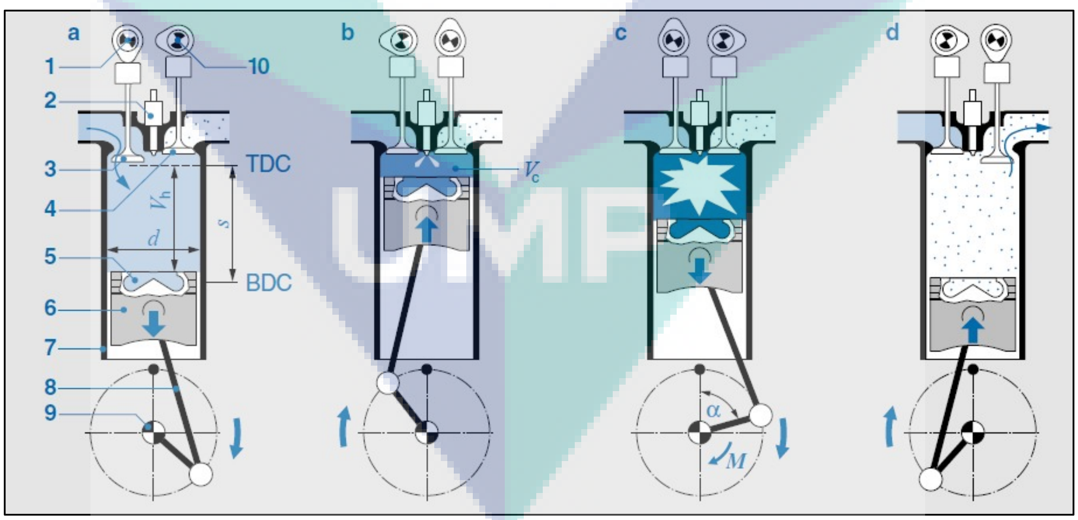
To make them what they are today, modern diesel engines have a turbocharger or a super- charger to increase the output power, high pressure injectors to decrease the fuel consumption, a cooled exhaust gas recirculation and an exhaust treatment with SCR, oxidation catalysts and particle filter to limit the emissions.

1.1.2 Basics

The basic idea of the diesel is the high compression of air in the combustion chamber. By injecting diesel fuel (Heat value ≈ 42.5 MJ/kg) with a very high pressure in the combustion chamber, the mixture of air/fuel will spontaneously ignite. The resulting force pushes the piston down and leads to a moment and movement at the crankshaft. Diesel engines can be used as two- or four-stroke engines.

Figure 1.1 represents the operation cycle of a four-stroke diesel engine. The single strokes are called as follow: induction stroke (a), compression stroke (b), ignition stroke (c) and exhaust stroke (d). In the first stroke the piston moves from the top dead center (TDC) to the bottom dead center (BTC). This movement leads to an increase of the capacity of the cylinder and a vacuum in the inlet manifold. The inlet valve is open and due to the down movement, fresh air flows into the cylinder. The exhaust valve is closed. The second stroke compresses the air. The inlet and exhaust valve are closed. Due to the compression, the air reaches a temperature of 900 ˚C. Right before reaching the TDC the fuel injection system injects the fuel with a pressure up to 2500 bar in modern diesel engines. With the third stroke, the injected diesel fuel begins to ignite and burns. The chemical energy of the diesel fuel is being converted into mechanical energy, which pushes the piston down and this leads to a torque at the crankshaft, due to the kinetics of the piston rod and crank web. The piston is reaches the BDC again. With the last stroke the exhaust is released. The exhaust valve is open. To release the exhaust, the piston is moves back to the TDC and presses the hot exhaust out of the cylinder. The intake valve is closed.

Figure 1.1: Cycle of a four-stroke diesel engine [4]



α	Crankshaft angle	1	Inlet-valve camshaft	6	Piston
d	Bore	2	Fuel injector	7	Cylinder wall
M	Turning force	3	Inlet valve	8	Connecting rod
s	Piston stroke	4	Exhaust valve	9	Crankshaft
Vh	Compression volume	5	Combustion chamber	10	Exhaust-valve camshaft
VC	Swept volume				

1.2 Emissions

Over the last years, the limit for emissions is getting tighter to protect the earth environment. In figure 1.2 the emission limits for NO_x and particle matter over the last years for the emission standards, which are released by the European Union, are shown. These emission limits were issued for light passenger cars with a diesel engine. The limits for gasoline engines are different because the formation of soot and NO_x is lower. For diesel engines, the NO_x and PM are the relevant emissions which have to be reduced. The Euro 1 standard for PM was introduced in 1992. Since then, the PM emission limit dropped until the actual limit in 2014 for the Euro 6 standard. The limits for the NO_x were introduced in the year 2000 with the Euro 3 standard. Since the Euro 4 standard, which has been introduced in 2005, the limits dropped by 20% compared to 2009 with the introduction of the Euro 5 standard, and up to 50 % with the introduction of the Euro 6 standard in the year 2014.

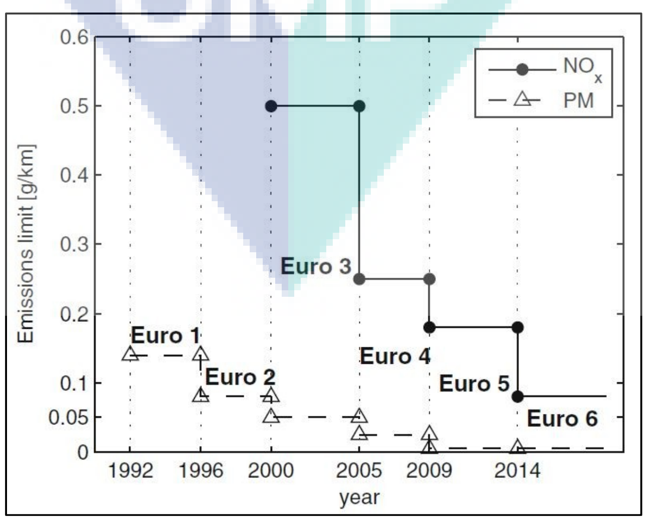


Figure 1.2: Changes of NO x and PM emission limits for diesel passenger vehicle cars since 1992 [5]

To understand what emissions are and how they are formed, the following section gives an overall view of the emissions which are produced during the use of a combustion engine. The burning of an air/fuel mixture, under ideal circumstances, produces water vapour and carbon dioxide. In equation 1.1 the formula for the ideal, complete combustion is shown [6].

$$n1CxHy + m1O2 \rightarrow n2H2O + m2CO2 \tag{1.1}$$

In this equation, the fuel represented with the expression C_XH_Y, reacts with the oxygen O_2 to harmless water H_2O and carbon dioxide CO_2. The single elements are in balance for the ideal combustion, if sufficient oxygen is available. These components have an amount of 10 to 20% of the total exhaust mass. The rest of 80% are atmospheric nitrogen and surplus oxygen. CO_2 is not classified as an emission because it is a natural gas which is a component of air. However it is responsible for the greenhouse effect and global warming. Regulations limit the CO_2 emissions and therefore it should be investigated. Naturally not every combustion takes part under ideal conditions and so some pollutants are produced. These are hydrocarbon (HC), carbon monoxide (CO), nitrogen oxide (NO_X) and particle matter (PM). These components represent an amount of 0.1% of the total exhaust mass in part-load. Figure 1.3 shows a scheme of the several components of the whole exhaust mass shown

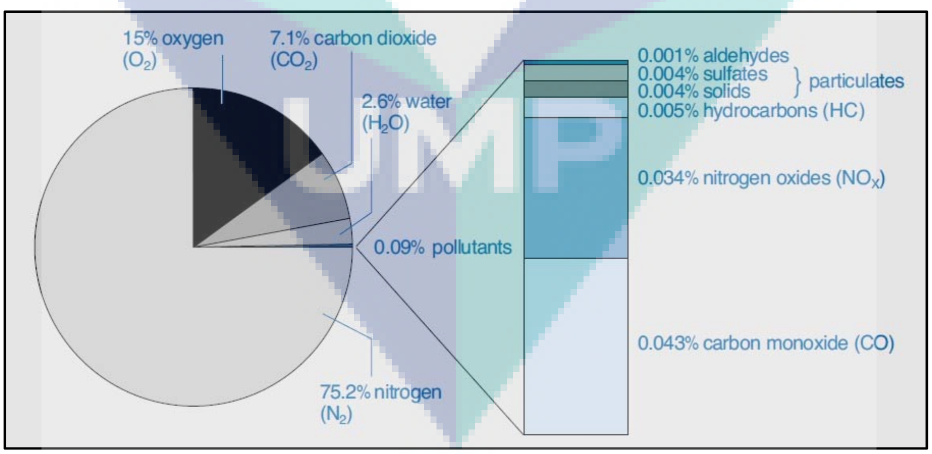


Figure 1.3: Exhaust-gas composition of a Diesel engine under part-load conditions (Figures in percent of weight) [4]

HC Hydrocarbons are the result of an incomplete combustion and are always found in the exhaust gas. They are formed in the combustion chamber where a high surplus of fuel exists and the wall temperatures are cold. These cold regions are especially near to the cylinder wall. In this region the combustion flame does not burn the surplus fuel. Furthermore HC is formed during a late and low velocity injection due to the fuel nozzle. The limits for the HC emissions are not as tight as the limits for NO_x and PM. To reduce the HC emission, less investigation work has to be done. Some hydrocarbons, like aromatic hydrocarbon, can lead to lung, skin or stomach cancer.

CO Carbon monoxide is the result of rich air/fuel mixture combustion due to a deficiency of air. Some CO is produced in conditions with air excess, but only minimal or is formed during a short period of rich operations. As diesel engines are always operated on the lean side of stoichiometric the emissions of CO are negligible low. Even if there is some CO left, the use of a diesel oxidation catalytic will convert the remaining CO to harmless substances like carbon dioxide CO_2 and water. For humans it is dangerous because it prevents the ability of blood to absorb oxygen, which leads to asphyxiation.

NO_x Nitrogen oxide is a combination of different nitrogen and oxygen compounds, such as nitrogen oxide (NO), small amounts of nitrogen dioxide (NO_2) and very small parts of other nitrogen-oxygen combinations, summed up as NO_x. It is a very unwanted emission and due to the harmfulness of NO_x, it is an aim for engineers to reduce the amount of NO_x, which are produced by the engines of the cars. In the atmosphere it destroys the ozone layer and is responsible for acid rain. For humans it is dangerous for the lung tissue and reduces the lung function. For the formation of NO are responsible a number of possible reactions [7].

$$
\begin{aligned}
O + N_2 &\rightarrow NO + N \\
N + O_2 &\rightarrow NO + O \qquad (1.2) \\
N + OH &\rightarrow NO + H
\end{aligned}
$$

The equations 1.2 show that oxygen can react with nitrogen to nitric oxygen and atomic nitrogen, Atomic nitrogen can react with oxygen to nitric oxygen and atomic oxygen. Furthermore the atomic nitrogen can react with hydroxyl radical to nitric oxygen and atomic hydrogen. These are only possible reactions and it cannot be predicted which

Reactions take place during the combustion. Another important fact for the formation of nitrogen oxides is the temperature of the combustion. A higher temperature leads to a higher NO_X emission. This process is shown in figure 1.4. The formation of NO starts actually at 1200 K.

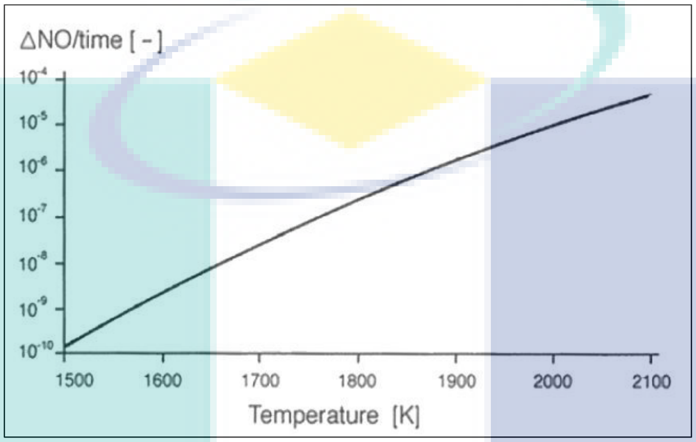


Figure 1.4: NO formation over the temperature [8]

PM Particulates are solid carbon soot particles and are generated in the fuel rich zone in the cylinder chamber during the combustion. Compared to the other emissions particles can be seen as smoke and they are undesirable odorous emissions. Soot is always formed during the combustion process. High formation of soot is formed especially at the point when the engine is under load at low engine speed. At this point, the maximum fuel is injected to get the maximum power of the engine and these results in a rich mixture. During idle speed the soot formation is negligible. The particle can be reduced dramatically with the use of swirl flaps, when the engine is under part-load. Particle can lead to cancer and to damage of the lung. To reach the limits of the NO_X and PM emissions, which are displayed in figure 1.2 on page 3, the engineers have to investigate methods to decrease the emissions of diesel engines. In scheme 1.5 the NO_X - PM trade-off is shown. In this scheme the conflict between the NO_X and the PM is shown. With the increase of the NO_X the PM decreases. Otherwise with an increase of the PM the NO_X decreases. Lowering both emissions doesn't seems technical feasible but with some measures it is possible to lower both emissions slightly. Within the use of an exhaust gas recirculation (EGR), a reduced compression ratio or inter-cooling it is possible to decrease the NO_X emissions. Now the conflict starts. The lower NOx emission leads to a higher soot emission. Within the use of a soot filter, it is possible to get nearer to the target. On the other hand it is

Possible to lower the soot emission with the use of variable valve timing, higher injection pressure, multiple injections or the formation of an injection sequence. All these measures therefore lead to an increase of the NO_X emission. This emission can be converted with the removal of NO_X from smoke gas which is called $DeNO_X$. By changing the beginning of the injection it is also possible to change the emissions. A good compromise is the water injection. With this measure the NOx and the soot emissions can be decreased evenly.

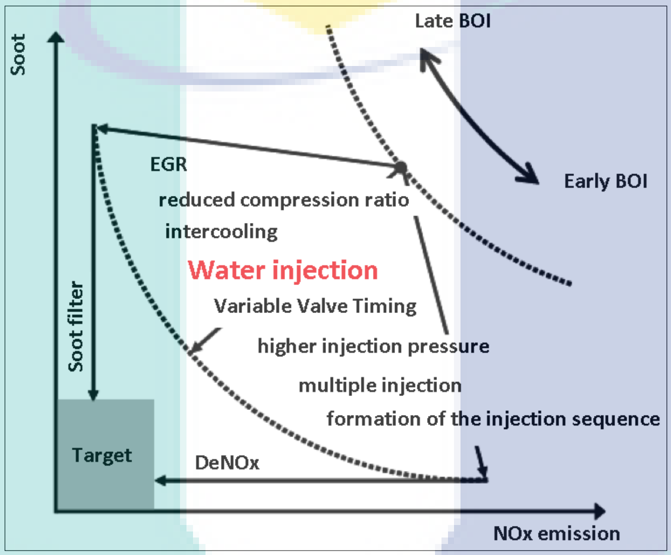


Figure 1.5: NO_X - PM trade-off with different measures to decrease the NO_X or PM emissions, based and translated from [9]

In some cases it is better to find a compromise of all these measures. So there is no need to design an engine that only reduces the NO_X emission to almost zero or the PM. If there is a compromise of the measures, it is may be possible to use a smaller soot filter or the use of flue gas denitrification.

1.3 Water Injection

The insertion of water to the combustion process goes back to the year 1894 [10]. Since that time many research projects have been undertaken. Installed in aircraft engines in World War II [11], the insertion of water lead to a cooling effect on the engine head and cylinder. Furthermore the aircraft engines could be operated at a higher indicated mean effective pressure and therefore the fuel consumption decreased. Other usage of water insertion is applied at marine diesel engines and stationary engines to produce electricity [12].

The water insertion in that case is used to decrease the NO_X and soot emissions and therefore it is possible to use a smaller exhaust treatment. In passenger cars the insertion of water is used to increase the performance [13]. Some companies provide an upgrade package for the tuner scene. To insert the water several methods were investigated during the last years [7] [14]. In the following three methods are described.

Diesel emulsion

Figure 1.6 describes one out of three methods to insert water to the combustion process. Outside the engine there is a blender mixing the diesel fuel with water. The water gets inserted to the combustion chamber via the fuel and the fuel nozzle. The advantage of this system is that there is no need of an extra water injection nozzle. The disadvantage is that there must be an emulsifier used to combine the diesel fuel with the water. According to that, it is very difficult to control the right amount of water in the fuel.Another challenge for the emulsion fuel is that after a certain time the mixture of fuel and water will disintegrate. Some investigations have shown that injecting water/fuel mixture leads to a decrease of NO and soot. These results from the injection of the water/fuel mixture to the zone where the NO is formed [15]. Further investigations have shown that using a 5 % and 10 % water emulsion in biodiesel, the decrease of the NO_X is possible as well. The engine brake power is not affected with this method [16]. Other researches projects have shown that diesel emulsion decrease the NO_X emissions, more than the use of normal diesel fuel without water [17]. In figure 1.10 the results of these investigations are shown

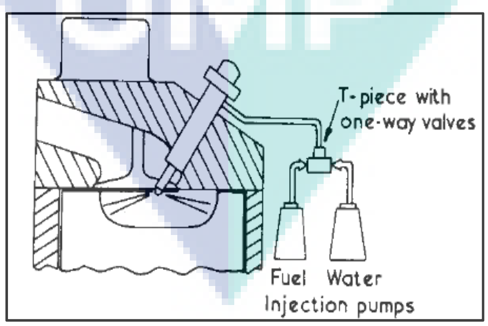


Figure 1.6: Emulsion [15]

Water direct injection

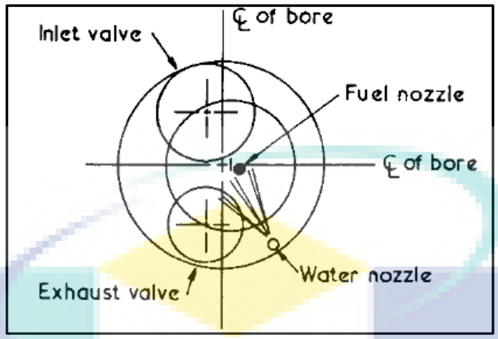


Figure 1.7: Direct injection [15]

In figure 1.7 another system is represented. Inside the combustion chamber another nozzle is installed, additional to the fuel nozzle. With this nozzle it is possible to inject water into the combustion chamber. The cylinder head has to be modified to install the water nozzle. The advantage is that it is possible to inject water at any desired time and the amount can be con- trolled easily by a computer controlled water pump. The disadvantage is that there might be not enough space in the cylinder head to install the additional water nozzle. So this system can possible not be used in certain applications. In some cases it is possible that the direct water injection leads to a dilution of the engine lubricating [15]. Investigations on this system have shown that there is a significant decrease of NO and soot [10]. Other research projects showed a decrease of the NO_X emissions is possible, as well as the HC emission, even in large medium-speed engines [18]. In figure 1.9 the results of these investigations are shown

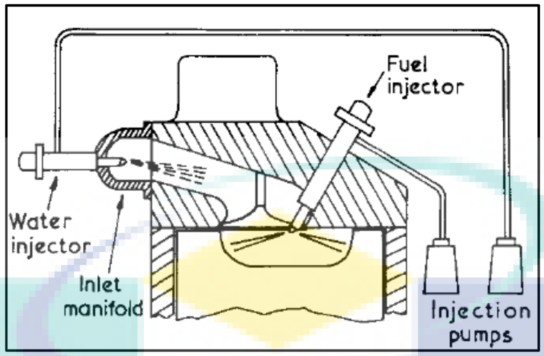


Figure 1.8: Injection to the intake manifold [15]

Injection via intake manifold

The third system is shown in figure 1.8. In the intake manifold an additional injector is installed to introduce the water to the intake manifold. To work with this system, the intake manifold has to be modified to install the water nozzle. Advantage of this system is that the amount of injected water can easily be controlled. If no injection is required, it can be switched off by set- ting the water amount to zero. The disadvantages are almost the same as the disadvantages of the direct injection. The injection of water can lead to a dilution of the engine lubricating. In some cases it is not possible to install the water injection system because there might be no space at the engine. Investigations to this system have improved the reduction of NO_X at low and high loads [19]. Other research projects show the same behaviour in service of engines. In figure 1.10 the results of these investigations are shown.

In figure 1.9 and 1.10 the effects of the three different measures to insert water are displayed especially for the NO emissions. Takasaki [20] investigated the use of the water direct injection into the combustion chamber. With the increase of injected water and at different injection times, the NO decrease up to 80 %. Subramanian [17] investigated the water injection via a diesel- water emulsion and the injection through the intake manifold. In comparison to the use of pure diesel, the NO emission for the diesel emulsion decreases around 65 %. With the use of the injection through the intake manifold, the same decrease of about 65 % is obtained.

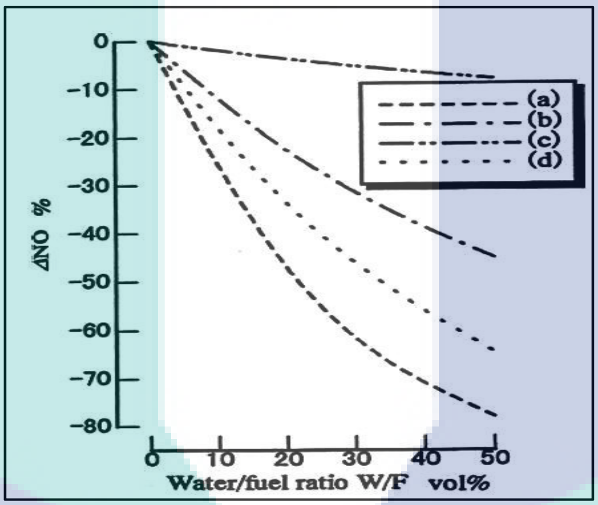


Figure 1.9: Water injection with the direct injection measure [20]

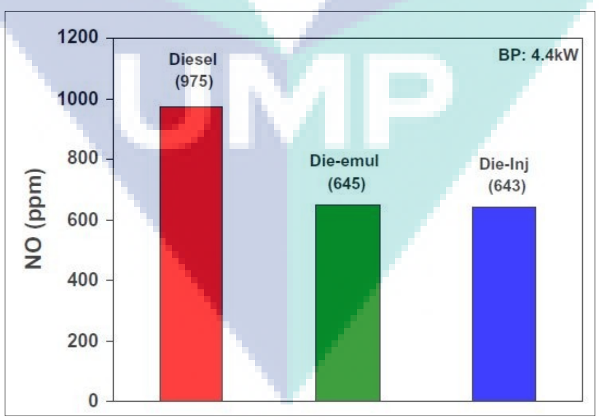


Figure 1.10: Comparison pure diesel, emulsified diesel and water injection [17]

All water insertion systems share the main aspect to reduce the pollution of combustion engines, especially the NO_X emissions. This leads to a higher volumetric efficiency. In some conditions a benefit of this is an increase in power and lower fuel consumption.

The use of these water insertion systems on gasoline engines prevents the knocking and increases the safety of mechanical parts. The advantages and disadvantages of all these systems have to be confirmed in tests and measurements on a test bed.

1.4 Thermodynamic Background

The main reason to insert water to the combustion process is to lower the emission and increase the performance. To understand how this is possible, some thermodynamic know- ledge in this section is described.

Responsible for the decrease of the emissions is a lower process temperature. The lower temperature is the result of the evaporation of water in the combustion chamber. The heat evaporation of water transfers the heat of the air into the water. This cools down the intake air. This causes an increase of the density and this leads to an increase of the mass in the combustion chamber (higher volumetric efficiency). The injected water absorbs the heat and lowers the pressure. The effect is lower work at the compression. Additional benefit of the lower temperature is the reduction of NO_X formation. To cool down the intake air, the correct humidity ratio must be in the intake system. Definition of the humidity ratio ω is the ratio between the mass of water vapour m_W to the mass of dry air m_a, equation 1.3 [21]

$$\omega = \frac{m_v}{m_a} \tag{1.3}$$

Furthermore the humidity ratio can be expressed with the partial pressure of the vapour of Wet air p_v and the total pressure of dry air p.

$$\omega = 0.622 \frac{p_v}{p - p_v} \tag{1.4}$$

with the gas constant of dry air (R_a = 287.08 *J/kgK*) and of vapour (R_v = 461.53 *J/kgK*)

$$\frac{R_a}{R_v} = 0.622 \tag{1.5}$$

At the saturation state, p_v is equal $p_{v,s}$ and x is equal x_s. Another important value is the relative humidity ϕ . It is the percentage ratio of the current pressure of vapor of wet air P_v to the saturated vapour pressure $p_{v,s}$. This ratio can be seen in equation 1.6.

$$\phi = \frac{p_v}{p_{v,s}} \tag{1.6}$$

The relative humidity is a very important value for the operation with water injection systems. If the vaporisation of the water in the intake manifold increases the relative humidity of the saturation point of 100 %, there is the possibility that there is condensate or rather fog. This condensate will lead to uncontrolled behaviour of the combustion process because condensate is not compressible. If the volume of the water is larger than the volume of the clearance volume, this leads to a very high cylinder pressure. To eliminate and act against this, the injection has to be controlled, not to push the relative humidity above 100 %. This information about the state of the air, either dry or wet, and the Mollier diagram give more information about the amount of water to inject and the heat which will be released. The Mollier diagram is shown in appendix A.1. To see how the water injection influences the performance of the engine, a look at the p-V-diagram is necessary.

In figure 1.11 the p-V-diagram is shown. The diagram is based on the Seiliger process and is only a comparison process of the real engine process. To understand how the water injection influences the performance, it is a good comparison. This comparison needs some assumption. The working medium is supposed as ideal gas and there is no flow loss during the gas exchange. The following change of states is involved in the Seiliger process. Between state 1 and state 2 an isentropic compression takes place. Isochoric heat propagation takes place between state 2 and 3. On the way from state 3 to 3' isobaric heat propagation takes place. Between state 3' and 4 an isentropic expansion happens. The last change of state is isochoric heat dissipation between state 4 and 1.

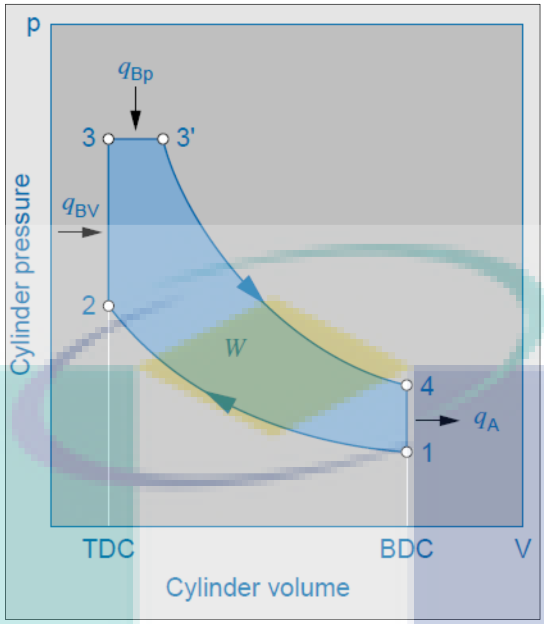


Figure 1.11: Seiliger process [4]

The blue area in figure 1.11 is the theoretical work *W,* which the engine provides to the car. The efficiency η of the engine is described as follows

$$\eta = \frac{q_A}{q_b}$$

Where q_A is the quantity of heat dissipated and q_B is the combustion heat. Furthermore In figure 1.11 and equation 1.8, q_{Bp} is the combustion heat at constant pressure and q_{BV} is the combustion heat at constant volume.

$$q_B = q_{Bp} + q_{BV} \tag{1.8}$$

$$q_{Bp} = m \cdot c_p \cdot (T_4 - T_3) \tag{1.9}$$

$$q_{BV} = m \cdot c_v \cdot (T_3 - T_2) \tag{1.10}$$

The injected water will cause a lower temperature of the whole curve in figure 1.11. Furthermore a greater specific heat capacity leads to an increase of the combustion heat.

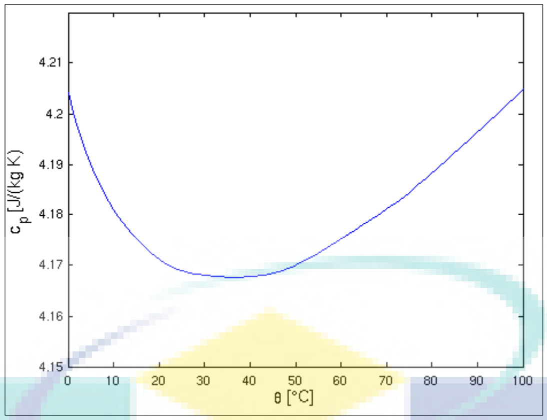


Figure 1.12: Specific heat capacity versus temperature of water [22]

In figure 1.12 the specific heat capacity of water is displayed. The specific heat capacity increases beyond the temperature of $40^{\circ}C$ and below $30^{\circ}C$. At the range of $30-40^{\circ}C$ the specific heat capacity is at the lowest point. The enthalpy of vaporization changes while the water vaporises in the combustion chamber. In equation 1.11 this relation is shown.

$$m_w \cdot \Delta H_{vap} = m_a \cdot c_p \cdot \Delta T \tag{1.11}$$

In this equation m_w is the mass of injected water, ΔH_{vap} the enthalpy of vaporization, m_a the mass of air and ΔT the change of the temperature. The calculation of the temperature change can be done by the following equation.

$$\Delta \mathrm{T} = \frac{m_{\mathrm{w}} \cdot \Delta \mathrm{H}_{\mathrm{vap}}}{m_{\mathrm{a}} \cdot \mathrm{c}_{\mathrm{p}}} \tag{1.12}$$

With an increasing mass of water the difference of the temperatures will increase. So the lowering of the temperature in the combustion chamber is increasing with more injected water. Furthermore the density of the intake air is responsible for the lowering of the temperature, due to the fact that the density changes with the influence of humidity.

$$\rho = \frac{\rho}{\mathrm{R_f} \times T} \tag{1.13}$$

In equation 1.14 p is the ambient pressure, R_f the gas constant of humid air and T the temperature. The gas constant of humid air R_f is furthermore calculated by the following equation.

$$Rf = \frac{R_a}{1-\phi \times p_d / p \times (1-R_a / R_v)} \tag{1.14}$$

Where R_a is the gas constant for dry air with a value of 287, 058 J/kgK [23], R_v the gas constant of vapor with a value of 461, 523 J/kgK [23], _ the relative humidity, p the ambient pressure and p_d the saturation pressure of water in the air.

2 Test Setup

2.1 Facilities

2.1.1 Engine

The engine is a modern type of engine with common rail direct injection, waste gate turbo charger, EGR, swirl flaps and an intercooler. For further research the valve lift was measured. Therefore the intake manifold was disassembled. The intercooler, the injectors, the wiring had to be disassembled as well. The valve lift curves are shown in appendix A.2. The oil which is being used is a standard oil 10W-40. The diesel comes from company Petronas and obtains 5 % biodiesel. Figure 2.1 shows the engine as it is being used on the test bed. In table 2.1 below the engine specifications are listed.

Figure 2.1: Engine on the test bed

Engine Type	In-line 4 cylinder, 4 stroke, Diesel
Maximum Power	103kW / 3600RPM
Maximum Torque	294Nm / 1400-3400RPM
Bore x Stroke	95,4 x 104,9 mm
Compression Ratio	17.5
Combustion Chamber Type	Direct Injection
Charge system	Turbocharged
Fuel system	Common Rail System
Valve Layout	Double Overhead

Table 2.1: Engine specification [24]

2.1.2 Test Bed

To control the load of the engine, an eddy-current type water cooled dynometer is used. The manufacturer is Dynalec Controls and the model is Eddy Current Dynamometer ECB-200F. The position of the throttle is controlled by the test bed tower in the control room. Indicators for the speed and the torque are displayed at the tower.

2.1.3 Measurement Equipment

Recording the data of the measurements are made by portable Windows XP based PC from the brand DEWETRON GmbH. The model Dewe-5000-CA-PROF is used to record the cylinder pressure. To measure the pressure in the cylinder a water-cooled precision cylinder pressure sensor from the company Kistler Instrumente AG, model 6041 a, is used. This cylinder pressure sensor is installed in the hole for the glow plug in cylinder 1. The detection of the TDC is done by a crank angle encorder from the company Kistler Instrumente AG, model 2613B. A Dewe-5000, from the company DEWETRON GmbH, is used to record the temperature of the thermocouple type k elements. Altogether 6 thermocouples are installed, see figure 2.2. To get the information of the actual torque and engine speed during the measurements, a test bed control tower from the company Dynalec Controls Pvt.Ltd. provides this information on a 4-digit, 7-segment display. The fuel consumption in liter is measured with a fuel meter from the company AIC SYSTEMS AG, model AIC- 1204. To display the actual fuel flow a display from the same brand is used, model BC 3034.

The exhaust gas is sucked into a portable gas analyser, which detects CO, CO2, O2, NO and NOx. The gas analyser is made by the company Kane International Ltd, model Kane Auto 4-3. The sampling of the particle matter is done by a high power air sampling pump from the company SKC Inc., model Flite 2. The samples of the particle matter are being weighed on a electronical analytical and precision balance from the company Sartorius AG, model BSA224S-CW. The resolution of this balance is 0,0001 g and is sufficient for the particle matter sample. After sampling the particle matter in the exhaust gas the sample is getting dried in an forced convection oven from the company BINDER GmbH Redline, model RF 53. The actual temperature and relative humidity of the ambient is measured with a USB data logger, from the company Lascar Electronics Ltd., model EL-USB-RT. The water injection system is controlled by an electronical control device, see Bachelor thesis by Edgar Huber [25]. In table 2.2 an overview of the above mentioned measurement devices are listed.

Device	**Model**	**Company**
Cylinder pressure recorder	DEWE 5000 CA PROF	DEWETRON GmbH
Temperature recorder	DEWE 5000	DEWETRON GmbH
Pressure sensor	6041 a	Kistler Instrumente AG
Crank angle encorder	2613B	Kistler Instrumente AG
Fuelmeter	AIC-1204	AIC SYSTEMS AG
Fuel flow display	BC 3034	AIC SYSTEMS AG
Portable gas analyser	Kane Auto 4-3	Kane International Ltd
Air sampling pump	Flite 2	SKC Inc.
Precision balance	BSA224S-CW	Sartorius AG
Convection oven	RF 53	BINDER GmbH Redline
USB data logger	EL-USB-RT	Lascar Electronics Ltd.

Table 2.2: Overview over the measurement devices

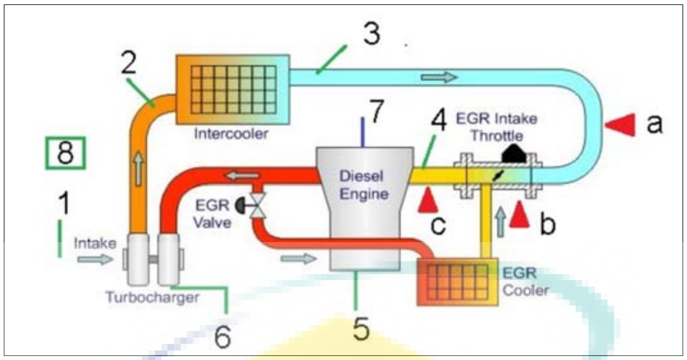


Figure 2.2: Schematic of the engine with the sensors and water injections locations

1	Temp intake air	5	Temp oil		
2	Temp before IC	6	Temp exhaust		
3	Temp after IC	7	In cylinder pressure		
4	Temp after WI	8	Relative humidity sensor		
a	Injector top	b	Injector side	c	Injector bottom

2.2 Experiment Description

The aim of the whole project is to reduce the emission, the fuel consumption and an increase of the performance. To reach this aim the behaviour of the engine has to be investigated on the test bed. Therefore, experiments with the engine have to be done. In order to get more information and data from the engine, to do more experiments, a first experiment has to be done. In table 2.3 the first experiment is shown. The BMEP is set to 2 Bar, which equals a load of torque on the engine of 44 Nm. The location of the injector is at the top to inject the water with the airflow. The beginning of injection starts at t0, which is 20 degree crank angle after the inlet valve opened. At the first experiment the beginning of the injection is a default value because the variation of this variable and the effects of the variation are not investigated at this time. After running the first experiment, more information about the location of the injector, the begin of the injection and the amount of water, after analysing the data, are available.

No	Ratio w/f [%]	m.water	m.fuel	eng.speed
1	0	0	14,35	2000
2	20	2,87	14,35	2000
3	40	5,74	14,35	2000
4	60	8,61	14,35	2000
5	80	11,48	14,35	2000
6	100	14,35	14,35	2000

Table 2.3: Experiment 1 with 2 Bar BMEP, 44 Nm torque (low part-load conditions)

In experiment 2 the BMEP is increased to 3.5 bar, which is equal to a load of torque to the engine of 83 Nm. This load point represents middle part-load conditions. The location is at the top. The beginning of the injection is set to t0, which is 20 deg CA after the inlet valve opened. The overview of the experiment 2 is shown in table 2.4.

No	Ratio w/f [%]	m.water	m.fuel	eng.speed
7	0	0	23,80	2000
8	20	4,76	23,80	2000
9	40	9,52	23,80	2000
10	60	14,28	23,80	2000
11	80	19,04	23,80	2000
12	100	23,80	23,80	2000

Table 2.4: Experiment 2 with 3.5 Bar BMEP, 83 Nm torque (middle part-load conditions)

In experiment 3 the load is varied and is set to 5 bar BMEP, which equals a load of torque on the engine of 119 Nm. The location of the injector is the same as in experiment 1 and 2, at the top. The beginning of the water injection is t0, which is as well as in experiment 1 and 2 at 20 degree CA after the inlet valve opened. The overview of the experiment 3 is shown in table 2.5.

No	Ratio w/f [%]	m.water	m.fuel	eng.speed
13	0	0	32,40	2000
14	20	6,48	32,40	2000
15	40	12,96	32,40	2000
16	60	19,44	32,40	2000
17	80	25,92	32,40	2000
18	100	32,40	32,40	2000

Table 2.5: Experiment 3 with 5 Bar BMEP, 119 Nm torque (high part-load conditions)

With these 3 experiments more data about the influence of the water injection should be available. After analysing the data, the next experiments can be started. The location of the injector is getting changed to different locations to investigate the influence of the different spray directions at these locations. Furthermore the begin of the injecting timing is investigated.

In experiment 4 the location of the injector is set to the side, which allows to inject the water vertical to the air flow. The BOI is set to t0. The BMEP is set to 5 bar BMEP. A overview over the experiment is shown in table 2.6.

No	Ratio w/f [%]	m.water	m.fuel	BMEP
19	Best of	-	-	5
20	Best of	-	-	5
21	Best of	-	-	5

Table 2.6: Experiment 4 with Location Side

In experiment 5 the location of the injector is set to the bottom, which allows injecting the water against the air flow. The BOI is set to t0. The BMEP is set to 5 bar BMEP. An overview over the experiment is shown in table 2.7.

No	Ratio w/f [%]	m.water	m.fuel	BMEP
21	Best of	-	-	5
22	Best of	-	-	5
24	Best of	-	-	5

Table 2.7: Experiment 5 with Location Bottom

In experiment 6 the variation of the injection timing is investigated. Therefore the BOI is set to 20, 40 and 60 deg CA after the inlet valve opened. The engine speed is constant at 2000 rpm. In table 2.8 the parameters of the experiment 6 are displayed.

No	Ratio w/f [%]	m.water	m.fuel	BOI [deg
25	Best of	-	-	20
26	Best of	-	-	40
27	Best of	-	-	60

Table 2.8: Experiment 6 with 5 Bar BMEP and different BOI times

The speed, the torque and the pedal position are read off at the test bed control tower and recorded into the measurement program file. To get the emission of each test point, a little amount of the exhaust gas is carried out to the gas analyser. The emission values for CO, CO2, NO and NOx displayed by the gas analyser are recorded to the measurement program file. The particle matter is measured by the smoke gas analyser. Clean, dry and pre-weight filters are exposed to the exhaust gas. After a time of 90 seconds in the exhaust flow, the filter has to be dried in an oven at 50 °C for 2 hours. During that time period the water steam of the exhaust is dried and only the particle matter is left. Weighing the samples again, the difference between the two masses can be used to calculate the particle matter. Furthermore the current ambient temperature and relative humidity are recorded and saved on the PC.

3 Mechanical Designs

The injection of water to the combustion process can be done by different methods, as described in chapter 1.3. This project focuses on the injection of water via the intake manifold. To inject the water, investigations have been made on which injector is used, where the injector is being located and how the water and pressure supply is been designed. The electronic control unit for the injector is planned, implemented and commissioned by Edgar Huber, see his bachelor thesis [25].

3.1 Planning and Application

The injecting of the water into the intake manifold needs an injector to spray the water. To spray the water two types of injector are selected. The first one is from a petrol multipoint injection system. The second one is from a petrol direct injection system. Petrol injectors are used because they don't need a high pressure pump compared to diesel injectors. To decide which of the two discussed injectors are used, the spray is being analysed. In picture 3.1 the spray of the multipoint injector is illustrated. The injector is operated with 4 bar and 12 Volt, which causes a current of 3.3 A.

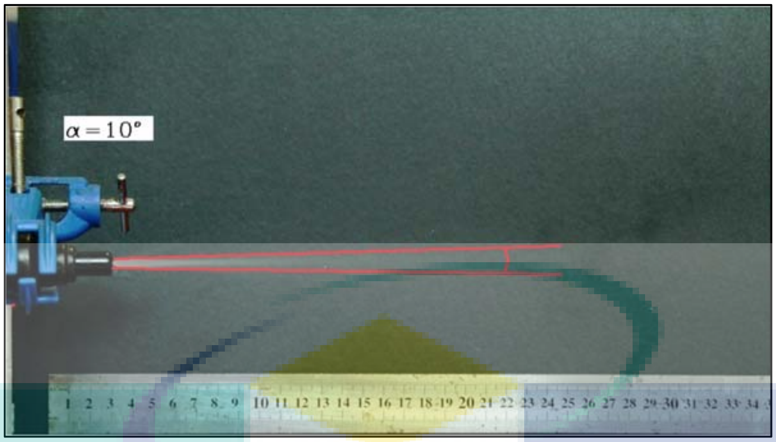


Figure 3.1: Multipoint injector with a spray angle of 10^o

In picture 3.2 the spray of the direct injector is illustrated. The direct injector is operated under the same pressure and current conditions as the multipoint injector. The most significant and visible difference is the spray angle and the dispersion range. While the multipoint injector has a spray angle of 10^o , the direct injector has a spray

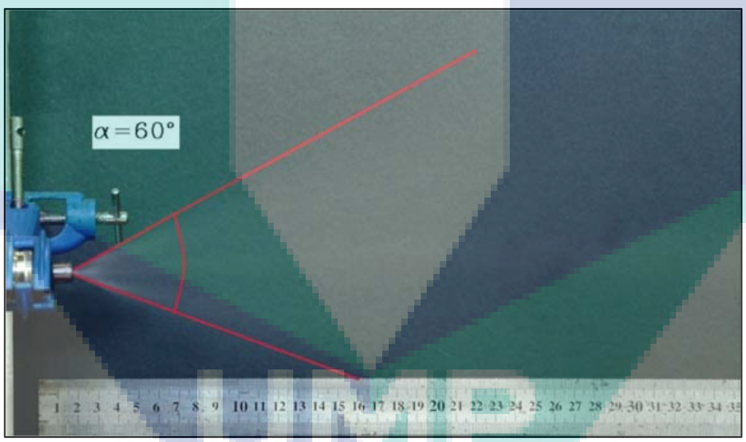


Figure 3.2: Direct injector with a spray angle of 60°

Angle of 60° . The dispersion range of the multipoint injector is a straight beam. This will lead to a wetting of the intake manifold wall. The dispersion range of the direct injector is significantly larger with a spray angle of 60° . Therefore and additional to the air flow, the injected water will be better absorbed by the air. This wide dispersion range is caused by the swirl effect inside the direct injector [26]. According to that, the choice of the injector is the direct injector. With this choice, the mounting and location of the injector is been designed. The injector at the intake manifold can be located in three positions. Each location has its benefits and so the mounting of the injector is designed for each location. The first one is designed to inject the water in the same direction as the air flow and is shown in picture 3.3. The second one is designed to inject the water vertical to the air stream and is shown in picture 3.4. The last location is designed to inject the water against the air flow and is shown in picture 3.5. Which of these locations could be the best solution for this project, will be investigated during the measurements and will be discussed later in chapter 6. For the implementation of the water injection system, it was necessary to disassemble the intake manifold. Through the use of a milling machine the intake manifold is being prepared for the injector. To mount the injectors to the intake manifold, unique holders are created. Furthermore water and the pressure supply are planned. To store a certain amount of water a pressure resistant tank was selected. To display the tank capacity a transparent tube was installed parallel to the tank. The connection for the pressure supply was installed on top of the tank. The connection for the water supply and the connection to the injector are installed at the bottom of the tank. The whole system is shown in figure 3.6. The type of water which is injected is distilled water to act against calcification. The pressure for the total application

comes from the domestic supply of the laboratory and has a maximum value of 7 bar, which is constant and sufficient for the injector to operate well.

Figure 3.3: Injection with the air flow

Figure 3.4: Injection vertical to the air

Figure 3.5: Injection against the air flow

Figure 3.6: Water tank for the water injection

The temperature of the air after the water injection is very important to measure the influence of the injection. This temperature gives information about the effects of the injected water. Therefore a temperature sensor is installed in the intake manifold, direct in front of before the intake valve of cylinder 1. The location will be displayed later.

3.2 Validation

The validation of the water injection system is a very important step to get high quality results, when driving the tests in full operating mode. Therefore pre-tests have to be done. The first test is the control for leakage of the water injection system under pressure. After a time of 1 hour, no leakage and pressure drop is detected and the test was successfully finished. The next test is to check the injected mass of water, which the injector dispenses during a certain time. Therefore the whole injection system with the injector, the tank, the connection and the electronic control device is build up on a separate test device. The pressure for the injector and the tank system is set to 5 bar. The injector spray direction is set towards a measuring cup. The opening time is controlled by the electronic control device via a PWM and the voltage for the injector is set to 13,9 Volt. The whole system is activated for 10 seconds. The opening time gets increased from 1 ms to 7 ms and depends on the adjusted engine speed, in this case to 2000

rpm. After the 10 seconds the measuring cup is weighed on a precision scale. In figure 3.7 the results are shown. The x-axis, which represents the opening time, is spread over the amount of injected water in 10 seconds. The injected water is
linear to the opening time in the range from 1 ms to 7 ms, which makes it easy to calculate the injected water per stroke. Therefore the weigh mass of water per 10 seconds is converted to mass of water per stroke by the following equation, where n is the engine speed in rounds per second.

$$m_{w/stroke} = \frac{m_{w/10sec}}{10*n} \tag{3.1}$$

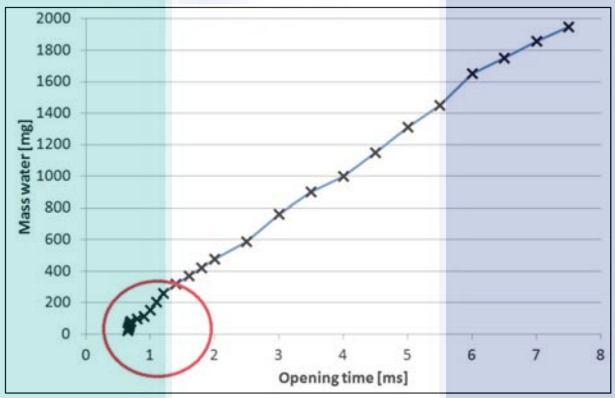


Figure 3.7: Validation of the injected mass of water per second

In figure 3.7 is the range below 1 ms marked with the red circle where the injector is not working linear. In this range the nozzle is operated in the ballistic section [4]. To open and close the nozzle a certain time is required. If the injector opening time is shorter, than the open and close time, the injector has no time to spray water. In figure 3.8 this section is shown. Below 0.6 ms the injector doesn't open. The time to lift and close the needle needs more time, than the injector can inject in that time. This behaviour is considered in the electronical control device.

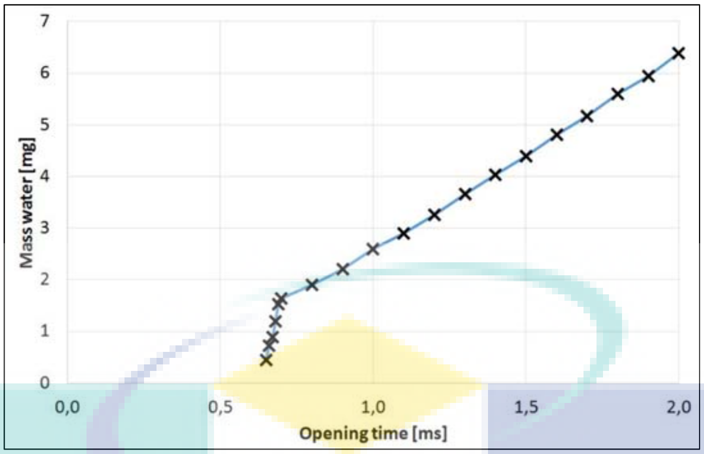


Figure 3.8: Focus of the injected mass of water per second below 2 ms opening time

With this test, the desired mass of water can be calculated to be injected later to the intake manifold. The now desired mass of water can be programmed into the electronic control device.

4 Preliminary and Optimization Work

The testing environment consists of the test bed, the engine and the measurement equipment. At the beginning of this project the engine, the test bed and the measurement equipment were in good condition the engine is a new one and had only been used a couple times on this test bed. The test bed itself was loaded with a smaller engine before. To realise this project, some improvements had to be made, to inject the water and operate the engine. In the following these improvements are described.

The engine is connected with a cardan shaft to the dynometer. The diameter of the shaft was too weak for the power of the current engine on full load and so it had to be replaced with a stronger cardan shaft from a lorry. As mentioned before, the shaft was used before for a smaller engine and therefore the shaft had the right properties. Due to a different diameter of the shaft, the mounting of the shaft on the dynometer and the engine had to be redesigned.

Figure 4.1: Crank angle sensor mounting

Another issue was the improvement of the mounting of the crank angle sensor. At first the mounting was connected with the test bed. This lead to measurement errors because the test bed stands still and the engine vibrates. To solve this problem, the mounting of the crank angle sensor was connected directly to the engine, so the crank angle sensor and the engine vibrate in harmony. The result is showed in figure 4.1

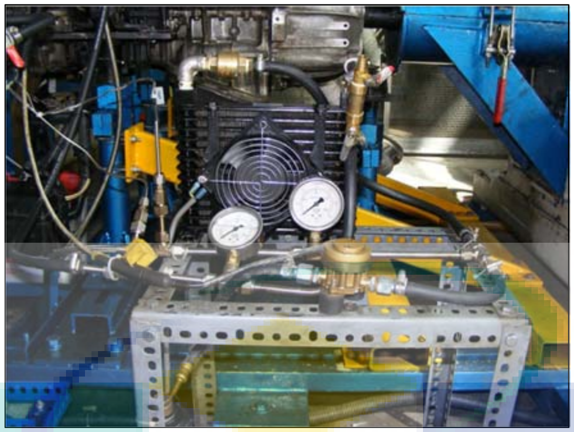


Figure 4.2: Fuel supply system

The fuel system of the engine is a common rail system. The temperature of the return flow is getting very warm, up to $60^{o}c$, noticed during some test runs before. Referring to [27], high input fuel temperatures lead to higher fuel consumption, a decrease in engine power and torque. To act against this, it is necessary to cool down the fuel temperatures. A fuel cooler was applied, shown in figure 4.2.

During a test run, an unusual engine coolant temperature at high load was recognized. The outlet water temperature didn't reach the normal value of $85^{o}c$. After investigating the coolant water supply, the high water pressure of the cooling tower pump was the reason. The thermostat of the engine was always opened, due to the high pressure of the cooling tower pump. So, the coolant water float directly back to the cooling tower and the coolant water had no chance to reach the normal coolant water temperature of $85^{o}c$. Installing a bypass solved this problem. A hose from the input to the output of the cooling tower was applied. Now the main water stream can flow directly back to the output. The engine gets only the amount of water he needs.

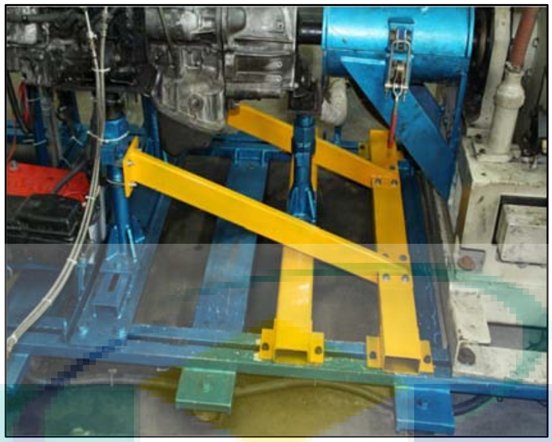


Figure 4.3: Stiffening of the test bed

Some smaller, but necessary, improvements have also been executed. The stiffness of the test bed was too weak for the current engine. The mounting of additional bars and supports solved this problem. To simulate the air stream of the intercooler, which is normally generated by the driving of the car, a fan was mounted directly on top of the intercooler. For safety reasons the ignition key had been installed in the control room. Therefore the wire had been extended and it is now possible to start and stop the engine from the control room.

To do the measurements, which are going to be done later in this project, some changes with the test bed were necessary. To measure the emissions and the particle matter a collecting point is installed after the turbo charger output. This collecting point gets now the raw emissions, which are produced during the combustion. At this point no exhaust after treatment has been done yet and the influence of the water on the emissions can be measured directly and undelayed. The connection between the collecting point and the gas analyser and the air sampling pump is realised with a tube and a hose.

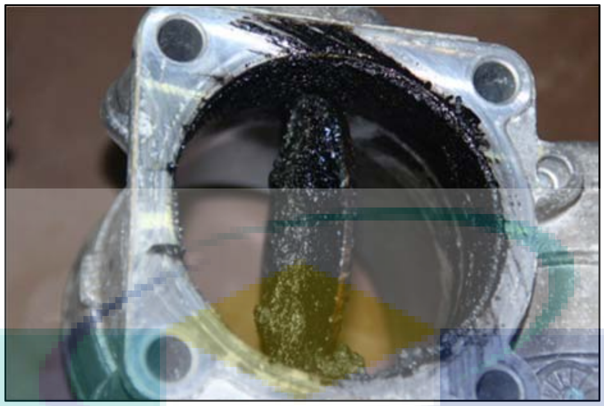


Figure 4.4: Deposits on the intake manifold throttle

The tube is bent to a spiral to cool down the exhaust gas. This is made to protect the gas analyser and the air sampling pump from the hot exhaust gas. A flexible hose is installed to disconnect the sampling line easily from the gas analyser and the air sampling pump. Between the tube and the hose a valve is installed to shut off the exhaust gas stream, if no emission or particle matter is measured. During the disassembly of the whole intake manifold, at some parts deposits out of oil and soot were detected. In figure 4.4 the intake-manifold throttle is shown, with the deposits on it.

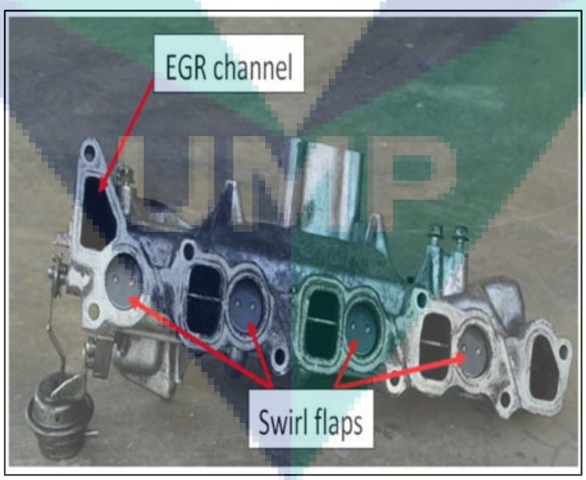


Figure 4.5: Inlet manifold with the swirl flaps and the EGR channel

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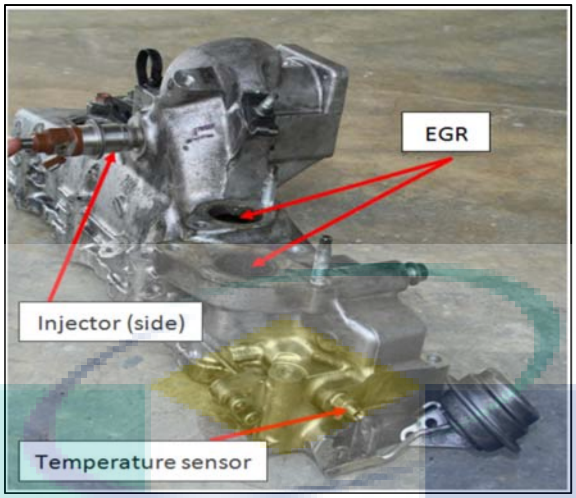

Figure 4.6: Side view of the inlet manifold with the mounting for the EGR, the additional Temperature sensor and the injector for the side location

In figure 4.5 the inlet manifold is shown. For every cylinder there are 2 inlet channels. One is a normal channel; the other is equipped with a swirl flap. They are normally closed, if the engine is driven in part-load. In figure 4.6 the same part is shown, this time from the side view. The location of the injector, the temperature sensor for the measurement of the influence of the water injection after the injector and the mounting and the channels for the EGR are shown in this figure 4.6.

UMP

5 Simulation

Simulations of the real measurements are very good to predict the behaviour of the engine or to calculate how the engine will behave in different situations and conditions. For example it is possible to run the engine at unreal 8000 rpm, or how high the emissions at that speed are. To get this information, it is necessary to simulate and calculate these situations. Therefore simulation software called GT-Power, an application from GT-Suite, is used in this project.

5.1 Introduction GT-Power

GT-Power is an application of GT-Suite and is used to simulate the whole engine to calculate the engine performance, acoustics and the control system. Other applications of GT-Suite are: GT-Cool to simulate the cooling system and the engine heat management, GT-Crank to design a crankshaft, GT-drive to simulate a whole vehicle, GT-Fuel to simulate the fuel system and the general hydraulic system and GTVtrain to design the valvetrain system. The main application is GT-ISE, in which ISE stands for interactive simulation environment and allows the user to build up the model. GT-Post is the post-processing tool of GT-Suite. All these applications and the main software GT-Suite are from Gamma Technologies, Inc [28].GT-Power is a one-dimensional simulation tool which uses the gas dynamics to calculate the heat transfer and the flow in each components of the engine model. To link these components, connection objects are used. The properties of the components are made by the user. To know all properties of the engine, it had to be disassembled, to get all information such as the inlet and outlet diameter and length of each component. In the situation of short time and the fact that other users wants to do their research on the engine, some estimation had to be done. So not all properties in the simulation are 100 % correct, but the estimation is quite good.

5.2 Simulation Parameter

For the simulation of the engine, several components were used to build up the engine model. In figure 5.1 the whole engine model is shown. In GT-ISE there are several object categories for the components. These are divided into the categories flow, mechanical, control, general and thermal. Due to the fact, that some information/data about the engine are not available, these information/data from an example of GT-Power are used. The example is a 4 cylinder, four-stroke, turbocharged direct injection 3.1 liter diesel engine, with common rail and EGR. The data of this example is a good derived model and is suitable for the simulation of the investigated engine.

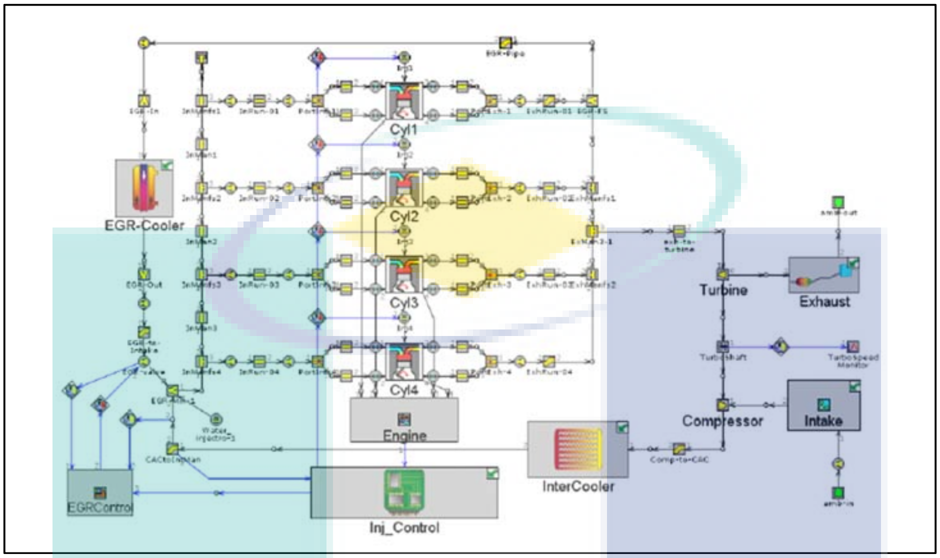

Figure 5.1: Simulation model of the engine in GT-ISE

References References describe the properties of the fluids and gases that are used in the engine. They are not showed on the simulation surface. The most important references for the simulation are gases and incompressible liquids. For the gases the medium vaporised diesel, water, oxygen and nitrogen are relevant. Following attributes must be given. The carbon, hydrogen, oxygen, nitrogen, sulfur and argon atoms per molecule. The reference object *h2o-vap* for example consists out of two hydrogen and one oxygen atom. Furthermore the critical temperature, pressure and the absolute entropy of the substance at a temperature of 298 K must be set. The fluids reference objects in the simulation are *diesel-combust* and *h2o-combust*. For these objects the attributes of the heat of vaporisation, the density and the absolute entropy are relevant. Some components in the simulation need initial state information. Therefore the objects *ambient*, *boost* and *exhaust-ambient* are relevant. In each object the pressure, temperature and composition are entered. The composition of *ambient* and *boost* is another object called *mixture-air* and consists of 76% nitrogen and 23% oxygen.

The exhaust-ambient consists of 97% *air* and 3% *diesel-vap*.

Intake The intake system in the simulation consists of the component intake including the air box and the component of the inlet environment. The air box is designed out of several bundled pipes, which represent a volume. Every pipe is characterized by the inlet diameter, the outlet diameter, the length, surface roughness and the medium flow through the pipe. In this case the medium is air, with an initial temperature of $30^{o}c$ and a initial pressure of 1 bar The inlet environment is characterized by the ambient pressure, the ambient temperature and relative humidity and altitude. For the ambient pressure a value of 1,013 bar , an ambient temperature of $30^{o}c$ and altitude of 0 m (sea level) are used for the simulation. The relative humidity is designed as a variable and can be changed in each simulation case.

Compressor In the compressor object, different attributes can be set. The most essential attribute is the *CompressorMap*. This is a 2-D-Table, where the rotation speed of the compressor is spread over the mass flow rate, pressure ratio and compressor efficiency.

Water injector To inject the water, a component called *InjRateConn Connection* is used to describe the instantaneous injection rate. Any fluid (fuel, water, etc.) can be injected with this component into a pipe or flowsplit. Follow attributes are set in this component. The mass flow rate is defined as a variable, to vary the amount of water during the simulation. The injected fuel temperature is set to $25^{o}c$, which represents the water in the current ambient. The fluid object is called *h2o-combust* and is a reference object. The attribute *Vaporized Fluid Fraction* is the mass fraction of the injected water that will vaporize immediately after injection. The water injector is linked to the EGR-Mix pipe, which is located at the real engine near to the intake manifold and a perfect place for the water injection.

Intercooler The intercooler is modelled by a bundle of pipes to simulate a large surface. This intercooler is an air-to-air intercooler and needs certain airflow to operate. The airflow is modelled through the vehicle speed simulated by a signal generator. The medium which flows through the intercooler is called boost. This is air under a desired pressure and temperature. In the simulation the pressure has a value of 2.2 bar absolute and a temperature of $70^{o}c$.

EGR-System The EGR-System consists of the components EGR Controller, EGR Cooler, EGR mix pipe, EGR valve and a EGR pipe from the exhaust manifold. In the EGR controller the EGR rate is set by the EGR-Rate-Map. The desired EGR rate is a function mapped over engine speed and requested BMEP. The BMEP is set by the accelerator pedal. The EGR Cooler is simulated by a heat exchanger with a Master and a Slave. These are characterized by the flow area, heat transfer area, inlet and outlet diameter and the length. The EGR mix pipe is modelled to collect the flow from the exhaust and the fresh air and leads the mixture to the intake manifold. Volume, surface roughness, wall temperature and the angles of the pipes to the coordination axis are set to model the mix pipe. The EGR valve is a simple orifice, where the diameter is a variable, set by the EGR controller per EGR-Rate- Map. The EGR pipe from the exhaust manifold is a simple pipe, with no special characteristics. During the simulation the EGR-Rate is set to a target of 9%. In the total intake air mass 9% gas is from the exhaust.

Intake manifold The intake manifold is a simple model with pipes and a flow split for the two intake valves. For each pipe the inlet and outlet diameter, the length, surface roughness and the medium inside that flow through the pipe must be provided.

Cylinder, valves and fuel injector The cylinder and the subcomponents intake valve, exhaust valve and fuel injector are used to predict the main issues of the engine, such as the formation of the emissions, the pressure and out of that the heat release and the p-V-diagram. For the intake valve the following attributes are relevant: the diameter of the valve, the valve lash, the cam timing angle, the lift array and the flow array must be set. With the last one the lift array is divided by the valve diameter. The lift array was measured at the real engine before the simulation and the data are involved in the simulation. The same is true for the exhaust valves. The data of the valve lift for the intake and exhaust are in appendix A.2. Attributes of the fuel injector are the injected fuel mass, injection timing, fuel object, injected fuel temperature. The injection timing (BOI) is mapped of the engine speed and the injection mass. The cylinder model is used to specify the attributes of the engine cylinders. The following attributes are relevant: flow medium, wall temperature, heat transfer object and combustion object. These objects will be described later in this chapter.

Engine crank train the engine crank train is used to simulate the kinematics of the crank train and therefore following attributes are relevant: engine type, engine speed, cylinder geometry, firing order and the start of the cycle. The engine speed is modelled as a variable.

Data for the cylinder geometry and firing order are taken from the engine manual of the Isuzu 4JJ1 [24].

Exhaust manifold the exhaust manifold is modelled almost the same way as the intake manifold, except a flowsplit pipe, which leads the exhaust to the EGR Cooler.

Turbine The turbine will predict the output power of the turbocharger, the mass flow rate and the outlet temperature. Attributes of the pressure, temperature, wastegate diameter and the turbine map must be set in this object.

Exhaust system the exhaust system consists of a catalyst and a muffler. For this project the use of these parts are not relevant due to the fact that the emissions after the exhaust manifold are investigated. In the simulation these parts must be used to run the simulation correctly.

Injection control the injection control is a another subsystem of the model. The main component is a signal generator which represents the position of the accelerator pedal in percentage. The requested fuel mass is a looked up in map of engine speed and pedal position. With the value of the fuel mass the injectors are triggered. The simulation uses some model to calculate the behaviour of the engine.

Friction model The Chen-Flynn Engine Friction Model is used to specify the parameters for the friction model. In equation 5.1 the formula is shown, according to which the simulation calculates the engine friction.

$$FMEP = FMEP_{const} + A \cdot p_{Cyl.max} + B \cdot c_{p,m} + C \cdot c^2_{P,M} \tag{5.1}$$

The following terms are entered into the simulation. $FMEP_{const}$ is constant with a value of 0.3 bar. A is the peak cylinder pressure factor and has a value of 0.004. B is the mean speed piston factor and has a value of 0.08 *bar/*(*m/s*). C is the mean piston speed squared factor and has a value of 0.0003 *bar/*(*m/s*)2. The term $p_{Cyl.max}$ is the maximum cylinder pressure in bar. $C_{p,m}$ is the mean piston speed in m/s.

Cylinder wall temperature solver A cylinder wall temperature solver is used to model the cylinder structure of the engine. This model uses the finite element to represent the cylinder liner, piston and cylinder head, including the valves. With the knowledge of the geometry of the engine, the model predicts the structure temperatures and the surface temperatures.

Heat transfer model The heat transfer model which is used is the Woschni model [6] [29]. In the following equation is Woschni´s correlation shown

$$h_C = 3.26B^{-0.2} \cdot p^{0.8} \cdot T^{-0.55} \cdot w^{0.8} \tag{5.2}$$

Where B is the bore, p the pressure, T the temperature and w the average cylinder gas velocity is. The average cylinder gas velocity is calculated with following constants. C_1 for the gas exchange period with 6*,* 18+0*,* 417 c_u/c_m, C_1 for the rest of the cycle with 2*,* 28 + 0*,* 308 c_u/c_m and C_2 with $3.24 \cdot 10^{-3}$

Combustion model The combustion model which is used in the simulation is a three term Wiebe function. The following parameters are entered into the simulation. The ignition delay is a map, where the engine speed is spread over the air-fuel ratio. The premixed fraction is set to 0.05 and the tail fraction is set to 0.15. The premix-, main- and tail-duration are maps where the engine speed is spread over the air-fuel ratio. The number of temperature zones in this model is set to "two zone model".

NO_x model To predict the formation of NO during the combustion, the calculation in the simulation is based on the extended Zeldovich mechanism. The following terms are entered into the simulation. The NO_x calibration multiplier, the N_2 oxidation rate multiplier, the N_2 oxidation activation energy multiplier, the N oxidation activation energy multiplier and the OH reduction rate multiplier are set to 1. These terms are the rate constants that are used to calculate the formation of NO [6] [29].

With all these settings the simulation run can be configured.

5.3 Simulation

The model is fully built and the simulation is ready to run the calculation. To configure the simulation the case specific input, simulation type and output data must be described. The time of the simulation is controlled and set periodically for each top dead centre event. Each case is calculated with 50 steps, while each step takes 3 seconds.

The experiments 1 to 3 are simulated. The ratio of the water/fuel, the pedal position, the mass of water and the engine speed are given into the simulation for the calculation. All the other parameters are constant and don't have to be changed. In appendix table A.1 all the simulation cases for the experiments 1 to 3 are shown. The data for the mass of water is calculated based on the fuel mass. In the simulation, the injection control is equipped with a fuel map. In this fuel map the injected fuel mass is spread over the engine speed and the pedal position. To get the right pedal position, another map is required. This map is the BMEP map, which is provided by the simulation as well. This map shows the requested BMEP over the engine speed and the pedal position. The now desired BMEP is looked up in the BMEP map. The now needed pedal position is looked up in the fuel map. Now the injected fuel mass is read out of the map at the right engine speed and pedal position. On this basis the injected water mass is now calculated.

The experiments 4, 5 and 6 cannot be simulated with this software. The location of the injector is connected to a one dimensional part and the simulation only calculates the difference of the mass flow in each part. In the simulation the solver does not make the difference if the injector is connected to the begin or to the end of a pipe. To simulate the influence of the different locations, a CFD simulation tool needs to be used. The simulation of the different injection times is not done, because too much estimation has to be done and a lot of information about the injector is not provided.

5.4 Results

The calculation is done and the results are viewable with the application GT-POST. The following charts and images are the results of the simulation, which are most important for the project. Furthermore a lot more data is available for further investigations and research projects. First the influences of the water injection to the pressure and the temperature are shown. After that, the possibility is shown to what extent the water injection can lower the emissions. All results show the NOx emissions. The NO emission content in this NOx emission and are not separated analysed, because they vary of about 2 ppm over the whole simulation. At the end the influence of the water to the performance is shown.

5.4.1 Influence of the water injection on the pressure

The p-V-diagram of the engine for experiment 3 is shown in figure 5.2 for a BMEP of 5 bar and 2000 rpm. At the low pressure range the influence of the water injection is not essential. The most significant difference between case 1 without water injection and case 6 with 100% water injection of the injected fuel is observed in the high pressure range for a BMEP of 5 bar. The peak pressure in figure 5.2 rises from 70 bar to 76 bar, with an increasing mass of injected water. A focus on this area is shown in figure 5.3. This behaviour is caused by the rise of the specific gas constant and an increase of the volumetric efficiency. Humid air with 100% RH has a greater gas constant as humid air with 60% RH. The gas constant of humid air with 100% RH is 461,40 J/(kg K), the gas constant of humid air with 60% RH is 371,05 J/(kg K) [23], which are calculated with the equation 1.14 in chapter 1 on page 12. Due to a higher volumetric efficiency _v , the pressure increases as well. The humid air with 100 % has a higher mass m, than the humid air with 60%. Expressed in equation 5.4, the rise of the gas constant R and the rise of the mass m is responsible for the rise of the pressure p.

$$\eta = \frac{m_a}{m_{th}} \tag{5.3}$$

$$p = \frac{m \cdot R \cdot T}{v} \tag{5.4}$$

In equation 5.3 m_{th} stands for the theoretical mass, this is in the cylinder. In equation 5.4 the lower temperature T, caused by the injected water, is decreasing the term less, as the gas constant R and the mass m. The volume V is assumed to be constant. If it is later desired to inject even more water than fuel, the peak pressure should be observed, to prevent a destruction of the engine due to a to high peak pressures. The p-V-diagram for the experiment 1 and 2 is not displayed here. The pressure rise is only minimal and thus only a little influence is visible.

In the p-V-diagram another result can be calculated. The area of the high-pressure range represents the indicated work. In equation 5.5 the formula for the indicated work per cycle $W_{c,i}$ is displayed

$$W_{c,i} = \oint p \cdot dv \qquad (5.5)$$

Where p is the pressure relative to the Volume V. With an increase of the pressure, following with the increase of the injected water mass, the work is increasing. The losing of work during the gas exchange is less than the profit of the injected water. In figure 5.3 a focus on the peak pressure is shown. With increasing water, the pressure rises. At a water/fuel ratio of 100 % the peak pressure has a value of 76 bar. With no injected water the pressure has a value of 70 bar.

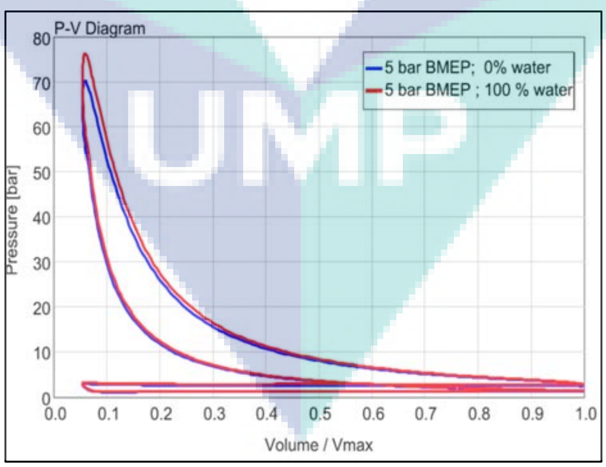

Figure 5.2: p-V-diagram for 5 bar BMEP, an engine speed of 2000 rpm and 0 % and 100 % water

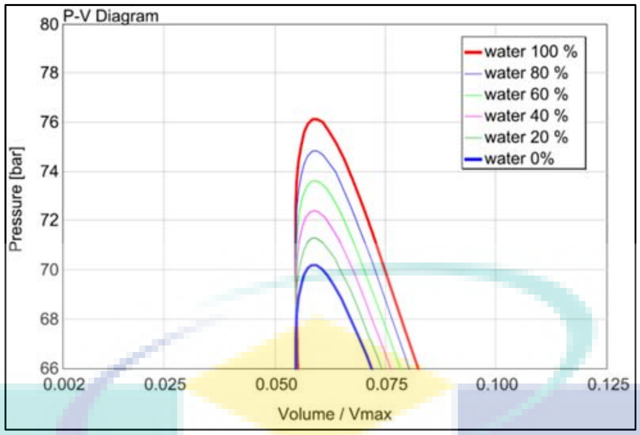

Figure 5.3: p-V-diagram for 5 bar BMEP and an engine speed of 2000 rpm, focus on the peak pressure

5.4.2 Influence of the water injection on the temperature

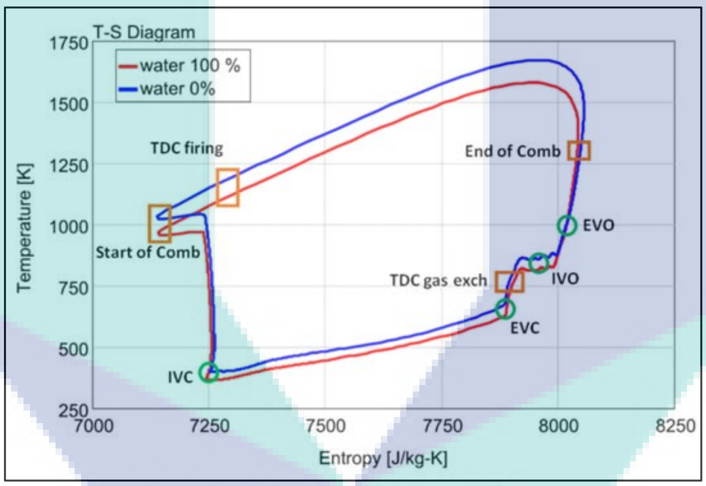

Figure 5.4: T-S-diagram for 5 bar BMEP and an engine speed of 2000 rpm for 0% and 100 % water, with the events of the valves and the engine

In the following section the T-s-diagram is analysed. With an increasing amount of water, the temperature decreases in the high and low pressure range. The entropy is higher in case 6 in the high pressure range compared to case 1. According to the second law of thermodynamics, entropy cannot be decreased. The entropy with 0% and 100 & water is almost constant in the low pressure range right after the exhaust valve opened. New fresh cool air is getting into the cylinder after the intake valve opened. During the gas exchange the temperature and the entropy are almost the same. Different temperatures are detected after the intake valve is closed and the compression is done. Right before the start of the combustion, different temperatures are detected due to the injected water. In figure 5.4 the T-S-diagram for experiment 3 with a BMEP of 5 bar is showed for a water mass of 0 % and 100 %. In this figure the influence of the water injection is more pronounced and the change due to the water injection is providing better viewable results, than the results of the experiment 1 and 2. In these two experiments the influence of the water is not as much pronounced as in figure 5.4.

In figure 5.4 the operation points of the valves are displayed, as well as the main events of the engine. With the analysis of the influence of the water injection to the pressure and to the temperature, the influence on the emissions and the performance can be further analysed.

5.4.3 Influence of the water injection on the emission

First the influence of the water injection on the NOx emissions are analysed, followed by the emissions for CO and CO2. The results for soot are not available for this combustion model. In the following figures the NOx concentration in PPM is spread over deg CA. The NOx is produced between the firing TDC and the gas exchange TDC. At the TDC of the gas exchange the engine gets fresh air and the NOx are afterwards pushed out of the cylinder. These NOx concentrations are not treated with an after-treatment and therefore they are far above the limitation for NOx emissions, but in order to see the influence of the water injection, this simulation result is very informative. In figure 5.5 the water injection can lower the NOx concentration from 170 ppm to 100 ppm, by 58%, which is a huge improvement. This is represented for the experiment 1 with a BMEP of 2 bar and an engine speed of 2000 rpm. In figure 5.6 the influence of the water injection can theoretically lower the NOx concentration from 460 ppm to 230 ppm, which is a rate of 50 % and a huge value of improvement in reducing the NOx emissions as well. This figure is representing the experiment 3 with a BMEP of 5 and an engine speed of 2000 rpm. Experiment 2 is not shown, the results are between experiment 1 and 3 and show the same behaviour as the other
two.

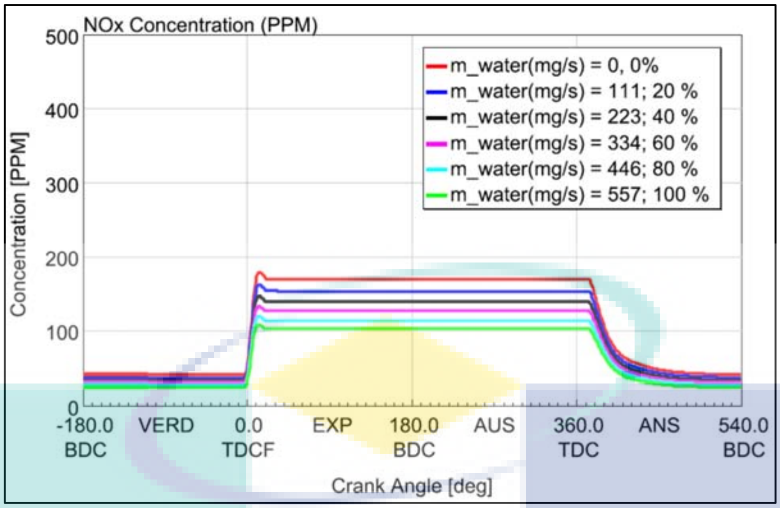

Figure 5.5: NOx concentration over deg CA for 2 bar BMEP and 2000 rpm

Additional to the NOx emissions, the exhaust contains CO and CO2. In figure 5.7 the CO concentration over the water/fuel ratio is shown. With an increasing amount of water, the CO concentration decreases. The amount of CO depends on the load, represented by the BMEP. With a higher load and water mass, a lowering in the CO emission is detected. The same behaviour is observed for the CO2 concentration, see figure 5.8. With an increase of the load and the water mass, a decrease of the CO2 is detected. Attention should be paid to fact, that CO2 is not a classified as a emission, it is a natural gas which is contained in the air. However it is responsible for the greenhouse effect and the global warming. Regulations limit the CO2 emissions and therefore it should be investigated.

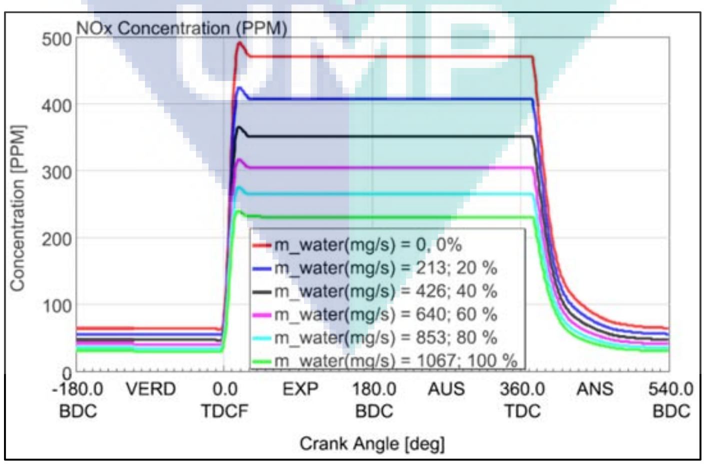

Figure 5.6: NOx concentration over CA for 5 bar BMEP and 2000 rpm

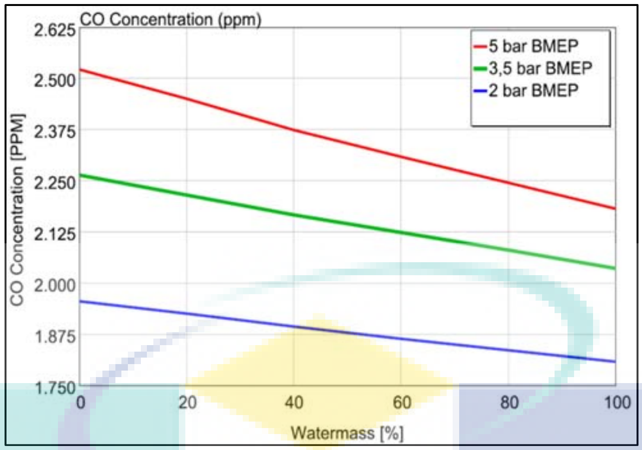

Figure 5.7: Brake specific CO emission over the water/fuel ratio

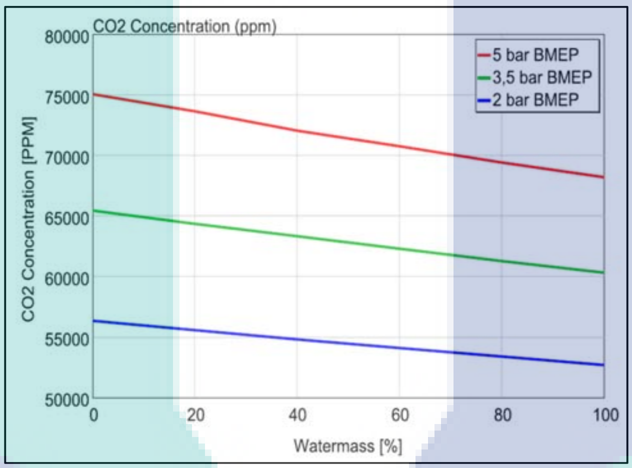

Figure 5.8: Brake specific CO2 emission over the water/fuel ratio

The CO concentration decreases from 2500 ppm to 2180 ppm with a decrease of the water mass for a load of 119 Nm, 5 bar BMEP. For 3,5 bar BMEP the CO concentration decreases from 2250 ppm to 2050 ppm with a increase of the water mass. In experiment 1 with 2 bar BMEP the CO concentration decreases from 1950 ppm to 1800 ppm. For the experiment 3 the CO concentration gradient is greater than in the other experiments.

5.4.4 Influence of the water injection on the performance

First the main noticeable fact for a car driver/user, the torque, is analysed. Following the reason for the influence on the torque, the analyse of the fuel consumption is shown in the following figures. In figure 5.9 the indicated torque in Nm for the load of 2, 3,5 and 5 bar BMEP over the injected water mass is represented. In figure 5.10 the brake torque in Nm for experiment 1 to 3 over the ratio water/fuel is displayed. The torque increases with an increase of the injected water mass for both types of torque. At a BMEP of 2 bar, the influence of the water on the torque is less than operating the engine with 5 bar BMEP. An increase of only 1 Nm is detected at 2 bar BMEP, which is far less than an increase of 6 Nm at 5 bar BMEP. With rising load, the water injection has more benefits on the torque.

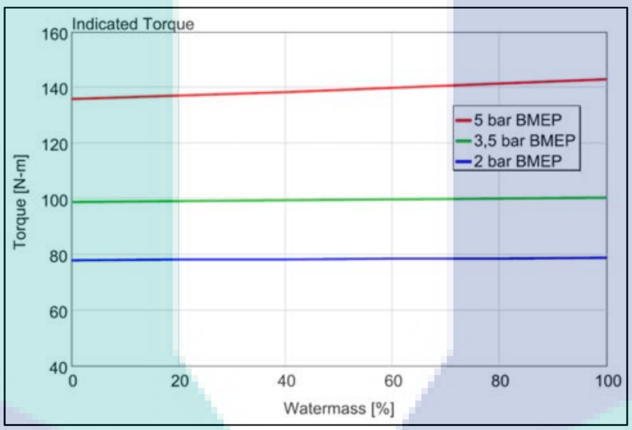

Figure 5.9: Indicated torque over the water/fuel ratio

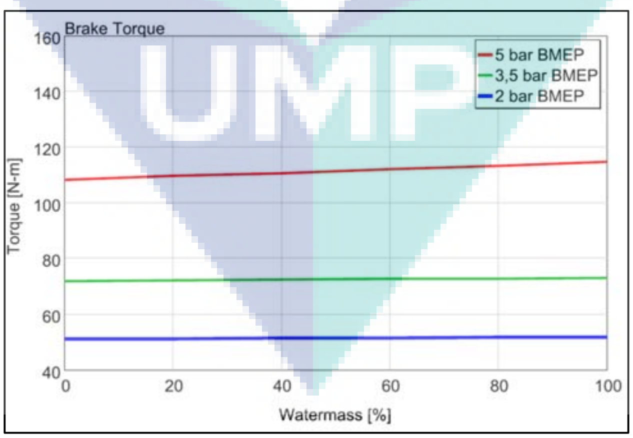

Figure 5.10: Brake torque over the water/ fuel ratio

The difference between the indicated torque and the brake torque is the friction torque. This is the torque which the engine must provide to move the crankcase, the pistons and the appearance of heat to the coolant system. For the driver of a car the increase of the brake torque is the main parameter, which he can feel, if an increase of the brake torque with the water injection is reached. The indicated power is only a parameter which the engine needs to overcome by the use of the fuel energy. The indicated and brake torque are based on the IMEP and BMEP, with the following equation:

$$T = \frac{p}{2 \cdot \pi \cdot N} \tag{5.6}$$

$$P = p \cdot V_d \cdot i \cdot N \cdot n_R \tag{5.7}$$

Where general the T is for the torque, P for the power, p for the pressure, V_d for the displacement of one cylinder, i for the number of cylinder and n_R the number of crank revolution. The indicated torque is calculated with the IMEP and the brake torque with the BMEP. Consequently, with an increase of the pressure, either IMEP or BMEP, the torque is increasing. The results of the simulation show this behaviour exactly. With the increase of the IMEP, see figure 5.11, and the BMEP, see figure 5.12, the torque is increasing.

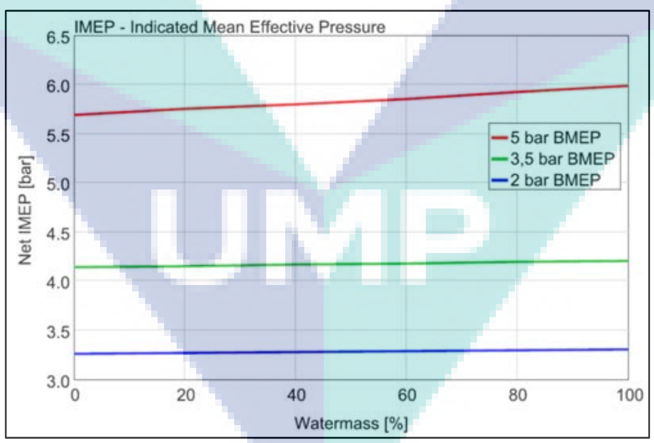

Figure 5.11: Indicated mean effective pressure over the water/fuel ratio for 2, 3,5 and 5 bar BMEP

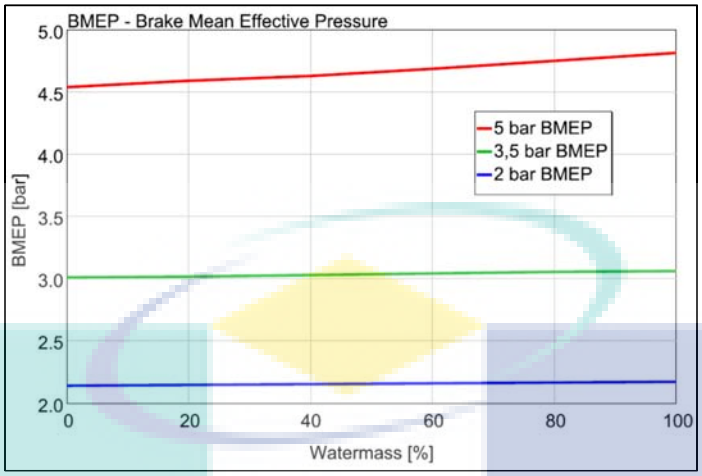

Figure 5.12: brake mean effective pressure over the water/fuel ratio for2, 3,5 and 5 bar BMEP

By the use of equation 5.7 the IMEP and BMEP can be calculated as well, if the desired power is available. The increase of the torque is furthermore caused by an increase of the volumetric efficiency and the brake efficiency, resulting from of the increase of the injected water mass. In figure 5.13 the volumetric efficiency is shown. By increasing the water mass, the volumetric efficiency increases for every load. At low part-load, represented by 2 bar BMEP, the influence of the water injection is lower, compared to the high-part load with 5 bar BMEP. The brake efficiency η_e in %, see figure 5.14, is the ratio between brake power *Pe* and the product of the fuel consumption per time *B* and heating value Q_{HV} . The following equation 5.8 shows the formula for the brake efficiency.

$$\eta_e = \frac{p_e}{B \cdot Q_{HV}} \tag{5.8}$$

The volumetric efficiency increases from 88% to 98% for 5 bar BMEP with an increase of the water mass from 0% to 100%. In experiment 2 with 3.5 bar the volumetric efficiency increases from 74% to 81% with an increase in the water mass. From 70% to 75% the volumetric efficiency rises in experiment 1 with 2 bar BMEP. Last but not least, which to some people is the most important benefit of the water injection is analysed. The brake specific fuel consumption is shown in figure 5.15.

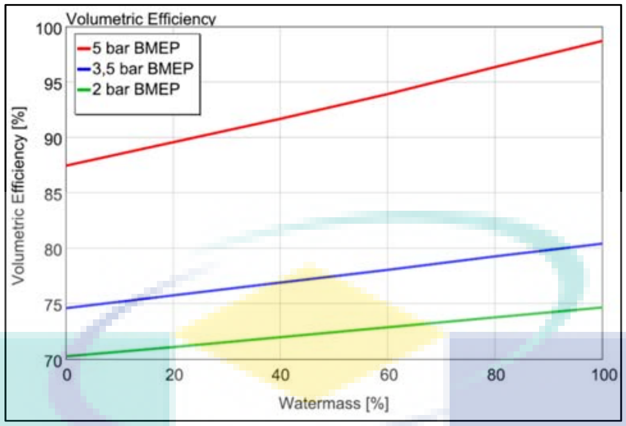

Figure 5.13: Volumetric efficiency over the water/fuel ratio for 2, 3,5 and5 bar BMEP and 2000 rpm

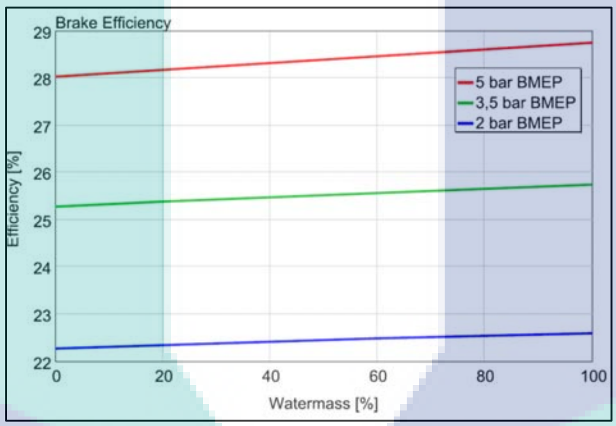

Figure 5.14: Brake efficiency over the water/ fuel ratio for 2, 3,5 and 5 bar BMEP and 2000 rpm

With an increase of water, the brake specific fuel consumption decreases. This results from of the increase of the brake power. In equation 5.9 the brake specific fuel consumption *BSFC* is shown

$$BSFC = \frac{FC}{P_e} \tag{5.9}$$

where the *FC* is the total fuel consumption per time and P_e is the brake power. With an increase of the brake power, which can be seen in figure 5.10 represented by the brake torque, the brake specific fuel consumption decreases. Now the results of the

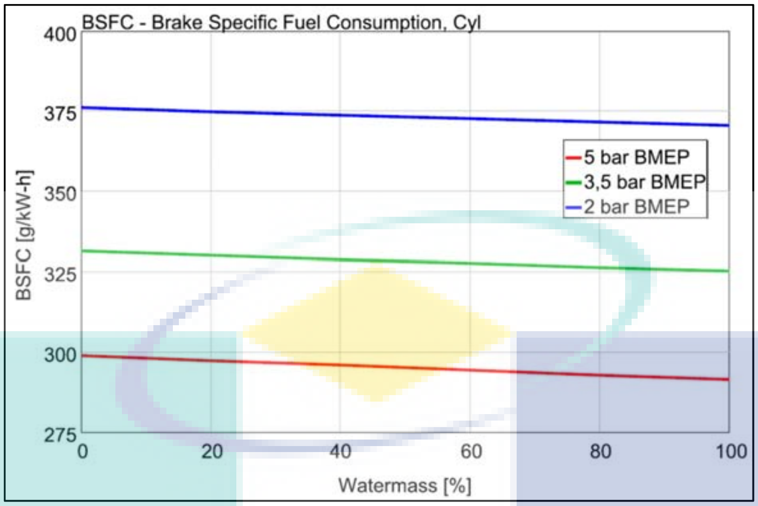

Figure 5.15: Brake specific fuel consumption over the water/fuel ratio for 2, 3,5 and 5 bar BMEP

Simulation are analysed and can be compared with the measurements accomplished at the engine on the test bed. In appendix A.3 to A.8 additional results for the experiment 3 with 5 bar BMEP and 0% and 100% water/fuel ratio are shown. In this experiment the most visible results are available.

6 Measurements and Results

The testing of the water injection system is done and the data is being analysed. First the influence of the water injection on the temperature is analysed. Next, the results of the influence of the water injection on the pressure is analysed. In comparison to the pressure, the heat release is analysed. Followed by the analyse of the emissions and at the end, the influence on the performance is analysed. At the end of each section, a comparison with the simulation is giving, if data are available. The first three experiments are analysed at first. Later the analyse of the data from the experiment 4,5 and 6 is presented.

In figure 6.1 the relative humidity and the ambient temperature at the test bed is shown. The curves are following the daily process of the weather in Malaysia. To operate every test bed run under the same conditions, the relative humidity and the ambient temperature are being watched before the test run. If the values where not the same, as the values at the last test run, certain time was waited before a new test run could started. Every day there was a section, where the ambient temperature is at about $28^{\circ}c$ and the relative humidity is between a range of 75% and 70%. This section is used to run the measurements.

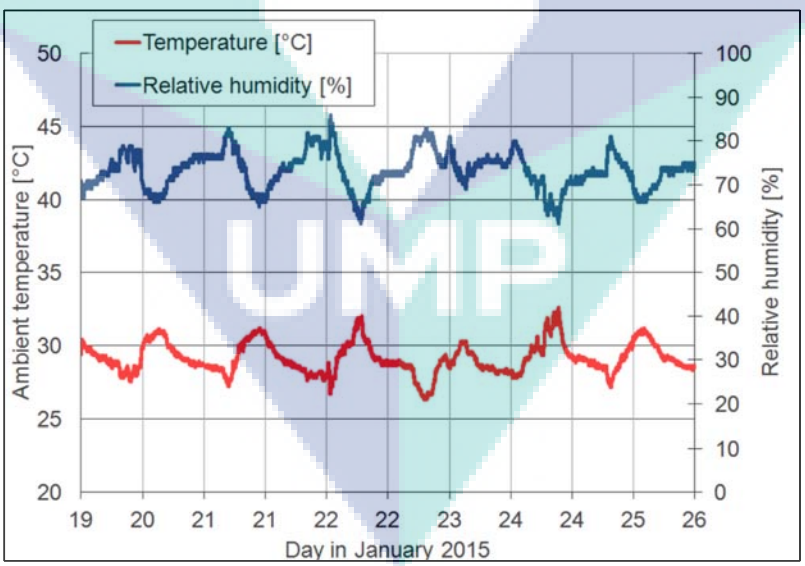

Figure 6.1: Relative humidity and ambient temperature over the whole test runs

6.1 Influence of the water injection on the temperature

In this section the influence of the injected water on the temperature is analysed. In figure 6.2 the temperatures of the exhaust and the temperature after the water injection are highlighted, for experiment 1 with 2 bar BMEP. These two temperatures decrease with the increase of injected water. The exhaust temperature decrease from $226^{\circ}c$ to $210^{\circ}c$, which is a difference of $16^{\circ}c$. The temperature after the water injection decreases from $85^{\circ}c$ to $70^{\circ}c$, with a difference of $15^{\circ}c$, which is almost as same as the exhaust temperature. The other temperatures are constant over the whole measurement cycle. The temperature after the intercooler has a value of constant $39^{\circ}c$. The temperature before the intercooler has a value of constant $50^{\circ}c$. The oil temperature is constant and has a value of $98^{\circ}c$. The fuel temperature is constant at a value of $31^{\circ}c$. To make sure that all measurement points are measured under the same conditions, the ambient temperature is displayed as well and is constant at $29.5^{\circ}c$ over the whole measure cycle. The water injection changes the temperatures of the exhaust and the point after the water injection. The other temperatures are not affected by the water injection. So the thermodynamic theory, see chapter 1.4 at page 10, is confirmed with these results. The evaporation enthalpy of the water absorbs the heat from the air in combustion chamber. The experiment 2 with 3.5 bar BMEP is shown in figure 6.3. The temperatures are displayed over the water/fuel ratio. The temperature of the exhaust decreases from $324^{\circ}c$ to $309^{\circ}c$, which is a difference of $15^{\circ}c$. The temperature after the water injection point drops from $109^{\circ}c$ to $94^{\circ}c$, which is the same difference as the exhaust temperature with $15^{\circ}c$. The other temperatures are constant during the whole test run. For the experiment 3 with 5 bar BMEP, the influence of the water injection on the temperatures, is shown in figure 6.4. The temperature of the exhaust and the point after the water injection is affected by the injected water. The exhaust temperature decreases from $404^{\circ}c$ to $383^{\circ}c$. This is a temperature drop of $21^{\circ}c$ in the exhaust. The temperature after the water injection decreases by $18^{\circ}c$, from $115^{\circ}c$ to $97^{\circ}c$. The other temperatures are constant and are not affected by the injection of water. The most significantly influence is detected at experiment 3 with 5 bar BMEP. At this engine load point, the temperatures decrease more than in the other engine load points with 3.5 and 2 bar BMEP. The simulation and the measurements show the same behaviour. The tendency of the temperatures shows the exact same process. The values of the temperatures are different, due to the fact of the estimation, which are done in the simulation model.

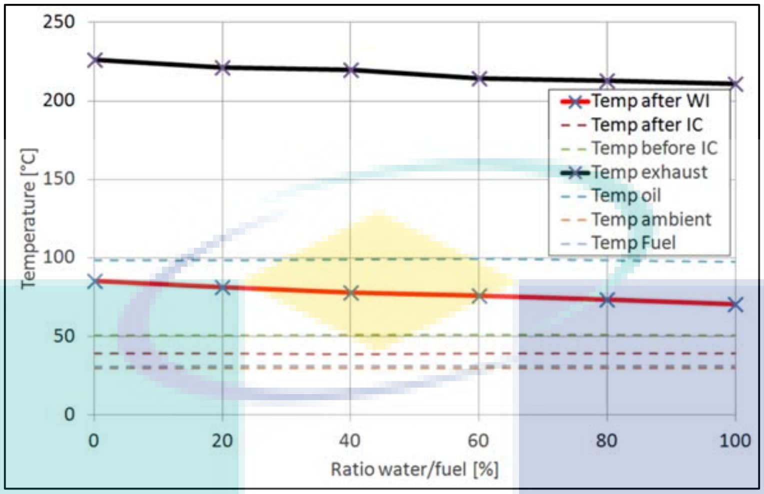

Figure 6.2: Temperature over the water/fuel ratio for 2 bar BMEP and 2000 rpm

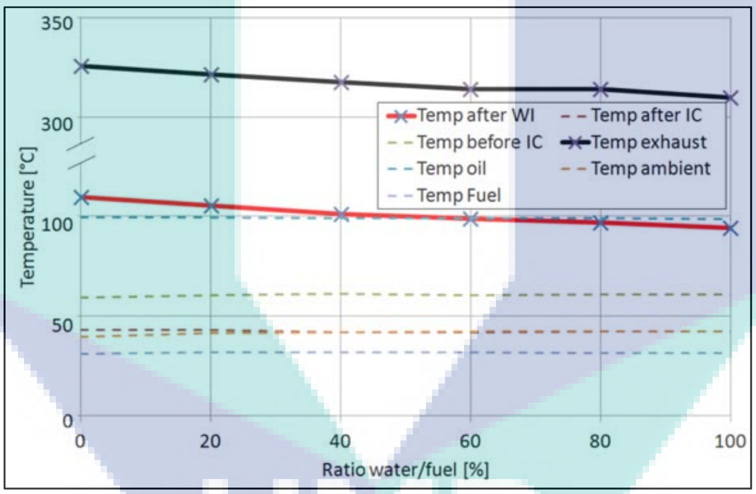

Figure 6.3: Temperature over the water/fuel ratio for 3,5 bar BMEP and 2000 rpm

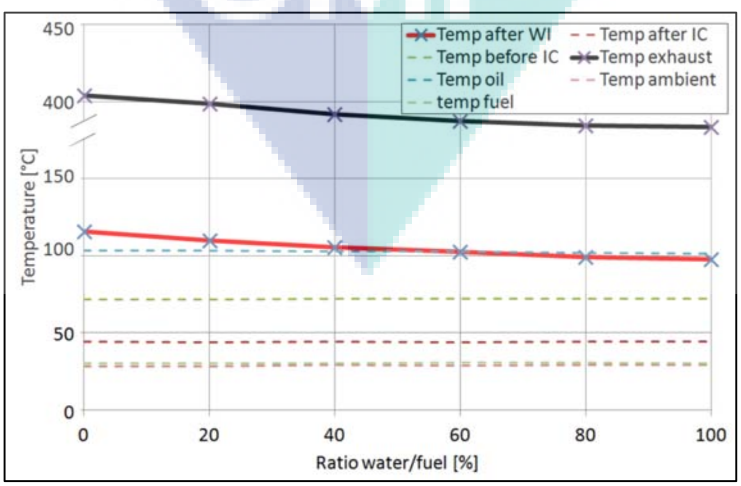

Figure 6.4: Temperature over the water/fuel ratio for 5 bar BMEP and 2000 rpm

6.2 Influence of the water injection on the pressure and heat release

In the following section the raw data from the DEWE-5000 are analysed and prepared with a software. Each experiment is described by its own for the pressure, the heat release and the cumulative heat release. Every figure is spread from -100 deg CA to 100 deg CA to compare the figures and to detect an event in each figure. So it is easily viewable what happens at which deg CA.

6.2.1 Results for low part-load 2 bar BMEP

The following figures show the cylinder pressure, the heat release and the cumulative heat release for 2 bar BMEP. In figure 6.6 the cylinder pressure over crank angle is shown for 0%, 40% and 100% water/fuel ratio. General there are 2 pressure peaks detected, one at 5 deg CA before TDC with 54 bar and one at 15 deg CA after TDC with 53 bar. These two peaks are the results of the compressions pressure and the pressure rise from the ignition of the injected fuel. In between these peaks the pressure drops to 45 bar. A difference of the pressure rise, with injecting water, is detected at 10 deg CA before TDC. The pressure with a water/fuel ratio of 0% increases at this point more, than the pressure with 40% and 100% ratio. The peak pressure with 0% injected water is at almost 54 bar, where the peak pressure of the curve with 100% injected water is 53 bar. The reason for the different pressure peaks is the temperature. The lowering of the temperature in the gas equation is lowering the pressure. The peak from the compression pressure is maximum at 5 deg CA before the TDC, this is caused by the heat transfer to the walls. After the TDC the pressure drops further down with the water injection, than the pressure without water injection. The combustion chamber is cooler and the ignition delay is greater. The fuel ignites later and has therefore more time to get ready for the ignition. This behaviour can be seen in figure 6.5. The ignition delay τ_{id} is depending on the temperature. With increasing temperatures, the ignition delay time decreases. After the ignition of the fuel the pressures for all water/fuel ratios are the same.

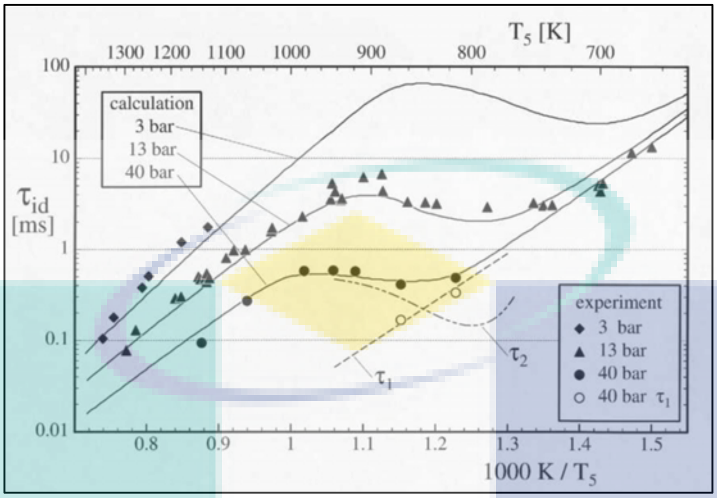

Figure 6.5: Ignition delay versus combustion temperature [30]

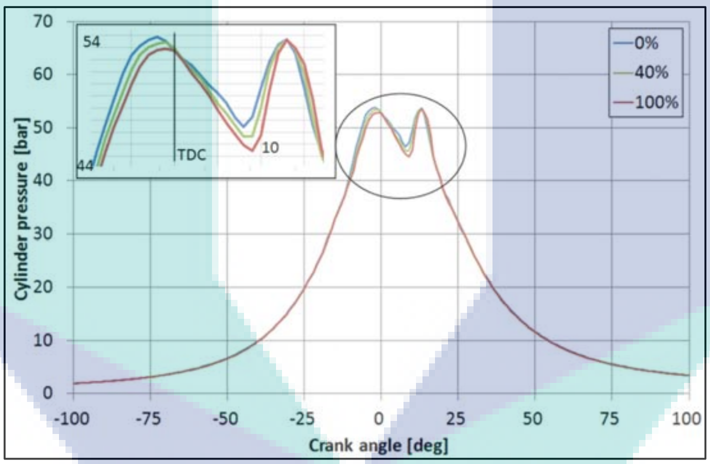

Figure 6.6: Cylinder pressure versus crank angle for 2 bar BMEP and a water/fuel ratio of 0%, 40% and 100%

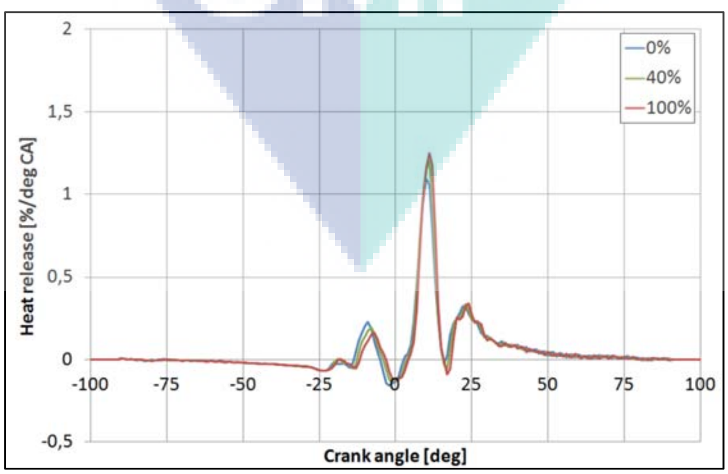

Figure 6.7: Heat release versus crank angle for 2 bar BMEP and a water/fuel ratio of 0%, 40% and 100%

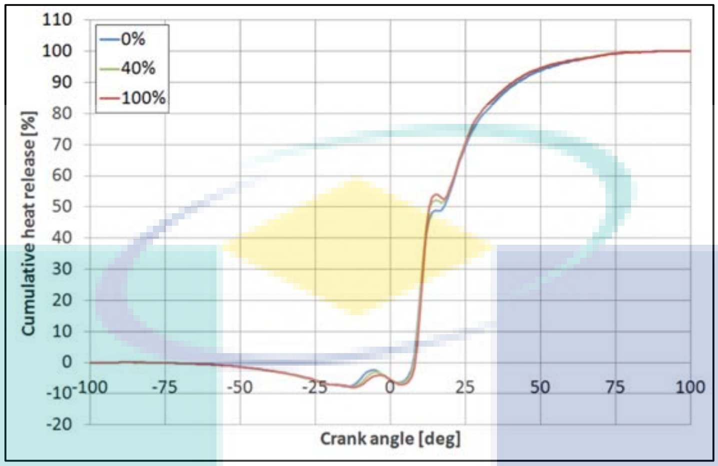

Figure 6.8: Cumulative heat release versus crank angle for 2 bar BMEP and a water/fuel ratio of 0%, 40% and 100%

In figure 6.7 the heat release for 2 bar BMEP and a water/fuel ratio of 0%, 40% and 100% is shown. The heat release is the rate at which heat is generated from the engine. In equation 6.1 the formula is shown, how the heat release is calculated.

$$\frac{dQ_H}{d\alpha} = \frac{\kappa}{\kappa - 1} \cdot p \frac{dV}{d\alpha} + \frac{1}{\kappa - 1} \cdot V \frac{dp}{d\alpha} \tag{6.1}$$

p and V are the cylinder pressure and the cylinder volume, _ is the crank angle and κ is set to 1,37 [6] for diesel engines. From 100 deg CA to 25 deg CA before the TDC the heat release is negative. This results from the heat transfer from the air to the walls and then to the coolant water. In the range of 25 deg CA to 5 deg CA before the TDC the heat release from the pre fuel injection is detected. A little amount of fuel is injected. After the pre fuel injection, the heat release of the main fuel injection start at 5 deg CA after TDC. This main fuel injection converts the most fuel energy into heat energy. At 20 deg CA after TDC the heat release of the post fuel injection starts. At the heat release point of the pre fuel injection a different behaviour with injected water is detected. The fuel ignites later with the injection of water. Without injected water the fuel ignites 12 deg CA before the TDC, with a water/fuel ratio of
40% the fuel ignites 11 deg CA before the TDC and with a injected water/fuel ratio of 100% the fuel ignites at 9 deg CA before the TDC. The ignition point moves near to the TDC with

an increase of the injected water, due to a decreased temperature in the combustion chamber, see ignition delay time in figure 6.5 on 46. The flame of the combustion needs more time to burn because inside the combustion chamber the temperatures are lower. Furthermore the heat release during the pre-fuel injection is less with injected water compared to the heat release without injected water. These loss is instead a benefit in the main fuel injection range. The heat release is higher in the main fuel injection range with water injection, than without water. This results from the now warm temperature in the combustion chamber and due to the greater ignition delay time, the fuel has more time to prepare for the combustion. For the post fuel injection range no difference between 0%, 40 % and 100% injected water is detected. In figure 6.8 the cumulative heat release is shown for 2 bar BMEP and water/fuel ratio of 0% and 40% and 100%. The 10% mass fraction burned position moves from 8,8 deg CA to 9,3 deg CA after TDC with an increase of the water/fuel ratio. The 50% mass fraction burned position moves from late to early, in numbers from 13 deg CA to 17 deg CA, due to the greater ignition delay time. In total the pressure and the heat release are only effected with small changes. There is no significant influence with the water injection detected at a BMEP of 2 bar and 2000 rpm.

6.2.2 Results for middle part-load 3.5 bar BMEP

The experiment 2 with 3,5 bar BMEP is now analysed for the cylinder pressure and the heat release. In figure 6.9 the cylinder pressure versus the crank angle is shown. There are two peaks detected, which are the results of the compression pressure and the pressure rise of the ignition of the fuel. The first peak has a value of 58 bar, the second peak a value of 59 bar. In between the pressure drops to 52 bar. The influence of the water injection is not significant viewable at the first peak and the values vary in the range of 0,2 bar, which is traced to a error in the recording. The second peak is affected by the water injection. The pressure drops further down to 51 bar with injected water and then rises up to 60 bar with a fuel/water ratio of 100%. Without water injection the pressure drops down only to 52 bar and then rises only up to 59 bar. This is caused by the increase of the injection delay time due to cooler temperatures in the combustion chamber. The fuel has more time to get ready for the ignition and then more prepared mixture is ready to ignite, which results in a higher pressure. If the water/fuel ratio will be increased further the peak pressure should be watched to prevent a destruction of the engine. In figure 6.10 the heat release versus the crank angle for 3,5 bar BMEP and water/fuel ratios of 0%, 40% and 100% is shown. At 25 deg CA before the TDC the heat release from the pre fuel injection starts. The energy of the fuel is converted into heat, to warm up the combustion chamber. Without water injection the heat release is higher compared to the heat release with a 100% water/fuel ratio. This results from the cooler temperature in the combustion chamber. With water injection the combustion chamber is cooler and the fuel needs more time to ignite, the ignition delay is greater. After the pre fuel injection the heat release is negative, which results from the heat transfer into the wall. The heat release of the main fuel injection starts 5 deg CA after the TDC. With increasing amount of water the curve of the heat release rises more, than the curve without water injection, which is caused by the greater ignition delay. The peak heat release values are the same for 0% ,40% and 100% water/fuel ratio. The heat release of the post fuel injection is not affected by the water injection. The oscillation at the end which starts at 25 deg CA after TDC are failures in the calculation made by the pressure recording system. In figure 6.11 the cumulative heat release versus the crank angle is shown for 3,5 bar BMEP and 0% ,40% and 100%. The 10% and 50% mass fraction burned position are not affected by the water injection at this engine load. The 90% mass fraction burned position is moving 3 deg CA to early with an increase of the injected water from 54 deg CA to 51 deg CA.

6.2.3 Results for high part-load 5 bar BMEP

In figure 6.12 the cylinder pressure, the heat release and the cumulative heat release are shown for 5 bar BMEP and water/fuel ratios of 0%, 40% and 100%. The cylinder pressure shows two peak values. The first peak is detected at 5 deg CA before the TDC, the second peak is detected 15 deg CA after the TDC. The first peak is from the compression pressure, an ignition of the fuel has not started yet. With a water/fuel ratio of 100% the pressure rises up 62,2 bar. Without water injection the pressure has a value of 61,7 bar. After this first peak the pressure drops to 57 bar, with 0% water/fuel ratio. If a water/fuel ratio of 100% is used the pressure drops even more to 56 bar. This is the result from the ignition delay. The combustion chamber is cooler and therefore the fuel needs more time to ignite, see figure 6.5 on page 46. The ignition of the fuel leads to an increase of the pressure again. The pressure rises

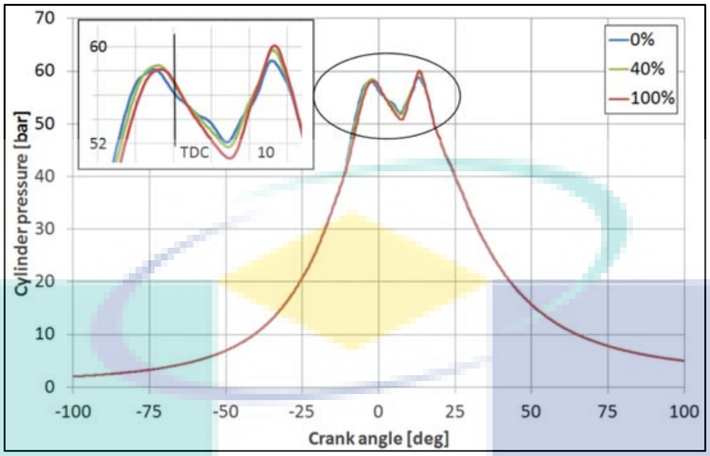

Figure 6.9: Cylinder pressure versus crank angle for 3.5 bar BMEP and a water/fuel ratio of 0%, 40% and 100%

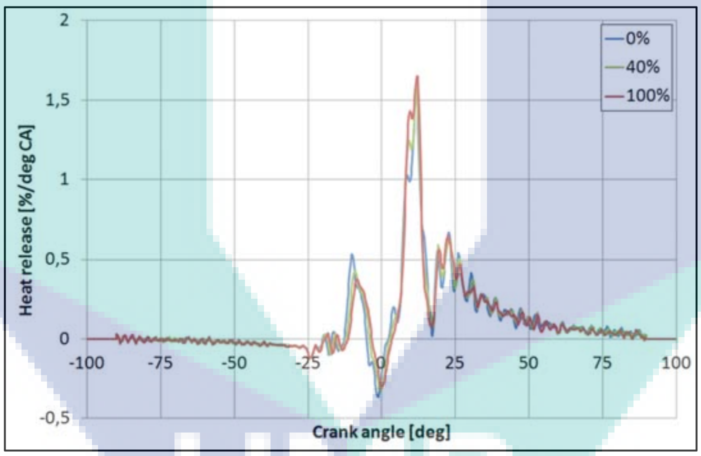

Figure 6.10: Heat release versus crank angle for 3.5 bar BMEP and a water/fuel ratio of 0%, 40% and 100%

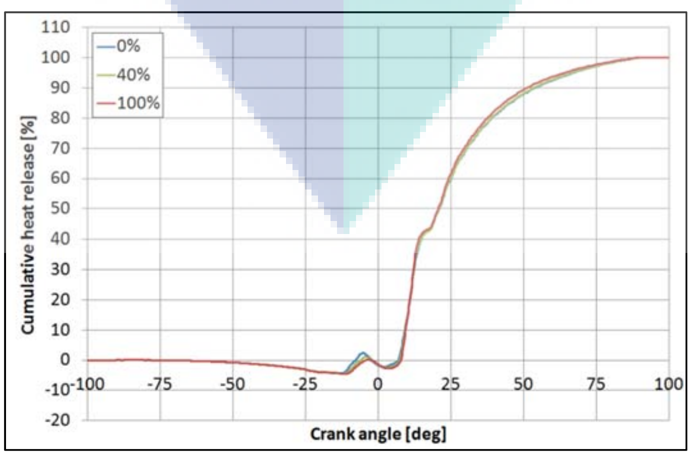

Figure 6.11: Cumulative heat release versus crank angle for 3.5 bar BMEP and a water/fuel ratio of 0%, 40% and 100%

up to 62 bar for the all kind of water/fuel ratio. The heat release for the experiment 3 with 5 bar BMEP and 0%, 40% and 100% water/fuel ratio is shown in figure 6.13. In the pre fuel injection zone the heat release starts 15 deg CA before the TDC without water injection and moves to 13 deg CA with 100% water injection. This is caused by the increase of the ignition delay. The heat release of the main fuel injection and the post fuel injection are not influenced by the water injection. In figure 6.14 the cumulative heat release for 5 bar BMEP and 0%, 40% and 100% is shown. The release of heat during the pre-fuel injection is detected at 15 deg CA before the TDC and with increasing water the heat release moves to 13 deg CA.
The 10% and the 50% mass fraction burned position change with increasing water 0.3 deg CA to late. The 90% mass fraction burned position moves 2 deg CA, from 54 deg CA to 52 deg CA after TDC with injecting water.

The pressure in the cylinder is not significantly affected by the water injection at this point of engine load. A pressure rise of 1 bar is only detected at low part-load where the temperatures are lower than at high part-load. If the load is getting increased in further measurements, the cylinder pressure has to to be watched and analysed, if with a increase in the injected water mass the cylinder pressure increases as well. The heat release shows that with injecting water the ignition point moves slightly too late. If a characteristic diagram is developed for the engine with the input of the informations of the water injection system, the fuel injection times can be optimized. The mass fraction burned position is not really effected by the water injection. So there is no need to do further investigation on this values.

Compared to the simulation, the measurements show a different behaviour. In the simulation the pressures increase in every experiment. In the measurements on the test bed the pressures increases only a small amount or are constant. One reason is that the simulation model is made of too much estimation. Another reason is that the simulation is done under ideal conditions, which are not available on the test bed.

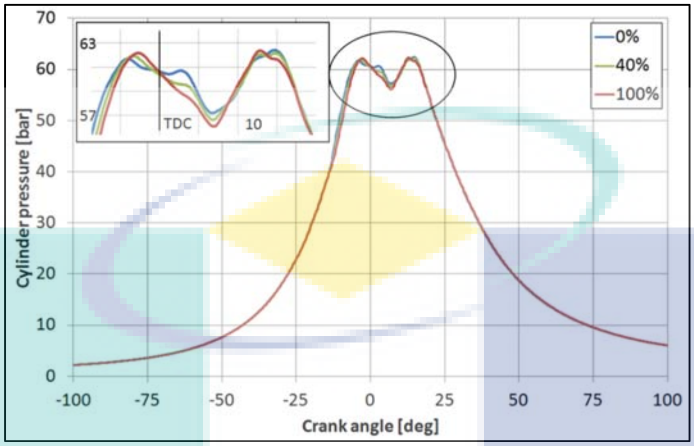

Figure 6.12: Cylinder pressure versus crank angle for 5 bar BMEP and a water/fuel ratio of 0%, 40% and 100%

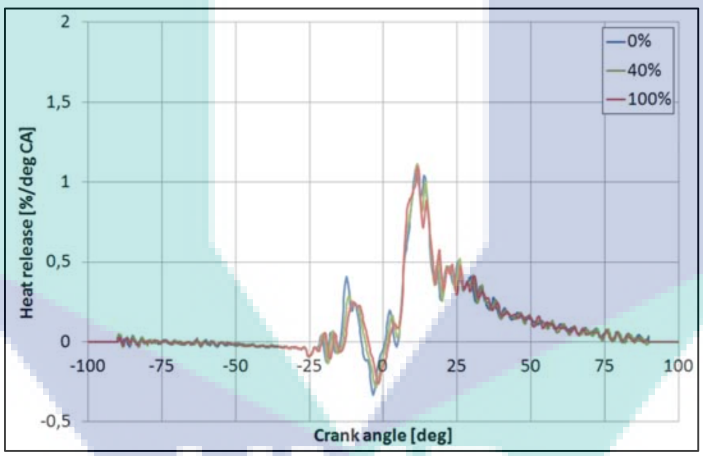

Figure 6.13: Heat release versus crank angle for 5 bar BMEP and a water/fuel ratio of 0%, 40% and 100%

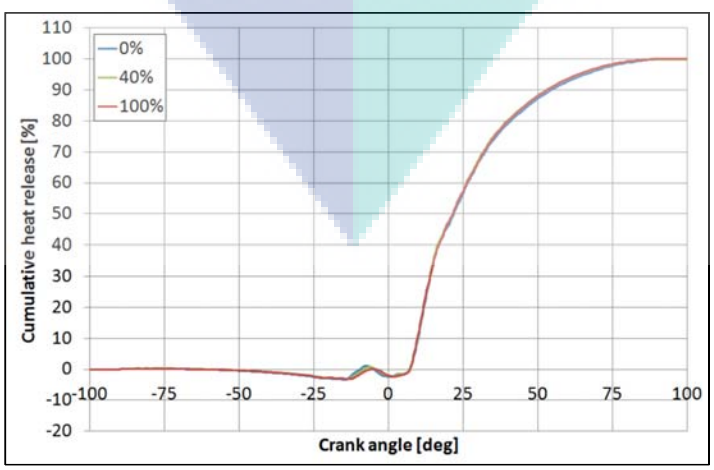

Figure 6.14: Cumulative heat release versus crank angle for 5 bar BMEP and a water/fuel ratio of 0%, 40% and 100%

6.3 Influence of the water injection on the emissions

In this section the influence of the water injection on the emissions are analysed and the results are presented. The most limited emission for diesel engines, the NOx are analysed at first. The NO emissions are not shown in the results. They are 2 ppm lower than all the NOx. So all NO emissions are summed up in the NOx emissions. A trade-off for NOx and PM is following. The emission CO is analysed as well though it is not a relevant emission for diesel engines because they are always operated on the lean side. For the sake of completeness the CO2 and the O2 emissions are analysed.

6.3.1 Nitrogen oxide and particle matter

In the following the influence of the water injection on the NOx emissions are analysed. Figure 6.15 shows the NOx emissions over the water/fuel ratio. For each water/fuel ratio the experiments 1, 2 and 3 are displayed in a column. With increasing water injected to the intake manifold, the NOx emissions decrease about two-thirds. At Experiment 1 with 2 bar BMEP the emissions decrease from 57 ppm to 16 ppm which is a 72% improvement. At experiment 2 with 3.5 bar BMEP the emissions decrease from 68 ppm to 23 ppm which is an improvement of 67%. At experiment 3 with 5 bar BMEP the emissions decrease from 89 ppm to 33 ppm which is an improvement of 63%. These huge improvements are possible due to the lowering of the temperatures. The lowering of the NOx emission is caused by the lower temperature due to the injection of water. In equation 6.2 the strong dependence of the change of temperature is described [6].

$$\frac{d[NO]}{dt} = \frac{6 \cdot 10^{6}}{T^{0.5}} \cdot \exp\left(\frac{-69.09}{T}\right) + O_2 \cdot N_2 \qquad (6.2)$$

The lower the temperature in the exponential term, the lower the NOx emission. This decrease of the NOx emission is a good result, but it has its drawbacks. With the decrease of the NOx emission the PM emission increased. This behaviour is explained in chapter 1 on page 6. In figure 6.16 the NOx - PM trade-off for 2, 3.5 and 5 bar BMEP is shown. With increasing water mass, the NOx emission decrease, but at the same time the PM emission increase. For the experiment 1 with 2 bar BMEP the PM emission do not increase or decrease significantly. The soot which is measured during the test drives are to less due to the fact that the engine forms only a few particle. The NOx emission therefore decreases. In experiment 2 with 3.5 bar BMEP the PM emission increase slightly with an increase in the water mass from 2,35 mg to 3,5 mg. At the same time the NOx emission decreased. At experiment 3 with 5 bar BMEP the PM emission increased as well, from 4.3 mg to 6 mg with an increase of the injected water mass. For the further experiments the "best of" of this trade-off is used. A point where the NOx emission are low and the PM emission are not high is at a water/fuel ratio of 40%. The comparison with the simulation shows the same behaviour. The NOx emissions are decreasing in the simulation and during the measurements at the test bed. The values are not exact the same, but the tendency is the same.

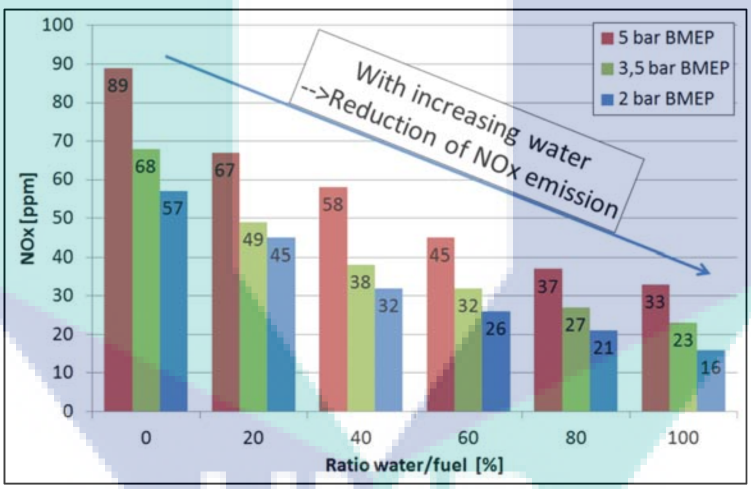

Figure 6.15: NOx emission over the water/fuel ratio for 2, 3.5 and 5 bar BMEP

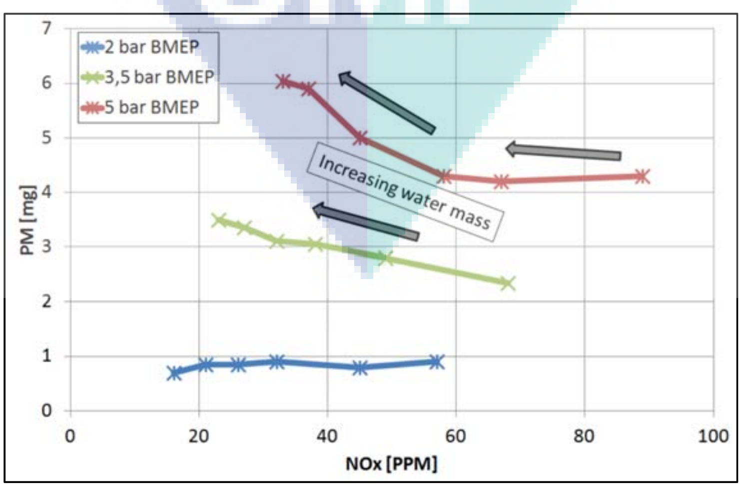

Figure 6.16: NOx - PM trade-off for 2, 3.5 and 5 bar BMEP

6.3.2 Carbon monoxide, carbon dioxide and oxygen

In the following figures the emissions are analysed which are formed during the combustion process. These are either very low or are not declared as a emission, but for the sake of completeness they are analysed as well.

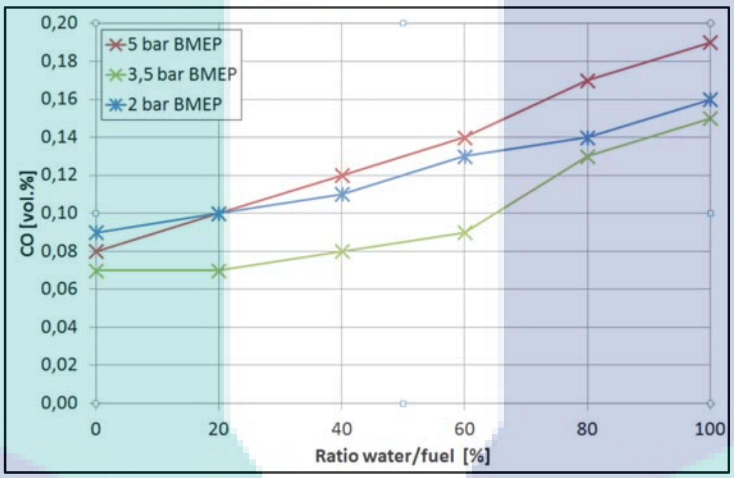

Figure 6.17: CO emission for 2, 3,5 and 5 bar BMEP and a water/fuel ratio range from 0% to 100%

In figure 6.17 the CO emission for 2, 3, 5 and 5bar BMEP and the water/fuel ratio range from 0% to 100% is shown. The CO emission unit is vol. %, which represents the total amounting % of the air. The CO emission increase very slightly from 0.09 vol. % to 0.16 vol.% for the experiment 1 with 2 bar BMEP. For experiment 2 with 3.5 bar BMEP the CO emission increase slightly from 0.07 vol. % to 0.15 vol. %. In experiment 3, with 5 bar BMEP the CO emission increase from 0.08 vol. % to 0.19 vol. %. These values are all low enough to convert them with a DOC in the exhaust treatment.

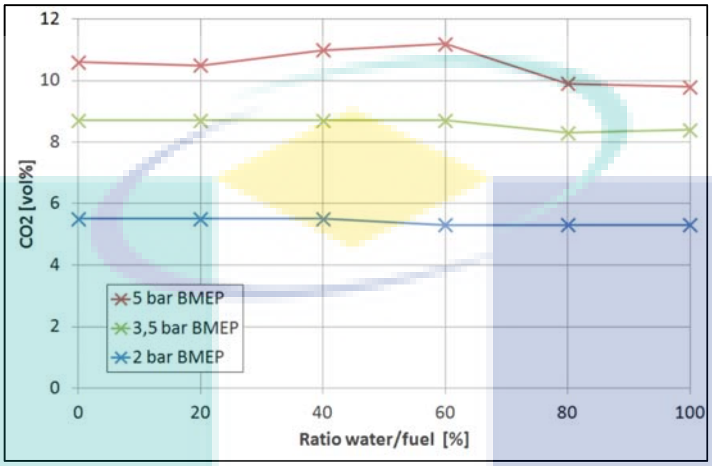

Figure 6.18: CO2 emission for 2, 3, 5 and 5 bar BMEP and a water/fuel ratio range from 0% to 100%

The CO2 emission which is responsible for the greenhouse effect and the global warming are shown in figure 6.18. With the increase of the water/fuel ratio, the CO2 are decreasing very slightly with an increase in the water mass. The CO2 emissions are not really effected by the use of the water injection.

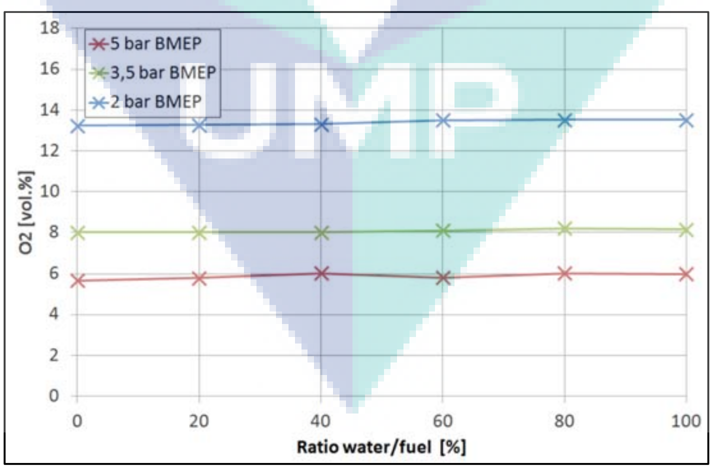

Figure 6.19: O2 emission for 2, 3, 5 and 5 bar BMEP and a water/ fuel ratio range from 0% to 100%

In figure 6.19 the O_2 emission is shown for 2, 3,5 and 5 bar BMEP. This emission is constant or very slightly increasing with the increase of the water injection mass. The value for the O_2 emission for 2 bar BMEP is at 13,5 vol.% and is slightly increasing with the increase of the water injection mass from 0% to 100%. For 3,5 bar BMEP the

O_2 emission is constant during the whole test run and has a value of 8 vol.%. The high part-load point has a O_2 emission value of 5, 8 vol. % with 0% water/fuel ratio and increases very slightly to 6,2 vol.% with a 100% water/fuel ratio.

Compared to the simulation the emissions show a different behaviour. In the simulation the emission for CO and CO_2 decrease with an increase in the water/fuel ratio. During the measurement on the test bed these emissions increased with the increase of the water/fuel ratio. One possible solution for this behaviour is that there is too much estimation in the simulation model. Furthermore the simulation run under ideal conditions due to that the boundary conditions are not available on the test bed.

In total analyse of the emission data from the test runs show a great decrease in the NO_x emission. The drawback is the increase of the PM emission. In order of the limitation for the NO_x emission these results are very good. Whit the use of a particle filter, the PM emission can be set to the limit as well. The CO, CO_2 and O_2 emission are low enough to negligible. The remaining emission is converted with a DOC.

6.4 Influence of the water injection on the performance

In this section the influence of the water injection on the performance is analysed. Therefore the IMEP, brake efficiency and specific fuel consumption for 2, 3, 5 and 5 bar BMEP over the water/fuel ratio is displayed in figure 6.20. With the use of the IMEP the indicated power can be calculated and with the use of the brake efficiency the brake power. In the following plots these values are represented by the IMEP and the brake efficiency. The first sub plot shows the IMEP, brake efficiency and specific fuel consumption for 2 bar BMEP. The IMEP decreases slightly from 3.6 bar to 3.3 bar with a increase of the water/fuel ratio. This is caused by fluctuations of the test bed and dyno. The brake efficiency decreases slightly as well. Without water injection the engine has a brake efficiency of 21.9%. With 100% water/fuel ratio the engine has a less brake efficiency with a value of 20.7%. The drift over the time of the test bed is responsible for this behaviour. Another reason is the heat up of the brake from the test bed. The control device in the test bed tower acts against this and changes the load of the brake. In these contents the specific fuel consumption increases. The lowering in the IMEP and therefore the torque the specific fuel consumption increases. In equation 6.3 the formula for the brake specific fuel consumption is shown.

$$BSFC = \frac{FC}{T \cdot 2\pi \cdot n} \tag{6.3}$$

With a decrease of the torque *T* the brake specific fuel consumption increases. This can be seen in figure 6.20 for 2 bar BMEP. The BSFC rises from 385 g/kWh to 407 g/kWh. In the second sub plot of figure 6.20 the IMEP, the brake efficiency and the specific fuel consumption for 3.5 bar BMEP is shown. At this load point the values are constant over the total range of the water/fuel ratio. The specific fuel consumption has a value of 322 g/kWh. The brake efficiency at 3.5 bar BMEP has a value of 26 %. At this engine load the indicated mean effective pressure has a value of 6.3 bar. All the previous value is not affected by the injection of water into the intake manifold. The fluctuations seen in the figure 6.20 for 3,5 bar BMEP are caused by the tolerance and drift over time from the test bed. The third sub plot in figure 6.20 shows the IMEP, the brake efficiency and the specific fuel consumption for 5 bar BMEP. On the x-axis the water/fuel ratio is displayed. All values of this plot are constant over the water/fuel ratio and are not affected by the water injection. The specific fuel consumption is constant with a value of 318 g/kWh. The brake efficiency has a value of 26.8%. The IMEP is constant over the whole water/fuel ratio and has a value of 7.6 bar. Summed up the previous values are not affected by the water injection at an engine load of 5 bar BMEP. The comparison with the simulation shows a different behaviour. In the simulation the IMEP and the brake efficiency increases and the specific fuel consumption decreases. The simulation is not influenced by the tolerance and fluctuations of the test bed.

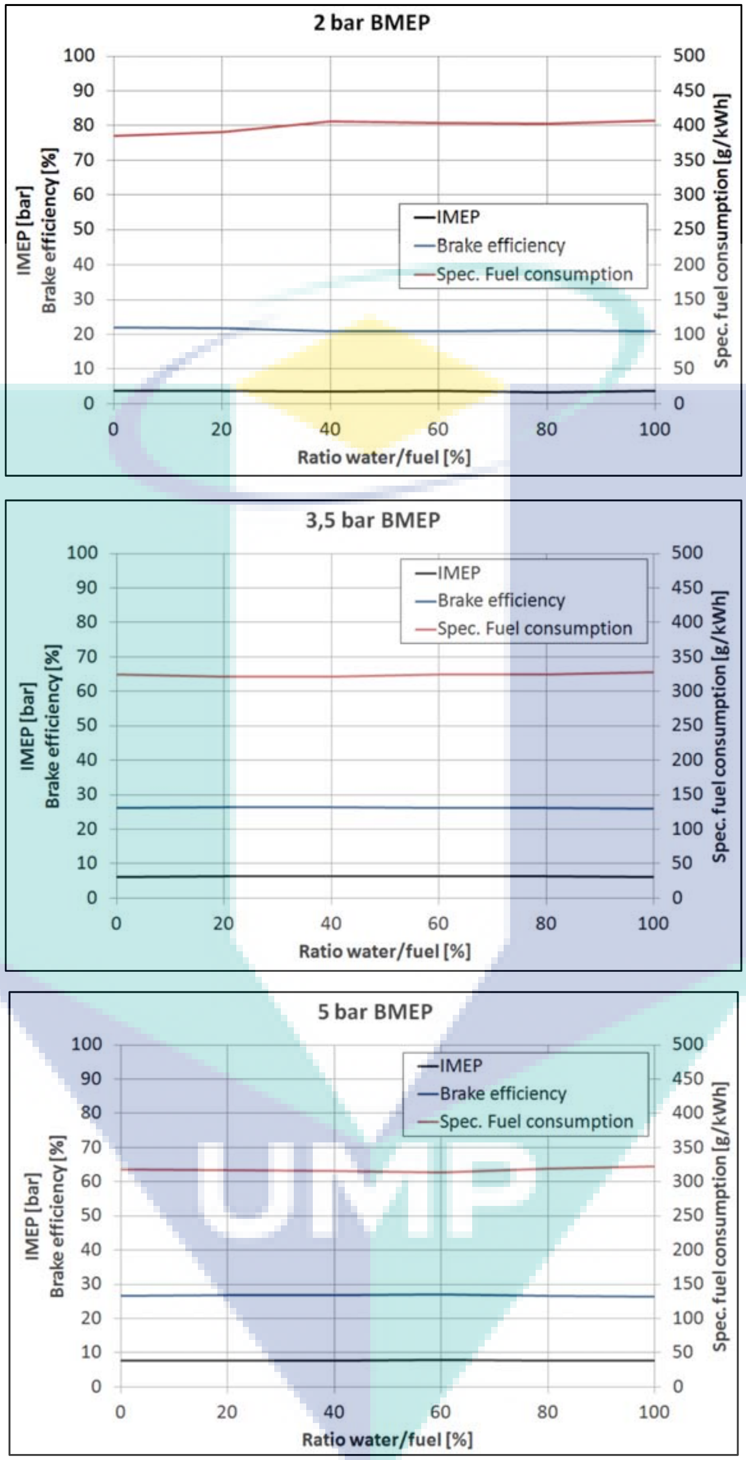

Figure 6.20: IMEP, brake efficiency and specific fuel consumption for 2,3,5 and 5 bar BMEP over the water/fuel ratio

6.5 Influence of different water injection locations

After analysing the first data, the next experiment could start. In this experiment the influence of the different water injection locations are analysed. The BMEP is set to 5 bar which is a load of 119 Nm and the engine speed is set to 2000 rpm. The engine is hold at this load point until the temperatures are constant. After all temperatures are constant the water injection system is turned on with the settings of a 100% water/fuel ratio. The temperatures are recorded for 180 seconds to see the dynamic behaviour of the water injection system. The locations are changed from the top position, to the side position and at last to the bottom position. In between the change of the location of the injector, the engine was shut down to do the mechanical work. After start up the engine, the same load point could not exactly been reached again. So there are slightly different values in the results between the next 3 experiments. In figure 6.21 the influence of the three water injection locations is shown. The temperature after the water injection is spread over a time of 180 seconds. All three temperatures start almost at the same point of about $110^{\circ}c$ and end at $93^{\circ}c$. A difference is detected between the start and the end temperature. The single temperatures decrease differently. The gradient of the bottom location is greater than the gradient of the side and top location. With the location of the injector at the bottom the temperature decreases faster. After 30 seconds the end temperature of $93^{\circ}c$ is reached and is constant the remaining time. If the injector is located at the side, the temperature after the water injection takes 90 seconds to reach the end temperature of $93^{\circ}c$. The slowest behaviour shows the location if the injector is installed at the top location. After a time of 150 seconds the end temperature of $93^{\circ}c$ is reached. The dynamic of the water injection is the best if the injector is located at the bottom. This is caused by the near physical distance between the injector and the intake valve. If it is possible to locate the injector even closer to the intake manifold the system would be even more dynamic. To close should the injector not be located to the intake valve to make sure that the water vaporises complete. In figure 6.22 the influence of the different water injection locations for the emission is shown. On the left y-axis the NOx emission in ppm is displayed, on the right y-axis the CO emission in vol. % is displayed. The x-axis shows the water/fuel ratio from 0% to 100%. The emissions are not really effected by the different water injection locations. In figure 6.22 the NOx emissions have almost the same values for every water injection location. The same goes for the CO emission where the values are the same for every water injection location. All other measured values show exact the same behaviour. They are not affected by the different water injection locations. The fuel consumption, the pressure, the IMEP, the torque, the other temperatures, the particle matter as well as the CO2 and O2 emission are not effected in this experiment.

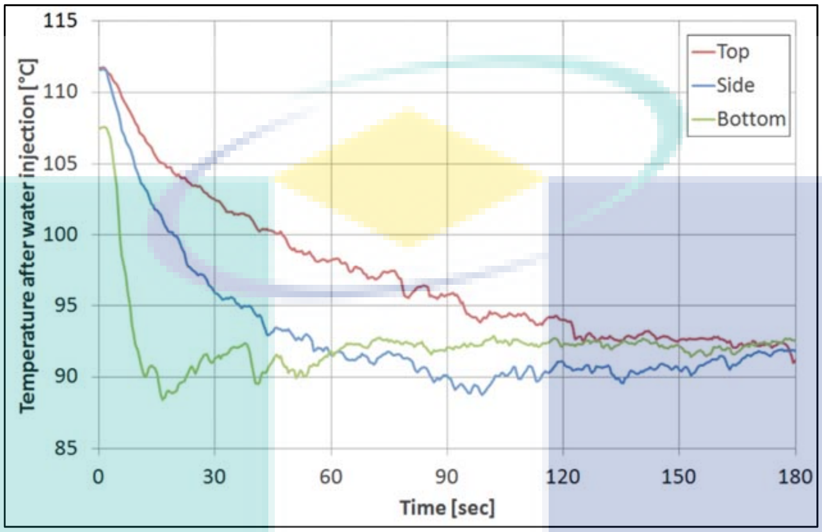

Figure 6.21: Dynamic behaviour of the three water injection locations over the time

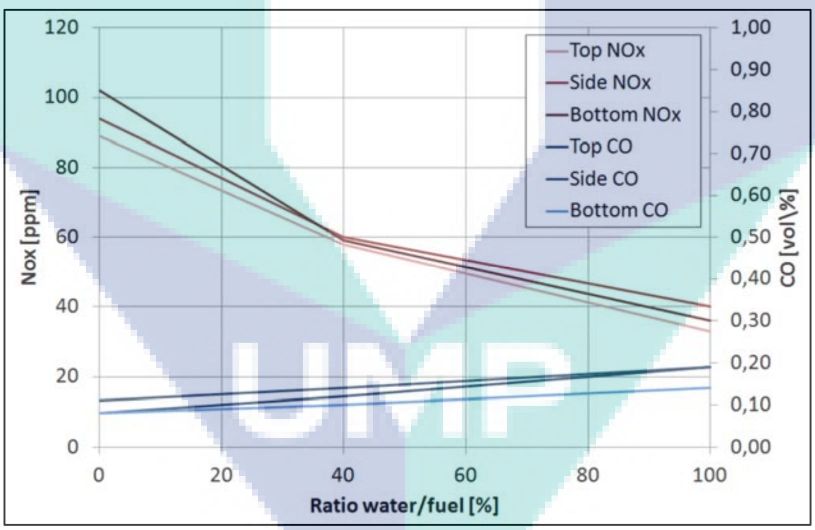

Figure 6.22: Influence of the different injector locations on the NOx and CO emission for 2000 rpm and 5 bar BMEP

6.6 Influence of different injection times for the water injection

In this section the influence of different injection times for the water injection is analysed. Therefore the begin of the water injection is changed from 20 over 40 to 60 deg CA after the inlet valve opened. The engine speed is set to 2000 rpm and the BMEP is set to 5 bar which is an engine load of 119 Nm. The begin of the water injection is controlled by the electronic control devise from Edgar Huber [25]. During the test run the begin of the water injection is being changed without stopping the engine. The investigation is made for a water/fuel ratio of 40%. In figure 6.23 the temperature after the water injection, the IMEP and the NOx emission are shown on the left y-axis. The CO emission is shown on the right y-axis. All values are displayed over the begin of the water injection. The begin of the water injection is expressed as deg CA after the inlet valve opened. The change of the begin of the water injection time is not effecting any value in the measurement. All values are constant and do not change the value during the change of the begin of the water injection time. The injected water vaporises due to the warm temperatures in the intake manifold and the temperature sensor detects only the water steam which is in the intake air. The little up and down in the figure is caused by the tolerance of the measurement equipment.

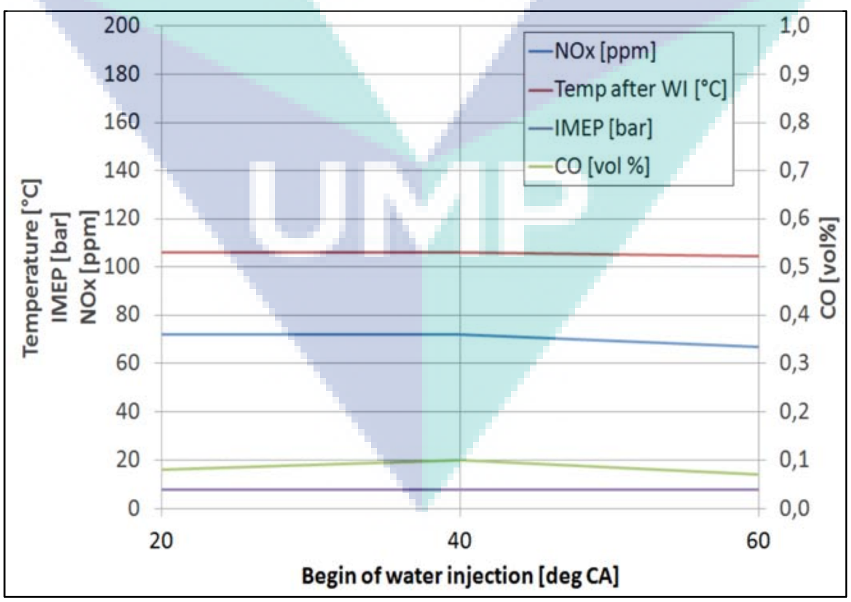

Figure 6.23: Influence of the begin of the water injection on the temperature after the water injection, the IMEP, the NOx and CO emission for 2000 rpm, 5 bar BMEP and a water/fuel ratio of 40 %

7 Conclusion and Perspectives

In this project the influence of a water injection system on the emissions of a diesel engine is investigated. The injection of water into the intake manifold shows a lot of advantages in order to influence the emissions. The drawbacks are small compared to the advantages. All experiments are down with 2000 rpm and three different engine load points. The water/fuel ratio is increased in a range of 0% to 100% in 20% steps. Injecting water to the intake manifold decrease the temperature in the combustion chamber, in front of the valves and the exhaust temperature after the turbocharger output. The temperatures in front of the valves can be decreased up to $20^{\circ}c$ with a water/fuel ratio of 100 % at a engine load of 119 Nm. With lower loads and the same water/fuel ratio the temperatures can be decreased up to $15^{\circ}c$. The temperatures in the combustion chamber and in the exhaust are decreasing in the same way. The other temperatures like the oil, water, fuel, air before and after the intercooler and the ambient are constant over the whole test runs during injecting water. This leads to a decrease in the emissions after the turbocharger. The lower temperatures in the combustion chamber inhibit the formation of NOx emission. With injecting water the NOx emission can be reduced about two-thirds. At the low partload point with 44 Nm the NOx emission could even been decreased of 72% after the turbocharger. This sounds at first nice, but a small drawback is included in this improvement of decreasing the NOx emission. With the decrease of the NOx emission the particle matter increase, which leads to a trade-off. If it is desired to decrease the NOx emission drastically the use of the water injection into the intake manifold is an efficient solution, with the drawback of higher PM emission. To get a balance between the NOx and PM emission a water/fuel ratio of about 40% should be used. The other emission CO, CO2 and O2 are increasing very slightly. This should not be a problem. The use of a diesel oxidation catalytic converter decreases the remaining CO. The pressure in the cylinder rises marginally with an increase of the injected water mass. The maximum peak pressure increases by 2 bar, with a 100% water/fuel ratio at a engine load point of 5 bar BMEP. If the water/fuel ratio is in further projects increased even more the pressure should be watched. The combustion process is lightly effected by the water injection. The burned mass fraction positions are moving 1 to 4 deg CA with a increase of the water/fuel ratio. An ignition delay is detected in the pre fuel injection zone of about 5 deg CA. The main and post ignition start a few deg CA later. The indicated mean effective pressure, the torque, the specific fuel consumption and the efficiency are not really affected by the water injection. In the figures they seem to change, but this is caused by fluctuation and tolerances from the test bed, the dyno and the control device. The consistency of the performance values are neither detected as an advantage nor a disadvantage during the tests. There is no need of an optimisation of the engine ECU.

All these three experiments are simulated before to check the behaviour of the water injection. The results show more or less the same tendency compared to the measurements on the test bed. Due too many estimations and tolerances of the measurement equipment and the test bed, the simulated values are different to the measurements on the test bed. If the injection location is changed an influence on the dynamic behaviour of the temperature after the water injection is detected. The nearer the injector is located to the intake valve, the faster the temperature decrease if the water injection started from a 0% water/fuel ratio to a 100% water/fuel ratio immediately. The temperature after the water injection is after 30 seconds steady if the injector is located towards the air stream. If the injector is set to the top location to inject with the air flow the temperature after the water injection is steady not until 150 seconds. The other measured values like the emissions, the combustion process, the performance and the other temperatures are not affected by the different injection locations.
In another test run the influence of different water injection timing are investigated. All test runs above are made with a injecting time 20 deg CA after the inlet valve opened. In this test run the injecting time is set to 40 deg CA and 60 deg CA after the inlet valve opened. After analysing the data absolutely no difference is detected between the injecting times. There is no influence on the engine with a change in the injecting times. Summed up the injection of water into the intake manifold decreases the NOx emission drastically, with the drawback of an increase of the particle matter. The combustion process and the performance are affected slightly or are even not affected by the water injection. Further projects can do a lot of research on this system. A variation of the injected water temperature can be investigated. Another parameter like the pressure of the injected water can be investigated or the use of a different injector from a diesel direct injector. To get more results the engine load and speed should be increased. With the additional data a characteristic map of the water injection can be created. In this map the injection pressure and the begin of the injection can be set over the engine load and engine speed. Another aspect which can be investigated is different injection positions or even more than one water injector. For example for every cylinder, or one in the intake manifold and one before the intercooler. The variation of the relative humidity and

temperature of the ambient can be changed to see how the water injection system reacts in other regions on the earth, where it is very cold or very hot and dry. The use of the water injection in engines for everybody has to be investigated after all these research projects are done. The use in passenger cars might take some time to implement and to get more information how the water effects the engine over a long term, or what happens with the water when the ambient temperatures drops below $0^{\circ} c$. Another problem is the storage of the water in the car. There must be a additional tank, which causes an extra place and weight in the car. It will take some years to get the water injection ready for series production and to apply in every car. In engines which are running a longer time over a constant speed and load, like in ship engines, generators or in a steady operation point it is a very good solution to decrease the emissions.

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Appendix

Case	Ratio w/f [%]	Pedal pos. [%]	m.water [mg/s]	Engine speed [rpm]

BMEP = 2 bar, T = 44 Nm

1	0	10	0	2000	Experiment 1
2	20	10	111	2000	
3	40	10	223	2000	
4	60	10	334	2000	
5	80	10	446	2000	
6	100	10	557	2000	

BMEP = 3,5 bar, T = 83 Nm

1	0	15	0	2000	Experiment 2
2	20	15	138	2000	
3	40	15	276	2000	
4	60	15	414	2000	
5	80	15	553	2000	
6	100	15	691	2000	

BMEP = 5 bar, T = 120 Nm

1	0	30	0	2000	Experiment 3
2	20	30	213	2000	
3	40	30	426	2000	
4	60	30	640	2000	
5	80	30	853	2000	
6	100	30	1067	2000	

UMP

Table A.1: Simulations cases for the experiments 1 to 3

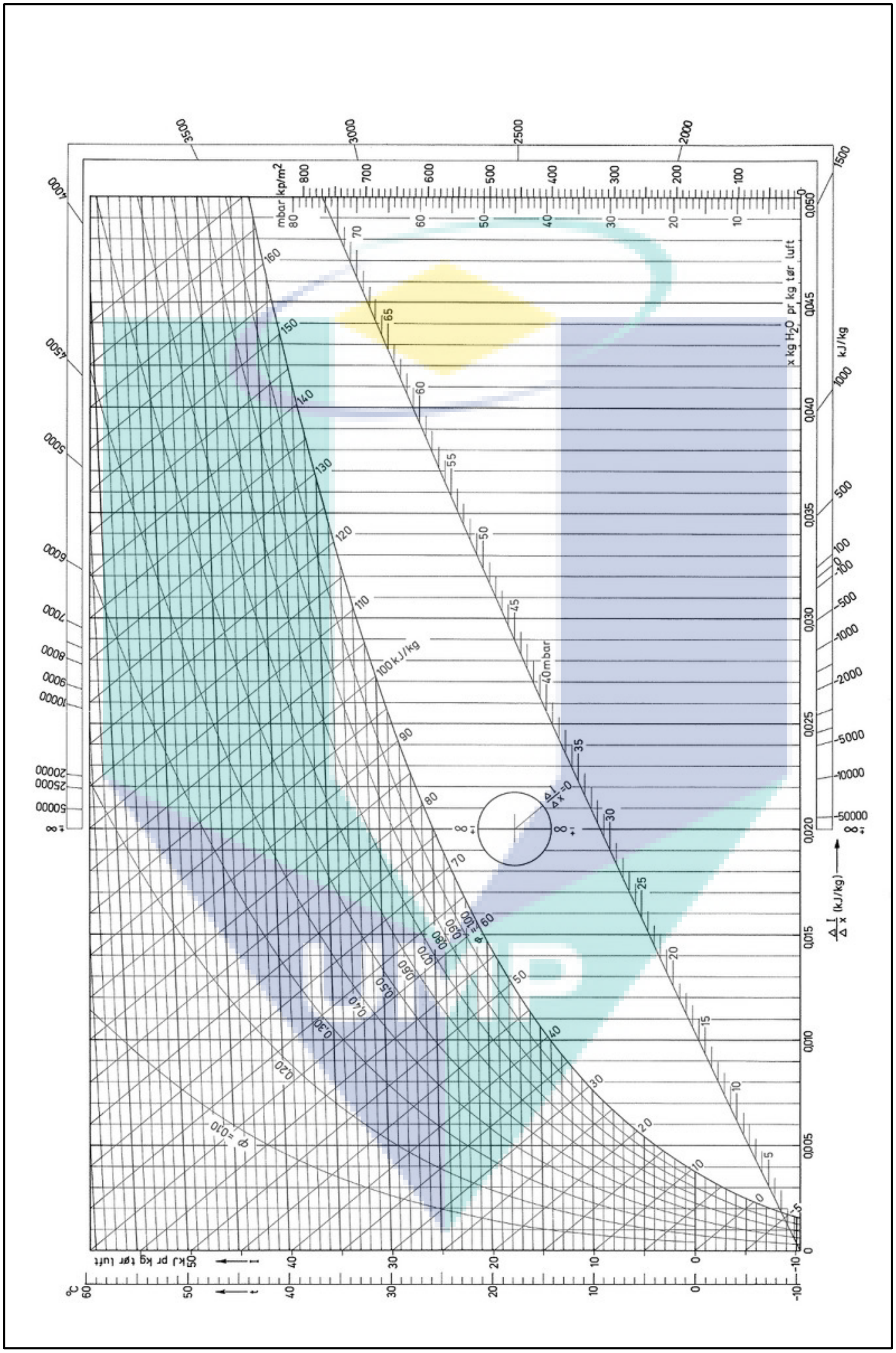

Figure A.1:Moillier diagramm for p = 1.013 *bar* [31]

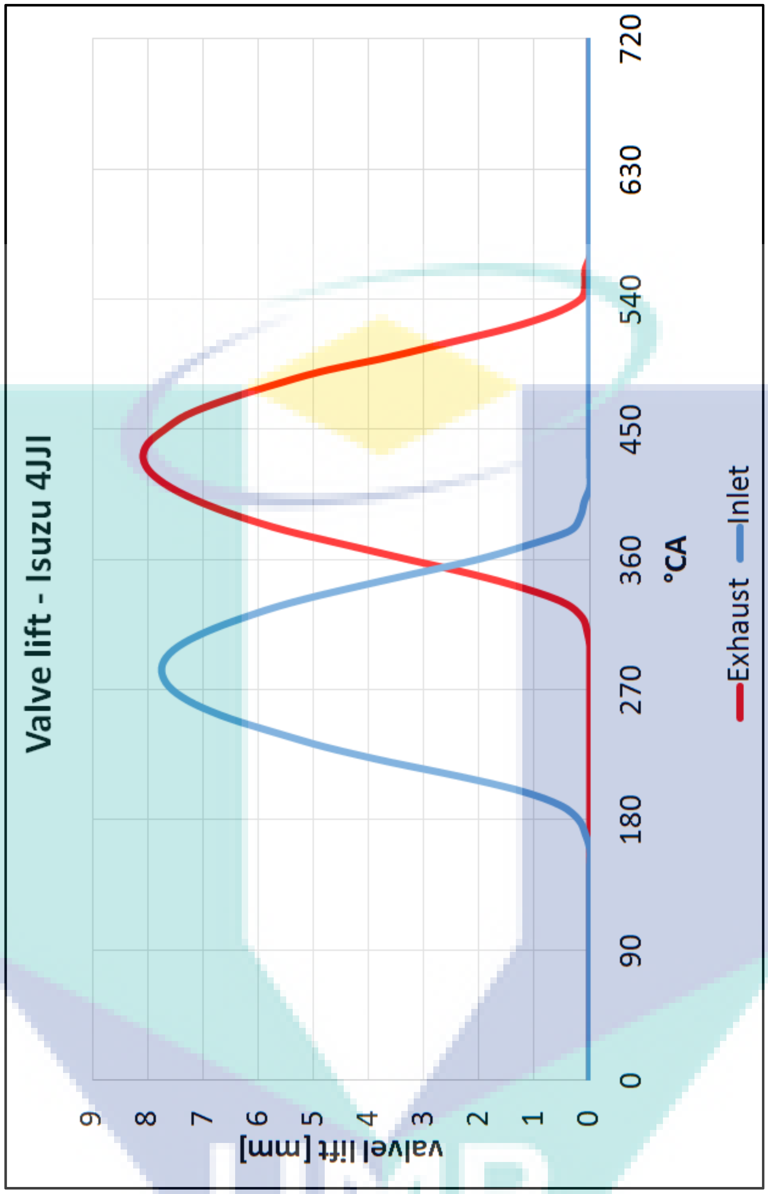

Figure A.2: Valve lift for cylinder 2

Cylinder #	1
Part Name	Cyl1
Volumetric Efficiency [%]	87.9
Volumetric Efficiency (m) [%]	79.9
Trapping Ratio	0.992
Burned Residuals Mass (SOC) [%]	13.9
EGR [%]	9.1
F/A Ratio (trapped)	0.038
Lambda, Effective	1.697
IMEP [bar]	5.68
PMEP [bar]	-1.33
ISFC [g/kW-h]	238.0
Indicated Efficiency [%]	35.2
Fuel Mass [mg]	28.1
Maximum Pressure [bar]	70.22
CA at Max. Pressure [deg]	6.6
dPmx/DCA [bar/deg]	2.095
Maximum Temperature [K]	1672

Figure A.3: 5 bar BMEP, 0% water/fuel ratio for cylinder 1

Intake Pressure [bar]	1.266
Intake Temperature [K]	361
Exhaust Pressure [bar]	2.443
Exhaust Temperature [K]	812
Heat.Tr. (frac. of F.E) [%]	22.5
Swirl at TDC	0.000
Swirl at BDC	0.000
NOx in ppm	471.64
Indicated Specific NO2 [g/kW-h]	5.00
Soot Concentration @ STP [g/m^3]	0.00
Indicated Specific Soot [g/kW-h]	0.00
HC in ppm	0.52
Indicated Specific HC [g/kW-h]	0.02
CO in ppm	2.51
Indicated Specific CO [g/kW-h]	0.02
CO2 in ppm	74955.60
Indicated Specific CO2 [g/kW-h]	759.25

Figure A.4: Continuation 5 bar BMEP, 0% water/fuel ratio for cylinder 1

Cylinder #	1
Part Name	Cyl1
Volumetric Efficiency [%]	98.9
Volumetric Efficiency (m) [%]	77.1
Trapping Ratio	0.989
Burned Residuals Mass (SOC) [%]	13.9
EGR [%]	8.9
F/A Ratio (trapped)	0.035
Lambda, Effective	1.703
IMEP [bar]	5.96
PMEP [bar]	-1.54
ISFC [g/kW-h]	234.0
Indicated Efficiency [%]	35.8
Fuel Mass [mg]	29.0
Maximum Pressure [bar]	76.08
CA at Max. Pressure [deg]	6.7
dPmx/DCA [bar/deg]	2.218
Maximum Temperature [K]	1577

Figure A.5: 5 bar BMEP, 100% water/fuel ratio for cylinder 1

Intake Pressure [bar]	1.321
Intake Temperature [K]	326
Exhaust Pressure [bar]	2.687
Exhaust Temperature [K]	766
Heat.Tr. (frac. of F.E) [%]	20.2
Swirl at TDC	0.000
Swirl at BDC	0.000
NOx in ppm	230.02
Indicated Specific NO2 [g/kW-h]	2.63
Soot Concentration @ STP [g/m^3]	0.00
Indicated Specific Soot [g/kW-h]	0.00
HC in ppm	0.49
Indicated Specific HC [g/kW-h]	0.02
CO in ppm	2.19
Indicated Specific CO [g/kW-h]	0.02
CO2 in ppm	68436.80
Indicated Specific CO2 [g/kW-h]	746.95

Figure A.6: Continuation 5 bar BMEP, 100% water/fuel ratio for cylinder 1

	Performance
Brake Power [kW]	22.7
Brake Power [HP]	30.4
Brake Torque [N-m]	108.2
IMEP [bar]	5.69
FMEP [bar]	1.16
PMEP [bar]	-1.33
Air Flow Rate [kg/hr]	180.4
BSAC [g/kW-h]	7957
Fuel Flow Rate [kg/hr]	6.8
BSFC [g/kW-h]	298.8
Volumetric Efficiency [%]	87.4
Volumetric Efficiency (M) [%]	79.4
Trapping Ratio	0.997
A/F Ratio	26.63
Brake Efficiency [%]	28.0

Figure A.7: Performance values for 5 bar BMEP, 0% water/fuel ratio whole engine

	Performance
Brake Power [kW]	24.0
Brake Power [HP]	32.2
Brake Torque [N-m]	114.8
IMEP [bar]	5.98
FMEP [bar]	1.18
PMEP [bar]	-1.54
Air Flow Rate [kg/hr]	203.6
BSAC [g/kW-h]	8471
Fuel Flow Rate [kg/hr]	7.0
BSFC [g/kW-h]	291.3
Volumetric Efficiency [%]	98.7
Volumetric Efficiency (M) [%]	76.9
Trapping Ratio	0.996
A/F Ratio	29.08
Brake Efficiency [%]	28.7

Figure A.8: Performance values for 5 bar BMEP, 100% water/fuel ratio whole engine