Investigation of biofuelled combustion and their

performance optimisation strategies for internal

combustion engines



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Thomas Bohl

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ABSTRACT

The increasing use of biofuels to replace fossil fuels as well as more stringent emissions regulations for internal combustion engines cause a challenge for the engine manufacturer to build engines that can cope with a large range of fuel properties, but still offer low fuel consumption and very low exhaust emissions.

In this work a heavy-duty diesel engine test bed has been built including the fuel and emission analysis equipment suitable for a wide range of biofuels. Also a constant volume spray vessel has been commissioned to optically investigate the macroscopic spray characteristics of different fuels. This vessel was built with the potential investigation of fuel combustion in the future. Four different biofuels, soybean oil methyl ester (SME), palm oil methyl ester (PME), used cooking oil methyl ester (UCOME) and hydrotreated vegetable oil (HVO) in blends of B10, B20, B50 and B100 have been tested, as they are potential candidates to replace mineral diesel in larger scales.

The main aim of this project was to investigate the spray and combustion characteristics of various biofuels, their impact on exhaust emissions and performance and the potential optimisation of the control strategy in a heavy-duty Euro V diesel engine. The engine tests revealed that for all biofuels the nitric oxide (NO_x) emissions increased compared to mineral diesel (B0), while particulate number (PN), carbon monoxide (CO) and total hydrocarbon (THC) were significantly reduced. The fuel consumption changed according to the heating value and with the three fatty acid methyl esters (FAME) full power was not reached.

The macroscopic spray tests showed that lower density fuels, such as HVO, have slower penetration speeds, but wider spray cone angle resulting in better fuel-air mixing conditions. As the engine fuel injection is based on a volumetric injection the heating value and fuel density are mainly influencing the spray characteristics on the engine.

In the last part the engine power has been successfully restored for all biofuels and the exhaust emissions have been reduced below the B0 benchmark limits by applying a new engine control strategy showing that the use of neat biofuels can be used on heavy-duty diesel engines without any modifications to the engine hardware and still passing the current emission regulations.

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DEDICATION

I dedicate this work to my beloved wife Katrin Bohl. You compromised so much over the last years to make this thesis possible. Thank you!

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LIST OF ABBREVIATIONS AND SYMBOLS

Abbre viations

ADC	Athens driving cycle	EU	European Union
ASTM	American standard for testing materials	FAME	fatty acid methylester
aTDC	after top dead centre	FBP	full boiling point
B0	neat mineral diesel without biofuel	FFA	free fatty acid
B10	10 % biofuel and 90% mineral diesel	FID	flame ionisation detector
B20	20 % biofuel and 80% mineral diesel	FQS	Fuel Quality Sensor
B50	50 % biofuel and 50% mineral diesel	GHG	greenhouse gas
B100	100 % biofuel and 0% mineral diesel	GtL	Gas-to-liquid fuel
BDC	bottom dead centre	GUI	graphical user interface
BMEP	brake mean effective pressure	HD	heavy duty
BNC	Bayonet Neill-Concelman	HHV	higher heating value
BP	British Petroleum	HRR	heat release rate
BSFC	break specific fuel consumption	HVO	Hydrotreated vegetable oil
bTDC	before top dead centre	ICE	internal combustion engine
BTE	brake thermal efficiency	ID	ignition delay
BTEX	benzene, toluene, ethyl benzene	IDI	indirect injection
CA	crank angle	IMEP	indicated mean effective pressure
CAD	crank angle degree	IP	injection pressure
CCD	charge-coupled device	ISB	a Cummins engine model
CFD	computational fluid dynamics	IT	injection timing
CI	compression ignition	LHV	lower heating value
CLD	chemiluminescence detector	NDIR	nondispersive infrared sensor
CO	carbon monoxide	NEDC	new European driving cycle
COV	coefficient of variation	NIR	nearinfrared
CN	cetane number	NO _x	nitrogen oxides (NO and NO2)
CPC	condensation particle count	NSF	non-soot fraction
CR	compression ratio	PAH	polycyclic aromatic hydrocarbon
CV	calorific value	PM	particulate matter
CVV	constant volume vessel	PME	palm oil methyl ester
DAQ	data acquisition	PN	particulate number
DI	direct injection	RGB	Red – Green - Blue
DoE	design of experiment	RME	Rapeseedoilmethylester
DPF	diesel particulate filter	RP	rail pressure
EGR	exhaust gas recirculation	RTFO	Renewable transport fuel obligation
ECU	engine control unit	SCR	Selective catalytic reduction
ESC	European stationary cycle	SME	soybean oil methylester

SN	smokenumber	TDC	top dead centre
SPCS	solid particle counting system	THC	totalhydrocarbon
SMD	Sauter mean diameter	TTL	transistor-transistor logic
SOC	start of combustion	UCOME	used cooking oil methylester
SOF	soluble organic fraction	VI	virtual instrument in Lab VIEW
SOI	start of injection	VOF	volatile organic fraction
STD	standard deviation		

Symbols

А	factor for spray penetration model					
В	exponent for spray penetration model					
c_p	heat capacity and constant pressure					
c_{v}	heat capacity and constant volume					
\mathbf{d}_0	nozzle diameter					
dm	mass of fuel					
dP	pressure difference					
dU_{s}	change of sensible energy in the piston					
g	gaseous					
h	enthalpy of fuel					
1	liquid					
\mathbf{M}_{inj}	injected mass perstroke					
$Q_{ch} \\$	chemical energy released by combustion					
$Q_{\rm ht}$	heat transfer					
\mathbf{Q}_{n}	net release rate					
р	combustion chamber pressure					
r _c	compression ratio					
r _p	pres sure ratio					
\mathbb{R}^2	coefficient of determination					
S	penetration length					
Т	injection duration					
t_{break}	break-up time					
$T_{\rm w}$	wall temperature					
V	combustion chamber volume					
V _{cr}	crevice volume					
W	mechanicalwork					
Х	proportional factor for spray model					
z	weighting factor					

α	load ratio
γ	ratio of heat capacities
$\eta_{\rm b}$	burning efficiency
η_{g}	efficiency of the cycle factor
η_{m}	mechanical efficiency
η_{th}	thermalefficiency
λ	air-to-fuel ratio
ρ	density
θ	crankangle

Chapter 1. Introduction

1.1 Background

The 2014 energy outlook forecasts an increase in population to 8.7 billion people by 2035 and an increase in primary energy use by 37 % between 2013 and 2035 [1]. Approximately 20 % of the total energy consumption is used in the transportation sector with a share of 95 % fossil fuel currently [2]. In 2035 still around 89 % of the transportation fuels will be derived from oil, but a share of 11 % will be non-fossil fuel alternatives, such as biodiesel and bioethanol [1].

Increasing fossil fuel consumption combined with depleting reserves and rising instability of the oil-producing countries leads to a significant growth of energy prices [3, 4]. Another concern of using fossil fuels and the increasing level of GHG emissions is global warming and ozone layer depletion. Predictions for the period of 1990 to 2020 state that road transportation and fossil fuel consumption will increase by a factor of three and the resulting emissions are a source of global warming. The energy demand will increase in all sectors, but most significantly in the use of automobiles [5, 6]. Biofuels are considered to have the potential to alleviate problems of fossil fuel depletion, fuel dependency and greenhouse gas emissions at some extend. However, currently the production of biofuels is too expensive to become a serious competitor to fossil fuels. Therefore most governments started to incentivise the production of biofuels from different feedstock. In 2006 the total support of biofuels including tax incentives was around 3.7 billion EUR [7].

To face the above mentioned problems, the European Union as well as the UK government has introduced regulations to support the use of biofuels. The EU Biofuels Directive (2003/30/EC) was adopted in May 2003 and set the target of 5.75% biofuel blend by 2010. In January 2007 this target was amended to 10% by 2020. The UK announced the Renewable Transport Fuel Obligation (RTFO) to commit transport fuel suppliers to blend their fuel with at least 5% biofuel [6, 8].

Furthermore, the European Commission has published directives to regulate and reduce the exhaust emissions from light- and heavy-duty vehicles. In Europe and many other countries in the world the vehicle manufacturers are committed to meet these EU emission standards [9]. Currently, nitrogen oxides (NO_x) , carbon monoxide (CO), total hydrocarbon (THC), non-methane hydrocarbons (NMHC) and particulate

matter (PM) emissions are regulated in these standards. Important unregulated emissions are aromatic, polyaromatic and carbonyl compounds, etc. [4, 10].

New vehicles introduced in the EU must meet current or planned standards. The latest standard for light commercial vehicles and passenger cars is Euro VI, which is illustrated in Table 1-1.

Stage	Date	Test	CO	НС	NOx	РМ	PN
			g/kWh				1/kWh
Euro VI	Jan 2013	WHSC	1.5	0.13	0.4	0.01	8.0×10 ¹¹
Euro VI	Jan 2013	WHTC	4	0.16 ^d	0.46	0.01	6.0×10 ¹¹

 Table 1-1: Euro 6 emission regulations for heavy-duty diesel vehicles [11].

The NO_x emission limits in Euro VI have been reduced from 2 g/kWh in the previous Euro V norm to 0.4 g/kWh [6]. This means that very clean combustion and a good aftertreatment system is necessary to meet these low standards. In terms of PM, the limit in Euro VI has been reduced by 50 % compared to Euro V, but it already forced manufacturer to install diesel particulate filters (DPF) in the aftertreatment system. However, the most important change in Euro Vb and VI is the particulate number (PN), which was not implemented in any of the previous regulations.

For heavy-duty vehicles with a 'technically permissible maximum laden mass' of over 3.5 tons the emission regulations differ from regulations for passenger cars as illustrated in Figure 1-1. The NO_x aftertreatment techniques, such as SCR, have already been implemented before Euro V, so that for Euro VI the main focus will be to reduce PM emissions [6]. The history for the European standards can be seen in Figure 1-1. The standards for light-duty and heavy-duty vehicles are not really comparable, since the emissions for light-duty vehicles are defined in g/km and for heavy-duty vehicles in g/kWh.



Figure 1-1: European Union (EU) emissions standards for passenger (left) cars and heavy-duty (right) diesel engines [12].

The increasing fuel demand, global warming and the tight emission standards force researchers and vehicle manufacturers to consider alternative fuels and improve the combustion performance. For this purpose, some research projects focus on higher fuel injection pressures, multiple injection strategy, optimised aftertreatment systems, including EGR, particulate traps and the controlling of the right injection timings [9]. While reviewing recent literature in this field it was concluded that little research has been carried out on injection optimisation of various biofuels when considering their differences in fuel properties, such as cetane number (CN) and fuel density. Also, a holistic study of combustion tests, spray analysis and engine recalibration using various types of biofuels, such as biodiesels and hydrotreated fuels, has not been carried out so far and provides a gap of research. This proposal is made to approach the use of various diesel-like biofuels and investigate the combustion performance in a heavy-duty DI diesel engine provided by Cummins UK Ltd.

1.2 Aims

The main aim of this research is to investigate the spray and combustion characteristics of various diesel-like biofuels, their impact on emissions and performance as well as the potential optimisation of the injection strategy in HD diesel engines

1.3 Objectives

The objectives for this thesis are divided as follow:

- 1. Conduct a review of various literature and research articles regarding the feasibility of various biofuels for commercial use, fuel injection and combustion of different diesel-like biofuels, and the impact of certain fuel properties on the emission formation.
- 2. Set up a state-of-the-art HD diesel engine test bed for combustion analysis and exhaust emission measurements of various biofuels.
- 3. Set up a comprehensive constant volume vessel (CVV) test rig including a high speed camera system as well as a high pressure fuel system to investigate the macroscopic spray characteristics of various biofuels.
- Investigate and analyse engine performance and exhaust emission (CO, NO_x, THC and PN) using various biofuels in a representative engine test cycle.
- 5. Investigate the macroscopic spray characteristics of various biofuels using optical diagnostic tools.
- 6. Carry out potential engine power restoration and engine recalibration using few fuel representatives.
- 7. Propose an optimised engine control strategy using biofuels to improve engine performance and minimise engine exhaust gas emissions.

1.4 Thesis outline

This research thesis focuses on the challenge of fossil fuel depletion and increasing pollutant emissions of power generation facilities. Diesel-like biofuels have the potential to overcome those problems, since it is made from renewable sources and it can reduce engine emissions significantly. The conducted work is summarised in the following chapters:

Chapter 1 – Introduction

Contains background information in the research field and explains the aims and objectives of this work.

Chapter 2 – Project fundamentals

Introduces the modern fuel injection system and the diesel engine combustion and emission formation process. The formation of the four measured exhaust emissions (PN, NO_x , THC and CO) is explained in detail before various biofuels are reviewed with focus on their production methods, fuel properties and combustion characteristics.

Chapter 3 – Methodology

Contains a detailed description of the two test rigs, which have been set up during this time to carry out the research for this project.

A four cylinder Cummins ISB Euro V diesel engine test rig has been built from scratch to investigate the biofueled combustion and emission formation.

A constant volume vessel (CVV) has been commissioned to investigate the macroscopic spray characteristics of various biofuels at engine-like injection conditions.

Chapter 4 – Engine tests using various biofuels

Compares the results of the engine tests of 16 different biofuel types plus reference diesel (B0) with focus on engine performance, formation of NO_x , PN, THC and CO emissions as well as combustion analysis using the crank angle based in-cylinder pressure.

Chapter 5 – Optical diagnostics of various biofuels

Compares the results of macroscopic spray characteristics of 8 different biofuel types plus reference diesel (B0) in terms of spray tip penetration, spray cone angle and surface area using a high speed camera system.

Chapter 6 – Engine power restoration and optimisation of biofuels

Investigates the feasibility of power restoration with neat SME fuel and assesses potential injection control optimisation using neat HVO and SME fuel by investigating the effect of changing start of injection (SOI) and rail pressure (RP) on NO_x and PN formation.

Chapter 7 – Conclusions and future work

Concludes the findings in the previous chapters and proposes a prototype solution for an optimised control strategy using various biofuels. Also, some recommendations for future work are listed.

Chapter 2. Project fundamentals

The chapter starts with the basic operation of a diesel engine – the four stroke engine cycle. The modern fuel injection system including common rail and solenoid and piezo injectors are introduced. Several attempts to improve fuel combustion and reduce engine emissions are discussed. Different sustainable biofuels and their power and emission performance are reviewed at the end of this chapter.

2.1 The diesel engine

The German engineer Rudolf Diesel is the pioneer of the very first diesel engine. The efficiency of 26.2 % was about twice as high as a contemporary power plant [13]. Nowadays internal combustion engines (ICE) are used in a huge variety of applications, such as generators, on-road vehicles, agriculture machinery, railways and ships. A diesel engine is a compression ignition (CI) engine where fuel and air is mixed together in the combustion chamber. The air is compressed into the cylinder first and due to the high pressures the temperature rises. Diesel is injected into the hot chamber and the fuel mixes with the surrounding air until the self-ignition point is reached and the combustion starts. The chemical energy of the fuel is released as heat and converted into mechanical energy by driving the piston reciprocally. The engine can operate in four- or two-stroke cycles, whereas the four-stroke engine each stroke is designed for air intake, exhaust, compression and expansion.

2.1.1 Efficiency

The efficiency of the diesel engine, equation (2-1), is defined as the effective available work divided by the energy supplied by the fuel [14].

$$\eta = \frac{W_e}{m_{fuel} \cdot CV_{fuel}} = \eta_{th} \cdot \eta_g \cdot \eta_b \cdot \eta_m \tag{2-1}$$

The efficiency can be calculated by knowing the thermal efficiency of the ideal process and the influences of the real process.

$$\eta_{th} = 1 - \frac{1}{r_c^{\gamma-1}} \left[\frac{r_p \cdot \alpha^{\gamma} - 1}{(r_p - 1) + \gamma \cdot r_p(\alpha - 1)} \right]$$
(2-2)

The parameter r_p is the pressure ratio during constant-volume heat addition, γ the ration of gas heat capacities c_p and c_v and α the load ratio.

The thermal efficiency of the ideal gas standard dual cycle considers mainly heat losses and is dependent on the pressure ratio, compression ratio r_c and the load ratio.

The efficiency of the cycle factor, η_g , includes the deviations between the real and the ideal gas, as well as wall heat losses and flow heat losses during the gas-exchange process.

 η_b considers losses due to the incomplete combustion of the fuel and η_m is the mechanical efficiency which includes friction losses from the piston, piston rings, bearings, oil pump, coolant pump and fuel injection pump. If the engine is turbocharged, this must also be included [14].

2.2 Modern fuel injection system

The fuel injection system is the heart of any engine. Here, most parameters can be changed to improve spray atomisation and thus increase the efficiency of the combustion. Over the last decades the fuel injection systems changed from mainly mechanically to now electronic-controlled injection systems. This opened opportunities for faster injection rates, higher pressures and multi-injection events. In addition, modification on combustion chamber geometry and the use of turbocharged air as well as exhaust gas recirculation have improved the efficiency and emission reduction drastically.

2.2.1 Direct injection and indirect injection system

The combustion chamber is the place where the engine combustion takes place. There are two main types of chambers, those with a direct injection (DI) into the chamber and those with an injection into a pre chamber called indirect injection (IDI). The fuel injection system is a very important part of the engine as this determines the air-fuel mixing process and influences combustion behaviour and emission formation. The air motion in DI engines is inherently smaller compared to IDI engines. However, this requires higher injection pressures with multiple-hole nozzles to achieve similar mixing rates. In the last decades the DI technology has widely replaced IDI engines as fuel savings of up to 20 % can be achieved while the higher combustion noise of DI systems has been reduced with introducing pilot

injection [14]. In DI engines the fuel atomisation, heating, vaporisation and mixing with the air must take place in a rapid succession to achieve good combustion performance. This sets special requirements on the exact fuel and air demand going into the chamber. With a decrease in engine size often an air swirl is used to achieve faster air-fuel mixing rates. The special shape of the intake air channel and the combustion chamber bowl creates an air vortex inside the cylinder which enhances the fuel-air mixing. The combustion chamber shape is usually a shallow bowl in the crown of the piston with a central multi-hole injector [15]. Koerfer et al. found that a turbocharged DI engine can achieve a specific power output of 70 kW/litre with a BMEP of over 25 bar when using conventional diesel [16].

The amount of fuel injected into the cylinder is approximately proportional to the power output of the system. However, high amounts of fuel can lead to incomplete combustion and the formation of excessive amounts of soot and therefore a lean fuelair ratio (and at part load a very lean fuel-air ratio) is necessary in diesel engines. This leads to more complete combustion and therefore higher efficiencies.

The ratio of air is usually expressed by the excess-air factor λ (lambda) related to the mass of air required for stoichiometric combustion. A $\lambda=1$ means that the air and fuel is stoichiometrically balanced and all the injected fuel will, in theory, be burned. If $\lambda>1$ then the mixture is lean and the air mass is greater than theoretically required. Turbocharged CI engines have lambda values between 1.15 and 2 at full load and in idle mode the value can be higher than 10 [14].

2.2.2 Common rail system

The fuel injection system is responsible for injecting the right amount of fuel at the right time, at the right pressure and with the correct atomisation profile. Early injection systems used in-line fuel injection and distributor injection pumps, which both control the injection event mechanically. These pumps are cheap but have limitation in injection pressures and accuracy of the injected time and amount. Nowadays, common rail systems or unit injectors using electronic control are widely used. The unit injector has the pumping element integrated in the injector and thus higher pressures and better mixture preparation can be achieved. Common rail systems can operate with even higher pressures and also the rail pressure can be controlled and modified very easily and quickly. Also, the rate and number of

injections as well as its timing can be controlled very precisely [14]. The common rail works as a pressure accumulator and decouples the pressure generation (high pressure pump) from the injection process (injector).



Figure 2-1: Schematic diagram of a common rail system [17].

Figure 2-1 shows the main components of a common rail system which are a high pressure pump, fuel rail with a pressure sensor, electronically controlled injectors and an engine control unit (ECU). The injection pump is usually a radial piston pump and with the current technologies rail pressures of up to 2700 bar are possible and required for Euro VI [18]. The control of pressure depends largely on the system. Some rails are equipped with a pressure control valve on the high pressure side and the excess fuel of the pump flows back to the low pressure circuit. This allows fast pressure changes such as in an event of a load change. This method is also called first generation common rail. In the second generation rail system the rail pressure is controlled by the suction side. The metering unit attached to the pump ensures that exactly the right amount of fuel is pumped to the rail to maintain the pressure. This reduces the pumping work and a pressure relief valve releases excess pressure in a fault situation or rapid load reductions. Another way of controlling the system is the two-actuator system where both former ideas are combined. The rail is equipped with a pressure control valve at the rail and a metering unit at the pump and thus uses the advantages of very accurate high pressure side control and suction side fuel delivery control.

With any common rail injection system the injection parameters such as injection timing, injection pressure and injection strategies (multiple injection events) can be adjusted very easily and flexibly [19].

2.2.3 Injectors

The injectors are connected to the rail by a high pressure pipe and are mounted either vertically or inclined in the combustion chamber and sealed with a copper washer. The start of injection and the injected fuel quantity are controlled electrically by the ECU. Presently either solenoid or piezo driven injectors are used in new-built vehicles.

Solenoid driven injectors are used in first and second generation common rail whereas piezo-injectors are used in third generation rails systems [20-23].

The operating concept of a solenoid injector is based on the acting forces being in balance.



Figure 2-2: Functional schematic of a solenoid-valve injector [14].

When the engine is running and the fuel pump operating the pressure inside the valve-control chamber and chamber volume increases (Figure 2-2) but since both pressures are equal and the resulting forces are balanced, the injector remains closed. To open the valve, the injector is triggered with a so called pickup current which will allow the solenoid valve to open rapidly. After a short period of time the current is reduced to a holding current. The outlet restrictor opens and the fuel in the valve-

control chamber flows back through the fuel return pipe. The inlet restrictor in Figure 2-2 prevents the sudden pressure compensation and therefore the pressure in the valve-control chamber drops, which leads to an unbalanced force on the needle and fuel injection commences. The flow rates through the inlet and outlet restrictors determine the speed of response of the injector. The speed of the needle movement is determined by the flow rate through the inlet and outlet restrictor. When the valve plunger reaches its upper stop the injector is fully open and the injection pressure is similar to the rail pressure and the quantity of fuel is proportional to the time that the valve is open. The valve shuts as soon as the trigger signal stops and the holdingcurrent drops to zero. At this point the spring presses the needle back into the seat and the outlet restrictor closes. The major drawbacks of solenoid injectors are the relatively high electric power required [24] and therefore higher fuel consumption and a low dynamic response [25]. A faster dynamic response can be achieved by optimising some magnetic and electric circuit parameters as tested by Oki et al. [26]. Also, a higher voltage of up to 80 V can be applied to obtain a sharper current rise which minimises the valve opening delay [27]. Another way to overcome the drawbacks of solenoid-driven injectors is solved with piezo-actuated injectors, where the converse piezoelectric effect is used [28].



Figure 2-3: Functional schematic of a piezo driven injector [14].

In 2000 the first vehicles with piezoelectric injectors left the production plants with the opportunity of up to seven split injections. Similar to the solenoid driven injector the pressures in the piezoelectric injectors are also in balance. From the rail fuel flows directly to the bottom part of the nozzle, where a constant force is exerted on it. The upper part of the nozzle is pressurised by the fuel which is fed via two restrictors into a control chamber which is closed by a servo valve. Since the piston area of the upper part is larger than the lower part, the forces are acting downwards and pushing the needle into the seat. An electric voltage on the piezoelectric actuator is required to trigger the injector and open a servo valve. Now the fuel inside the control chamber can escape and the pressure in the control chamber drops instantaneously, but the pressure around the needle keeps constant. Consequently, the resulting force pushes the needle out of its seat and fuel is injected. To stop the injection the voltage is switched off and the piezo actuator contracts. The servo valve can move back and the pressure through the two throttles builds up again. As soon as the pressure reaches the rail pressure the spring and the resulting force pushes the needle back into its seat and the nozzle outlets are closed. In the same manner than the solenoid injectors the speed of the nozzle movement is determined by the two restrictor diameters. The forces inside the injector to move the servo valve can be very high and go up to 1000 N [29].

To achieve a fast movement of the piezo actuator of about 100 µs the electric field must be energised to strength of 2 kV/mm. This can be realised with voltages at about 200 V and currents smaller than 30 A [29]. Some researchers compared conventional solenoid driven injector with piezo-driven and found that piezo-driven injectors achieve higher injection flow rates, faster needle response times and have a better ability to control the injection rates [24, 28, 30]. This allows a shorter period between the injection and less fuel can be injected during pilot and post injections [31]. This leads to a lower fuel consumption and quieter combustion. An investigation into the microscopic and macroscopic fuel characteristics of piezo and solenoid driven injectors found that piezo-driven injectors have a shorter injectors [32]. Spray tip penetration is only marginally affected but the spray cone angle is larger with piezo-injectors due to better atomisation performance. A more detailed description of the electronic control of common rail systems has been summarised by Hawley et al. and Guerrassi et al. [33, 34].

2.2.4 Nozzle configuration

The fuel injection is one of the most important parts in the combustion process and will heavily influence the fuel atomisation, evaporation and mixing quality with air as the grade of air-fuel mixing has a significant impact on the emission formation during the combustion. In almost all modern diesel engines multi-hole injectors are used accommodating many holes around the axis of the nozzle to deliver the fuel amount through small orifices. Multi-hole injectors enhance the fuel evaporation rate as smaller orifice diameter lead to smaller fuel droplets which can evaporate and ignite quicker [35, 36]. It was further reported that a multi-hole spray has an equivalent penetration distance compared to a single-hole nozzle with the same overall total hole area [35, 37]. Bergstrand and Denbratt looked into the effect of nozzle orifice diameter on diesel engine combustion performance and reported that smaller orifice diameter cause smaller Sauter mean diameter (SMD) resulting in improved evaporation and mixing rate, shorter ignition delay and better fuel economy [38]. Using higher injection pressures and smaller orifice diameter is an effective way to reduce particulate matter [39]. Increasing the number of nozzle holes will increase the particle distribution due to higher likelihood of particle collision with each other [40]. Also injectors with more holes provide better air utilisation which leads to reduction in NO_x and soot at the same time [41]. A better fuel-air mixture and emission reductions can also be achieved by increasing the injection pressure [42]. Pierpont and Reitz performed a study on the effects of injection pressure and nozzle geometry in a DI engine and concluded that the nozzle angle had almost no effect at higher load [43]. Higher injection pressures leads to smaller SMD of the fuel droplets and will therefore significantly reduce the particulate emissions. However higher injection pressures also increase the spray tip penetration and can cause wall impingement resulting in higher unburned hydrocarbons. An accurate spray prediction becomes important to carefully optimise the spray penetration at various engine speeds and loads. The most important macroscopic spray characteristics are the penetration tip length and the spray cone angle. Both can be influenced by many parameters, such as injection pressure, cylinder pressure, and nozzle diameter. But also the fuel properties, such as viscosity and surface tension affect the spray characteristics of different fuels. Various different fuel spray models have been developed to accurately predict fuel injection.

The most popular diesel spray models have been established by Warkuri et al. in 1960, Dent in 1971 and Hiroyasu et al. in 1980 [44-46]. Most newly developed models to adapt modern technologies, are still using the basic principles of those models, such as Naber and Siebers, whose developed model applies the theory of Wakuri [47].

2.3 The combustion process

The combustion process in CI engines is very complex and dependent on many parameters, such as fuel injection, mixing of fuel and air, injection pressure and timing. The efficiency of combustion is related to the conversion rate of the chemical energy of the fuel into useful mechanical energy. The speed of heat release depends on in-cylinder pressure and the type of combustion occurring. A very typical in-cylinder pressure and heat release profile can be seen in Figure 2-4 and Figure 2-5 and the combustion can be divided into four main steps.



Figure 2-4: Hypothetical pressure diagram for a compression ignition engine [13].

A-B: Ignition delay. The ignition delay is the period between the start of injection (SOI) and the start of combustion (SOC). During this time the spray breaks into smaller droplets and mixes with the air until the self-ignition point is reached. The duration of this period is very important for the quality of the combustion and is dependent on air temperature, size of injection droplets and the fuel-air mix properties [48].

B-C: Premixed combustion. In this phase, the air-fuel mixture which has been mixed during the ignition delay will be burned rapidly. This causes a very sharp rise of the cylinder pressure, but the duration is rather short. Obviously, the longer the ignition delay period lasts the stronger the premixed combustion will be. A short ignition delay period is desired since lower peak pressures will decrease noise level and reduce damage to engine components.

C-D: Mixing-controlled combustion. This section is the major type of combustion in CI engines and takes much longer than the premixed phase. However, the speed of combustion is slower which is indicated in a smaller in-cylinder pressure and heat release rate. Mixing-controlled combustion is determined by the mixing rate of the new injected fuel and surrounding air. Several steps are involved, such as liquid fuel atomisation, vaporisation, mixing of fuel vapour and pre-flame chemical reaction. The slowest of all these steps, fuel vapour air mixing, determines the overall speed of this combustion process [15].

D-E: Late combustion. The heat release endures while the cylinder volume expands and the last unburned fragments of fuel or soot particles are burnt. The rate of combustion becomes slower since the temperature drops as the expansion stroke commences.



Figure 2-5: Typical DI engine heat-release-rate diagram [15].

To achieve maximum power the peak pressure should occur at approximately 10-20 crank angle degree (CAD) after top dead centre (aTDC) [13]. This requires a start of injection at about 8-10 CAD before TDC [15]. However, the exact injection timing is very much dependent on the engine load, fuel type and compression ratio.

Researcher and engine manufacturer strive to find the best possible injection parameters to increase combustion efficiency and reduce emissions to the lowest potential limit.

2.3.1 Derivation of heat release rate

For combustion analysis the in-cylinder pressure is a very valuable parameter as much information, such as ignition time delay and heat release rate, can be derived [49]. The heat release profile offers an insight into the start of combustion, ignition delay, as well as the burning characteristics of different fuels in the premixed and diffusion phases [50]. The in-cylinder pressure can be measured with a piezoelectric pressure transducer. The transducer contains a quartz crystal and through a diaphragm it is exposed to the pressure. A change in pressure leads to a change in the compression of the crystal and an electric charge is induced. An attached charge amplifier is used to magnify the output voltage. To reference the pressure, crank angle degree instead of time is used to give accurate information about the combustion timing. The cylinder pressure changes as a result of piston movement, combustion, heat transfer through the chamber walls, air intake and exhaust gas flow as well as piston ring leakage (crevice). Piston movement and combustion are of course the main effects, but by using a "heat release" approach based on the first law of thermodynamics the heat transfer and leakage effects can be included. This helps to relate the chemical energy of the fuel released during combustion to the cylinder pressure change. Figure 2-6 shows an open system boundary for the combustion chamber [51].



Figure 2-6: Single open system [52]

The equation for such an open system is:

$$\delta Q_{ch} = dU_s + \delta Q_{ht} + \delta W + \sum h_i \cdot dm_i$$
(2-3)

Where δQ_{ch} is chemical energy released by combustion, dU_s is the change of sensible energy in the piston, δW is the mechanical work done by the piston and δQ_{ht} the heat transfer to the chamber walls. The last term is the sum of energy entering and leaving the system, such as the fuel.

For direct injection engines the cylinder is a single open system. When the intake and exhaust valves are closed, the only mass flows across the system boundary are the fuel and crevice flow. However, the crevice flow has been neglected for the derivation of the heat release rate. Thus the net release rate becomes the difference between the chemical energy (heat released by combustion) and the heat transfer from the system, which is equal to the work done by the piston plus the rate of change in sensible internal energy of the cylinder contents.

$$\frac{dQ_n}{d\theta} = \frac{dQ_{ch}}{d\theta} - \frac{dQ_{ht}}{d\theta} = p \cdot \frac{dV}{d\theta} + \frac{dU_s}{d\theta}$$
(2-4)

The sensible internal energy of the cylinder content can be further treated as an ideal gas and applying the logarithmic laws and substituting some ideal gas parameters the net heat release rate can be written as:

$$\frac{dQ_n}{d\theta} = \left(\frac{\gamma}{\gamma - 1}\right) \cdot p \cdot \frac{dV}{d\theta} + \frac{1}{\gamma - 1} \cdot V \cdot \frac{dp}{d\theta}$$
(2-5)

The detailed derivation of the heat release rate can be found in the book of Heywood [15]. The appropriate values for γ during combustion are not well defined, but in diesel heat-release analysis γ is usually a constant value between 1.3 and 1.35 [53]. Now the only two variables are the volume and pressure change per CAD. Both can be taken from the pressure transducer and the crank angle encoder.

For many engineering applications equation (2-5) is sufficient for combustion analysis. However, some more sophisticated methods have been introduced including crevice and leakage effects.

$$\frac{dQ_{ch}}{d\theta} = \left(\frac{\gamma}{\gamma - 1}\right) \cdot p \cdot \frac{dV}{d\theta} + \frac{1}{\gamma - 1} \cdot V \cdot \frac{dp}{d\theta} + V_{cr} \\
\cdot \left[\frac{T'}{T_W} + \frac{T}{T_W \cdot (\gamma - 1)} + \frac{1}{b \cdot T_W} \cdot \ln\left(\frac{\gamma - 1}{\gamma' - 1}\right)\right] \quad (2-6) \\
\cdot \frac{dp}{d\theta} + \frac{dQ_{ht}}{d\theta}$$

Figure 2-7 shows an example of what the heat release profile might look like. The lowest curve shows the net heat release rate versus crank angle. The addition of the heat transfer and crevice effect gives the chemical heat release, whereas the proportion of the crevice flow is very small. The curve at the top of the diagram is the fuel within the combustion chamber times its lower heating value.



Figure 2-7: Heat release analysis showing the effects of heat transfer, crevices , and combustion inefficiency [54].

2.4 Diesel engine emissions

An internal combustion process is very complex. Mineral diesel is a fractional distillate of crude oil and thus consists of many different hydrocarbons with diverse bonds, chain lengths and geometries. During combustion the bonds break and new, smaller molecules are formed and heat is released. Consequently many side reactions

take place and some undesirable by-products occur, which are called emissions. The main emissions are nitrogen oxides (NO_x), particulate matter (PM), unburned total hydrocarbons (THC) and carbon monoxide (CO) and have to be lower than specific limits set by the European Commission. In CI engines the NO_x and PM emissions are usually higher than in SI engines, whereas the CO and THC emissions are slightly lower. In California, for example, only 12% of the total fuel is used by heavy-duty diesel trucks but they are responsible for more than 50% of the NO_x and PM emissions [55].

2.4.1 Particulate matter

Particulate matter is the only emission which is not chemically but physically defined. Particles are tiny solid or liquid particles and consist mainly of carbon, but also inorganic elements, such as metal, nitrogen, hydrogen and sulphates.

The weight of the PM in diesel engine is 10-100 times higher than in gasoline engines, but in terms of numbers both should be considered. Particulates are now strictly regulated in most countries and are usually classified as g/km in light-duty vehicles and g/kWh in heavy-duty vehicles. The mass emitted by vehicles has been decreased since the 1970s, by more than two orders of the magnitude due to improved combustion and sophisticated aftertreatment systems.



Figure 2-8: Typical particle size and distribution from combustion engines [56].
As illustrated in Figure 2-8, particulate matter can be categorised by three main types according to their average diameter. The size distribution is lognormal and the smallest particles are called 'Nuclei mode' and the diameter is usually smaller than 50 nm. Those small particles are often also called nanoparticles. The next larger type is the 'Accumulation mode' with particles in the range of 50 nm to 1000 nm which are classified as fine or ultrafine particles. The largest particles have diameters of more than 1 µm (microns) and form the coarse mode. The solid line in Figure 2-8 illustrates the concentration distribution whereas the dotted line represents the total mass contribution. Even though the particle concentration in the nuclei mode is very high compared to the other modes the total mass contribution to the particulate matter emissions is quite low. Modern diesel engines emit less particle mass than older ones, however, larger numbers of PM are produced [57]. The smaller particles raised even more concerns over arising health problems. The US Environmental Protection Agency distinguishes between coarse particles PM10 with a diameter of smaller than 10 μ m and fine particles PM2.5 with a diameter of smaller than 2.5 μ m. It states that the smaller the particle size the deeper the particle can get into the lungs or even the bloodstream and can cause serious health problems such as asthma, heart attacks and decreased lung function. [58]. Mark et al. pointed out that particles with a size of 100 – 2000 nm can stay in the atmosphere for approximately 8 days [59]. Wilson and Shu discovered that the half-life of fine particles can even be up to weeks, whereas coarse particles have an approximate half-life of minutes to hours [60]. The half-life is the time at which half the particles are still present similar to the half-life of radioactive decay.

PM composition

The three modes of particles are connected to each other by mass transfer and a growing process during combustion. Growth usually occurs by agglomeration with other particles, but also through acquisition of fresh particles from the gas phase.



Figure 2-9: Some typical particles likely to be emitted by an internal combustion engine [61].

Figure 2-9 shows the connection of the three different particle modes. During combustion in fuel rich conditions the oxidation is hindered and replaced by pyrolytic reactions causing the fuel molecules to break down and form soot precursors, which undergo a nucleation step to form the small nuclei particles of less than 3 nm. The soot precursors are assumed to be mainly polycyclic aromatic hydrocarbon (PAH) with a ring structure and added alkyl groups and unsaturated hydrocarbons such as acetylene [62]. The nucleation mode particles are assumed to be in a volatile state and thus have a spherical shape. Some researchers suggest that the particles could also be solid or consist of a solid core [61]. These very first molecules consist mainly of carbon and hydrogen atoms with a carbon-hydrogen ratio of 10:1 [63].

Composition of PM

PM can be divided into volatile and non-volatile fractions. Thereafter, 5 clear subgroups can be named: carbonaceous, ash, sulphates, organics and nitrates. The organic fraction is referred to as soluble organic fraction (SOF) or volatile organic fraction (VOF) depending on whether the mass has been obtained by extraction and dissolution or by heating and volatilising. This is the most complex fraction and contains everything from aromatics to esters and alcohols. The sulphate fraction is

water-soluble and its main component is sulphuric acid, which is hygroscopic and therefore contains significant amounts of water. To achieve accurate results in measuring the amount of sulphate gravimetrically the humidity conditions must kept at the same standards. Since most fuels nowadays only contain traces of sulphur, the sulphate fraction is usually very small. The nitrate fraction is water-soluble as well and nitric acid is its main component. In the past not much attention has been given to this fraction, but this could change since good after-treatment systems and different fuel properties have been introduced to eliminate the other fractions. NO_x emissions are related to the nitrate fraction, because nitric acid can be formed by a reaction of water and NO_2 .

$$2 NO_2 + H_2 O \rightarrow HNO_2 + HNO_3 \tag{2-7}$$

$$3 HNO_2 \rightarrow HNO_3 + 2 NO + 2 H_2O \tag{2-8}$$

The reactions have been investigated and indeed a small amount of HNO_2 and HNO_3 were found in the exhaust [64]. However, how the nitric acid transfers into the particulate phase is less well understood. The carbonaceous fraction consists predominantly of carbon and is often called soot. The ash fraction contains largely inorganic elements and predominantly metals. In a typical diesel engine the particulate emissions are made up of 1-2% ash fraction, 10-90% organic fraction, 10-90% carbonaceous fraction, <5% sulphate fraction and <1% nitrate fraction for engines between idle mode and full load [61]. The exact proportions are not identifiable since no clear boundaries exist between the fractions and the analytical techniques are not able to divide them without affecting each other. For example, sulphuric acid can react with metals in the ash phase and form metal sulphates, which can be both water soluble or non-soluble. Also, a 'double determination' of the organic fraction can be found in methods used for PM and hydrocarbons [65].



Figure 2-10: Routes for the formation of particulate matter [61].

Figure 2-10 shows that the main sources of particulate matter are to a large proportion the fuel and lubricant oil, but also the material disintegration of engine components and potential impurities in the intake air.

The carbonaceous fraction forms in the engine via pyrolysis of the fuel. The organic fraction arises from fuel molecules, which escaped during combustion or soot growth. The ash has its source from heavy metals in the fuel or the disintegration of the material. The sulphate and nitrate fraction have their origin chiefly in the fuel and lubricant properties.

2.4.2 Nitrogen oxide

The formation of nitric oxide (NO) and nitrogen dioxide (NO₂) result from many different processes. The nitric oxide is the only nitrogen oxide which is produced directly during combustion, whereas the nitrogen dioxide is usually formed when the gas cools down.

Thermal or Zeldovich mechanism

Zeldovich studied the NO formation of near-stoichiometric fuel-air mixtures intensively and based on his research he derived equation (2-9) and (2-10) [66]:

$$0 + N_2 \rightleftharpoons NO + N \tag{2-9}$$

$$N + O_2 \rightleftharpoons NO + O \tag{2-10}$$

Lavoie et al. added a third reaction, called extended Zeldovic mechanism, consisting of a hydroxyl radical reaction, which contributes significantly to the nitrogen oxide formation [67].

$$N + OH \rightleftharpoons NO + H \tag{2-11}$$

The reactions take place at very high temperatures of 1800 K and higher and the kinetics of formation increase with higher temperatures due to a very high required activation energy (319 kJ/mol) [68]. The forward and reverse reaction rate constants have been experimentally measured by many researchers and were gathered by Bowman et al. [69]. Thermal NO is formed in both premixed and diffusion controlled combustion, but since the pressure is very high the flame reaction zone is extremely thin and the retention time is short. Thus, the NO formation in the post-flame front dominates and it can be concluded that the combustion and formation process are decoupled. The concentrations of O, O_2 , OH, H and N_2 are assumed to be at the equilibrium values. This results in a simplified rate of reaction written as [15]:

$$\frac{d[NO]}{dt} = \frac{6 \cdot 10^{16}}{T^{1/2}} \exp\left(\frac{-69,090}{T}\right) \cdot [O_2]_{eq}^{\frac{1}{2}} \cdot [N_2]_{eq}$$
(2-12)

It is evident that the rate of NO formation is strongly influenced by high temperatures and high oxygen concentrations.

Fenimore or prompt mechanism

Another mechanism, discovered by Charles P. Fenimore, explains the NO formation in the flame zone long before the thermal NO can be produced [70]. Since the formation is so rapid it was named "prompt NO". The mechanism is based on the reaction of hydrocarbon radicals with nitrogen to form cyano (CN) molecules and amines (NH_x). These amines and cyano compounds react further via intermediate reactions to nitrogen oxide. An example mechanism using hydrogen cyanide is given below [68]:

$$CH + N_2 \rightleftharpoons HCN + N$$
 (2-13)

$$C + N_2 \rightleftharpoons CN + N \tag{2-14}$$

$$HCN + 0 \rightleftharpoons NCO + H$$
 (2-15)

$$NCO + H \rightleftharpoons NH + CO$$
 (2-16)

$$NH + H \rightleftharpoons N + H_2 \tag{2-17}$$

$$N + OH \rightleftharpoons NO + H \tag{2-18}$$

A more detailed description about the chemistry and reaction mechanism can be found at Miller and Bowman [71, 72].

N₂O-intermediate mechanism

Melte and Pratt proposed the first intermediate mechanism from molecular nitrogen and oxygen [73]. Both gases enter the combustion mainly through the intake air and at elevated pressures, low combustion temperatures and fuel lean mixtures (Φ <0.8) the mechanism can contribute up to 90% of the NO_x formation [74]. In CI engines the proportion of the intermediate NO_x contribution is estimated to be at about 30%.

$$0 + N_2 + M \rightleftharpoons M + N_2 0 \tag{2-19}$$

$$H + N_2 0 \rightleftharpoons N0 + NH \tag{2-20}$$

$$0 + N_2 0 \rightleftharpoons N0 + N0 \tag{2-21}$$

A more detailed description of the N_2O intermediate mechanisms have been discovered and summarised by a number of researchers [75-78].

2.4.3 Unburned total hydrocarbon

The hydrocarbon emissions are rather small compared to those found in spark ignition engines due to high air-fuel ratios. The volatility of hydrocarbons is closely related to chain lengths and geometry. The smallest hydrocarbon is methane with a boiling point of -161.5 °C. The hydrocarbon emissions from diesel engines can be measured using a hot particulate filter at 190 °C and a heated flame ionization detector (FID). All hydrocarbons with a boiling point above 190 °C therefore remain in the liquid state and will be added to the particulate matter group [15].

The main sources of unburned hydrocarbons are an incomplete combustion process and to a minor extend the evaporation of lubricant oil. Three main mechanisms have been identified by which the fuel escapes the full oxidation process and the magnitude of THC emissions is mainly determined by the quality of the fuel-air mixing process.

The first mechanism is due to a lean fuel-air ratio in some areas in the combustion chamber. This occurs more likely at light loads with very low fuel-air ratios. Ignition can only occur when the fuel-air ratio is within the combustible limits. Before combustion starts and fuel is injected into the cylinder a fuel-air equivalence ratio develops and forms different combustion regions as illustrated in Figure 2-11.



Figure 2-11: Schematic diesel engine fuel spray showing equivalence ratio contours at time of ignition [15].

While injection commences the fuel-air ratio in some areas is already too lean to auto ignite or sustain a fast reaction front. The combustion flame front cannot propagate any further and fuel pyrolysis and partial oxidation takes place instead. Thus the temperature drops locally and is lower than in stoichiometric mixtures, which leads to incomplete combustion. Some of these hydrocarbons have a chance to escape cylinder without being burned. The magnitude of this depends on the ignition delay period as well the load and speed of the engine [79]. Ladommatos et al. showed that the ignition quality of the fuel, expressed through the cetane number, is related to THC emissions when start of injection or start of combustion was held constant [80]. A second mechanism is a poor mixing region leading to rich fuel-air ratios above the combustible limits, such as the central area of the spray as shown in Figure 2-11. After the mixture ignites the flame front propagates through the combustible mixtures. However, towards the end of the injection event there is insufficient time to mix the residual fuel with air and parts of the fuel survive until the exhaust stroke. This

happens predominantly at high loads where injection last long into the expansion stroke. Also, especially older injectors tend to keep significant amounts of fuel in the nozzle sac volume and this fuel enters the combustion chamber very late and evaporates after combustion has terminated.

A third mechanism is flame quenching or fuel impingement on the cylinder wall. Heywood pointed out that hydrocarbon emissions are very sensitive to oil and coolant temperatures and an increase from 40-90 °C can reduce the THC emissions by 30%. Depending on the degree of fuel impingement on the cylinder wall, this can be a significant source of THC emissions. Some researchers also investigated the effect of different lubricant oils on the THC emissions. De Albuquerque et al. tested different lubrication oils in a spark-ignition engine and found that fuel is absorbed by a thin oil layer at the cylinder wall during expansion and is released during the exhaust stroke. This oil film absorption process differs depending on the oil and decreases with increasing oil viscosity. Synthetic oils produce lower hydrocarbon emission levels than semi-synthetic or mineral oils [81].

2.4.4 Carbon monoxide

Carbon monoxide is only a minor issue in diesel engine emissions and its main determent is the lack of air in certain regions within the combustion chamber. In a locally fuel rich region there is not enough oxygen for a complete oxidation of the fuel and CO is formed. However, once these CO molecules enter fuel lean regions they most likely react further to CO_2 during the combustion. Another reason for CO formation is the dissociation of carbon dioxide at very high temperature. The driving force is the Boudouard equilibrium which moves towards CO at higher temperatures and to CO_2 at lower temperatures. A third mechanism is the intermediate product formation of CO during combustion. This happens, for example, if the gases are quenched on cold surfaces and the reaction suddenly stops and further oxidation does not continue. Despite all these mechanism the overall CO formation is very low and within the latest Euro 6 legislation the limits for CO are at 0.5g/km for passenger cars and 1.5 g/kWh for heavy duty diesel engines, which can be easily achieved with the current technologies.

2.4.5 Unregulated emissions

The main focus of emissions lies in the regulated emissions described above. However, a considerable amount of unregulated pollutants are formed during combustion, which are mainly aldehydes and polyaromatic hydrocarbons (PAHs). The dominant aldehydes in internal combustion engines are formaldehyde and acetaldehyde when using diesel or biodiesel [82]. These aldehydes are usually intermediate products of hydrocarbons or oxygenated compounds in the fuel [83]. Cheung et al. investigated unregulated emissions using biodiesel and a standard Euro 5 Diesel engine and found that the aldehyde emissions decreased with increasing biodiesel concentration [84]. Some reasons are given by Guarieiro et al. and they suggested that the aldehyde content might be higher when using waste cooking oil, because aldehydes could already be formed during the frying and cooking process. Also waste cooking oil consists of many short chain chemicals, which are more likely to react to aldehydes [85]. Some researcher also suggested that aldehydes can be formed by glycerol during the transesterfication process of biodiesel [86, 87].

Aromatic hydrocarbons are often toxic, mutagenic and carcinogenic and contribute to the formation of tropospheric ozone which is clustered as a pollutant near groundlevel of the atmosphere [88].

1,3-butadiene is a carcinogen and its main pollution source are internal combustion engines [89]. Furthermore, it can be converted to genotoxic products through a photochemical reaction in the atmosphere with nitrogen oxide [90]. The main source of 1,3-butadiene are cyclohexane radicals, which break into straight chain radicals while ethylene as a co-product is formed [91].

More common unregulated pollutants are benzene, toluene, ethyl benzene and xylenes called BTEX and the main emission source is also motor vehicles from structural changes within the fuel [92, 93]. Krahl et al. identified BTEX emissions as carcinogenic, mutagenic and teratogenic [94]. Further investigation of BTEX has been intensively studied and analysed by Takada et al. [83]. They observed higher benzene emissions at lower loads and exhaust gas temperatures. The same is valid for toluene and xylene and it is explained by an easier oxidation at higher temperature. Therefore it was concluded that lower engine loads give higher BTEX emissions. Most researchers agree that PAH emissions decrease with increasing biodiesel concentration [95-101]. While Hansen et al. concluded that PAH can be

formed during combustion, the majority explained the decrease in PAH with the absence of aromatics in biodiesel [99]. Very few authors did not find reductions in PAH emissions when using biodiesel, although most of the publications have not been peer-reviewed [102-104].

2.5 Combustion optimisation

Due to very strict emission legislation engine manufacturers are constantly trying to reduce the emission limits. In Europe and some countries in Asia and South America the current legislation, Euro 5 and Euro 6, have very low limits for NO_x and PM. Euro I was introduced in 1992 and the current Euro VI legislation was introduced in 2014 with the limits for PM being tightened drastically. In addition, the usage of different biodiesel and blends could result in worse combustion performance without an optimised control strategy for these fuels. This forces researchers and engine manufacturers to quickly develop and adopt new technologies to comply with the new standards.

Over the last 20 years two major ways have been identified to reduce engine exhaust emissions. In-cylinder methods, such as (cooled) exhaust gas recirculation and improved injection strategies can be applied. The major focus in this thesis will be on different injection strategies. The alternative method of emission reduction is achieved through using aftertreatment systems, such as a diesel particulate filter (DPF), selective catalytic reduction (SCR) system, lean NO_x trap, and diesel oxidation catalyst. For current Euro V and Euro VI standards the in-cylinder and after-treatment technologies have been considers together to fulfil NO_x and PM emission regulations. In many European countries the improved fuel quality with higher cetane numbers and lower sulphur contents have also contributed to cleaner exhaust gas emissions. This is discussed further in the following section.



Figure 2-12: Trade-off between PM emissions and NO_x for two levels of swirl [105].

The major optimisation challenge is the emission trade-off between NO_x and PM. As illustrated in Figure 2-12, most combustion control technologies can only reduce NO_x while PM is increased or vice versa. Increasing the exhaust gas recirculation rate or retarding injection timing favours NO_x reduction, but leads to higher smoke numbers [106]. In contrast, higher injection pressures leads to lower soot emissions, but higher NO_x emissions [107, 108].

Multiple injection events and modification of the chamber and piston design can increase the swirl-level and combustion efficiency and shift the curve closer to the origin. However, there are natural limitations for the swirl level and the number of injections per cycle.

2.5.1 Multiple injection events

With new electronic injection technology using a common rail and solenoid or piezodriven injectors more than one injection can take place per cycle. Currently, the highest possible numbers of events are 8 injections [14]. Usually the injection is split into at least a pilot and a main injection, where about 15% of the fuel is injected in the first event [109]. Many researchers and engine manufacturers are currently investigating split injections to reduce both NO_x and PM at the same time. The main optimisation parameters are the quantity of fuel in the event and the dwell time between each injection [109]. To further understand the mechanisms of emission reductions, multidimensional computations were carried out to model different injection schemes and predict the emission levels with experimental data [106]. This reduces the amounts of experiments required and helps to find the best injection strategy for each engine. Multiple injections have a significant effect on diesel engine pollutants and shift the NO_x -PM trade-off curve closer to the origin [110-114]. In general pilot injections are believed to reduce the combustion noise [115] and shortens the ignition delay of the main combustion [116]. It further lowers the rate of heat release which inhibits NO_x formations [117].

Shundoh et al. used high pressure injection in combination with pilot injection and found that NO_x was reduced by 35% and smoke by 60-80% without any increase in the break specific fuel consumption [118]. Nehmer et al. investigated the effect of different fuel quantities in the pilot injection using a Caterpillar single cylinder engine and pointed out that the pilot injection reduces the peak pressure in the cylinder and therefore reduces the combustion temperature and NO_x emissions. [119]. It also had a positive effect on the combustion duration in the power stroke which results in lower soot formation. Tow et al. added a third injection and investigated the dwell time between the injection events [120]. They found that at higher engine loads with double injection and long dwell times, particulate matter was reduced to a third without an effect on NO_x and only a small increase in fuel economy compared to single injection. Adding a third injection straight after the end of the main injection reduced particulate by 50 % and NO_x by 30 %. The main reason for lower emissions is because less fuel is injected during the main pulse and the overall combustion temperature is reduced [121, 122]. The time between pilot and main injection is a key parameter and approximately 10 CAD seems to be ideal for double injections.

Also, Kumar et al. found that late post injections increase the exhaust gas temperature and are can be used for DPF regeneration [123].

2.5.2 Multiple injection events combined with EGR

Many researchers have shown that EGR is an effective tool to reduce the NO_x emissions, but increases soot formation emissions [124-126]. A fraction of the used gas from the exhaust re-enters the inlet system and thus reduces the amount of oxygen and nitrogen in the cylinder. The presence of carbon dioxide and water vapour increases the specific heat capacity and lowers the combustion temperature significantly. The combination of exhaust gas recirculation and multiple injections can further reduce the emissions. Zhang investigated the effect of pilot injection and EGR on soot, NO_x and combustion noise [127]. He pointed out that reducing the injection amount in the pilot event and increasing the dwell time between the injections reduces the soot emissions, while NO_x was reduced by EGR. Uchida et al. combined pilot injection with EGR and found a little advantage for the NO_x-BSFC trade-off after improving swirl ratio and combustion chamber design. However, they also found that at light load conditions the soot formation increases slightly [128]. Pierpont et al. assessed the combination of EGR and multiple injections and were able to achieve reduce both NO_x and soot emissions when using double and triple injections. The fuel consumption was to a minor extent higher than for the single injection system [107].

Reitz and Montgomery concluded that a simultaneous reduction of PM, NO_x and BSFC is possible if the number of injections, timing of injection, fuel mass ratio and EGR rate are together optimised [129, 130]

2.6 Biofuels

According to the International Energy Agency the energy consumption in the world will rise by more than 30 % from today until 2030 [131].

Also energy for transportation, which counts for 30 % of the total consumption, will grow drastically. The sectors major energy source is fossil fuel oil and it accounted for 22 % of the total CO_2 emissions in 2008 [132].

The increasing CO_2 emissions, as well as depleting reserves of fossil fuels, force governments to reduce the dependence on mineral fuels and implement policies to strengthen alternative fuels. In May 2003 the European Parliament introduced the biofuel Directive 2003/30/EC, which instruct the countries to supply 2% of the transport fuel by renewable sources in 2005, 5.75 % by 2010 and 10 % by 2020 [133].

Researchers all over the world are looking for solutions of using different kind of fuels, such as vegetable and animal oils, as well as hydrogen and alcohol from renewable sources.

For CI engines, biodiesel seems to be a good option to replace mineral diesel. The interest stems from the fact that biodiesel can be used without major engine modification in a diesel engine and also reduces some pollutant emissions. In Europe the total biodiesel production in 2004 was 1.9 million tonnes and this figure rose to 4.9 million tonnes in 2006 and 8.6 million tonnes in 2011 [134, 135]. The capacity for biodiesel production is above 23 million tonnes annually and Europe is the worldwide leader in both biodiesel production and capacity [135].

The greatest barrier for biodiesel as a commercial fuel is its high costs compared to conventional fuel. The oil feedstock is the main contributor to the cost and can be less expensive when using frying oils, soap stocks, fat and greases for example or expensive when using high-quality vegetable oils, such as rapeseed, soybean and palm oil. However, for the low cost feedstocks the amount of free fatty acid (FFA) is very high which causes problems in the transesterfication process and the cold-start properties could be unacceptable [134]. When selecting a feedstock for biodiesel production, the most important factors are the oil percentage and the yield per hectare [136].

The main feedstock for biodiesel differs between countries and continents. In Europe the major feedstock is rapeseed oil while in the US soybean oil is mainly used. Edible oils such as rapeseed, soybean, coconut and palm oil are called "first generation fuels", because they were the first oils used as a feedstock. Currently most biodiesel in the world is produced from edible oils, which caused a large "food-vs.-fuel debate" as well as concerns about the destruction of vital soil resources and deforestation [136].

Nowadays much more attention is given to fuels produced from non-edible sources such as Jatropha curcas and croton as well as recycled and waste oils such as waste vegetable oils and lignocellulose material. Often they are called second or even third generation fuels. In this thesis the use of the term first generation fuels refers to all biofuels made from edible feedstocks and second generation biofuels when made from non-edible oils. More sophisticated methods, such as gas-to-liquid, hydrotreating or fuels made of lignocellulose will be referred to as third generation fuels.

2.6.1 Biodiesel production methods

The production of biodiesel is multifarious. Most feedstocks require an oil extraction process, where the oil contained in the fruit and seed is extracted. The products of the extraction are the crude oil and the residues of the seeds, such as the kernel cake, which is still very energy-rich and can be used in gasification or anaerobic digestion plants. Before extraction the seeds are usually dried to reduce the water content. This is conventionally done by sun drying over a period of 3 weeks or in an oven at about 105 °C [136]. Three main methods have been identified for oil extraction. The most conventional method is the mechanical extraction using a mechanical press, where up to 80 % of the oil can be extracted. Each press is designed and optimised for specific seeds and the yield rate will decrease when other seeds are used in the press. Further treatment, such as filtering and degumming are required after the extraction process.

A more sophisticated process uses solvent extraction, a technique where a liquid solvent is used to remove the solid constituent. Many factors influence the yield, such as particle size, type of solvent, temperature and agitation of the solvent. Small particles are preferred because the relative surface area between solid and liquid is bigger, which will enhance the extraction time. Higher temperatures increase the rate of yield, but more energy is consumed compared to the mechanical press. This makes solvent extraction only economic at large scale production of more than 50 tonnes biodiesel per day [137].

Another very promising extraction method is enzymatic extraction, where suitable enzymes are used to crush the core of the seeds and extract the oil. This method is very environmental friendly and does not produce any volatile organic compounds. However, the extraction time is quite long and the enzymes are very sensitive against temperature and pH and very expensive to replace. For the production of biodiesel many methods are known. The main challenge for all these methods is to overcome the high viscosity, low volatility and polyunsaturated character of the oils [138].

Lit et al. conducted a comparison of four different diesel production methods and concluded that the transesterfication process is most promising, due to its high conversion efficiency and low cost [139]. The transesterfication process consists of a number of consecutive steps (see Figure 2-13). In this reaction the triglyceride (the raw oil) is converted stepwise into di-glyceride, mono-glyceride and finally glycerol. Due to the higher density of glycerol it separates from the produced biodiesel quite easily. The by-product glycerol can be used either as a fuel to be burned or as a feedstock in the cosmetic industry. For the conversion of the triglyceride an alcohol is needed. Methanol or Ethanol is used as a reactant due to their low cost. The mass ratio of oil or fat to alcohol is about ten to one and in the presence of a catalyst this can produce ten units of biodiesel and one unit of glycerol. In terms of molar mass three molecules of alcohol are required for each mole of triglyceride to produce one mole of biodiesel and one mole of glycerol [140].

The transesterfication reaction can be carried out either with or without a catalyst. A catalyst stimulates the reaction, but does not take part in it itself. Furthermore, the catalyst enhances the solubility of the alcohol and thus increases the rate of reaction.

Alkaline catalysts are NaOH, NAOCH₃, KOCH₃ and KOH, whereas sodium and potassium hydroxide are favoured in terms of the yield to cost ratio. When using alkaline catalysts the biodiesel reaction is approximately 4000 times faster compared to using acid catalysts. Furthermore, higher biodiesel yield and lower impurity levels can be achieved. The main drawback of alkaline reactions is that the FFA level is limited to a maximum of 3 %, because FFA and alcohol can form soap and water, which prohibits the formation of biodiesel. In addition, the reaction is very energy intensive, recovery of glycerol is difficult and the catalyst has to be removed from the products [141, 142].

Acid catalysts include sulphuric, hydrochloric and phosphoric acids. It is generally accepted that acid catalysts are less sensitive against FFA and water. Sometimes acid reactions are even used to reduce the FFA content and then the reaction continues in an alkaline environment. The yields with acid catalysts are very high, but the reaction speed is relatively low (3-48 h). The catalytic reaction is generally preferred to non-catalytic reactions. However, the relatively low retention times due to weak solubility and the after treatment system, including glycerol and catalyst separation, is still an issue, which prohibits the breakthrough of biodiesel mass production.



Figure 2-13: Biodiesel production mechanism [143]

2.6.2 Fuel properties of biodiesel

Since biodiesel has various production techniques and is produced from quite differently scaled plants, it was important to install a standardisation of the fuel quality to guarantee a stable engine performance [144]. The quality of biodiesel has to comply with the international biodiesel standard specifications, which include either the American Standards for Testing Materials (ASTM) or the European Union Standards for biodiesel fuel (EN 14214) [145]. The properties of biodiesel are characterised by physiochemical properties, such as calorific value (MJ/kg), cetane number, density (kg/m³), viscosity (mm²/s), cloud and pour points (°C), flash point (°C), water, sulphur and glycerine content and many more. These properties depend

on the type of feedstock and its production process [146, 147]. The following section describes some properties more in detail.

Cetane number

The cetane number (CN) is one of the most important parameters for diesel fuels to define the ignitability of each fuel. Better ignition abilities are always associated with high cetane numbers [148, 149]. It has a significant effect on the combustion performance, including noise level and exhaust gas composition [150]. The CN value based on two liquids, namely hexadecane with a CN of 100 and is heptamethylnonane with a cetane of 15 [151]. In general biodiesel have higher CN values than conventional diesel, which often results in higher NO_x emissions and lower soot concentrations [152, 153]. Values of CN which are too low, below 45, cause have poor engine combustion with excessive wear of components and poor cold-start abilities [150]. Animal fats tend to have higher cetane numbers than vegetable oils and the longer fatty-acid carbon chains and the more saturated molecules are contained in the fuel the higher its CN value [154]. The highest cetane number is believed to be from palm oil methyl ester and the lowest from grape biodiesel with cetane numbers of 61 and 48 respectively [155]. Since Euro III the cetane number of mineral diesel has to be at least 51.

Density

The fuel density is a mass measurement per volume unit and thus influences the volumetric fuel consumption. Most biodiesel have slightly higher densities than conventional diesel. The density is an important parameter, since fuel injection operates on a volumetric basis [156].

According to Gis et al. lower fuel densities lead to lower emissions of PM, THC and CO while less power is being produced [150]. No explanation for the drop in emissions has been mentioned, but a correlation between higher cetane number and lower fuel density has been found by other researchers, which can explain the lower emissions with lower fuel density [157].

Viscosity

The viscosity is a very important parameter, because it is an indication of the flow ability of the fuel. It is a measure of the internal friction, which represents the resistance occurring through the fuel flow. Therefore, it has a major effect on the fuel injection system, the spray atomisation and vaporisation. The viscosity is strongly dependent on the temperature and thus can result in large differences between cold-start and steady-state operation [136]. Some biodiesel become too viscous if the temperature is too low. Since the viscosity is such an important factor the European Union and the United States have set standards for viscosity ranges of 1.9-6.0 mm²/s and 3.5-5.0 mm²/s for ASTM D445 and EN ISO 3104, respectively [158, 159].

If the viscosity is too high the fuel does not atomise and vaporise properly and result in high THC emissions through incomplete combustion. It can also create carbon deposits on the injector, valves and combustion chamber.

Conversely, low viscosity fuels have poorer lubrication properties, which increase material wear. Also, too fast atomisation can cause a very strong pre-mix combustion phase and have a negative impact on the emissions as well. A difference in viscosity of biodiesel compared to conventional diesel fuel changes the response time of the injection, such as the needle lift timing and the amount of fuel injected might be not optimised for the applied injection strategy.

Calorific value

The calorific value of a fuel is a measure of the available energy per unit mass or heating value when a fuel is oxidised. It is one of the most important properties in terms of fuel economy and thermal efficiency, even though there is no direct link to the spray and combustion characteristics of the fuel [160]. Fuels have a higher heating value (HHV) and lower heating value (LHV). In this thesis the LHV and the CV have the same meaning and its quantity is determined by bringing all the combustion products back to the original pre-combustion temperature excluding the evaporation of the produced water molecule. The lower heating value treats water as a vapour and thus assumes that the energy to vaporise the water is not recovered. The HHV can be measured with a constant volume bomb calorimeter filled with oxygen [161]. According to Demirbas the HHV can also be estimated from some physiochemical properties, such as viscosity, density and the flash point [146].

The HHV of biodiesel is between 39-41 MJ/kg compared to diesel fuel which is approximately 45.5 MJ/kg [159].

The main reason for the difference is that diesel oil consists of approximately 86 % carbon and 14 % hydrogen, whereas biodiesel has an oxygen content of

approximately 11 % and thus lower carbon and hydrogen proportions of 77 % and 12 % respectively [150]. The higher the percentage of carbon and hydrogen in the fuel, the higher the calorific value. The calorific value of hydrogen is 120 MJ/kg, and carbon is 32.7 MJ/kg [150].

2.6.3 Biodiesel combustion

The different fuel properties of biodiesel, such as viscosity, cetane number, surface tension, density and calorific value will influence the spray and combustion characteristics and have an important effect on exhaust gas emissions.

Studies have shown that fuels with higher viscosities can lead to reduced atomisation quality of the injected fuel. Consequently, the average droplet diameter of the spray and the breakup time is increased [162, 163]. This leads to incomplete combustion and carbon deposits on the cylinder walls as larger droplets need more time and energy to evaporate and combust. To avoid these phenomena many parameters, such as injection timing (IT), injection duration, injection pressure (IP), compression ratio (CR), EGR ratio and fuel temperature can be adjusted to optimise biodiesel use. Many researchers already investigated the influence of changing these parameters in CI engines fuelled with various biofuels. It is generally accepted that combustion of biodiesel leads to a shorter ignition delay, a lower heat release rate (HRR) and loss in brake specific power [164] [165].

Combustion performance

Gumus et al. investigated the combustion performance of biodiesel in a four-stroke single cylinder CI engine. He found that pressure and heat release rates are lower compared to mineral diesel, which was explained with the higher viscosity and lower volatility of biodiesel [166]. This however is questionable and the increase in cetane number and shorter ID is a better explanation.

Further the combustion duration increases due to poorer atomisation and slow combustion of the fuel, which is affected by the higher viscosity, flash point and lower volatility. Also the longer injection duration required with biodiesel will cause longer combustion duration as the injector needle opening time reaches further into the expansion stroke. Xue et al. summarised the results of many researchers working on biodiesel in CI engines and concluded that the use of biodiesel leads to reduced engine power, but that drivers may only be able to recognise it during full load operation [10]. The main reason for this power loss is derived from the LHV and high viscosity of the fuel, due to poor fuel injection atomisation. However, other authors state that higher viscosities are beneficial, since better lubrication reduces friction losses in the engine [167]. The type of feedstock should not have a significant effect on the power loss. Many researchers concluded that the injection parameters, such as injection pressure and timing, are a crucial factor for the power output and fuel consumption of a CI engine [168-170]. Xue et al. also concluded that fuel consumption is usually increased when using biodiesel [10] and only a few researchers actually observed lower fuel consumptions [171]. Different feedstocks and different production processes also influence the amount of fuel consumed. For example, Sahoo et al. compared the BSFC of Jatropha, karanja and polanga methyl esters and reported a difference in BSFC for the different fuels [172]. The use of a turbocharger also has a significant effect on the fuel consumption, which was explained by the improvement of air-fuel mixing, higher air temperature and the increased air charge in the cylinder [173]. In addition, an optimised injection strategy around injection pressure and timing can improve the brake specific fuel consumption [168, 174]. Some researchers concluded that visual carbon deposits are lower for biodiesel due to lower soot formation [175, 176] and engine wear is reduced [177]. However, more long-term tests should be carried out to verify these statements.

To determine the combustion efficiency of biodiesel, emission production is another indicator of combustion quality. Thus, much research has been carried out to measure the main emissions PM, NO_x , THC and CO during biofuel combustion.

Combustion emissions

The majority of research work shows that the PM emissions are reduced when the engine is fuelled with biodiesel. Wu et al. tested five different biodiesel on a Cummins ISBe6 DI engine and found that PM was reduced up to 69% on average [178].

Many authors believe that the reduction of particulate matter is mainly dependent on the higher oxygen content of the fuel. However, the absence of aromatic and sulphur compounds is also responsible for lower PM emissions [179-181]. Lin et al. tested the influence of different feedstocks on soot formation and concluded that biodiesel with shorter fatty acid carbon-chain lengths are favoured [147]. A large amount of research has also focused on the engine operation conditions, such as load, speed, injection strategy, cold start, etc. Many have shown that PM emissions increase with increased engine load mainly due to reduced air-fuel ratios [180, 182, 183]. In addition, at higher engine speeds PM emissions are reduced, since higher turbulences of the air charge improves the fuel-air mixing [184, 185]. Several research articles also describe the impact of EGR on PM emissions and the fact that higher EGR ratios increase soot formation due to less oxygen available in the air [174, 186].

Most research indicates that the use of neat biodiesel increases NO_x emissions. For example an increase of 15% in NO_x emissions was observed with biodiesel at high load due to higher combustion temperatures [183]. The main reason for NO_x emissions are high temperatures and higher oxygen concentrations during combustion. The latter is higher because of the oxygen content within the fuel [180]. Some researchers believe that NO_x emissions are only higher for biodiesel in certain types of engines or engine operation conditions. Staat and Gateau compared mineral diesel and biodiesel on two different engine test cycles and concluded that NOx emission of biodiesel increase by 9.5 % with the ECE R49 test cycle and reduce by 6.5 % with the AQA F21 transient test cycle. From an extensive literature review it has been concluded that biodiesel NO_x emissions are generally higher when tests are performed on a test bench as when measured during vehicle tests [187]. Main reason is the higher engine loads imposed on test beds compared to vehicle tests. In a different study on a single cylinder DI engine the NO_x emissions at fixed speed and increasing load have been investigated [188]. It was reported that the NO_x concentration was higher at all engine loads, but the difference increased with increasing load figures. Lin et al. tested pure fish oil biodiesel at fixed torque and concluded that NO_x concentration of biodiesel is higher at all engine speeds, but the difference decreased with increasing engine speed [189]. No explanation has been given, but it is possible that at higher engine speed the effect of the ignition delay diminishes.

The CN has a strong impact on NO_x formation. While only very few researchers believe that higher CN values increase NO_x emissions, most agree that high CN lead to an earlier start of combustion and thus reduces the premix combustion phase, combustion temperatures, and NO_x formation [178, 179]. Engine operating conditions have been investigated to explore NO_x formation. Karabektas tested RME biodiesel in a natural aspirated and a turbocharged CI engine and found that the NO_x emissions were higher with turbocharging due to a higher air to fuel ratio and higher combustion temperatures [173].

McCormick et al. tested the combustion of biodiesel on a new engine equipped with common rail techniques and on an old engine and concluded that the increase in NO_x emissions were less significant at the new engine [190]. The injection strategies have an impact on the NO_x formation as well. According to Tsolakis et al. the retardation of injection timing results in reduced NO_x , but increased CO, PM and THC emissions [174]. They further studied the effect of EGR on exhaust gas composition. They found that EGR was more effective when using biodiesel and biodiesel/diesel fuel blends compared to conventional diesel fuel use. Sharma et al. varied the injection pressure at full load and reported that this leads to a significant drop of NO_x emissions [170].

According to most research studies, CO and THC emissions are reduced when using biodiesel fuels in CI engines [174, 191, 192]. Krahl et al. investigated the performance of RME compared to mineral diesel and found a 50 % reduction in CO emissions [193]. Wu et al. tested five different biodiesel fuels and reported a reduction between 4-16 % in CO and a reduction between 45-67 % in THC [178]. Only a few research studies have reported an actual increase in CO and THC when using biodiesel and explained their results with the higher viscosity, lower volatility and the poorer spray characteristics of the fuel, which leads to poorer mixing and combustion [194].

Wu et al. found further that the CN and oxygen content of the fuel has an effect on the CO and THC emissions [178]. Higher oxygen content in biodiesel causes lower THC and CO emissions due to better air-fuel mixture and more complete combustion. Higher CN also reduces the THC and CO emissions, but no explanation has been given.

Changing the injection strategy also affects the pollutant formation. Some authors reported that an advance in injection leads to earlier biodiesel combustion and lower CO and THC formation [167, 195]. Conversely, retarded injection events result in higher CO and THC biodiesel emissions [174].

Karabektas, who tested biofuels on an natural aspirated and a turbocharged engine, reported an increase in CO without turbocharging as a result of less air being available for the air-fuel mixing in the combustion chamber [173]. In addition, most studies showed that with increased engine load the CO emissions are increased, whereas the opposite trend is reported about THC resulting in lower THC emissions at higher engine load [167, 186, 192]. At higher engine load the air to fuel ratio decreases and less oxygen is available for complete combustion and more CO is formed. Also, at higher engine load the cylinder gas temperature is higher and the fuel evaporation rate is higher resulting in lower THC emissions due to unburned hydrocarbons.

2.6.4 Soybean oil methyl ester (SME)

The soybean or soya bean is a legume native to East Asia, where it was first cultivated for food production between 1700 and 1100 BC. Soybean is not just popular for its relative high oil content, but also for the high protein content of 38% [196].

The protein yields per area of land are higher than that of many other plants and the quality of the proteins is comparable to proteins from animals. Thus, soybean is often used for animal feed and as a meat and milk replacement for vegetarians and lactose intolerance sufferers. The cultivation of soybean has increased from about 17 million tonnes per year in 1960 to 265 million tonnes per year by 2010 [196, 197].

2.6.4.1 Production of SME

Soybean oil is a vegetable oil and consists of a mixture of saturated and unsaturated fatty acids with chain lengths predominantly between 16 and 22 carbon atoms. About 14 % of the fatty acids are saturated, 29 % are mono-unsaturated and about 60 % are poly-unsaturated [198]. The major constituent is linoleic acid (C18:2) with 51%, followed by the mono-saturated oleic acid (C18:1) with 29 %. The other significant proportions are palmitic acid (C16:0), stearic acid (C18:0), and linolenic acid (C18:3) with 10 %, 4 % and 7-10 %, respectively [198]. Traditionally soybeans were mostly produced for food, however, the proportion used for biodiesel production is growing rapidly. Currently soybean methyl ester (SME) is the best developed and the most produced biodiesel in the world, followed by methyl esters produced from rapeseed and palm oil. SME is produced in large scale facilities in many countries, such as the US, South America, China and India.

According to the Institute of Sustainable Resources, the crop yield of soybean plants is approximately 3 tonnes per hectare, which is equivalent to around 640 litres of biodiesel [199]. The oil content of soybeans is between 18-24 %, and lower compared to rapeseed and sunflower crops for example, but the viscosity and lower heating value (LHV) are slightly higher [200].

The production of SME is very similar to RME. After the soybeans are cleaned and dehulled, the beans are crushed and the oil is extracted. This can be done by mechanical pressing or solvent extraction, but only with the latter method high yields of oil can be achieved. The cracked soybeans are pressed into flakes and then solvent-extracted with hexane. The remaining oil content in the cake is less than 1 %. The crude oil is refined by removing phospholipids (degumming) and by eliminating the remaining free fatty acids (neutralisation). For transesterfication, which is the most common biodiesel production process, one litre soybean oil is stirred with 10.2 g potassium hydroxide and 200 ml methanol at 45 °C to produce about a litre SME and 200 ml glycerol [200].

Although soybean's oil content is rather low, SME is still advantageous to other fuels, because soybeans can be produced without any nitrogen additives. Pimentel et al. reported that nitrogen fertiliser is one of the most costly inputs in crop production [201]. The production of soy costs about \$537 per hectare. For 1 t of oil about 5,556 kg soybeans are required, which is equivalent to approximately 2 hectare and \$1,074 of production cost. Adding the costs for transesterfication, the total cost for 1000 kg biodiesel is \$1,212 and \$0.84 (£0.55/l) [201].

Haas et al. calculated the production costs of a large scale production plant (38 million litres) producing SME using a process simulation software [202]. They found that the single greatest contributor of the biodiesel price is the feedstock itself, which accounts for 88% of the total costs. Their final production costs of SME were estimated at \$0.53 per litre and thus lower compared to Pimentel [201, 202].

SME	Unit	Source					
		Α	В	С	D	E	F
Cetane number	-	47	54.1	45.7-56	48	46	51.5
Density at 15 °C	kg/l	0.87	0.883	0.88	0.884	0.889	0.881
Viscosity at 40 °C	mm ² /s	5.2	4.512	4	3.958	4.1	4.27
Net heating value	MJ/kg	38.81	37.4	37.16	37.25	37.26	37.388
Flash point	°C	166	123	-	-	-	-

 Table 2-1: Properties of SME from various sources [145, 200, 203-206].

2.6.4.2 Combustion, spray and emission characteristics of SME

Canakci and Van Gerpen tested SME in a John Deere 276T model and measured exhaust emissions and combustion parameters [205]. Results implied that NO_x and CO_2 increased by 13.1 % and 1.8 %, but CO and THC decreased by 18.2 % and 42.5 %, respectively. A lower smoke number of SN 0.41 was also observed. They further found that the start of combustion for SME was 3.1 CAD earlier than that of diesel fuel and the ignition delay of 0.751 CAD was shorter than with conventional diesel fuel. Both results were attributed to the higher cetane number for SME of 51.5 compared to diesel fuel of 42.6.

Scholl and Sorenson investigated the combustion behaviour of SME in a DI engine and concluded that many combustion parameters, such as ignition delay, peak pressure, and rate of pressure rise are close to that of diesel fuel combustion at the same engine conditions [206]. They further studied the combustion and emission characteristics of SME using different fuel injector nozzle diameters. It was found that the ignition delay of SME is more sensitive to the nozzle size than diesel fuel. In addition, CO and PM emissions are slightly lower, THC emissions are significantly lower and NO_x emissions are higher for SME compared to conventional diesel fuel.

Celikten et al. compared the performance and emissions of an engine fuelled with RME and SME for different injection pressures [207]. Tests were carried out for a four cylinder diesel engine with injection pressures of 240, 300 and 350 bar. It was found that with increased injection pressures smoke number (SN) and CO emissions were reduced, while NO_x increased. At all injection pressures, SME had the lowest engine torque and power, but the highest BSFC. However, SN and CO emissions were less compared to the RME. Some studies have looked more in depth into spray and combustion analysis. Lee et al. studied the atomisation and ignition characteristics of SME in a common rail diesel engine by using a spray visualisation system and phase Doppler particle analyser [162]. They concluded that viscosity, surface tension and CN were higher as the blend ratio of biodiesel increased. The spray tip penetration increased with higher blends due to higher viscosities and lower volatilities. Also, the ignition delay became shorter and peak pressures combustion higher.

2.6.5 Palm oil methyl ester (PME)

Palm oil methyl ester (PME) is a fatty acid methyl ester (FAME) with a very similar production process and fuel properties to other FAMEs, such as SME and RME. Palm oil (Elaeis guineensis and Elaeis oleifera) is widely cultivated in the tropical belt of Africa and South-East Asia. It is an edible oil and 90 % of the palm oil is used for cooking and the remaining 10 % for non-food consumption mainly in Malaysia, Indonesia, Singapore, Thailand and Brazil [136, 159]. The oil yields of the palm tree with 6000 *l*/ha per year is very high compared to many other oil bearing plants, such as soybean and rapeseed, and about 30-60 % of oil from the fruit of the oil palm can be derived for oil production [136]. There are significant differences between palm oil and palm kernel oil in terms of their physical and chemical properties. While palm oil mainly contains palmitic (C16:0) and oleic (C18:1) acids, palm kernel oil contains mainly lauric acid (C12:0) and has a saturation level of 89% [208].

Worldwide palm oil contributes approximately to 1 % of the biodiesel production and the total production of palm oil increased from 18.5 million tons in the year 1998 to 27.8 million tons and 51 million tons in 2003 and 2011 respectively [208, 209]. Many treat PME as an unsustainable biofuel because of tropical deforestation for large palm oil plantations [210].

2.6.5.1 Production of PME

According to Balat, palm oil consists of approximately 49 % saturated and 51 % nonsaturated fatty acids with chain length between 16 and 18 carbon atoms. The acidic components are 42.6 % palmitic (C16:0), 40.5 % oleic (C18:1), 10.1 % linoleic (C18:2) and 4.4 % stearic (C18:0) acids [158]. The saturation level is much higher compared to many other vegetable oils, which also improves the oxidation stability of the oil and may result in lower NO_x emission.

Palm oil has the potential to be cultivated with lower GHG emissions per produced mega joule than rapeseed and Jatropha due to the higher productivity per area. However, if palm plantations require a conversion from rain forest the resultant GHG emissions exceed levels from fossil fuel production. In contrast, palm trees cultivated on grassland leads to biofuel acting as a GHG sink [210].

The cost of PME in a production plant with an annual capacity of 143 million litre is approximately 0.82 \$/litre, whereas the palm oil feedstock would be 0.73 \$/litre

[211]. The capital costs for the plant, the costs for methanol as well as the distribution and blending process only accounts for \$0.06, \$0.05 and \$0.04 per litre, respectively. A marginal income can be generated with selling glycol as a by-product. In Malaysia, one of the world's largest palm oil producers, the local prices of net palm oil and PME production are \$0.39 and \$0.60 per litre, respectively, compared to \$0.26 / litre for commercial diesel [212].

PME	Unit	Source					
		Α	В	С	D	Ε	F
Cetane number	-	54.6	64	64.4	-	64.3-70	50-52
Density at 15°C	kg/l	0.864	0.878	0.8744	0.873	0.875	0.877
Viscosity at 40°C	mm2/s	4.5	-	-	4.41	4.3-4.5	4.25
Net heating value	MJ/kg	-	40.063	40.03	-	37.16	-
Flash point	°C	135	-	-	144	-	170

Table 2-2: Properties of PME from various sources [136, 145, 178, 213-215].

2.6.5.2 Combustion, spray and emission characteristics of PME

Wu et al. tested PME together with four other biodiesel in a 6 cylinder Euro III engine and measured all the main emissions at 1500 rpm with different load conditions [178]. They reported that the non-soot fraction (NSF) of PME was similar to that of diesel fuel, but all other biodiesel showed higher NSF numbers. The high CN of PME could be responsible for the large NSF difference between different methyl esters. Compared with diesel fuel, PME reduces dry soot emission by 80–82 % on average and shows the highest reduction compared to the other test fuels. The same applies for THC emissions. However, the NO_x emissions of PME increased by 10-23 % on average and were therefore higher than for other methyl esters. Results indicate that PME has a very high CN of 64 and could lead to lower NO_x values.

Ng et al. [216] carried out emission tests of PME B100, B50 and mineral diesel at different speed and load conditions. The result showed that BSFC for PME is higher than diesel fuel, which can be explained by the lower heating value of PME. They also found that with neat PME the NO_x, THC and SN decreased by 5.0 %, 26.2 % and 66.7 %, respectively, due to cleaner combustion, higher CN and better soot oxidation. They concluded that PME significantly reduces tail pipe emissions and is a viable fuel to replace fossil diesel. Karavalakis et al. [217] studied the regulated

and unregulated exhaust emissions in conjunction with fuel economy of diesel fuel and PME with B5, B20 and B40 blend ratios. The new European driving cycle (NEDC) and the Athens driving cycle (ADC) has been applied on a Euro 3 light duty vehicle using a chassis dynamometer and the results showed an increase of NO_x emissions when using biodiesel with its peak value at B20 over both cycles (13.7 % and 23.2 % over the NEDC and ADC, respectively). The increase in FAME blends caused an increase in CO emissions with the highest value at 11.78 % for B20 over NEDC and 11.62 % for B40 over ADC. The unburned hydrocarbon emissions increased with biodiesel over the NEDC, while over the ADC the increase in FAME caused a decrease with the maximum at B40 (about 26.47 %). The same trend was observed for PM emissions.

Wang et al. [213] studied the macroscopic spray characteristics of PME at injection pressures up to 3000 bar by recording injection delay, spray angle and penetration, project spray area and volume with a high speed CCD camera. The results show that biodiesel give narrower spray angles, but larger penetrations under different injection pressures, due to higher viscosity of the biodiesel. The smaller spray angle for PME also led to smaller spray area and spray volume, which in return causes poorer air entrainments and larger SMDs of the fuel.

Gao et al. [214] investigated the experimental and simulated PME spray characteristics. The macroscopic spray characteristics penetration, cone angle and spray tip speed were measured for various biodiesel blends in a constant volume vessel using a high-speed CCD camera. They concluded that the spray penetration and tip speed increases and spray angle decreases with increasing blend ratios. The results of their study are shown in Figure 2-14.



Figure 2-14: Spray characteristics of PME blends and fossil diesel [214].

They concluded that even though the surface tension and viscosity of the biodiesel are higher than those of the conventional diesel fuel, the microscopic and macroscopic spray properties of B5, B10 and B20 blended fuels were almost identical to diesel. Nevertheless, the Sauter mean diameter (SMD) of biodiesel blends distinctly increases for 20% biodiesel mix. This indicates that the characteristics of fuel atomization will differ with higher ratios of biodiesel.

2.6.6 Hydrotreated vegetable oil (HVO)

Hydrotreated vegetable oil (HVO) is a fairly modern bio-based diesel fuel from renewable sources produced by a process called hydrotreating. Any vegetable oil or animal fat can be used as a feedstock, and where currently commercial vegetable oils, such as rapeseed, soybean and palm oil are used, the future feedstocks are most likely to be used cooking oil, inedible oils or waste animal fats. Hydrotreated fuels are also called renewable diesel fuels and the term biodiesel is usually avoided since that is commonly used for fatty acid methyl esters (FAME) produced by transesterfication. HVOs are mixtures of paraffinic hydrocarbons without sulphur or aromatic contents and they are characterised by a higher CN and lower density than conventional diesel. Furthermore, no engine modification or additional service of the engine is necessary and up to 30 % HVO can be added into EN 590 fuel, and even more to ASTM D975 to still meet the fuel standards. Even neat HVO fuel has already been used for public transport, such as city buses [218].

2.6.6.1 Production of HVO

HVO can be produced from many different feedstocks without affecting the fuel quality. Currently the cheapest available feedstocks in sufficient quantities are rapeseed, sunflower and soybean oil as well as palm oil [218]. The HVO production contains hydrotreating, where hydrogen is used to saturate fats and oils with unsaturated double bonds and remove oxygen by forming water as a by-product called dehydration. Therefore, vegetable oils with higher saturation levels, such as palm oil, consume less hydrogen then highly unsaturated oils and fats. An illustration of the production process is shown in Figure 2-15.



Figure 2-15: Production process of HVO [219].

In the first step the triglyceride is hydrogenated and broken down into monoglycerides, di-glycerides and carboxylic acids. These intermediates are then formed into n- and *iso*-alkanes by either hydrodeoxygenation (with no carbon removal) or decarboxylation and decarbonylation (both removing a carbon from the initial intermediate) [219].

The by-products are water, carbon monoxide, carbon dioxide as well as naphtha, which is a group of liquid hydrocarbons to be used for heating and energy requirements. The CO and CO_2 very often react further to produce methane, another useful by-product. Reaction temperatures and pressures are between 300 and 360 °C and 50 to 180 bar pressure, respectively, in the presence of a zeolite catalyst [220].

A change in reaction temperature can influence the composition of the products to their desired state.

The first commercial scale HVO plant with a 170,000 t/year capacity was built in summer 2007 at Neste Oil's Porvoo oil refinery in Finland [218]. Two years later Neste started a second plant with the same capacity. In 2010 and 2011 two large scale plants in Rotterdam and Singapore with an annual capacity of 800,000 t have been commissioned. Currently HVO plants are integrated into oil refinery plants, but with increasing demands larger stand-alone units are expected to become competitive as well. The costs of producing HVO are in some studies stipulated to be about 50 % the transesterfication processing costs [221]. Kalnes et al., however, stated that the overall economics will depend on feedstock costs and by-product revenues [222]. Sunde et al. reported in their studies that HVO made from waste or by-products outperforms FAMEs and biomass-to-liquid fuel with respect to costs and environmental life cycle impacts [221]. However, feedstock availability and logistics are currently limiting factors and other raw materials must be used.

HVO	Unit	Source					
		Α	В	С	D	E	F
Cetane number	-	80-99	98-99	88.2	70-90	81.8	
Density at 15°C	kg/l	0.78	0.7825	0.779	0.78	0.7758	
Viscosity at 40°C	mm2/s	2.5-3.5	3-3.5	2.985	-	2.65	
Net heating value	MJ/kg	44	44	43.991	44	43.86	
Flash point	°C	-	-	-	-	-	

Table 2-3: Properties of HVO from various sources [218, 221, 223-225].

Hydrotreated vegetable oil have high lower viscosities, lower cloud points and therefore better storage and cold flow properties than FAMEs. Also the relatively high heating value and CN makes HVO a very high-quality fuel for the transportation sector. A drawback for HVO is that lubricity is poorer and additives have to be used to increase lubrication properties [226]. A summary of different HVO properties can be found in Table 2-3.

2.6.6.2 Combustion, spray and emission characteristics of HVO

Some studies have tested HVO with different engines to gain an understanding of its combustion and exhaust formation behaviour. Kuronen et al. have tested neat HVO on two heavy duty engines and two city buses and compared the results with EN 590 diesel fuel [227]. For HVO the particulate mass (PM) was reduced between 28 and 46 %, NO_x was reduced by 7 to 14 % and THC and CO emissions decreased by 0-48 % and 5 to 78 %, respectively. In a later study they used a 6 cylinder 8.4 litre DI engine at several speeds and loads and changed the crank timing of the injection by -6 to +6 CAD [218]. They found that by retarding the injection the smoke-NO_x tradeoff shifts towards higher soot and lower NO_x values. Also, retarding the injection resulted in higher BSFC, but much lower NO_x emissions. They concluded a clear reduction in NO_x and soot emissions as well as fuel consumption on a mass basis, but lower volumetric fuel consumption due to the lower fuel density. Pflaum et al. investigated emission formation of neat HVO and mineral diesel on a two litre four cylinder CI engine bench and on a chassis dynamometer test [228]. Their results revealed that HVO had the potential to reduce PM, THC and CO emissions up to 50 % as it is free of aromatic compounds. The NO_x emissions, however, only showed a slight reduction compared to fossil diesel. Ratanen et al. tested several HVO blends (5, 15, 20 and 85 vol%) and compared them with conventional diesel [229]. They pointed out that both regulated and non-regulated emissions decreased with increasing HVO ratios. However, a clear reduction of NO_x was not observed.

Very few papers related to spray characteristics have been published so far. Hulkkonen et al. compared the macroscopic spray characteristics of HVO and mineral diesel [230]. An injector with two different nozzle diameters of 0.08 and 0.12 mm in a common rail fuel system with rail pressures of 450, 1000 and 1980 bar was used. They concluded that neither the type of fuel, nor the orifice diameter had an effect on the spray penetration. They further found that the cone angle of HVO is higher than that of diesel, probably due to lower viscosity of HVO. The spray angle also increases with a larger orifice diameter, but diminishes with higher injection pressures. Finally they concluded that the macroscopic spray characteristics of HVO are similar to gas-to-liquid (GtL). The effect of neat HVO on macroscopic spray parameters in a DI engine were studied by Sugiyama et al. [231]. Their results revealed that the SMD, spray penetration and spray angle were similar for mineral diesel and HVO. Chen et al. investigated the microscopic and macroscopic spray behaviour of HVO and other biofuels and concluded that HVO has a much smaller SMD than diesel fuel and that the effect of injection pressure on spray angle was not obvious for all tested fuels [232].

2.7 Summary

In this chapter, the spray and combustion characteristics as well as the formation of emissions when using different biodiesel have been reviewed. It started with the basic operation principle of the diesel engine and the modern common rail fuel injection system with either solenoid or piezo-driven injectors. The combustion process, including premixed and diffusion combustion and their impact on exhaust emissions have been discussed.

The four main emissions are particulate matter (PM), nitrogen oxides (NO_x) , total hydrocarbons (THC) and carbon monoxide (CO) and are regulated in European standards, such as Euro 5 and 6. Especially the limits of NO_x and PM emissions are very challenging for engine manufacturer and optimised combustion is essential to keep those emissions low.

Later on different opportunities to reduce engine emissions, such as multiple injections, exhaust gas recirculation (EGR) and changes in injection timing and pressure have been reviewed.

Biofuels are alternative fuels and current European directives strongly promote their use. The focus was on the four test fuels used in this thesis, soybean methyl ester (SME), palm oil methyl ester (PME), used cooking oil (UCOME) and hydrotreated vegetable oil (HVO). All these fuels are forecasted to have the potential of being produced in very large-scale for the transportation sector all over the world.

Even though they can be used in a diesel engine, mostly without any modification, the use of those fuels with standard engine parameters might cause worse spray and combustion performances and thus increase exhaust emissions.

One of the main challenges for researchers and engine manufacturers in the future is to find flexible strategies to run engines on a wider variety of different fuels and ensuring low exhaust emissions, good drivability and low fuel consumption at any time. Many approaches are currently being investigated and more sophisticated investigations of the impact of different fuels are being analysed. Continental AG, for example, developed a fuel quality sensor (FQS) to detect the main properties of the fuel, such as CN, density and viscosity to give flexible signal into the ECU to for optimised engine operation [233]. This tool can be helpful to feed certain fuel parameters into the engine management system, if flexible control strategies are established.

In this work the spray and combustion characteristics of various biofuels are being investigated and injection strategies are being developed including the feasibility of the FQS integration. The applied methods and equipment are described in detail in Chapter 3.

Chapter 3. Methodology

3.1 Test Fuels

The test fuels used in this project were B0 reference diesel, hydrotreated vegetable oil (HVO), soybean methyl ester (SME), palm oil methyl ester (PME), and used cooking oil methyl ester (UCOME) in blends of B10, B20, B50 and B100.

The rationales for the choices of fuels and blend ratios have been discussed with the external partners Cummins and BP. B0 diesel was chosen as a reference fuel and benchmark throughout this project. Also, it is used for blending the various biofuels. SME is one of the most commonly used biodiesels in the world. It is particularly popular in North America and therefore of great interest for Cummins. PME is cultivated in (sub-) tropical conditions and its high saturation level lets PME stand out from other biodiesel sources. UCOME is a highly sustainable fuel and therefore of great interest for future engine applications. HVO is relatively new on the market, but its fuel properties are very promising and Cummins have had many customers enquiring about the use of pure HVO fuel in their HD engines. The reasons to choose B10 and B20 are interesting in terms of short-term development of the fuel market. According to the European Directive 2003/30/EC all EU countries are required to reach a share of 10 % renewable energy in the transportation sector by 2020. This figure is likely to increase in the next decades and currently the biodiesel blend ratio of commercially available diesel is between 5 - 7 %. Most engine manufacturers provide warranty for their engines with diesel of up to 20 % biodiesel (B20). Therefore, the investigation of B20 biofuel is important and worth investigating. On the other side of the spectrum some companies or bus operators are requesting to run their fleet on much higher biofuel percentages. Cummins provided the information that B50 and B100 are popular blend ratios to use and the investigation of these blend ratios is therefore recommended.

A total amount of 200 1 of each feedstock and blend ratio has been kindly supplied by BP resulting in a total fuel amount of 3,600 1 including 400 1 reference fuel and dispatched in 50 1 drums for easier manual handling. A sample of every fuel type has been taken and analysed by BP laboratory in Bochum, Germany. The main fuel parameters for all neat fuels are displayed in Table 3-1. The full fuel property analysis is attached in Appendix A.

Fuel properties	B0 Ref Diesel	HVO B100	РМЕ В100	SME B100	UCOME B100
LHV [kJ/kg]	42,853	43,902	37,320	37,230	37,200
Cetane Number	51	>75	64.9	53.7	58.4
Density [kg/m ³] at 15 °C	840.4	780.1	876.5	885.2	879.5
Viscosity [mm ² /s] at 40 °C	2.82	3.02	4.55	4.18	4.35
Aromatics [%]	27.5	0	0	0	0
Oxygen content [%]	0	0	10.6	10.6	11.0

 Table 3-1: Main fuel properties of neat test fuels.

To store the fuel according to the Oil Storage Regulations, a fuel drum store was purchased from Denios with a secondary containment of 1,000 litres. A photo of the store unit and their fuels is shown in Figure 3-1. To ensure that during the winter the fuel is stored above 0 °C two heating panels have been mounted inside the fuel storage preventing the storage temperature to drop below 4 °C.


Figure 3-1: Fuel storage unit with integrated heating elements.

3.2 Engine Test Bench

The engine, which was built and used in this research, is a four-cylinder Cummins ISBe5 heavy duty, direct injection, four-stroke diesel engine with a high pressure common rail fuel system. The common rail system has a maximum pressure of 1800 bar and was connected to solenoid injectors. Further, the engine was equipped with a waste gate turbocharger and an air to water intercooler. The main engine specifications are summarised in Figure 3-2.



Figure 3-2: Engine specification

Engine	
Power (PS)	207
Peak torque (Nm)	760
Governed speed (rpm)	2500
Displacement (L)	4.5
Bore (mm)	107
Stroke (mm)	124
Cylinders	4
Emission	Euro 5
Injection system	Common Rail
Aspiration	Turbo-charged

Dynamometer	
Operation principle	Eddy current
Max Power (kW)	230

The engine had a rated power output of 155 kW with a maximum torque level of 760 Nm between 1400 and 1800 rpm and was connected to an eddy-current W230 dynamometer to control torque and speed of the engine. The displacement volume was 4.5 L with a bore diameter of 107 mm and a stroke length of 124 mm.



Figure 3-3: Engine test bed

A photo of the test rig is shown in Figure 3-3. The engine was controlled from a separated control room for safety purposes. A picture of the control room is shown in Figure 3-4.



Figure 3-4: Engine cell control room.

The exhaust gas was connected to a Horiba MEXA-1600D gas analyser to measure NO_x , CO, CO₂, O₂ and THC emissions. In the analyser, the NO_x was measured with a chemiluminescence detector (CLD), HC with a flame ionisation detector (FID), CO/CO₂ with a non-dispersive infra-red analyser (NDIR) and O₂ with the magnetic-pneumatic method [234].

A Horiba MEXA-1000 SPCS particle counter was used to count the particle numbers in real time using Laser Based Condensation Particle Counting (CPC) [235].

To record engine operation several pressure sensors and thermocouples were attached to the test bench.

For detailed combustion analysis an AVL water-cooled, high-speed pressure transducer QC34C was mounted in cylinder 3 to record the in-cylinder pressure. An optical crank angle encoder 365C from AVL was mounted at the free end of the crank shaft to provide crank-angle based timing information for the pressure transducer. From the pressure and crank angle data the indicated mean effective pressure (IMEP), heat release profile and combustion duration were derived. The pressure sensor signal was converted from an electrical charge into a proportional voltage signal and amplified through an AVL 1-channel charge amplifier and recorded through a high-speed NI data acquisition card PCI-6251. An external fuel conditioning system was installed to cool down the fuel return flow and measured the brake specific fuel consumption (BSFC) with an integrated Coriolis flow meter.

The DaTAQ Pro control software was used to control engine speed and torque as well as record fuel consumption and engine operation conditions. A Fuel Quality Sensor (FQS) from Continental was installed in the fuel inlet line to record basic fuel properties by NIR spectroscopy. The absorbance values of the NIR are used to find correlations between different fuel types and possible re-mapping parameters. The schematic engine set up is illustrated in Figure 3-5. The results of the engine tests are presented in Chapter 4.



Figure 3-5: Engine test bench set up.

3.2.1 Engine and dynamometer control software

To achieve reliable engine testing results an engine test bed with a sophisticated control software is essential. The engine and dynamometer were connected via an ordinary prop shaft which has no significant friction and other transmission losses. The dynamometer was not just used to measure speed and torque of the engine, but also to simulate the driving environment if the engine were installed in a powertrain system. This means that all these components needed to be controlled by the dynamometer control software:

- Fuel delivery system
- Cooling and combustion air flow including temperature control for water and oil
- Throttle control and braking
- Engine electrical system
- Engine starting system

The software, which controlled most of the engine components, was DaTAQ Pro supplied by DSG. The schematic of the control system is illustrated in Figure 3-6. DaTAQ Pro supports up to 512 input channels, being a mix of low and high speed analogue inputs, such as thermocouples, pressure sensors, speed and torque inputs. Sampling rates run at up to 40 kHz per card, dependent on the type of I/O hardware.



Figure 3-6: Dynamometer control system

To monitor engine operation, thermocouples and pressure sensors mounted on the engine were connected to the transducer box. The location for the thermocouples and pressure sensors are listed in Table 3-2.

Table 3-2: Pressure and temperature sensors on test bed.					
Pressure sensors	Location	Code	Range		
Block water pressure	At engine block	PTX 1	0-2.5 bar		
Air intake pressure	Air intake manifold	PTX 2	0-5 bar		
Oil pressure	Behind oil filter	PTX 3	0-10 bar		
Crankcase pressure	At blowby hose	PTX 4	0-2.5 bar		
Exhaust restriction pressure	Exhaust manifold	PTX 5	0-5 bar		
Thermocouples	Location	Code	Range		
Water inlet temp.	Water inlet connection	T0001	0-1200 °C		
Water outlet temp.	Wateroutletconnection	T0002	0-1200 °C		
Oil temp.	Below oil pan	T0003	0-1200 °C		
Air intake temp.	At air intake manifold	T0004	0-1200 °C		
Fueltemp.	At fuel outlet hose	T0005	0-1200 °C		
Intercooler water out temp.	After intercooler water side	T0006	0-1200 °C		
Exhaust gas temp.	At exhaust manifold	T0007	0-1200 °C		
Dynamometer cooling water temp.	At dynamometer cooling outlet	T0008	0-1200 °C		
Cooled fuel return temp.	After fuel return cooler	T0009	0-1200 °C		

The test bed was operated with an electronic throttle control, whereby a voltage signal between 0 V and 5 V was send from the ECU to the throttle pedal. The control software allowed several control settings. Most commonly the throttle and dynamometer control was set to direct and a direct throttle percentage and direct dynamometer feedback was set. Most of the time, however, the throttle was set to speed-mode and the dynamometer to torque-mode. Then the PID control of the throttle and dynamometer feedback adjusted both parameters accordingly within a few seconds. In this mode, the user was able to change torque without affecting the speed and vice versa.

Furthermore the user had the opportunity to create a test plan for the engine by setting conditions, such as ramp time, settling time and step duration at specific engine conditions. Up to 100 test points can be set and then the dynamometer operates through the test plan automatically to ensure steady state conditions without major external influences. In each test step the user can data log the engine conditions over a period of time or in interval sequences and calculate the average values for each condition.

3.2.2 Test bench cooling system

For the engine test bench several separate cooling circuits were used to ensure sufficient cooling at the appropriate temperatures. All cooling circuits were closed loop systems with a 50 % glycol mixture to reduce corrosion of the parts and pipes. An external 300 kW cooling system with 6 cooling fans in the building's backyard was designed to provide enough cooling for both engine and dynamometer. Plate heat exchanger in the engine cell with 1.7 m² heat exchange area ensured that the closed loop cooling systems could transfer the produced heat to the external cooling system. A control system ensured that the cooling fans kicked in as soon as the water temperature exceeded 27 °C to provide cold water.

The cooling set up in the engine cell is illustrated in Figure 3-7.



Figure 3-7: Engine and dynamometer cooling arrangement.

For the engine cooling the engine water pump was used to pump the water through the loop. A self-controlled 3-port valve controlled the water inlet temperature to 40 °C by mixing warm water from the water outlet into the cold inlet stream. The water outlet temperature was set to 80 °C by the thermostat on the engine. The volume flow of the cooling water was therefore dependent on the engine load and increased with higher engine power output. Another closed loop cooling circuit was used for the dynamometer and intake air cooling. Instead of having an engine water pump a conventional centrifugal pump PENTAX CM90 was used to force the water through dynamometer and intercooler in parallel arrangement. The water inlet temperature for both devices had to be as cold as possible – depending on the outside temperature. The volume flow of the intercooler was controlled manually according to the air outlet temperature, which was aimed to be around 50 °C, by a small cock valve after the water pump. The remaining water flowed through the dynamometer to provide as much cooling as possible. The dynamometer had no specific cooling inlet temperature requirements, as long as the outlet temperature did not exceed 65 °C. An automatic engine shut down was in place for when the water temperature exceeded 70 °C. The water pressure should not be higher than 0.5 bar to prevent any damages and an analogue pressure gauge was fitted to the system to monitor any pressure irregularities. The specifications of the pump are given in Table 3-3.

Table	3-3:	Specifica	ations of	f external	water	pump.
						F

Specification CM90 water pump				
Power (kW)	0.75	Flow Rate (I/min)	90 – 20	
Inlet/Outlet Sizes (mm)	25	Delivery Head (m)	1 – 22	

The actual flow rate of the water pump was around 70 l/min. The water pump was electronically connected to the dynamometer control software via the 'System

Annunciator' to ensure that the pump had to run in order to start the engine. Furthermore, if the pump stopped during engine operation it would cause the engine to stop immediately to prevent any damages to the system.

To cool down the intake air a finned air-to-water heat exchanger made of aluminium was installed. The air in- and outlet connections were 3 inches and therefore had the same size as the compressor outlet and the intake manifold.



Figure 3-8: Intercooler drawing

3.2.3 Test bench fuel system

Since different biodiesels were tested on the engine the fuel system was a very important part of the setup and a detailed procedure for switching fuel was necessary to ensure that the system was drained sufficiently and no contamination of different fuels took place.

Also, the fuel consumption was a crucial parameter to assess the performance of the engine. For this reason a new fuel metering system from DSG was purchased to measure the fuel consumption using a Coriolis mass flow meter.

The schematic diagram of the fuel system is illustrated in Figure 3-9.



Figure 3-9: Engine fuel system arrangement.

A 40 1 tank on top of the fuel metering unit provided sufficient fuel for a complete test sequence. This tank was mounted to be the highest point in the fuel system and the fuel was pumped into the tank using a drum hand pump. From there the fuel was fed by gravity into the fuel metering unit, where it passed through a Coriolis fuel meter to measure the fuel mass flow. The tank and fuel unit including all fittings were made of stainless steel to comply with good practise biodiesel handling. The fuel hoses from the fuel panel to the engine and return were made of a special rubber which was resistant to B100 fuel. Two fuel drains were added to the fuel inlet and return stream to be able to drain the fuel system completely before a new fuel was being used for the system. A fuel filter for each fuel type was used to reduce the degree of contamination in the system.

A procedure for changing fuels without any fuel waste was established. After the tests for a specific fuel are finished the following procedure was carried out:

• Turn off engine and open the drain valves of the inlet (V_5) and return (V_6) stream as well as on the panel V_4 to drain the tank and the fuel pipes into an

empty fuel drum. Also open ventilation valves V_2 and V_3 to ventilate panel. Unscrew the fuel filter and drain. If a new feedstock is used take a new filter, otherwise filter can be re-used. If B50 or B100 fuel is used take a "waste filter" and flush fuel system with red diesel.

- Fill drained fuel in an empty container and label "used XYZ fuel" properly.
- Close the drain values (V_{4-6}) and fill the next fuel into the tank. If new feedstock is used leave return drain value V_6 open.
- Warm up the engine for at least 20 minutes at 1600 rpm and 50 Nm and ensure ventilation valves V_2 and V_3 are open to allow air to exit the fuel system. If new feedstock is used the return flow is collected in a separate bucket. The fuel return from the injectors and common rail is designed to flow straight back to the engine and cannot contaminate the new fuel in the day tank. After 20 minutes the fuel pipes should be free from any contamination. The Fuel Quality Sensor can be used to double check that no contamination happened.
- Close ventilation valves V₂ and V₃ and also close return drain valve V₆ if new feedstock has been used.

Fuel quality sensor

Continental AG developed a sensor to measure basic fuel properties, such as viscosity, cetane number, lower heating value, bio, and aromatics percentage of diesel fuels based on NIR spectroscopy. Their latest prototype named A4 was provided for this project, which used optical absorption in the near infrared spectral range for 76 different wavelengths. As illustrated in Figure 3-9, the sensor sat in the fuel inlet pipe with a maximum flow rate of 100 *V*h and 10 bar pressure. Two txt.-files were generated after measurement. The first file called the prediction file contained the user-friendly outputs, such as date, time, cetane number, lower heating value, etc. and the second file called spectra file contained the wavelengths and the corresponding raw absorbance and temperature corrected absorbance values. The sensor was connected to a 12 V power supply (car battery) and to an interface box and from there via USB to the computer.

The FQS A4 sensor user interface is shown in Figure 3-10.



Figure 3-10: Continental Fuel Quality Sensor (FQS)

3.2.4 Engine exhaust system

The Cummins engine was fitted with a 'wastegate' turbocharger behind the exhaust manifold. The wastegate is a simple turbine bypass valve to adjust the turbine speed, and thus the compressor speed, by diverting a portion of the exhaust gas around, instead of through the turbine. This limits the amount of power delivered to the compressor and thus limits the pressure boost ratio of the compressor as well. The wastegate actuator was used to set the wastegate according to the ECU settings. Across the turbine the exhaust temperature dropped significantly from around 700 °C to less than 350 °C. A DAF silencer for the LF45 truck was added to the exhaust system to reduce the noise level emitted from the engine. For exhaust gas measurements two 6 mm ports were added to the exhaust system to connect several exhaust analysers, which are explained in detail below. The main measured emissions were THC, CO, NO_x and the particle number.

Horiba MEXA-1600D Gas analyser

The MEXA-1600D is an exhaust gas measuring system for all kind of diesel and petrol engines, in which the gas is directly sampled out of the main stream and the concentration continuously measured. The collected sample was heated to 190 °C before measuring to fulfil emission standards and also to reduce any adsorption and desorption effects due to high boiling HC. Further the air-fuel ratio can be calculated and EGR systems can be measured as well. The gas analyser consisted of different modules to measure different gas components:

AIA-260: CO/CO₂ Analyser:

The AIA-260 analyser was designed to measure the CO and CO_2 concentration in the exhaust gas via non-dispersive infrared (NDIR) method. A non-heated sample of the gas was dried in the sample handling system before it flows into the analyser. Different gas molecules with different atoms absorbed infrared energy at certain wavelengths and the grade of absorption was proportionate to the gas concentration at constant pressure. The relationship between the degree of absorption and sample gas concentration is given by the following absorption equation:

$$a(\lambda) = -\log\left[\frac{I(\lambda)}{I_0(\lambda)}\right] = \epsilon(\lambda) \cdot C \cdot L$$
(3-1)

Where, λ is wavelength,

 $\epsilon(\lambda)$ is molar absorption coefficient in λ of the gas,

C is the concentration of component,

L is sample thickness (path length of sample cell radiation.

Since the water concentration in the exhaust stream was high and water is absorbent and interferes easily, the sample was dehumidified before it entered the NDIR analyser.



Figure 3-11: Example of an absorption spectrum using NDIR - here for water [236].

FCA-266: THC/NO_x Analyser

The FCA-266 was designed to measure THC and NO_x concentration in the engine exhaust gas via flame ionisation detection FID for the organic HC and chemiluminiscence for the inorganic NO_x molecules. FID uses a hot H₂ flame to generate ions by the heat energy, which are proportional to the amount of carbon atoms in the sample. This method is widely used in the industry to measure organic emissions because of its high sensitivity to almost all HC compounds. The sample gas was mixed with the H₂ and burned with oxygen in a burner. The burned gas together with the formed ions left the burner and passed two electrodes with DC voltage supplied. The ions migrated to the electrodes and were detected as current and the detector output was nearly proportional to the number of carbon atoms. However, a selectivity of different HC compounds was not possible. The ionisation reaction within the flame is illustrated by the following formula:

$$CH^* + O^* \to CHO^+ + e^-$$

$$X^* = radical$$
(3-2)

For the concentration of NO and NO_x , a chemiluminescence detector (CLD) was used, which is a common method because of its sensitivity to NO and non-interference with other components. The CLD principle takes ozone gas to produce

 NO_2 in a reactor by oxidizing the NO. A part of the produced NO has a higher energy level than usual and in the next moment this excited molecules release this excited energy as light when returning to the ground state. This phenomenon is called chemiluminescence and the degree of emitted light is proportional to the NO concentration is the sample gas. Thus, the light emission intensity has to be measured.

$$NO + O_3 \rightarrow NO_2^* + O_2$$

$$NO_2^* \rightarrow NO_2 + hv$$
(3-3)

The light emission of NO_2 can be influenced by co-existing gas molecules colliding with the excited NO_2 molecules. Then, the energy is not released in form of light, but the collision absorbs the addition energy.

$$NO_2^* + M \to NO_2 + M \tag{3-4}$$

The probability of energy loss in general depends mainly on the concentration and size of the co-existing molecules (CO₂ and H₂O are quite large compared to N₂ or O₂). To significantly reduce the collision probability the reactor is maintained in a vacuum state. Present NO₂ in the raw sample cannot be measured by CLD since no energy excitement will take place. Therefore, a NO_x converter is used to convert all NO₂ to NO before measurement.

$$NO_2 + C \rightarrow NO + CO$$

$$2NO_2 + C \rightarrow NO + CO_2$$
(3-5)

IMA-261: O2 Analyser

The third analyser was designed to measure the O_2 concentration in the exhaust stream with the magnetic-pneumatic method. This method was built on the fact that O_2 is affected stronger to a magnetic field than other gases due to its paramagnetic characteristics. However, NO and NO₂ may interfere this analysis, since both gases have similar magnetic properties. When AC current flows in an electromagnet an alternating magnetic field occurs between both poles. When the gas sample streams through this field the pressure around the poles changes according to the oxygen concentration since this paramagnetic molecule is highly attracted by the changes in the magnetic field. The pressure rise is shown by the following formula:

$$p = \frac{1}{2} H^2 \cdot x \cdot C \tag{3-6}$$

- *H*: Magnetic field strength
- *x*: Magnetic susceptibility of paramagnetic susceptible gas (O₂)
- C: Concentration of paramagnetic susceptible gas (O_2)

The pressure change can be detected by a capacitor generating an alternating signal with alternating capacity.

Horiba MEXA-1000SPCS Particle Counter

The MEXA-1000SPCS is a real time solid particle counter within a specific range for engine exhaust gases according to the particle count measurement standards. It consists of a volatile particle remover (primary diluter PND1), a vaporiser (secondary diluter PND2) and a detector (condensation particle count CPC) [237]. It is recommended that the volatile particles are removed and only the solid particulates are measured, because the occurrence count of Nucleation mode particles due to concentration of volatile substances such as SOF and sulphates is thought to be affected greatly by the dilution conditions, which makes it difficult to achieve reproducible measurements. A sample of the exhaust gas was collected from the dilution tunnel first and coarse mode particles were removed through a cyclone separator with a sharp cut point of 2.4 µm to 10 µm. In the next section the sample gas was diluted with the heated diluter (PND1), where the air was heated up to 150 °C or higher and no more volatile particles should be produced. Then the sample was heated up to 300-400 °C in order to vaporise all volatile molecules. In the cooling diluter (PND2) the gas concentration was reduced by cooling down the gas to prevent formation of particles by re-condensation. Lastly, the sample was injected into the CNC (condensation particle counter). There, butanol vapour in a supersaturated state helped the particles to form cores and grow to larger particles, which then were counted by laser light. A detector measured all particles which were bigger than 23 nm. The range of the particle counter was limited by the cyclone and the detector, which was here 23 nm to 10 µm.

3.2.5 Engine data acquisition system for combustion analysis

In-cylinder pressure based combustion analysis is a common tool in modern engine research since this method is reliable, robust and fairly cheap. The core part of this method is a piezoelectric pressure transducer, which can measure the pressure in very fast frequencies and also very accurately over long periods of time. Quartz has a very good piezoelectric behaviour and the electrical charge is exactly proportional to the force acting on the crystal. Apart from the piezo pressure transducer the combustion analysis method consists of four more parts, which are a charge amplifier, a data acquisition device, a crank angle encoder and an indicating device.

In this research an AVL high speed piezoelectric pressure transducer QC34C was used. The sensor was water-cooled to ensure long lifetimes and had excellent thermodynamic behaviour. The measuring range went up to 250 bar with a natural frequency of up to 69 kHz. The sensor had a good sensitivity of 19 pC/bar, but also a robust shock resistance of 2000 g maximum. The small voltage output signal had to be magnified and conditioned before it was used. This was done by the AVL 2P2E Channel amplifier, which also compensated dynamic drifts of the sensor signal. The output signal of the amplifier was -10 V to 10 V on a BNC socket. To reference the pressure signal an external clock in terms of a crank angle encoder was used.

In engine research it is common practice to use an angle based reference instead of a time reference, since changes in engine speed will not be affected by the angle location. The AVL 365C crank angle encoder was used in this project, which provided high precision digital signals. The encoder consisted of a disk marked with slots to use the reflection of light as an optical measuring principle.

This is a very common used system in engine indicating technology due to its high precision in various operating conditions. The angle mark resolution was 0.5 degree crank angle. There was one track on the marker disks with 720 pulses for the angle information which included trigger pulse information per revolution too for synchronisation purposes. The encoder were able to measure speeds up to 20,000 rpm in a mechanically stressed environment, such as torsion and vibration. The amplifier voltage signal and the crank angle digital signals were connected to a high speed National Instrument DAQ card PCI-6251. The card offered up to 16 analogue inputs with 1.25 MS/s for 1 channel or 1 MS/s with multi-channel on a 16-bit resolution. Also 24 digital input/output channels could be used at 10 MHz. Since

every engine cycle had two TDC signals, the camshaft signal was used to determine the TDC before the air intake stroke.

A LabVIEW VI was programmed by the author and used for the DAQ indication, which acquired the analogue signal from the charge amplifier and the digital signals in a clock counter. The number of test cycles were set to 100 cycles and a path was chosen for data logging. The VI also acquired another analogue input signal for the intake manifold to compensate the sensor drift of the piezo pressure transducer. A screenshot of the LabVIEW VI is shown in Figure 3-12.



Figure 3-12: LabVIEW Front End for in-cylinder pressure acquisition.

The VI generated a .txt output file with 144,000 voltage readings (assuming 100 cycles), which were fed into a MATLAB program to calculate IMEP, heat release rate, mass fraction burned and fuel efficiency. The program consisted of a graphical user interface (GUI) where the .txt file were loaded into together with other engine parameters, such as speed, torque, fuel consumption and cycle numbers. Since different fuels were used in this work, a drop-down menu for each fuel was added to load the fuel specific properties automatically into MATLAB. At the start of the execution a specific set of voltage data was shifted to the very back according to the correct TDC position. After that the 100 cycles were averaged and smoothed using a moving average of 5 periods. The average and smoothed voltage data were converted to pressure using a fixed conversion factor generated by the charge amplifier. The

pressure difference between the cylinder pressure at BDC (before the compression stroke) and the averaged intake pressure was used to calculate and remove the pressure drift caused by the piezo sensor.

Data analysis started with calculating the indicated mean effective pressure (IMEP) for the averaged cycles, and also for every individual unsmoothed cycle to determine the COV (coefficient of variation). In the next section the specific heat and polytropic exponent were calculated for the heat release analysis based on the ideal gas equation. The basic first law heat release model was used to calculate heat release and determine start and end of combustion [15]. Furthermore with inputting engine speed, torque and fuel consumption the mechanical efficiency, fuel conversion efficiency and brake specific fuel consumption (BSFC) were calculated to assess the performance of the engine. The heat release rate, the cumulative heat release and the in-cylinder pressure were plotted and exported to a jpg-file. Also the in-cylinder pressure, engine volume and heat release were written crank angle based to an Excel spreadsheet. A screenshot of the MATLAB GUI is illustrated in Figure 3-13.



Figure 3-13: MATLAB GUI for calculating heat release rate and mass fraction burned.

Drift Compensation

Since piezoelectric sensors only carry out relative measurements and not absolute pressures, it was necessary to reference the output to a known value, such as the intake manifold pressure with the cylinder pressure at BDC. Thus, during data acquisition of the piezo voltage output, the intake manifold pressure over all cycles was recorded as well. The pressure was averaged and the pressure difference between the cylinder pressure at BDC and intake manifold pressure was added onto the measured value. An example for drift compensation on a motored pressure trace is illustrated in Figure 3-14.



Figure 3-14: Example of pressure drift compensation.

TDC Offset estimation

Finding the correct TDC offset for an internal combustion engine is fundamental for correct and detailed combustion analysis. A minor error in the TDC offset can lead to large errors in IMEP and Heat Release calculations. Brunt and Emtage concluded that the IMEP error from an incorrect angle is between 3 and 10% per degree [238]. Thus to ensure an error of smaller 1% the phase angle has to be correct within 0.1 CAD. Several methods to determine the offset are known, and the most common one is using a motored cylinder pressure curve by unplugging the injector cable. The main difficulty when using a motored pressure curve for TDC identification is that the peak pressure occurs slightly before the actual TDC position due to heat transfer and mass losses. The crank angle between measured and actual TDC is named loss angle. Researchers found that the peak pressure for common IC engines appear around 0.3 - 0.5 °CA before TDC [239]. For this engine the pressure and crank angle

data have been recorded in a motored (non-fired) cycle over several engine speeds. For each test the average in-cylinder pressure over the 50 cycles and the variance have been calculated. In all tests the peak pressure is shortly before TDC when the offset was set to 101.5 °CA aTDC. The calculated IMEP is also at its minimum close to 0 bar and the log(p)-log(V) diagrams show very straight lines for the compression and expansion stroke as illustrated in Figure 3-15.



Figure 3-15: Illustration of a log(p)-log(V) diagram at motored conditions with the correct TDC offset

3.3 Constant volume vessel

A medium pressure, high temperature combustion vessel was used to investigate the spray and combustion characteristics using high speed direct photography. The vessel is made of Inconel alloy and is resistant against corrosion and oxidation and suited for extreme environments subjected to pressure of 100 bar and temperatures of up to 1000 K. The constant volume vessel (CVV) has four optical accessible windows with 100 mm viewing size and further consists of an external 4.5 kW ceramic band heater to achieve background temperatures of up to 700 °C and an external 1.4 kW chiller to cool down the windows and pressure transducer. A small 3-bladed propeller driven by an electric motor agitates the air inside the vessel and produces a swirl representing engine cylinder conditions.

The four windows 90° apart from each other offer the opportunity for sophisticated optical diagnostics, such as Mie scattering or Schlieren photography. The fuel injection system contains an air-driven high pressure fuel pump, where the fuel can be pressurised and stored in a common rail up to 1800 bar. Different injectors with different nozzle sizes can be mounted onto the vessel and can be triggered by a TTL signal generated in an external trigger box. To capture spray images a charge-coupled device (CCD) camera PHANTOM V710 "monochrome" was mounted to one of the four windows with very high sampling speeds of up to 1,500,000 frames per second (fps).

This CCD camera uses a wide aspect ratio CMOS sensor with a maximum resolution of 1280 x 800 (20 μ) pixels at up to 7,500 fps. The frame speed and total recording time could be increased by reducing the resolution. For the spray and combustion tests the resolution was set to 256 x 256 pixels at 70,000 fps and a total recording time of 2.19 seconds. The camera was triggered with the same TTL signal that triggers the injector. A Nikon AF-S Nikkor lens with a focal length of 70-200 mm and a maximum aperture of f/2.8 was attached to the camera.

A 500 W xenon lamp on the opposite window ensured constant background light for the camera. The background pressure and temperature of the vessel was monitored and controlled by a control panel supplied by the vessel manufacturer. The CVV set up including high pressure fuel system and optical diagnostic devices are illustrated in Figure 3-16.



Figure 3-16: Constant volume vessel set up.

A high speed National Instrument data acquisition card PCI-6071E was used to acquire the pressure signal from the AVL pressure transducer QC34C as well as generating TTL signals to trigger CCD camera and injector. A LabVIEW DAQmx VI was programmed to generate the TTL signal for camera and injector simultaneously and set injection duration and dwell times accordingly. The VI was also used for the injector calibration. The results are presented in Chapter 5.

3.3.1 Injector calibration

The different injectors and injector nozzles were calibrated for every single fuel to know the total injected volume and mass for each fuel at different injection pressures and injection durations. A matrix for each fuel type was established to interpolate and extrapolate to different injection conditions.

The injected mass was important for spray and combustion tests and changed with fuel properties, such as bulk modulus, density and viscosity, and injection condition, such as rail pressure and injection duration. This affected parameters as injection delay and spray cone angle, penetration distance and atomisation characteristics and was used to explain differences and similarities of various fuels. Each test fuel was calibrated at pressures between 600 and 1800 bar and injection durations between 0.6 ms and 2.2 ms.

The calibration rig was made of an aluminium modular profile system with Perspex windows as shown in Figure 3-17.



Figure 3-17: Injector calibration setup.

In the centre a steel disc was machined to hold the injector in a vertical position. A plastic measuring cylinder was placed under the disk to capture the injected fuel droplets and vapour. The injectors were connected to the high pressure common rail system via a high-pressure flexi-pipe. The injectors were electrically connected to a solenoid driver box. This driver box required a positive TTL signal to trigger the injector. This was achieved by using the NI PCI-6071E card and LabVIEW. In "Calibration-Mode" the number of samples, frequency and injection duration were chosen. For every calibration 500 injections were captured and weighted. The frequency of injection was set to 10 Hz to have a clear distinction of the individual

injections. A screenshot of the frontend LabVIEW programme is illustrated in Figure 3-18.



Figure 3-18: Injector calibration VI - LabVIEW FrontEnd.

3.3.2 Spray image processing

The images obtained from the CCD camera were saved as 16 bit .tif files on the computer. A MATLAB GUI was designed to automatically measure cone angle, penetration length and spray area of a batch of spray images. The graphical user interface (GUI) was developed to load the images into the front end and operate the program user-friendly.

Figure 3-19 illustrates the two main spray characteristics, cone angle and penetration length, measured by the programme. The "Spray Tip Penetration Length" was defined as the maximum length between the nozzle exit and the tip of the spray.



Figure 3-19: Measurement points of macroscopic spray [240].

The "Cone Angle" was defined as the angle formed by two tangential lines touching the outer boundaries of the spray on both sides and joining together at the nozzle exit. The definition for the spray angle is a bit vague and spray cone angles can vary depending on the author's criteria. Another common definition is to measure the cone angle from the nozzle exit to the spray boundary at half the penetration distance. The spray area was defined as the area covered by the fuel plume at chamber conditions. To calculate the spray area the binary image with the same threshold limit as for the penetration and cone angle was used to sum up all black pixel within the spray plume.

For the image processing one image was loaded into the program to set the processing conditions. For this the image was rotated in a position where the spray is vertical having the nozzle at the top. In the second and third stage the image was cropped to remove the edge of the visual field and converted to a binary image. After that the image was thresholded to create distinct boundaries to the spray plume ensuring the exact points for the parameter measurement to take place. To eliminate any unwanted background noise a mask was created by subtracting a blank image from the spray images with the same rotation and thresholding conditions. A sequence of the steps is shown in Figure 3-20.



Figure 3-20: Image processing steps (left to right).

The images were then further processed to obtain spray tip length, cone angle and spray area.

The furthest vertical point of the spray to the top of the image was defined as the maximum spray tip. The angle between the two imposed tangents on the outer boundaries of the spray intersecting at the nozzle exit was measured as the spray cone angle. The principle is illustrated in Figure 3-21 for all three measurements, penetration distance, surface area and cone angle.



Figure 3-21: Annotated spray measurements

3.4 Engine restoration and optimisation

When using various biofuels it is important that the engine is still able to generate the torque according to the engine certification. Higher as well as lower torque generation is not desired as the certification process needs to be repeated, which is time and cost intensive. Engine restoration is therefore required to make sure that with all tested biofuels the maximum engine power can be achieved. The main fuel property changing the engine power output is the heating value of the fuel. Restoring engine power therefore is mainly determined by the amount of fuel injected at full load. Fuels with lower heating values need to inject more fuels at full load to be able to achieve similar engine power as mineral diesel.

Biofuels often have very different fuel properties, which also affect the engine spray and combustion characteristics and consequently change the composition of the exhaust gas emissions. In a step called engine optimisation certain injection parameters were modified in order to influence the spray and combustion characteristics, such as injection pressure or injection timing.

In this project the engine was remapped for two biofuels, HVO and SME, by changing the start of the main injection and rail pressure. The results from the macroscopic spray were used to backup and support the trends found in this optimisation stage. It was aimed to remap the engine to achieve similar emission levels as with conventional diesel, while power output remained unchanged and fuel economy was improved. The results are presented in Chapter 6.

Chapter 4. Engine tests using various biofuels

Biofuel engine tests were carried out using the European Stationary Cycle (ESC), a standardised test cycle for steady-state engine operation. This cycle was used as it is widely recognised in the field of diesel engine testing and the engine parameters and results were compared to the Cummins ESC test results. The ESC is characterized by high average load factors and very high exhaust gas temperatures [11].

The engine speed shall be held up to ± 50 rpm and the specified torque shall be held within ± 15 Nm at each test point. A weighting factor for emission measurement is given as illustrated in Figure 4-1, which is an indicator of engine running times at particular speeds and engine load.



Figure 4-1: The ESC for steady-state emission measurements (adopted from [11])

The maximum rated torque of the engine was 760 Nm, but as the dynamometer could only achieve a torque limit of no more than 750 Nm, 730 Nm was selected as the maximum engine torque. The engine speed and torque values for all fuels are given in Table 4-1. The test points of the ESC are randomly distributed and arranged in a matrix containing three different speeds and four different load levels. In the following chapters the three different engine speeds, 1490 rpm, 1855 rpm and 2220 rpm, are referred to as low, medium and high engine speed, respectively. The torque

values in the Table 4-1 represent the 25 %, 50 %, 75 % and 100 % of the total load and the load terms rather than the torque values are referred to in the following chapters.

Test Point	Reference Speed	Reference Torque	Load
#	[rpm]	[Nm]	[%]
1	1490	730	100
2	1855	380	50
3	1855	565	75
4	1490	385	50
5	1490	575	75
6	1490	200	25
7	1855	730	100
8	1855	200	25
9	2220	650	100
10	2220	180	25
11	2220	495	75
12	2220	335	50

Table 4-1: ESC Test matrix for base engine tests.

The 12 test points were programmed into the DSG DaTAQ Pro software with a ramp time for each test point between 10 and 20 seconds and a holding time of 120 seconds. A screenshot of the trend file for the full test cycle using B0 reference fuel is plotted in Appendix B.

After approximately 60 seconds at the test point all main parameters were stable and the emission analyser was triggered to measure the exhaust gas composition. Also the crank angle based in-cylinder pressure and intake boost pressure were recorded over 300 consecutive engine cycles. Some pre-chosen ECM parameters, such as injection timing and rail pressure were logged via Calterm III the Cummins in-house engine management software. A test protocol was used to manually record important engine parameters and can be found in Appendix B.

At the start of a test, the engine was warmed up for 20 minutes until cooling water inlet and outlet temperatures were constant at 40 and 80 °C, respectively. A full ESC test took about 40 minutes and three repetitions were carried out per fuel type. The tests started off with B0 reference fuel and all four biofuels (HVO, SME, PME and UCOME) and the four blend ratios tested in a random order.

When the new test fuel was of the same kind but a different blend ratio the same fuel filter was used but drained and filled with the new fuel. The fuel return from the fuel injector, fuel pump and common rail flowed through a fuel return cooler and fed back straight to the engine to prevent contamination in the fuel tank. After 20 minutes of warm up the old fuel was completely burned off and the new test run was ready to go. When using a new kind of fuel a new fuel filter was used.

The main aim of the engine tests was the comparison of different biofuels and investigation of differences in power and emissions from the B0 benchmark fuel.

4.1 Reference fuel – B0 benchmark

To be able to compare engine performance and emissions of various biofuels, it is important to have a confident benchmark using mineral diesel according to the current fuel standard. B0 reference fuel - without any biofuel content - was tested to determine engine performance and exhaust emissions as the benchmark for this test rig. The chapter is divided into the three sections; engine performance, exhaust emission and combustion analysis results.

4.1.1 Engine performance

The most important parameters to assess engine performance are power, torque and fuel consumption. The fuel consumption is usually expressed as the brake specific fuel consumption (BSFC) or brake thermal efficiency (BTE), which is inverserely proportional to the BSFC. The BSFC is the ratio of required fuel over the produced brake power of the engine and a good measure of the fuel efficiency of any engine. A heavy-duty diesel engine with a good fuel economy consumes approximately 200 g/kWh using EN590 mineral diesel fuel. The brake thermal efficiency by definition is the brake power divided by the initial fuel input. The BTE is a measure for the overall combustion efficiency and completeness of combustion of the fuel. A common diesel engine has an efficiency between 30 and 40 %, which means that around one third of the fuel energy is converted into useful mechanical energy. The remaining two thirds are lost by heat dissipating in the exhaust gas or in the engine cooling circuit. A small percentage is also lost through friction losses of moving or rotating engine components. The net torque and net power over the engine speed range are good indicators to identify the optimum working range of the engine. Engine manufacturers provide wide-open-throttle (WOT) curves for customers to define the engine's full potential and its maximum produced power at various engine speeds. The full load curve of the test engine is illustrated in Figure 4-2. The highest available torque is 760 Nm at engine speeds between 1400 and 1800 rpm. The maximum power of 151 kW is generated at 2300 rpm, which is the rated speed of the engine.



Figure 4-2: Full load curve of test engine.

Figure 4-3 shows the power output and BSFC of B0 reference fuel over the full ESC cycle. The highest BSFC occurred at point #10, high engine speed and 25 % engine load (2220 rpm and 180 Nm) with a BSFC of 247.7 g/kWh. The lowest BSFC was achieved at low and medium engine speed and 75 % load (1490 rpm & 1855 rpm and 575 Nm & 565 Nm) with a BSFC of 204 g/kWh.



Figure 4-3: Power output and fuel consumption over ESC cycle.

Figure 4-4 shows that the BSFC is high at low engine loads and lowest at 75% load. Higher engine speeds result in higher BSFC, while low engine speeds in this particular engine result in low BSFC. The average fuel consumption of all 12 test points is 216.1 g/kWh.



Figure 4-4: Brake specific fuel consumption of B0.

The standard deviation of the fuel consumption was below 2 % and has therefore not been further considered due to the very accurate fuel metering system ($\Delta F=\pm 0.15$ %). To determine the cycle-to-cycle variations the coefficient of variation (COV) was calculated based on the acquisition of 300 engine cycles. The COV was less than 1 % for all test points, which shows that the engine ran smoothly. The highest COV was found at high speed and 25 % load with 0.57 % variation. Generally, low and medium engine speeds show very low COV values of below 0.4 %.

4.1.2 Regulated engine emissions

The regulated engine emissions NO_x , CO, THC and PN were measured during the ESC over a period of 120 seconds. The measuring principles are explained in section 3.2.4. In diesel engines, reducing NO_x and PM is the most challenging aspect for engine manufacturers. In the current Euro VI standard particulate mass (PM) and particulate number (PN) are regulated. In this work only PN was measured. For heavy duty diesel engines, the emissions are usually presented as specific emissions in g/kWh, but also the emission concentration in ppm is common. The concentration can be converted to the specific emissions by knowing the exhaust gas flow rate. For the NO_x emissions usually a wet to dry conversion is necessary to take into

consideration the changes in intake air temperature and humidity. In this work the conversion factor was calculated to be 0.97 on average, however, since the relative humidity has not been recorded accurately, a conversion factor of 1 was used for all tests, meaning no wet to dry conversion was carried out.

To calculate the emission mass flow rates (g/h) based on the concentration, assuming a gas density of 1.293 kg/m^3 at standard conditions, equations (4-1) to (4-3) have been used:

$$NO_x = 0.001587 \cdot NO_{x(ppm)} \cdot \dot{m}_{exh} \quad in g/h$$
(4-1)

$$CO = 0.000966 \cdot CO_{(ppm)} \cdot \dot{m}_{exh} \quad in g/h \tag{4-2}$$

$$THC = 0.000479 \cdot THC_{(ppm)} \cdot \dot{m}_{exh} \quad in g/h \tag{4-3}$$

The above NO_x, CO and THC values as well as the \dot{m}_{exh} are presented in g/h. The specific emissions including the weight factor can be calculated according to (4-4) to (4-6):

Specific
$$NO_x = \frac{\sum NO_x(n) \cdot WF(n)}{\sum P(n) \cdot WF(n)}$$
 in g/kWh (4-4)

Specific
$$CO = \frac{\sum CO(n) \cdot WF(n)}{\sum P(n) \cdot WF(n)}$$
 in g/kWh (4-5)

Specific
$$THC = \frac{\sum THC(n) \cdot WF(n)}{\sum P(n) \cdot WF(n)}$$
 in g/kWh (4-6)

where WF is the weight factor in %, P the power in kW and n the mode point.

Figure 4-5 shows the NO_x emissions of B0 as concentration (ppm) and as the specific emissions (g/kWh). When discussing engine exhaust emissions the units have to be treated with care. While the specific NO_x emissions are very high at low engine loads for example, the corresponding NO_x concentrations are relatively low. Therefore all emission results are consistently presented as specific emissions (g/kWh) which is the unit used in the current Euro 5 and Euro 6 regulations for heavy-duty engines.



Figure 4-5: NO_x emissions (ppm and g/kWh) over ESC cycle.

Figure 4-6 shows the specific NO_x emissions for the various engine speeds and loads. The NO_x emissions decrease with increasing engine load and generally also with decreasing engine speed. The highest NO_x emissions occur at low engine speed and 25 % load with 12.1 g/kWh mainly due to very low exhaust gas flow and low power output. The average weighted specific NO_x emission for all 12 test points is 8.76 g/kWh, which is the benchmark for the comparison of different biofuels.



Figure 4-6: Specific NO_x emissions for various engine speeds and loads.

Figure 4-7 presents the specific PN emissions of B0 diesel. Especially at high engine speed at low and full engine load the PN emissions are highest with 133.6 and 164.8 trillion particles per kWh, respectively. The high PN value at full load results most likely from the low air to fuel ratio and a long combustion duration giving the

particles more time to form during the diffusion combustion phase. The average weighted specific PN emission for all 12 test points is 6.19×10^{13} per kWh, which is the benchmark for the comparison of different biofuels.



Figure 4-7: Specific PN emissions for various speeds and loads.

Figure 4-8 presents the specific CO emissions of B0. The trends are similar to the PN emission trends with the highest CO emissions at high engine speed and low load of 3.7 g/kWh. The full load points generate high CO values due lower air to fuel ratios. At low engine loads, the CO emissions are high due to rapid combustion and short combustion durations thus CO does not oxidise further to CO_2 . The average weighted specific CO emission for all 12 test points is 1.18 g/kWh, which is the benchmark for the comparison of different biofuels.



Figure 4-8: Specific CO emissions for various speed and loads.

Figure 4-9 presents the specific THC emissions of B0. THC emissions decrease with decreasing engine speed and increasing engine load. The reasons for high THC emissions at low load and high speed is most likely the low cylinder gas temperature not being high enough to provide a sufficiently high evaporation rate of the fuel and a significant portion of unburned liquid fuel can pass through the exhaust system. The cylinder temperature decreases with higher engine speed and lower engine load, similarly to the THC emissions. Another reason for high THC might be wall impingement at high engine speed as the rail pressure is significantly higher at high rather than low speed with 1310 bar and 1150 bar, respectively. The weighted specific THC emission for all 12 test points is 0.06 g/kWh, which is the benchmark for the comparison of different biofuels.



Figure 4-9: Specific THC emissions for various speeds and loads.

To gain a better understanding of the emission formation at various engine conditions, a detailed combustion analysis based on the in-cylinder pressure data is required. In the next section some important injection and combustion parameters are presented and the heat release rate HRR profiles of B0 diesel derived from the in-cylinder pressure data are introduced.

4.1.3 Combustion analysis results

During engine tests, the main ECU parameters as well as the crank-angle based incylinder pressure have been recorded to provide more detailed information about the combustion process during the combustion cycle. Parameters, such as start of main
injection (SOI), dwell time, rail pressure (RP) and injection quantity have been recorded with the in-house engine management software from Cummins called Calterm III. The heat release rate (HRR), start of combustion (SOC), and mass fraction burned profiles have been calculated based on the in-cylinder pressure.

The SOI for all 12 test points is shown in Figure 4-10. The following plots are 2D surface plots across the linear discretised grid. At higher engine speed, the fuel injection is advanced with its earliest injection at full load at 7.8 CAD bTDC. At medium engine speed the fuel injection is between two and three degrees retarded compared to high speed, but still advanced with respect to low engine speed. However, at low and medium engine speed the injection time retards with increasing engine loads of 2.8 and 4.4 CAD bTDC, respectively.



Start of main injection [CAD bTDC]

Figure 4-10: SOI for B0 diesel fuel at various speed and load conditions.

The RP for all test points is shown in Figure 4-11. Overall, the RP increases with increasing engine speed and engine load until the maximum rail pressure of 1800 bar is reached at full speed and full load.



Figure 4-11: Rail pressure for B0 fuel at various speed and load conditions.

At all test points, a small fuel quantity between 3 to 5 mg was injected as a pilot injection with a dwell time of 5 CAD before the main SOI. At high engine speed, a post injection event was added shortly after the main SOI was complete.

Based on the in-cylinder pressure data, the SOC was calculated using a MATLAB program. The SOC for all test conditions have been plotted in Figure 4-12. The combustion starts earlier at high engine speeds and retards with decreasing engine speed. Also, the SOC advances with increasing engine load. At full load, the ignition delay (ID) is very short due to the high internal gas temperature and the fuel ignites while pilot injection proceeds. At low engine load, the ID is longer and combustion starts around three CAD after pilot injection finishes.



Figure 4-12: SOC of B0 fuel for various speed and load conditions.

The peak in-cylinder pressure is plotted in Figure 4-13 and as a first approximation is proportional to the engine power output. The peak pressure increases with increasing engine load for all three engine speeds. A high peak pressure means high cylinder temperatures and therefore increasing NO_x emissions. At high engine speed, the location of the peak pressure is closest to TDC and it moves away with decreasing engine speed and increasing engine load.



Figure 4-13: Peak cylinder pressure at its location of B0 fuel for various speed and load conditions.

Figure 4-14 shows the HRR based on the in-cylinder pressure data at high engine load (75 % load). The HRR is presented in kJ per seconds rather than kJ per CAD as the profiles appeared more consistent in this time domain, furthermore time is a more appropriate metric as it is independent of engine speed. The injector opening signal was plotted for the pilot injection and the start of the main injection to visualise the SOI, SOC and ID. The SOC occurs earlier at high engine speed and retards with decreasing engine speed, which is agreement with the contour plot in Figure 4-12. Figure 4-14 shows the HRR based on the in-cylinder pressure data at high engine load (75 % load). The pilot injection causes a small peak in HRR before the main injection starts. This shows that the pilot fuel quantity is sufficient to cause ignition and the combustion chamber temperature rises. The peak of the pilot HRR is higher at high engine speed (2220 rpm) due to the higher fuel injection rate and longer ID at this pilot phase.

The maximum HRR is highest for medium engine speed (1855 rpm) and high engine speed (2220 rpm) with around 2000 J/s, and lowest for low engine speed (1490 rpm) with less than 1800 J/s. At medium and high engine speed the produced NO_x emissions were 8.1 and 8.2 g/kWh, respectively, while the NO_x emissions at low engine speed were only 7.1 g/kWh due to the lower HRR. The PN was more than twice as high at high engine speed as at medium engine speed, possibly due to the post injection at high engine speed and increased combustion duration. The PN at low engine speed is higher than at medium engine speed and the reason could be a lower air to fuel ratio at lower engine speed due to the lower intake boost pressure.

The SOC of the pilot injection takes place while fuel is still injected for low and medium engine speed. However at high engine speed, the SOC of the pilot event takes place just after the end of the injection. Due to the lower gas temperatures at high engine speed, the ID is extended and a more rapid combustion event occurs. The ID of the main injection is reduced at all three engine speeds and SOC takes place very shortly after SOI. The short ID reduces the premix combustion phase and time for NO_x formation.



Figure 4-14: HRR at 75 % load and low, medium and high engine speed.

Figure 4-15 shows the HRR based on the in-cylinder pressure data at low engine load (25 % load). The SOC of the pilot injection takes place after the end of the pilot injection. Thus the ID is very long resulting in a very high premix combustion phase for the pilot injection. During the main injection, the combustion starts much earlier due to the increased cylinder temperature by the pilot combustion. No clear distinction between premix and diffusion phases is observed and the predominantly a premix-controlled combustion takes place resulting in high NO_x emissions. Similar to the high load profile, the maximum HRR for the main injection is significantly lower at low engine speed, although the NO_x emissions are very similar at all three engine speeds. The SOC of the main injection is a few CAD retarded at low engine speed, which is expected to further reduce NO_x emissions. A reason for the high NO_x emissions at low engine speed cannot be explained. At high engine speed the PN

emissions are three and four times higher than at medium and low speed, respectively. The longer combustion duration and low cylinder temperature are believed to the reason for the high particulate number.



Figure 4-15: HRR at 25 % load and low, medium and high engine speed.

The in-cylinder pressure data have revealed that the HRR is an effective tool to explain NO_x and PN formation. Figure 4-16 shows the relationship between the peak HRR and the NO_x concentration in ppm. A high peak HRR indicates a more rapid combustion, which usually results in high cylinder temperature and high NO_x formation rates. This further illustrates that the NO_x formation can be controlled by changing injection parameters, such as RP and SOI, which will in turn influence ID and SOC significantly. A direct relationship for PM, CO and THC emission cannot be established as the local cylinder gas temperature and local air-to-fuel ratio are mainly affecting the particulate formation, which are not expressed in a HRR profile. In general it can be stated that a locally low engine temperature and a locally low airfuel ratio favours the formation of PN, CO and THC emissions, while NO_x formation is decelerated.



Figure 4-16: Linear relationship between peak HRR and NO_x concentration.

4.2 Biofuel engine tests

The main focus of this work is the investigation of various biofuels on engine performance and emissions. While section 4.1 focuses on the engine performance and emissions at different engine loads and speeds, in this section the effects of different fuel types are investigated.

4.2.1 Engine performance

The engine testing procedure with biofuels was identical to the B0 engine tests. However, at the three full load test points, the maximum engine torque was reduced for some fuels to prevent engine stalling as full power could not always be achieved. Achieving full power with B0 offered no problems, however the three biodiesel fuels struggled to achieve full load. Figure 4-17 shows the averaged power output of all three full load points for B0 mineral diesel and the four neat biofuels. While the power loss of HVO is negligibly smaller than the target value of 135.6 kW, all three FAMEs produce about 9 % less power at the full load test points.



Figure 4-17: Average power output at full load condition.

The engine fuel injection system is based on a volumetric injection meaning that a fixed volume is injected rather than a fixed mass quantity. Thus, fuel density plays an important role in the fuel injection. The LHVs and fuel densities of the various test fuels are displayed in Table 3-1. Although HVO has a higher mass-based LHV than B0 diesel, the volumetric LHV is 5 % lower than mineral diesel due to the lower fuel density of HVO. The volumetric LHV of biodiesel is 8.5 % lower for SME and 9.2 % lower for PME and UCOME. Figure 4-18 shows the maximum produced power versus the blend ratio of the fuel. When reducing the biodiesel blend ratio the maximum power output of the engine increases until 99 % of the maximum power is produced at a blend ratio of B10 (10% biodiesel and 90% mineral diesel). Other researchers testing various biodiesel on a diesel engine test bed experienced similar trends in the loss of power and also explained their results with the differences in volumetric heating value. Kaplan et al. tested sunflower oil methyl ester and mineral diesel in a 2.5 L 53 kW diesel engine and measured a loss in power of 5 - 10 % depending on the engine speed [241]. Cetinkaya et al. tested waste-oil methyl ester in a four cylinder Renault Megane diesel engine and the loss in torque was between 3 and 5 % using biodiesel [242]. Similar results were achieved by Lin et al using PME in a non-turbocharged engine at full load [98].



Figure 4-18: Average engine power at full load with increasing biodiesel blend ratio.

All ESC test points below full load can produce the desired engine torque by injecting more volume into the combustion chamber resulting in slightly longer injection duration and higher rail pressures when using biofuels. Longer injection duration and higher injection pressures affect the spray characteristics of the fuel injection and also affects the heat release rate. The spray characteristics of the different fuels are discussed in Chapter 5.

The ECU calculates an engine torque value on the basis of injection parameters (such as injection duration and RP) which are calibrated for mineral diesel. Therefore, the total injection quantity and indicated engine torque displayed in the engine management system deviates from the actual torque value demanded by the dynamometer. At full load the intake pressure signal (MAP sensor) and total injection quantity are reaching their limits at this specific engine speed and the inbuilt torque limiter restricts the injection quantity to prevent the system running too rich with fuel. Figure 4-19 shows the ECU data log for an ESC test run with SME B100 fuel. The black numbers are the indicated torque values calculated by the ECU based on the fuel injection parameters, based on the calibration of mineral diesel. The red values are the actual engine torque output demanded by the dynamometer.



Figure 4-19: Variation of indicated and real torque.

For all 12 test points the indicated torque is exactly 11 % higher than the real torque showing that at full load the in-built torque limiter restricts the maximum torque according to the full load curve in Figure 4-2.

The fuel consumption is another indicator of the engine performance. Figure 4-20 presents the BSFC of all four neat biofuels in comparison to B0 mineral diesel.



Figure 4-20: BSFC of various test fuels (full load excluded).

Since all fuels have different maximum power outputs, the three full load points have been excluded in this diagram only showing the BSFC for the nine test points between 25 % and 75 % engine load. The average fuel consumption of B0 is 217.5 g/kWh, which is 2.3 % higher than the BSFC of HVO and about 14 % lower than the

BSFC of the three FAMEs at B100. The differences in BSFC can be explained with the different calorific values of the fuels. While the mass-based LHV of HVO is 2.4% higher than B0, the mass-based LHV of PME, SME and UCOME is 12.9 %, 13.1 %, and 13.2 % lower, respectively.

Figure 4-21 shows the linear relationship between fuel consumption and blend ratio. Other fuel parameters, such as bulk modulus, viscosity or surface tension do not affect the fuel consumption significantly. Similar findings in BSFC have been reported by Hansen and Jensen testing a 6 cylinder 170 kW diesel engine with RME B100 and B0 and a 14 % increased BSFC were found [99]. Canacki and van Gerpen also mentioned a 14 % increase in BSFC with yellow grease biodiesel and SME compared to petroleum diesel fuel [205].



Figure 4-21: Fuel consumption with increasing biofuel blend ratio.

For all 12 test points, the highest fuel savings compared to B0 were achieved at full load with a 4.3 % lower BSFC with HVO and 10.1 % higher BSFC with PME, SME and UCOME. However, the absolute engine power output was lower for biodiesel than for B0 fuel. Minor savings have been accomplished at high engine speed and 25 % load for HVO (only 0.9 % lower BSFC) and at high engine speed and 50 % load for all three FAMEs with about 14.8 % higher BSFC compared to B0.

The brake thermal efficiency is a measure of how much fuel energy has been converted into useful brake engine power. As Figure 4-22 shows, the BTE is 38.8 % for B0 and HVO and slightly higher for all three FAMEs with its maximum efficiency of 39.2 % for SME. As the BTE is reciprocal to the BSFC for a specific

fuel, the highest and lowest savings compared to B0 are identical to the BSFC savings. Most researchers report no significant variations in brake thermal efficiency when using various types of biodiesel [205, 243, 244].



Figure 4-22: Thermal efficiency of various test fuels.

It can be concluded that all biofuels can be used on the engine with acceptable fuel consumptions and conversion efficiencies. However, full power cannot be achieved due to differences in fuel density and LHV. The limitations are caused by the ECU torque limiter. Full power restoration for biodiesel is carried out in Chapter 6.

4.2.2 Regulated engine emissions

The regulated engine emissions NO_x , CO, THC and PN were measured during the ESC. The measuring principles are explained in section 3.2.4 and all analysers were serviced and calibrated before use. In diesel engines, NO_x and particulate matter have the most challenging emission regulations and are the most difficult for engine designers to reduce to acceptable limits. For heavy duty diesel engines, the emissions are usually presented as specific emissions in g/kWh. The conversion from concentration to specific emissions is explained in section 4.1.2.

In Figure 4-23, the specific NO_x emissions of B0 and the four neat biofuels are shown. All emissions diagrams include the emission values at full load; bearing in mind that full load was not achievable with biodiesel.



Figure 4-23: Specific NO_x emissions of various test fuels.

The NO_x emissions of HVO are very close to the B0 mineral diesel benchmark. Although many fuel properties of HVO and mineral diesel are different, the fuels are chemically similar and chemical kinetics will be similar during the combustion phase. The main reason for the slightly higher NO_x value for HVO is likely to be caused by the higher cetane number of HVO (CN>75) leading to an earlier start of combustion with the peak cylinder pressure being closer to TDC resulting in higher cylinder temperatures. The higher combustion temperatures increase the rate of thermal NO_x formation. On the other hand, the higher CN leads to a shorter ID, and less fuel is premixed resulting in a shorter pre-mix combustion phase, where NO_x molecules are mainly formed. A more detailed explanation can be given when looking into the HRR profiles in subchapter 4.2.3. All three biodiesel fuels emit higher NO_x emissions than mineral diesel fuel with the smallest increase of 0.9 % for PME followed by 4 % for UCOME and 5.7 % for SME. In general biodiesel produces more NOx due to the 11% bonded oxygen in the fuel. The additional oxygen increases the thermal rate of reaction for the NO_x formation and shifts the thermal equilibrium towards a higher NO_x concentration. PME shows significantly lower NO_x values than UCOME and SME due to the higher CN of 64.9 compared to 58.4 and 53.7, respectively. The higher CN leads to a shorter ID and less fuel is burnt in the premix-controlled combustion phase, where NO_x is primarily formed. Sharp et al. investigated the NO_x emissions of SME in a transient FTP cycle and reported an increase in NO_x of 4-13 % compared to mineral diesel [245]. Verhaeven et al. tested

UCOME and RME in five different heavy and light duty vehicles and the NO_x emissions were up to 24 % higher than mineral diesel [246]. Wu et al. used five different biodiesel from croton, palm, soy, rapeseed and waste cooking oil and the NO_x emissions were about 10-23% higher, but varied between different fuels [178]. They also concluded that cetane number must have a significant effect on the NO_x emissions and PME produced less NO_x than SME and UCOME. Chang et al. as well as Grabowski et al. both reported in their work that higher biodiesel saturation causes higher CN and lower NO_x emissions [95, 247].



Figure 4-24: Specific NO_x emissions with increasing biofuel blend ratio.

Figure 4-24 presents the specific NO_x emissions with varying blend ratio. A linear increase in NO_x with increasing blending percentages would be expected, but instead the averaged NO_x values vary strongly between B10 and B50. A non-linear relationship between NO_x and blend ratio has also been observed by other researchers [172, 180, 248, 249].



Figure 4-25: Change in start of combustion with increasing biofuel blend ratio.

A reason of the varying NO_x values is the ECU control management where small changes in the injection strategy can have a significant effect on the combustion behaviour and emission formation of the fuel. Figure 4-25 for example shows the derived SOC at different blend ratios and a non-linear trend is with increasing biodiesel ratio is illustrated. For example, the SOC of SME advances by approximately 0.5 CAD when increasing the blend ratio from B0 to B10. At SME B20 the SOC retards to 17.7 CAD bTDC showing a similar trend to the NO_x emission curve in Figure 4-24.

The specific CO emissions are shown in Figure 4-26. Compared to B0 fuel the CO emissions are reduced for all fuels with the highest reduction of 51.8 % for PME, followed by UCOME and SME with 45.2 % and 44.9 %, respectively. The CO emissions of HVO are reduced by 28.6 %. Main reason for the reduction in CO emissions for HVO is the lack of aromatics (aromatic content of mineral diesel is 27.5 %). Aromatics are hydrocarbons with alternating single and double bonds forming a ring of usually 6 carbon atoms. Their binding energy is much higher than that of straight hydrocarbons making it more difficult to oxidise the molecule causing higher CO formation rates. The biodiesel fuels also contain no aromatics slightly lower C-H ratios resulting in low CO emissions. However, the main source for lower CO emissions is the bonded oxygen in the fuel reducing fuel-rich zones and promoting CO oxidation.

Most authors, who conducted similar work, observed similar reductions in CO emissions. Krahl et al. tested RME on a DaimlerChrysler diesel engine using the ECE R49 cycle and a 50 % decrease in CO has been reported with biodiesel [193]. Peterson and Reece tested several biodiesel in a turbocharged diesel engine and a reduction of 50 % across all biodiesel types were observed [250]. Some researchers found lower reductions in CO emissions [88, 251].



Figure 4-26: Specific CO emissions of tested fuels.

The highest reduction of CO was found for all three FAMEs at full load with up to 80 % lower CO at all engine speeds. At low load (25 % load) however the smallest reductions were observed with even a 5 % increase in CO emissions for SME at medium engine speed (1855 rpm).

Similar results are presented by Choi et al. who tested SME in a single cylinder engine. They observed no differences in CO emissions at low load, but CO reductions with increasing engine load [252].

The specific THC emissions are presented in Figure 4-27. While B0 emits on average around 0.06 g/kWh of hydrocarbons, it decreases by more 60 % for HVO B100 to 0.02 g/kWh. The reductions of THC for the three FAMEs are even greater with being 95 % lower than the B0 fuel benchmark. At most test points the THC value was even below the detection limit of the analyser and only at high speed and low load some THC emissions were measured for PME, SME and UCOME. Many researchers have also reported sharp reductions in THC [100, 253, 254]. There are several explanations for the strong reductions for THC with increasing blend ratios.

First of all the full boiling point (FBP) of mineral diesel is about 365 °C, while the FBP of HVO is around 360 °C and about 330 °C for PME, SME and UCOME. This means that biodiesel is more volatile than mineral diesel and evaporates quicker at high cylinder temperatures. Since usually the non-vaporised fuel gets into the exhaust stream, it can be assumed that the lower emissions are partly affected by the lower full boiling point. This can be supported by the fact that the highest THC emissions occur at higher speeds and lower loads. These are conditions were the least heat is released and cylinder peak cylinder pressure is relatively low. Therefore the cylinder temperature is lower and the rate of fuel evaporation rate is reduced. The absence of aromatics might also contribute to lower THC emissions for HVO and FAME as cyclic hydrocarbons require more energy to oxidise. Also, for the FAMEs the additional oxygen content bonded in the fuel helps to reduce THC emissions as the combustion is more complete and less fuel-rich zones are available to stop the oxidation of hydrocarbon. Rakopoulus et al. mentioned in their review that an increase in oxygen, either with oxygenated fuels or oxygen-enriched fuels will improve combustion and reduce THC emissions [255].



Figure 4-27: Specific THC emissions of tested fuels.

Due to the sharp reductions some researchers also doubt the credibility of the THC analyser. Usually flame ionization detectors (FID) are used and the sensitivity of detecting oxygenated molecules, which are present in biodiesel combustion, may be lower. Furthermore the exhaust sampling pipe is heated up to 190 °C to avoid condensation of hydrocarbons. As the volatility of biodiesel is lower it could be

possible that larger unburned molecules do not evaporate at 190 °C and higher sampling temperatures are required for the molecules to reach the flame ionisation detector (FID) [99, 206, 254].

Figure 4-28 shows the PN emissions of the different test fuels. The average specific PN emissions of mineral diesel were 61.9 trillion particulates per kWh. Similar to CO and THC emissions the PN was reduced sharply for PME, SME and UCOME with more than 75 % to around 13 to 15 trillion particles. The PN reduction of neat HVO fuel was 14 % compared to the benchmark. The highest reductions for all test fuels were achieved at higher engine loads with more than 80 % lower PN for FAME and 20 % for HVO. At low load the reductions were smaller, but still considerable with just about 60 % reduction for PME, SME and UCOME. At high engine speed and full load the PN emissions increased by 5 % for HVO B100.



Figure 4-28: Specific PN emissions of tested fuels.

In the literature, similar reductions in PM emissions were observed. Canacki and van Gerpen tested SME and UCOME and observed a reduction of 65 % for both fuels [205]. However, the reductions in the literature are pretty much all calculated on a mass basis, while the reduction in this work is based on the particulate number. Very few studies have been carried out comparing PN emissions between HVO, FAME and mineral diesel. However, with the reduction in particulate numbers it can be assumed that the particulate mass was also reduced. Several publications show that with biodiesel combustion the average particle size shifts towards smaller particles. Those are known to be more harmful as the retention time in the atmosphere is much

longer and the likelihood of penetration into the respiratory system increases. Krahl et al. compared the particle size distribution of rapeseed methyl ester with mineral diesel and reported an increase in particles between 10 – 40 nm and a decrease in particles above 40 nm [193]. In a later study they also measured the size distribution of SME and PME fuels and came to the same conclusion using two different measurement techniques namely scanning mobility particle sizer (SPCS) and electrical low-pressure impactor [256]. The main reason for the reduction in both, particulate mass and number, is the absence of aromatics in HVO and FAME. Aromatics are known to be soot precursors and the effect of aromatic content on the PM emissions have been studied amongst others by Schmidt and van Gerpen [254] and Azetsu et al. [257]. The oxygen content of the biodiesel fuel also promotes a cleaner combustion and reduces fuel-rich zones. Frijters and Baert established a correlation between the oxygen content in biodiesel and the PM emissions [258].



Figure 4-29: Specific PN emissions with increasing biofuel blend ratio.

Figure 4-29 presents the specific PN emissions in relation to the biofuel blend ratio. A trend of PN reduction can be found for all biofuels over the whole blending range. At the B20 ratio, a steep drop in PN emission was observed for all fuels except SME. The reason for slight variations with changing blend ratios is believed to be caused by slight changes of the injection strategy in the engine management system. A detailed investigation of the combustion analysis has been carried out in the next section. In general it can be concluded that the emissions of all biofuels are either slightly above the benchmark or significantly below. While NO_x emissions are

slightly higher than B0, the other measured emissions are much lower than the benchmark. This means that NO_x emissions could potentially be reduced when recalibrating the ECU, or by adopting an alternative aftertreatment strategy.

4.2.3 Combustion analysis results

In this section, the results of the in-cylinder pressure data are analysed and some ECU parameters are investigated to try to explain sudden changes in exhaust emission with increasing blend ratio.

The indicated mean effective pressure (IMEP) was calculated for all fuels based on the in-cylinder pressure. The average IMEP of the ESC excluding full load was 13.15 bar for mineral diesel. The average IMEP of HVO B100 was similar with 13.03 bar, while neat PME, SME and UCOME produced a lower average IMEP of 12.71 bar, 12.60 bar and 12.65 bar, respectively. As the BMEP, which is proportional to the brake power output, was constant for all fuels at 10.19 bar, the mechanical energy loss is lower for biodiesel resulting in a higher mechanical efficiency of around 80 % instead of 77 % for mineral diesel. The mechanical efficiency increased with increasing biodiesel concentration. The reason for the higher mechanical efficiency with biodiesel is very controversial. While Ramadhas et al. reported that the increase in mechanical efficiency is due to better lubrication of the fuel [167], Muralidharan et al. mentioned that the difference in fuel properties may affect the mechanical energy loss [259]. The higher bulk modulus and fuel viscosity might reduce the energy consumption of the fuel pump when pumping fuel through the fuel lines and pressurising it to the commanded rail pressure. The higher viscosity might reduce friction losses and possibly less fuel needs to be pumped into the rail as less fuel flows back through the fuel return pipes.

Mineral diesel fuel generates an average peak cylinder pressure of 128.8 bar at 9.8 CAD aTDC and a maximum HRR of 155.4 Joule/CAD at 6.7 CAD aTDC at an average speed of 1855 rpm and an average load of 50 % equivalent to a torque of 368.3 Nm. At the same conditions, HVO generates an average cylinder peak pressure of 130.4 bar at 9.7 CAD aTDC and max HRR of 156.6 Joule/CAD at 6.4 CAD aTDC. The higher and more advanced peak pressure for HVO results in higher peak temperatures and higher NO_x formation. The atomisation and evaporation rate of HVO is stronger than mineral diesel due to its lower distillation curve and improved

spray characteristics investigated in Chapter 5. This explains the slightly higher NO_x emissions despite the higher cetane number of HVO. The three FAMEs generated lower peak pressures and peak HRR of 125 bar and 153 J/degree, respectively, for PME, SME and UCOME. That shows that the premix-controlled combustion phase was less developed and peak cylinder temperatures were lower resulting in slower NO_x formation at this stage. However, the additional oxygen content in the fuel shifted the NO_x equilibrium towards higher NO_x concentrations. The main reason for lower combustion temperatures and pressures was the lower LHV of biodiesel. At the SOC the premixed fuel burned rapidly but as the heating value of biodiesel was rather low, the HRR was also reduced. To compensate the lower heating value the injection duration was prolonged to achieve the same brake power and the diffusion-controlled combustion phase increased.



Figure 4-30: Heat release rate of various neat fuels at low engine speed and 75 % load.

Figure 4-30 shows the HRR of B0 mineral diesel fuel, HVO B100 and SME B100 at low engine speed and 75 % load. The start of the pilot combustion was mainly determined by the CN showing an earlier SOC with HVO. The SOC of the main injection took place at the same time for all fuels. Due to the pilot injection the internal gas temperature was high resulting in a small ID for all fuels. The peak heat release rate is similar for B0 and HVO, but a much lower for SME showing the trend described above. The volumetric LHV of B0 fuel, HVO and SME B100 are 36 MJ/l, 34.2 MJ/l and 33 MJ/l, respectively. This should cause a decrease in the peak HRR for HVO and SME. The higher peak HRR of HVO indicates that HVO can compensate for the lower volumetric heating value much better than FAME and a very similar heat release curve is obtained. Since SME cannot compensate the LHV value as well as HVO, the injection duration for SME is longer resulting in a weaker, but longer diffusion combustion phase. This extended combustion phase is supplied by fresh oxygen bound in the fuel until injection ceases and formed CO, THC and PN can oxidise into CO_2 and water.

The heat release curve of PME and UCOME are very similar to that of SME explaining similar trends in emission reductions.



Figure 4-31: Cumulative HRR of various neat fuels at low engine speed and 75 % load.

Figure 4-31 shows the cumulative HRR of B0, HVO and SME at low engine speed and 75 % load. A slightly lower cumulative heat release curve was observed for SME fuel compared to B0 and HVO fuel. The gap increased after TDC until around 390 CAD aTDC where the difference between B0, HVO and SME reaches a minimum. At the end of the combustion stroke the gap between FAME and diesel increased again and the total heat release of SME is about 3.5 % lower than B0 and HVO fuel. This shows that the indicated engine power is lower for FAME to generate the same brake power output resulting in a higher mechanical efficiency for biodiesel as shown above. Figure 4-32 shows the HRR at low engine speed and 25 % engine load. The three low load points were the only exception where the peak HRR was higher for the FAMEs than for B0 and HVO fuel, while the peak pressure of biodiesel was still lower.



Figure 4-32: HRR of various neat fuels at low engine speed and 25 % load.

The peak HRR of the pilot combustion is higher at low load than at high load, due to more fuel being injected during this event. The start of the pilot combustion is advanced for HVO due to higher CN. B0 and SME fuel have the same ID and also similar cetane numbers of 51 and 53.7, respectively. The ID of the main combustion is very small for all fuels due to the high cylinder gas temperature.

The injection duration only lasts a few CAD at low load and no separation of premixed- and diffusion-controlled combustion phase is visible. This explains the high specific NO_x emissions particularly at low engine load as premixed combustion is the predominant combustion phase.

The HRR at low engine speed and 75 % load (Figure 4-30) is similar to the HRR at high engine speed and 75 % load as illustrated in Figure 4-33. However, compared to Figure 4-30, the two combustion phases are not as distinctive at high engine speed. The specific NO_x emissions were higher at higher engine speed although the peak HRR is lower at high engine speed. When converting the HRR from J/CAD to J/s the HRR at high engine speed, however, will be higher and the higher NO_x emissions

can be explained. Also, the difference in peak HRR between FAME and diesel fuel was smaller at high engine speeds and at this particular test point the relative increase in NO_x emissions was 7.2 %, 12.8 % and 11.9 % for PME, SME and UCOME and therefore higher than the average NO_x increase.



Figure 4-33: Heat release rate of various neat fuels at high engine speed and 75 % load.

The peak in the heat release curve at the end of combustion is a post injection, which only occurred at high engine speed.

The SOC of the pilot injection is advanced for HVO fuel due to the high CN. The difference in ID between HVO and B0 fuel is larger than at low engine speed and high load. With increasing engine speed and decreasing engine load the ID seems to increase and the CN causes a stronger effect. Additional scrutiny has been carried out by Yates et al. [260] and Schaberg et al. [261] establishing a relationship between the ID and the global cylinder temperature and pressure.



Figure 4-34: Difference in ID between GtL and B0 across cylinder temperature and pressure [261].

Figure 4-34 shows the difference in ID between GtL and mineral diesel fuel. At higher cylinder temperatures and pressures the differences in ID diminishes. The same trend can be observed in Figure 4-35 showing the relationship between ID and the brake specific peak cylinder pressure at fixed engine speed of 2220 rpm. The peak pressure per power output has been chosen as the biodiesel have lower full power outputs resulting in different and misleading peak pressures at full load. The ID increases with decreasing engine load and the differences between the fuels is related to their CN.



Figure 4-35: Effect of cylinder conditions on ignition delay at high engine speed.

The slightly higher NO_x emissions reported in section 4.2.2 can now partly be explained with the understanding of the injection strategy and combustion analysis.

With decreasing engine load, the effect of CN increases and the differences in ID grows between HVO and B0 fuel. At lower engine load a high CN has a stronger effect, which is reflected in the NO_x differences between HVO and B0 diesel fuel.

While at 75 % and 50 % load the NO_x emissions of HVO were higher than B0 diesel with 2.2 % and 0.7 %, respectively, the NO_x emissions at 25 % engine load were 1.7 % below the corresponding B0 diesel NO_x emissions.

For the three biodiesels, the same observation is made. With higher CN of biodiesel, the NO_x emissions decrease due to shorter ID, especially at low engine loads. Xingcai et al. tested diesel and ethanol-diesel blends and showed that when adding cetane improver to the fuel the NO_x emissions reduce significantly [262].

Mccormik et al. tested various biodiesel with different chemical structures and concluded that biodiesel with a higher saturation level have a higher CN, lower density and usually worse cold flow properties [157]. They measured the NO_x emissions of 21 different biodiesel and only the fully saturated and hydrogenated biodiesel produced NO_x emissions lower than the benchmark. Figure 4-36 shows their correlation between CN, density and NO_x emissions.



Figure 4-36: Effect of density and CN on NOx emissions for FAME [157].

As the PN emissions were not affected by changes in CN they concluded that the additional NO_x emissions are not driven by the thermal NO_x formation mechanism, but by the prompt mechanism instead. Two independent analysis by Miller et al. and McCormick et al. showed that the double bonds in the more unsaturated fuel

molecules affect the chemistry of the combustion and potentially form more hydrocarbon radicals, which lead to an increase in prompt NO formation [71, 157]. The thermal NO_x mechanisms are still present and with higher HRR and higher cylinder temperatures the NO_x formation rate increases. Figure 4-37 shows that the general trend is similar for all fuels and that in the premixed-controlled combustion phase the NO_x formation is mainly determined by the thermal mechanism. The

phase the NO_x formation is mainly determined by the thermal mechanism. The prompt NO_x formation is more applicable in fuel-rich zones in the diffusion combustion phase. At low engine load, the predominant combustion is premixed-controlled and the prompt NO_x formation is reduced. When comparing SME B100 and B0 diesel fuel with similar CN it was observed that the differences in NO_x emissions decrease with decreasing engine loads due to the absence of prompt formation. The difference in NO_x emission using SME B100 fuel was 4.9 %, 5.2 %, 7.2 % and 8.3 % at 25 %, 50 %, 75 % and 100 % engine load respectively.



Figure 4-37: Linear relationship between heat release rate and NO x concentration for all fuels.

It can be concluded that all three neat biodiesel fuels show smaller IMEPs across the test cycle resulting in higher mechanical efficiencies possibly due to the lower fuel pumping loss when pumping biodiesel with the higher viscosity and better lubrication characteristics. Although HVO has a lower volumetric heating value than mineral diesel, the peak HRR is higher for HVO while FAME generates lower peak HRRs due to the lower heating value. The spray tests confirm that HVO fuel mixes

very well with the surrounding air and a stronger premixed combustion phase is expected. This explains the higher NO_x emissions with HVO despite a lower aromatics content and higher CN. HVO can compensate for the drop in volumetric heating value well, while biodiesel requires longer injection duration to compensate the lower LHV. The lower HRR reduces NO_x emissions, but the bonded oxygen content is countering this effect. At low engine loads, no visible diffusion combustion is present, but a strong premix combustion phase occurs, resulting in high specific NO_x emissions with all fuels. The ID of all fuels diminishes with increasing cylinder temperatures and pressures, while the differences in NO_x emissions are still present. This proves that the thermal mechanism cannot be responsible for the increased NO_x emissions for biodiesel. However, a theory has been proposed saying that the unsaturated double bonds in the methyl esters can cause higher levels of hydrocarbon radicals, which are responsible for prompt NO_x formation.

Chapter 5. Optical diagnostics of various biofuels

In this chapter, the spray characteristics of various biofuels are investigated and compared with B0 mineral diesel. The chapter starts with the injector calibration to obtain the fuel quantity per stroke for each biofuel in relation to injection duration and injection pressure. For the spray tests, a high speed CCD camera was used to capture the spray images in a very short time of less than 2 ms. The macroscopic spray cone angle, spray tip penetration and spray area have been analysed using a MATLAB program. A more detailed description of the setup and processing techniques is described in section 3.3. For the spray tests only B50 and B100 fuels were tested and compared with B0 mineral diesel. The B10 and B20 series were not considered in this chapter.

5.1 Injector calibration

To determine the injected fuel quantity per stroke, a single-hole nozzle with an orifice diameter of 0.16 mm was selected. Calibration was carried out at 25 °C and 80 °C fuel temperatures, injection pressures between 600 and 1800 bar and injection durations between 0.6 ms and 2.2 ms. The reason for choosing these injection conditions were that injection pressures and durations are similar to the injection conditions on the engine test bed in Chapter 4. The calibration rig set up is shown in Figure 5-1.



Figure 5-1: Injector calibration rig.

High temperature injection was achieved by wrapping a robe heater around the injector. The fuel return temperature straight after the injector return outlet was measured and assumed to be identical with the fuel injection temperature. The same fuel injection system was used for the spray tests. The calibration test matrix with the B0 calibration data is given in Table 5-1.

Nozzle: 0.16mm											
		Injection duration [ms]									
FUEL. BU KEI DIESE		0.6		1		1.4		1.8		2.2	
ar]	600	1.19	2.38	3.48	6.95	5.79	11.59	4	15.95	5.02	20.15
		1.21		3.46		5.79		3.98		5.04	
		1.17		3.48		5.81		3.98		5.05	
ure [b	900	2.14	4.25	4.86	9.75	7.76	15.48	5.29	21.13	6.58	26.36
		2.18		4.89		7.73		5.28		6.59	
		2.05		4.88		7.73		5.28		6.6	
Injection press	1200	2.64	5.38	5.99	11.96	9.38	18.35	6.16	24.69	7.43	29.72
		2.71		5.98		9.06		6.18		7.43	
		2.72		5.97		9.08		6.18		7.43	
	1500	3.04	6.07	6.58	13.16	10.57	21.09	7.1	28.35	8.01	32.04
		3.03		6.55		10.53		7.09		8.02	
		3.04		6.61		10.54		7.07		8	(1)
	1800	3.19	6.38	6.98	14.05	11.27	22.56	7.76	30.76	8.52	0
		3.19		7.07		11.25		7.69		8.48	34.0
		3.19		7.02		11.32		7.62		8.5	(1)
Number Injections		500 50		50	0	500		250		250	

 Table 5-1: Template of injector calibration matrix with B0 calibration data.

To measure the injected mass per stroke, the fuel was injected either 250 or 500 times in a measuring cylinder filled with fuel absorber and the weight of the cylinder was measured before and after the injection event. Each calibration has been repeated three times, averaged and divided by the total number of injections to gain the mass per single injection.

The calibration results for B0 reference diesel fuel at 25 °C have been plotted in Figure 5-2 for various injection durations.



Figure 5-2: Injector calibration curves for B0 reference fuel.

The injected mass increases significantly, almost proportionally with increasing injection durations. Also, the increasing injection pressure effects the injected fuel quantity. The relationship between injection quantity and injection pressure has a form of a logarithmic function. For each fuel a generic equation with the two variables injection duration and injection pressure has been established to predict the injected fuel quantity within 5 % accuracy. The upper and lower limits for the equation are the calibration limits of 0.6 to 2.2 ms and 600 to 1800 bar for injection duration and rail pressure, respectively. The equations for B0 reference fuel are shown in equation (5-1) and (5-2) and represented as black solid lines in Figure 5-2.

$$M_{inj} = 6.73 \cdot T^{1.185} \cdot \ln(RP) - 5.519 \cdot T^2 + 28.58 \cdot T + 1.74$$

for T \le 2 ms (5-1)

$$M_{inj} = (6.73 \cdot T^{1.185} \cdot \ln(RP) - 5.519 \cdot T^2 + 28.58 \cdot T + 1.74)$$
$$\cdot \left(1 + \frac{1200 - RP}{6000}\right) \qquad \text{for } T > 2 \text{ ms} \qquad (5-2)$$

where T is the injection duration in ms, RP the rail pressure in bar and M the injected mass in mg/stroke.

Since the prediction at high injection duration was less robust than at low injection pressure, a slightly modified version of equation (5-1) was used for injection durations above 2 ms (equation (5-2)).

The relative prediction error between the real value and the calculated prediction is shown in Table 5-2. Except for very low injection pressure and short duration, where the error is 11%, the predicted and real value match within 5 % accuracy.

B0 Fuel		Injection pressure [bar]							
		600	900	1200	1500	1800			
s]	0.6	111%	97%	96%	99%	105%			
Injection ıration [m	1	104%	102%	99%	102%	104%			
	1.4	100%	101%	101%	98%	100%			
	1.8	96%	98%	100%	98%	98%			
qr	2.2	100%	100%	101%	101%	98%			

Table 5-2: Relative uncertainty between predicted and actual injected mass.

The equation to calculate the injected mass is different for each biofuel to provide the best possible fit. For all fuels, it was possible to keep the predictions within the 5 % error range except for a few conditions near the edges, mostly low pressure and short duration.

A comparison of the injected fuel quantity at 0.6 ms injection duration for all pure fuel types is shown in Figure 5-3.



Figure 5-3: Injected fuel quantity at various injection pressures using different biofuels.

At low injection pressures, the fuel viscosity has a major effect on the injection quantity. In particular the fuel injection of PME fuel was noticeably different and injection frequency was interrupting at low fuel temperature due to the high fuel viscosity. This observation diminished at higher fuel temperature and lower viscosity. Only at high injection pressure of 1800 bar, SME and UCOME fuel showed the highest fuel quantity. At lower injection pressures, the effect of the higher viscosity is reducing the fuel quantity per stroke despite the higher fuel density. HVO fuel has a very low density and therefore the injected fuel quantity is lower compared to biodiesel and B0 fuel. However at 600 bar injection pressure, the injected quantity is high for HVO fuel and the low viscosity seems to be more predominant than the fuel density. Especially at high injection pressures, the injector calibration was critical as the fuel temperature heated up due to the energy of compression. As it was difficult to control the temperature rise at high injection pressures, these values contain a specific uncertainty, which cannot be expressed by the standard deviation.

5.2 Spray tests

In DI engines, the spray characteristics associated with injecting fuel into the combustion chamber determines the combustion behaviour and thermal efficiency. An adequate air-fuel mixture is important to achieve clean combustion and good engine performance. Fuel injection, atomisation and evaporation are complex two phase flow processes and a comprehensive understanding is essential to reduce exhaust emissions to a minimum. Injecting fuel into a nitrogen atmosphere enables investigation of the fuel spray without combustion or significant evaporation. The spray characteristics of biofuels are different from mineral diesel due to differences in fuel properties, such as density and viscosity. An understanding of the differences in the spray behaviour is helpful to optimise combustion and explain emission formation.

For the spray tests, the same injector with the 0.16 mm single-hole nozzle was used throughout. The four parameters that have been changed during the injection tests were vessel chamber temperature, chamber pressure, rail pressure (RP), and injection duration.

The test matrix with the 16 different test points is shown in Table 5-3.

Chamber temperature [°C]	Chamber pressure [bar]	Rail pressure [bar]	Injection duration [ms]
25	1	600	2.2
25	1	600	0.6
25	1	1800	2.2
25	1	1800	0.6
25	60	600	2.2
25	60	600	0.6
25	60	1800	2.2
25	60	1800	0.6
100	70	600	2.2
100	70	600	0.6
100	70	1800	2.2
100	70	1800	0.6

 Table 5-3: Test matrix for spray tests based on high and low levels for four parameters.

During the test phase it was decided to not further consider the four test points at high chamber temperature and ambient chamber pressure as fuel condensation on the quartz windows resulted in very poor picture quality and the test points at ambient pressure did not add much valuable information to the overall spray tests. The optical quartz windows had to be cleaned at regular intervals to ensure good picture quality as the fine fuel particles adhered to the windows and worsened the view after too many injections. At high chamber pressures, the cleaning intervals were increased due to the higher pressure reducing the effect of fuel atomisation.

The chamber pressure was set to 60 bar at cold chamber temperature and set to 70 bar at 100 $^{\circ}$ C chamber temperature to achieve similar air densities of approximately 65 kg/m³.

A 500 W xenon light source was used to give a uniformly distributed illuminated background and a V710 Phantom monochrome CCD camera was used to capture the images of the fuel injection. The device was borrowed from the EPSRC instrument pool and was able to take up to 1.4 million images per second on the lowest resolution. Figure 5-4 shows the picture of the spray test set up.



Figure 5-4: Equipment setup for spray tests.

The camera resolution was set to 256 x 256 pixels resulting in a capturing speed of 70,000 fps with an exposure time of 5 ns. Camera and injector were triggered simultaneously using the same output channel on the DAQ card. Both devices required a TTL signal to trigger. However, the camera trigger signal rested of a fixed 5 Volt signal and image capturing started when the signal was pulled down to 0V on the falling edge by the incoming 5V TTL signal. In the software, the high speed camera had to be set into capturing mode before the receiving a trigger signal. In this mode, the camera filled its internal buffer with the image data. When the camera was triggered the internal buffer was once filled with the captured images before and after the trigger signal. At each of the above test points, five injections were captured and the video sequence was cropped from the start of the trigger up to the point where the spray plume hits the edge of the visual field. The obtained images were saved as .tif files with 16 bits per channel giving a range of 0 - 65535 values for each pixel. An example of the captured image is shown in Figure 5-5.



Figure 5-5: Example of a captured raw image.

The batch of images was loaded into the MATLAB program, cropped and thresholded to a binary image with only black and white pixels. The threshold limit for the grey scale was set to 65,500 to capture most of the grey pixels resulting in a strong developed spray. In the next step the penetration length, cone angle and spray area were analysed and imported into Excel. Figure 5-6 shows some raw images with the correspondent penetration distance and cone angle of B0 fuel at 100 °C ambient temperature, 70 bar back pressure, 1800 bar injection pressure and 0.6 ms injection duration.



Figure 5-6: Raw spray images with corresponding penetration length and cone angle.
5.3 Spray tip penetration

The penetration values were averaged over usually five sets of images.

Figure 5-7 and Figure 5-8 show the spray curve of B0 and the neat biofuels at 25 $^{\circ}$ C and 100 $^{\circ}$ C, respectively.



Figure 5-7: Spray penetration curve at 25 °C chamber temperature and 60 bar backpressure.

At 25 °C the spray penetration of PME and UCOME was very poor. Both fuels have a high viscosity of 4.55 mm²/s and 4.35 mm²/s at 40°C, respectively. At ambient temperature this viscosity is even higher and causes the very poor spray characteristics. Further, looking at the spray images it can be clearly seen that less fuel was compared to other fuels at the same conditions. The results for the injector calibration show that the average amount of fuel quantity injected at 25 °C and 1800 bar was 6.06 and 6.59 mg/stroke, respectively for PME and UCOME. SME has similar fuel properties, but a slightly lower viscosity of 4.18 mm²/s. However, the injected fuel quantity per cycle was 6.87 mg/stroke and the penetration curve is similar to that of B0 and HVO. The visible start of injection is very similar for all fuels, but slightly advanced for B0 and HVO fuel. However, it should be mentioned that the difference in the start of injection will probably be negligible for engine tests as at engine speeds of 1500 rpm the difference is about 0.13 CAD between two images.



Figure 5-8: Spray penetration curve at 100 °C chamber temperature and 70 bar backpressure.

At 100 °C chamber temperature, the fuel temperature was approximately 80 °C as the fuel injector heated up by heat transfer from the vessel. With higher fuel temperature, the fuel viscosity decreases and fuel density increases, favouring penetration velocity and cone angle development. Figure 5-8 shows that the five fuels have very similar penetration curves and the poor injection characteristics of PME and UCOME erased. The start of visible fuel injection retarded by less than 0.03 ms for each fuel, except HVO where the fuel injection advances by around 0.014 ms compared to 25 °C. The standard deviation (STD) of the penetration was not included in the graphs, but the averaged STD of each fuel is illustrated in Table 5-4.

Table 5-4: Average S ID of different test fuels.				
Fuel	25°C	100°C		
B0 Reference Diesel	±0.958 mm	±0.858 mm		
HVO B100	±1.012 mm	±0.655 mm		
PME B100	±1.972 mm	±0.884 mm		
SME B100	±0.969 mm	±0.812 mm		
UCOME B100	±0.590 mm*	±0.938 mm		

Table 5-4: Average STD of different test fuels

*only three data sets available

The values in the table indicate that at lower temperature the STD increased due to higher fuel viscosity causing a more unstable injection. Especially the STD of PME fuel at 25 °C was almost twice as high as the other fuels.

As the start of the visible fuel injection differed with the different fuel types, the start of injection was referenced to zero for better spray comparison. Figure 5-9 shows the spray tip penetration of mineral diesel and the four neat biofuels (B100 series) for the high injection pressures of 1800 bar (only every fifth data point has been plotted to have a better distinction between the curves).



Figure 5-9: Spray penetration at 100 °C and 70 bar backpressure referenced to zero.

The high density fuels, SME, PME and UCOME have longer tip penetration than B0 diesel fuel and the spray penetration of HVO was lower due to the lower fuel density. This is in agreement with the work of other researchers investigating the macroscopic characteristics of biofuels [204, 263, 264].



Figure 5-10: Spray evolution of various biofuels at 25 °C chamber temperature.

Figure 5-10 and Figure 5-11 present the spray evolution process of all tested fuels at 1800 bar injection pressure and 60 and 70 bar backpressure, respectively.

The poorer injection characteristics of PME and UCOME at 25 °C can be noticed and after 1 ms the spray is already broken up from the nozzle. This shows that the main reason for less fuel mass injected is not the delayed start of injection, but the earlier end of injection. It can be assumed that as soon as the energising time of the injector solenoid drops the high viscosity fuels stop injecting immediately, while the lower viscosity fuels continue injecting fuel a bit longer.



Figure 5-11: Spray evolution of various biofuels at 100 °C chamber temperature.

At high vessel temperature, the penetration tip is longer for all three FAMEs, PME, SME and UCOME; however at the start of injection, all fuels still have similar penetration trends and the differences appear more with proceeding injection. Also it can be seen that the cone angle becomes constant over time. At 1 ms after injection the three FAMEs are already broken up from the nozzle while HVO and mineral

diesel still show a complete spray plume at the tip of the nozzle. The earlier injection breakup of the three FAMEs is explained with the higher fuel viscosity of biodiesel. Figure 5-11 also indicates that the cone angle and the total spray area has increased compared to 25 °C in Figure 5-10.

5.3.1 Spray tip penetration model

In the past, researchers developed or adopted theoretical models of spray penetration experimental results and reliability to verify evaluate their with the phenomenological spray models [45-47, 204]. Phenomenological models are integrations of reduced physics-based methods and experimental data, in spray applications the combination has resulted in many derivatives of such models with very few differences aside from the relevance of data used in their calibration. One very well established model is that of Hiroyasu and Arai, which was adopted to analyse the experimental penetration curves and to evaluate the validity of this model for the use of different biofuels [46].

This model was selected for further analysis in this work above the many other candidates because historically it is so well established for fuel spray applications (*i.e.* accessibility within CFD codes, textbooks *etc.*), simplicity and its distinctive input parameters. Furthermore, it was considered that whilst other phenomenological spray models [45-47, 204] might also be appropriate, the assumptions, numerical approaches *etc.* are similar enough that the general outcomes and conclusions of this work would be identical.

In the 1970s, Dent and Hay et al. published detailed comparisons of different spray models and concluded that models predicting the relationship between penetration distances to the square root of time give the best accuracy [45, 265]. Although Dent established a model applying the gas jet mixing theory, while Wakuri's model related the spray angle to appropriate physical parameters, both models still incorporate the basic parameters pressure difference, gas density, nozzle diameter and square root of time [44, 45, 266]. It was Hiroyasu and Arai who first applied the two-zone theory saying that the penetration length is proportion to the time, t until the spray breaks up and then advance at a rate proportional to the square root of time. Many recently established spray models have been derived from these initial assumptions that the spray is influenced by the diesel injection system parameters and by the environment

where the spray is injected into. With this many models are more or less derived from this original relationship established by Hiroyasu and Arai and they all carry these basic principles in their codes.

Although the theoretical model was verified at low back pressure and injection pressure only, the general principle of the model was believed to be valid for higher pressures as well. They established a correlation between the spray penetration and the time by considering two effects influencing the penetration process. At the start of the injection the fuel density and injection pressure dominates the spray penetration, while with ongoing injection the air entrainment process in the chamber becomes more dominating. Hiroyasu et al. stated that the time when the chamber conditions become more dominating than the injection conditions is the breakup time t_{break} [46]. The spray penetration up to the breakup time was considered to be a linear function of time, while the penetration after the initial breakup progresses with the square root behaviour. The linear and root function equations are shown in equation (5-3) to (5-5) [46]:

$$S_1 = 0.39 \cdot \left(\frac{2 \cdot \Delta P}{\rho_l}\right)^{\frac{1}{2}} \cdot t \qquad for \ t < t_{break} \tag{5-3}$$

$$S_2 = 2.95 \cdot \left(\frac{\Delta P}{\rho_g}\right)^{\frac{1}{4}} \cdot \sqrt{d_0 \cdot t} \qquad \qquad for \ t \ge t_{break} \qquad (5-4)$$

where
$$S_1 = S_2 \rightarrow t_{break} = \frac{29 \cdot \rho_l \cdot d_0}{(\rho_g \cdot \Delta P)^{\frac{1}{2}}}$$
 (5-5)

S is the penetration (m); t, time (s); ΔP , pressure difference across the nozzle (Pa); ρ_l , liquid fuel density (kg/m³); ρ_g , the ambient gas density (kg/m³), and d_0 the nozzle diameter (m). Using the above equations the breakup time has been calculated and is shown in Table 5-5

Fuel	25°C	100°C
B0 Reference Diesel	0.0354 ms	0.0343 ms
HVO B100	0.0329 ms	0.0318 ms
PME B100	0.0369 ms	0.0364 ms
SME B100	0.0373 ms	0.0366 ms
UCOME B100	0.0370 ms	0.0365 ms

Table 5-5: Breakup time of the tested fuels according to Hiroyasu and Arai.

The comparison of the theoretical model and the experimental data at 1800 bar is illustrated in Figure 5-12 and Figure 5-13 (only every third data point was plotted to have a better distinction between the curves).



Figure 5-12: Experimental spray data and Hiroyasu model for B0 diesel at 1800 bar injection pressure.



Figure 5-13: Experimental spray data and Hiroyasu model for biofuels at 1800 bar injection pressure.

In the Hiroyasu model, S_1 before break-up time seems still valid, but S_2 changes at higher injection and higher back pressure. As the difference between the experimental data and the theoretical correlation is not constant for all fuels, it seems likely that at least one fuel parameter will have a significant effect on the spray penetration. For the linear correlation before break-up time, Hiroyasu and Arai have taken the fuel density into account, but for the root function after break-up time, the fuel density has not been considered. A hypothesis has been established saying that the fuel density will influence the penetration after the break-up time. To prove this new correlation, factors have to be determined and the weight of the fuel density within this equation has to be derived. According to Hiroyasu and Arai the two correlation factors for the root function are:

$$A_1 = 2.95 \cdot \left(\frac{\Delta P}{\rho_g}\right)^{\frac{1}{4}} \cdot \sqrt{d_0}$$

$$B_1 = 0.5$$
(5-6)

Thus the S_2 root function can be written as:

$$S_2 = A_1 \cdot t^{B_1} \tag{5-7}$$

The values for A_1 and B_1 as well as the fuel density at 80 °C have been listed in Table 5-6.

Fuel	Density [kg/m³]	A ₁	B ₁
B0 Reference Diesel	786.5	47.58	0.5
HVO B100	730.0	47.58	0.5
PME B100	835.0	47.58	0.5
SME B100	840.0	47.58	0.5
UCOME B100	838.0	47.58	0.5

Table 5-6: Fuel density of tested fuels at 80 °C and Hiroyasu correlation factors.

The values for A_1 and B_1 are constant for all fuel types, as ΔP , ρ_g , and d_0 do not change when changing fuels. At 1800 bar injection pressure, 70 bar back pressure, 100 °C chamber temperature and 0.16 mm nozzle diameter the values for A_1 and B_1 are 47.58 and 0.5, respectively.

The experimental spray data has been plotted vs time (see Figure 5-9) and a trend line in the format of the power equation (5-7) has been added with the experimental factors A_2 and B_2 .

Table 5-7 shows the correlation factors A_2 and B_2 together with the coefficient of determination R^2 for the trend lines of all fuels. As R^2 is almost 1, it can be stated that the trend lines describing the experimental data curve well.

Fuel	A ₂	B ₂	R ²
B0 Reference Diesel	53.087	0.5386	0.9982
HVO B100	51.337	0.5474	0.9973
PME B100	53.524	0.5333	0.9988
SME B100	53.602	0.5524	0.9984
UCOME B100	53.807	0.5443	0.9984
HVO B50	52.298	0.5389	0.9948
PME B50	53.570	0.5426	0.9870
SME B50	53.203	0.5515	0.9828
UCOME B50	53.427	0.5398	0.9902

Table 5-7: Correlation factors and coefficient of determination of fuel penetration trend lines.

In the next stage a correlation between the new factors A_2 and B_2 and the old factors A_1 and B_1 has to be established. The B_2 factor seems constant and just distributed randomly around a fixed value, thus no fuel parameter dependency is assumed at this stage. However, for A_2 it can be seen that HVO has the lowest parameter of 51.3 followed by diesel of 53.1 and 53.5 to 53.8 for the three neat FAMEs, respectively. The A_2 value for the B50 series are always between B0 and the equivalent B100 fuel. The density dependency has been expressed in the following equation:

$$A_2 = A_1 \cdot X_2 \cdot \rho^z \tag{5-8}$$

where z is the weighting factor of the fuel density effecting the penetration and X_2 is a proportional factor to compensate the higher injection pressure and chamber density. Rearranging and linearising the equation and taking the 47.58 for A_1 and A_2 values from Table 5-7 the equation can be written as:

$$\frac{A_2}{A_1} = X_2 \cdot \rho^z \tag{5-9}$$

This equation with the ratio of A_2/A_1 has to be valid for all fuels in Table 5-8 and one constant for X_2 and z representing the whole range of fuels has to be determined. A common way to solve this equation is a graphical solution by converting it to a linear equation with the slope m and the axis intercept as the two unknown variables. As equation (5-9) is an exponential function it can be converted to a linear equation by taking the natural logarithm:

$$ln\left(\frac{A_2}{A_1}\right) = ln(X_2) + z \cdot \ln(\rho)$$

$$y = b + m \cdot x$$
(5-10)

Fuel	ln(ρ)	A ₂ /A ₁	In(A ₂ /A ₁)
B0 Reference Diesel	6.681	1.1157	0.1095
HVO B100	6.602	1.0790	0.0760
PME B100	6.721	1.1249	0.1177
SME B100	6.731	1.1266	0.1192
UCOME B100	6.725	1.1309	0.1230
HVO B50	6.642	1.0992	0.0945
PME B50	6.701	1.1259	0.1186
SME B50	6.706	1.1182	0.1117
UCOME B50	6.703	1.1229	0.1159

Table 5-8: Linearised x- and y-values for the graphical equation solution.

This linear equation is plotted for all fuels in Table 5-8 with $\ln(A_2/A_1)$ as the y-value and $\ln(\rho)$ as the x-value. Figure 5-14 shows the graphical solution and a linear trend can be clearly noticed.



Figure 5-14: Graphical solution of the density dependency on spray penetration.

The trend line has a coefficient of determination of 0.9481, which is considered high and can sufficiently indicate a linear correlation. The slope of the trend line is 0.3387 and the y-axis intersect is -2.1562. Thus z and X_2 can be determined as:

$$m = z = 0.3387 \tag{5-11}$$

$$b = ln(X_2) \to X_2 = e^{-2.1562} = 0.1158$$
 (5-12)

The new and modified penetration equation can now be written as:

$$S_{2} = 2.95 \cdot \left(\frac{\Delta P}{\rho_{g}}\right)^{\frac{1}{4}} \cdot \sqrt{d_{0}} \cdot 0.1158 \cdot \rho^{0.3387} \cdot t^{0.5432}$$

$$for \ t \ge t_{break}$$

$$(5-13)$$

The exponential factor for the time has changed from 0.5 to 0.5432 and is the average value of B_2 in Table 5-9.

Fuel	Real A ₂	Calculated A ₂	B ₂
B0 Reference Diesel	53.087	52.941	0.5386
HVO B100	51.337	51.549	0.5474
PME B100	53.524	53.672	0.5333
SME B100	53.602	53.862	0.5524
UCOME B100	53.807	53.738	0.5443
HVO B50	52.298	52.254	0.5389
PME B50	53.570	53.309	0.5426
SME B50	53.203	53.405	0.5515
UCOME B50	53.427	53.342	0.5398
Average:			0.5432

Table 5-9: New correlation factor for modified Hiroyasu and Arai spray model.

The new factors A_2 and B_2 are now valid for all fuels at higher injection pressures and chamber densities of around 1800 bar and 60-70 kg/m³, respectively. The new spray models have been plotted in Figure 5-15 and Figure 5-16 for B0 diesel fuel and the four neat biofuels (only every third data point has been plotted to have a better distinction between the curves). It can be concluded that the modified model matches the experimental data very well for different types of fuels with changing fuel densities.



Figure 5-15: Comparison between experimental spray data and modified Hiroyasu model for B0 diesel.



Figure 5-16: Experimental spray data and modified Hiroyasu model for various biofuels at 1800 bar.

With having a new mathematical equation for the second stage of the penetration, the breakup time defined by the intersection of the linear and the root equation has changed and can be calculated based on the following equation:

$$t_{break} = 0.5392 \cdot \left(\frac{\sqrt{d_0 \cdot \left(\frac{\Delta P}{\rho_g}\right)^{\frac{1}{4}} \cdot \rho_l^{0.3094}}}{\left(\frac{\Delta P}{\rho_l}\right)^{\frac{1}{2}}} \right)^{2.1891}$$
(5-14)

Figure 5-17 shows the breakup time of the test fuels using the original and the modified penetration model. The spray breaks up quicker in the modified model and especially HVO B100 and HVO B50 with low densities tend to break up much earlier than the FAME fuels.



Figure 5-17: Breakup time of various fuel for original and modified spray model.

For validation purposes the model has been applied to the spray tests at low injection pressure of 600 bar. As nozzle geometry, fuel density and chamber density are constant throughout the tests just ΔP changes from 1730 bar to 530 bar giving a new, smaller pre-exponential factor for A_2 . The results of experimental and modelled penetration data have been shown in Figure 5-18 for B0 and PME, representatively for all fuels.



Figure 5-18: Comparison of original and modified spray model at 600 bar injection pressure.

5.4 Spray cone angle

Figure 5-19 presents the spray cone angle for the five test fuels at 1800 bar injection pressure, 60 bar backpressure and 25 °C fuel and air temperature. Each curve shows the average values over five test repetitions.

The cone angle of PME stands out while the other fuels have quite stable angles after the initial stage of the spray. As already mentioned earlier the high viscosity of the PME seems to have a very negative effect on the spray atomisation. UCOME and PME both have a slow rise in cone angle at the very initial stage of the spray starting at around 12 degree while the other fuels are show spray angles of around 20 degree straight away. One possible explanation is again the high viscosity which does not allow much atomisation at the very beginning resulting in a narrow jet injecting into the chamber.



Figure 5-19: Spray cone angle evolution of tested fuels at 25 °C chamber temperature.

Figure 5-20 presents the spray cone angle for the five test fuels at 1800 bar injection pressure, 70 bar backpressure and 100 °C chamber temperature and 80 °C fuel temperature. All tested fuels show a very similar cone angle with HVO being slightly higher. UCOME, PME and SME fuel show a slow rise in cone angle just after the start of injection. A jet-shaped spray at the very beginning is assumed here caused by the higher viscosity of the fuel.



Figure 5-20: Spray cone angle evolution of tested fuels at 100 °C chamber temperature.

Figure 5-21 compares the average cone angle from the start to the end of injection of all five test fuels at 25 °C and 100 °C chamber temperature. At the higher temperature the respective cone angle is higher for all fuels except mineral diesel.



Figure 5-21: Average spray cone angle at high and low fuel temperature and 1800 bar injection pressure.

Also the average cone angle for the three FAMEs is clearly smaller at both temperatures than HVO which is in line with the findings Valentino et al. and Guan et al. saying that higher fuel viscosities lead to a decrease in spray cone angle [264, 267].

Fuel	600 bar	1800 bar
B0 Reference Diesel	20.03	21.56
HVO B100	21.33	22.29
PME B100	19.81	21.1
SME B100	20.28	20.88
UCOME B100	19.89	21.62

 Table 5-10: Average spray cone angle at high and low injection pressures.

Table 5-10 presents the average cone angle of all neat fuels for 600 bar and 1800 bar injection pressure at 100 °C. With higher injection pressures the cone angle increases slightly. The higher pressure causes higher kinetic energy in the spray and smaller fuel droplets are formed, which are more likely to atomise and scatter in the chamber. At 600 bar injection pressure HVO provides the highest cone angle of all

fuels followed by the three FAMEs and B0 with very similar cone angles of about 20 degree. The high cone angle of HVO can be explained with the low viscosity of the fuel.

5.5 Spray area

Figure 5-22 presents the projected spray area of all five test fuels at 1800 bar injection pressure and 25 °C air temperature. The spray area is an indicator for the quantity of the fuel-air mixing process as a higher spray area means more surface contact between fuel and surrounding air. The projected spray area for PME breaks up very early during the injection stage followed by UCOME resulting in more stratified air-fuel mixture. While PME shows a very narrow cone angle at this injection condition causing the smaller spray area, the main factor for UCOME is the shorter penetration tip length as the cone angle of UCOME was similar to the other fuels. HVO and B0 show the highest projected spray area with linear increase over time. Both fuels have a low viscosity at 25 °C, which leads to better spray atomisation and therefore better fuel-air mixing. SME shows the best spray behaviour of the three biodiesel, but its atomisation characteristics is still lower than B0 mineral diesel and HVO fuel due to the higher viscosity resulting in a narrower spray angle.



Figure 5-22: Spray area of tested fuels at 25 $^\circ \! C$ chamber temperature.

Figure 5-23 presents the projected spray area of all five test fuels at 1800 bar injection pressure and 100 °C chamber temperature. With higher fuel temperature the viscosity of all fuels reduced significantly showing a linear trend in spray area over time for all fuels. Thus it can be concluded that all fuels have similar air-fuel mixing properties for the same injection conditions. While the three FAMEs and B0 achieve a large contact area of fuel and surrounding air by a long spray tip penetration (see Figure 5-9), HVO fuel achieves the same contact area by having a larger cone angle than the other fuels (see Figure 5-20).



Figure 5-23: Spray area of tested fuels at 100 °C chamber temperature.

However, in an internal combustion engine the injection conditions will not be identical for all fuels as different densities and heating values cause different amount of fuel energy to be injected at the same injection duration and pressure. The fuel quantity injected at these injection conditions has been measured earlier on during the injector calibration. With knowing the calorific value of each a fuel the total injected energy has been calculated and a linear injection of mass versus time over the full injection duration has been assumed. As the injection duration is 0.6 ms the injected energy increases from 0 at the start of injection linearly to the full amount after 0.6 ms. This is not the most accurate assumption as the injection response has not been considered and the actual injection duration might be different to the 0.6 ms. However, the injection delay is very similar for all fuels when a common-rail

system is used and the final ratio of area per unit energy is still true and represents real engine conditions. The injected mass and energy of all test fuels are shown in Table 5-11.

Fuel	Injected mass [mg]	Calorific value [kJ/kg]	Injected energy [J/stroke]
B0 Reference Diesel	6.32	42,853	271
HVO B100	5.61	43,902	246
PME B100	7.00	37,320	261
SME B100	6.97	37,230	259
UCOME B100	6.92	37,200	257

Table 5-11: Injected mass and energy of different fuels at 1800 bar and 0.6 ms injection duration.

Figure 5-24 presents the spray area per Joule versus the time for 1800 bar injection pressure, 70 bar backpressure and 100 °C chamber temperature. At the start of the injection, all fuels follow the same trend and with commencing fuel injection the specific projected spray area develops faster for HVO followed by the three FAMEs and is less distinctive for B0.



Figure 5-24: Energy specific projected spray area for tested fuels at 1800 bar.

The main reason for HVO showing the largest energy specific spray area is the low density resulting in less fuel and therefore less energy being injected at a set duration,

although the calorific value of HVO is very high. The energy specific spray area of conventional diesel is least developed as the density of the fuel is very high resulting in a high injection volume and mass and the high heating value further increases the effect as less fuel is required on an engine to achieve the same power output as biodiesel with a higher specific spray area.

Furthermore, with the spray area being an indicator for the quantity of the fuel-air mixing process it can be concluded that HVO shows a very high tendency for a good initial fuel-air mixture resulting in a high premix portion when the fuel ignites, which increases NO_x emissions and reduces PM formation. At this point it should be noted that this is just a simplified illustration as in reality the fuel concentration of the spray is radially distributed going from its highest value at the centre to its minimum at the spray edge. The high CN of HVO, however, will work against this effect as an earlier ignition causes provides less time for the premixing. For SME, PME and UCOME the trend will be similar and the good higher ratio of spray area per unit energy would cause stronger premix combustion, but the high viscosity of the fuels results in larger fuel droplets diminishing the fuel-air mixture and therefore the premix combustion. SME has the lowest viscosity of all three FAMEs, followed by UCOME and PME and also SME shows the highest area per unit energy ratio in Figure 5-24 followed by UCOME and PME, respectively. This is in agreement with the comprehensive biodiesel review by Lapuerta et al. showing that higher saturated FAMEs such as PME have higher viscosities and lower NO_x emissions and increasing the rail pressure will reduce the mean droplet size of the fuels resulting in lower NO_x emissions [153]. Further work of the effect of NO_x emissions for different biodiesel has been carried out by the authors and SME showed highest NO_x emissions followed by UCOME and PME indicating the same trend as shown in Figure 5-24 and a correlation between fuel viscosity, saturation level, fuel-air mixing quality and NO_x emissions can be assumed.

Chapter 6. Engine power restoration and optimisation of biofuels

When using various biofuels with different fuel properties, the engine performance and exhaust emissions differ from the original certification obtained using mineral diesel fuel. Differences in heating value and fuel density are mainly responsible for changes in power output, while the fuel's chemical structure, CN and viscosity are the main sources of differences in exhaust gas emissions. As engine manufacturers go through a time and cost intensive process to certify an engine for a specific power output, it is important that with the use of alternative fuels the same power output can be obtained. Since most biofuels have a lower LHV than mineral diesel fuel, full power is often not achievable with these fuels. Therefore engine restoration is important to ensure that the full-load curve can be met using biofuels. The exhaust emissions are likely to be different to the original engine certification and emissions above the benchmark could potentially cause problems in the aftertreatment system. Small differences in exhaust emissions can be compensated by an aftertreatment system, but large differences can cause severe losses in the exhaust gas conversion efficiency and/or exhaust gas filters are more likely to be blocked. With an optimised injection strategy the emissions can be controlled, which often means that some emissions will be reduced while others might increase. The most popular example is the PM-NO_x trade-off, where one side can be reduced to the disadvantage of the other. Mani et al. optimised the injection timing of a single cylinder four stroke diesel engine running on waste plastic oil and reported a reduction in NO_x emissions when retarding the injection [268]. Similar observation has been made by Ganapathy et al. resulting in lower NO_x, but higher BSFC, CO, THC and PM emissions when the fuel injection is retarded [269]. When advancing the injection the opposite effect was found. Pierpont and Reitz changed the rail pressure when testing mineral diesel on a Caterpillar heavy-duty engine and concluded that an increase in rail pressure increased NO_x emissions and vice versa [107]. They further noted that retarding the injection timing will shift the NO_x -PM trade-off curve towards lower NO_x emissions.

When optimising the fuel injection, many parameters are available resulting in various combinations of design parameters. Two of the most effective injection parameters are the start of the main injection (SOI) and the rail pressure (RP). Both

have a significant impact on engine power performance and exhaust emissions [270, 271].

For the restoration tests, SME was used and an attempt was made to restore full power by fuel injection optimisation. According to the results in Chapter 4 the power loss was approximately 10 % with all three FAMEs. If power can be restored successfully using SME, then power might also be restored for PME and UCOME. For the optimisation tests, HVO and SME have been chosen. Chemically HVO is different from the other three biodiesel blends and therefore the optimisation of HVO was particularly interesting and additionally very few engine tests have been carried out using HVO. SME was chosen to represent the three FAMEs and since the results of the exhaust emissions in Chapter 4 revealed that the combustion of SME caused the highest emissions of NO_x and PN, the successful optimisation of SME would show that exhaust emissions might also be optimised for PME and UCOME fuel.

6.1 Engine restoration with SME B100

The drop of full power for biodiesel has already been addressed in section 4.2.1 showing that the lower volumetric heating value of all biofuels requires injection of more fuel on a volumetric basis leading to longer injection durations. Within the engine management system, a standard calorific value and fuel density is stored, which represents mineral diesel and based on these two parameters the generated engine torque is calculated by an empirical solution including the rail pressure and injection duration of the injection events. However, there are two possible solutions to overcome the issue of underestimating the indicated engine torque and preventing it from generating full power with fuels having lower LHV. The best practise solution is to have two flexible parameters for fuel density and LHV in the engine management system to calculate the required fuel quantity and corresponding engine torque. This method, however, requires reliable input signals of density and LHV to find the optimum injection conditions on the engine map.

As mentioned in section 3.2.3 Continental have developed a Fuel Quality Sensor (FQS) to continuously measure the main fuel parameters, such as biofuel percentage, heating value, density and aromatics content with near-infrared (NIR) spectroscopy. The sensor can be mounted in-line with the existing fuel pipe and every 90 seconds an absorption spectrum of the fuel is acquired which then is compared with a huge

database of absorption spectra on a chip in the sensor and the correct parameters are given out within better than 10 % error. A prototype of this sensor has been provided by Continental together with the software and a feasibility test has been carried out testing the different biofuels and blend ratios.

more o 11 De mutor of 1 Q5 outputs from feur mines					
PME	Bio blend ratio	Cetane number	Fuel density	LHV	
B10	31 %	-9 %	2 %	2 %	
B20	13 %	-8 %	2 %	2 %	
B50	13 %	-4 %	2 %	-2 %	
B100	3 %	9 %	1%	N/A	

Table 6-1: Deviation of FQS outputs from real values

Table 6-1 shows the FQS results with all PME blends in comparison with the fuel properties provided by BP. At all blend ratios, the detected CN and fuel density were within a 10 % error band. The LHV was measured accurately for blends of B10 to B50, but no data were measured for B100. Reasons for not measuring the LHV at B100 are probably the insufficient data base, which needs to be fed by Continental with more B100 absorption spectra. The detection of the biofuel percentage was inaccurate at lower biofuel blend ratios, but this value is not compulsory for engine restoration and has been neglected. As the practical implementation of this design was not in the scope of this project, the engine restoration has been carried out by temporarily lifting the torque tables by 15 % to overcome the miscalculated indicated torque values. As shown in Figure 4-19, the calculated torque values were 11 % higher than the real torque values across all engine loads. The three red coloured test points in Figure 6-1 were attempted to be restored. The engine test setup and data logging procedure was identical to the engine tests in Chapter 4.



Figure 6-1: Restoration test points (red) of ESC test cycle.

SME B100 was used for the restoration tests, but any other biodiesel fuel with similar fuel density and LHV value can be used. Table 6-2 shows the three full load points with their maximum torque values. The maximum torque was therefore lifted from 730 Nm to 840 Nm for point #1 and #7 and from 650 to 748 Nm for test point #9.

Test Point	Engine speed [rpm]	Max torque [Nm]	ECM torque limit (+15 %) [Nm]
#1	1490	730	840
#7	1855	730	840
#9	2220	650	748

 Table 6-2: Three full load points with their full torque values and overwritten torque limit (right column)

Figure 6-2 shows the restoration results for all three full load points in comparison with B0 diesel fuel and the original maximum torque of SME. At low engine speed (1490 rpm), half the power loss was restored resulting in a power drop of only 4.1 %. At medium engine speed (1855 rpm), the engine torque increased from 680 Nm to 703.5 Nm using SME fuel and the resulting power loss after restoration was reduced to only 3.6 %. The smallest power drop was achieved at high engine speed (2220 rpm), increasing the torque from 590 Nm to 640 Nm equivalent to 98.5 % of maximum torque with mineral diesel.



Figure 6-2: Comparison of generated torque before and after restoration.

Observations of the engine management data show that the new torque values at low and medium engine speed are equivalent to 790 and 800 Nm, respectively. It seems that another safety measure was implemented in the ECU to prevented overloading the engine beyond 800 Nm. The specific fuel consumption of the restored full load points are shown in Figure 6-3. After torque restoration, the BSFC increases by about 1.2 % at low and medium engine speed and 1.6 % at high engine speed. An increased BSFC is expected as the injection duration was prolonged and reached further into the expansion stroke. Thus, the diffusion combustion phase increases and energy is lost by hot gas entering the exhaust stream instead of converting into useful mechanical power. The relative increase of the BSFC between the restored figures with SME and B0 diesel benchmark are therefore 10.7 %, 12.7 % and 11.4 % for low, medium and high engine speed, respectively. The mass-based LHV of SME B100 is 13.1 % lower than B0.



Figure 6-3: Comparison of specific fuel consumption before and after restoration.

The specific NO_x and PN emissions are shown in Figure 6-4 and Figure 6-5. Across all three engine speeds, the specific NO_x emissions reduced with SME when power was restored. The reason for the reduction is found in the HRR shown in Figure 6-6 and Figure 6-7. When restoring engine power, the injection duration increases at constant rail pressure. The initial injection parameters, such SOI and RP, were left unchanged, resulting in the same fuel-air mixing process and ID. Thus, the initial rise in HR and peak HRR was identical for original and restored SME combustion as illustrated in Figure 6-6. This is in line with the findings in section 5.2 showing that longer injection durations show no measurable effect on the initial spray patterns or fuel-air mixing process. The lower specific NO_x value results from the higher engine power output due to the longer fuel injection towards the end of the power stroke. At medium and high engine speed the NO_x emissions of the restored SME tests were slightly higher than the B0 benchmark, whereas the NO_x emissions at low engine speed is lower. The lower NO_x measurement of 6.22 g/kWh for restored SME at low engine speed could be caused from the lower temperature and higher humidity of the intake air at the beginning of the tests. While the B0 benchmark and SME original tests have been carried out at a steady-state engine cell environment with constant intake air temperature and humidity, the full load restoration tests were carried out at different ambient air conditions as the cooling system was struggling to maintain constant temperatures at full power. Therefore restoration tests were carried out at colder engine cell conditions, especially at low engine speed.



Figure 6-4: Comparison of specific NO_x emissions before and after restoration.



Figure 6-5: Comparison of specific PN emissions before and after restoration.

The PN emissions of the restored SME tests follow the opposite trend to the NO_x emissions. The specific PN emissions increased with restored power and longer fuel injection. The HRRs in Figure 6-6 and Figure 6-7 show that the start of the injection is unchanged for all three engine speeds, but injection duration of the main injection was prolonged for the restoration tests. The HRRs of the original SME and restored SME are very similar until about 20 CAD aTDC when the restored curve deviates from the original curve towards the B0 diesel combustion profile.



Figure 6-6: Comparison of heat release rate before and after restoration at low engine speed.



Figure 6-7: Comparison of heat release rate before and after restoration at high engine speed.

It can be concluded that power can be restored to achieve more than 95 % of the original power output when using biodiesel fuel. A hidden torque limiter is prohibiting full recovery due to safety interlocks in the ECU as incorrect torque values are calculated based on mineral diesel fuel density and LHV. Inputting flexible fuel density and LHV parameters into the engine management system might solve this issue to obtain full power recovery. The changes in NO_x and PN emissions did not provide any additional challenges as specific NO_x emissions were reduced, while the increased PN emissions were still lower than the benchmark B0 values.

6.2 Engine optimisation of SME B100

For the engine optimisation, test points #6, #2 and #3 have been chosen out of the ESC (see Figure 6-8) to represent low engine load, medium engine load, and high engine load, respectively.

For each test point a Design of Experiment (DoE) was created by varying fuel rail pressure (RP) and start of the main fuel injection (SOI). Both parameters were overwritten with Calterm III, the ECU software provided by Cummins. Altering the injection pressure will affect the fuel spray penetration and atomisation. Some fuel properties of SME, such as density and viscosity, are different to B0 and therefore the spray characteristics of the fuel injection are different with SME. The injection pressure has been reduced by 150 and 300 bar of its original value. The reason for reducing is that a drop in NO_x emissions is desired and based on the findings of other researchers it can be expected that lower injection pressures will cause lower NO_x emissions [107].

The start of injection (SOI) has been advanced and retarded by ± 1.5 and ± 3 CAD, respectively. By changing the main SOI, the dwell time between pilot and main injection was kept constant and the start of the pilot injection was changed according to the main SOI. The SOI is one of the major injection parameters affecting the incylinder pressure and heat release characteristics. Advancing the injection timing means that fuel is injected earlier into the chamber, resulting in lower cylinder pressure and temperature at the time of injector excitation. This will have significant effects on the spray penetration and atomisation characteristics investigated in Chapter 5. Retarding the injection has the opposite effect and fuel is injected closer to TDC, where cylinder pressure and temperature are higher. This will cause the fuel spray to penetrate less into the chamber, but spread wider in the radial direction providing a better fuel-air mixture. However, it can be expected that due to the higher temperature, the fuel-air mixture will ignite faster thus resulting in a shorter ignition delay.



Figure 6-8: Optimisation points (red) of ESC.

Figure 6-9 shows the full DoE with all 15 test points in a randomly distributed order. Each test point was run for 2 minutes to obtain stable fuel consumption and emissions data. The set of 15 test points has been repeated three times.



Figure 6-9: Design of experiment of the optimisation tests.

Table 6-3 shows the actual rail pressures (in bar) and SOI (in CAD bTDC) of all three test points.

Poi	nt 6	Point 2		Poi	nt 3
RP	SOI	RP	SOI	RP	SOI
1000	2.45	1210	4.2	1450	8.0
1150	2.45	1210	5.7	1600	6.5
1000	0.95	1360	4.2	1450	6.5
1150	5.45	1060	4.2	1300	3.5
850	2.45	1360	2.7	1300	8.0
1000	6.95	1360	8.7	1450	3.5
1000	5.45	1210	2.7	1300	5.0
1150	3.95	1210	8.7	1300	6.5
1150	0.95	1060	8.7	1600	5.0
1000	3.95	1360	5.7	1450	5.0
850	6.95	1210	7.2	1600	3.5
850	5.45	1060	2.7	1450	2.0
1150	6.95	1060	5.7	1300	2.0
850	3.95	1360	7.2	1600	2.0
850	0.95	1060	7.2	1600	8.0

Table 6-3: Randomly distributed test points of DoE.

The main focus of the engine optimisation was to understand the effects on NO_x and particulate number (PN) as they are both most significant in diesel engines. The THC and CO emissions have been measured, but not included in this chapter, since their emissions levels were not critical.

6.2.1 NO_x emissions with varying RP and SOI

Figure 6-10 illustrates the specific NO_x emissions with varying SOI at the three test points. The error bars are two standard deviations around the mean value giving a confidence level within this range of 95.5 %. A linear trendline has been fitted through the points showing a good correlation between the NO_x emissions and the start of injection.



Figure 6-10: Specific NO_x emissions with varying start of injection.

When retarding the SOI, the NO_x emissions were reduced. This can be explained using the peak pressure in Figure 6-11. When the injection retards the peak pressure shifts further away from TDC into the expansion stroke. This reduced the peak pressure and cylinder temperature. As the NO_x formation rate is highly dependent on the temperature, retarding the injection reduces the specific NO_x emissions for SME.



Figure 6-11: Influence of peak pressure on NOx formation.

Using the results obtained in Chapter 5, the macroscopic fuel spray behaviour will change with changes in the injection timing. When the injection advances, fuel injection occurs earlier in the compression stroke where the cylinder air density is lower. Thus the spray will penetrate further and the cone angle will be narrower. Retarding the injection means that at the time of fuel injection the piston is closer to TDC and the air pressure and density is reaching its maximum. A shorter spray penetration, but wider cone angle is expected resulting in a better air-fuel mixture. The better fuel-air mixing as well as the higher cylinder temperature results in faster fuel ignition and shorter ID as shown in Figure 6-12. This is another indication of lower NO_x emissions at retarded injection.



Figure 6-12: Effect on start of injection on ignition delay.

The specific NO_x emissions with varying RP are presented in Figure 6-13. When the RP was reduced, the NO_x emissions decreased. At lower injection pressure the fuel spray contained less kinetic energy to atomise and the fuel evaporation rate reduced. Therefore, the fuel-air mixing process was slower and at the start of auto ignition less premixed gas was available to burn rapidly causing NO_x to form. Also, at lower injection pressures injection rate was lower and less fuel was injected at the time of ignition.



Figure 6-13: Specific NO_x emissions with varying rail pressure.

The error bars show two standard deviations around the mean value and a linear trend with decreasing RP can be observed. To be able to predict the NO_x emissions the linear trendlines have been used to fit a model as a function of SOI and RP. The two linear trendlines have the format of equation (6-1) and (6-2).

$$[NO_x]_{SOI} = A \cdot \Delta SOI + B \tag{6-1}$$

$$[NO_x]_{RP} = A \cdot \Delta RP + B \tag{6-2}$$

The coefficients of determination are shown in Figure 6-10 and Figure 6-13 for all three test points. The linear fit is very good for all test points (R^2 above 0.9), except the NO_x vs RP correlation at point #2 with an R^2 value of 0.63.

Specific NOx emissions [g/kWh] at point #2 - raw data

Specific NOx emissions [g/kWh] at point #2 - modelled



Figure 6-14: Specific NOx emissions at its fitted model at point #2.

In Figure 6-14 the raw NO_x data and the fitted model for point #2 are presented in a contour plot showing that the general trend of higher NO_x with higher rail pressure and advanced injection can be represented well with linear trendlines. The contour plots for point #6 and #3 are very similar and the R² value even proved to have a better fit.

6.2.2 PN emissions with varying RP and SOI

Figure 6-15 illustrates the specific PN emissions of the three test points with varying SOI. The error bars are two standard deviations around the mean value giving a confidence level within this range of 95.5 %. As a linear trendline was not very accurate, a quadratic trendline has been fitted through the points showing a good correlation between the PN emissions and the SOI.



Figure 6-15: Specific PN emissions with varying start of injection.

No clear trend towards advanced or retarded injection was observed for all different test points. While at low engine load (point #6) the tendency goes towards higher PN emissions when retarding injection, the opposite effect can be observed at high load (point #3) emitting more PN when injection is advanced. The explanation for this trend can be found when looking into the in-cylinder pressure data below.

Figure 6-16 presents the PN emissions in relation to RP. When injecting fuel at higher pressure, the fuel breaks up faster into smaller droplets and the air-fuel mixing process improves. However, the relationship is not linear and it seems that when reducing the RP the PN emissions increase non-linearly. This is in agreement with the study of Su et al. investigating the effect of injection pressure on spray SMD and particulate emissions. They found that the correlation between injection pressure and SMD is linear, but the effect of SMD on particulate emissions is non-linear and follows a quadratic or exponential trend [272].


Figure 6-16: Specific PN emissions with varying rail pressure.

Table 6-4 shows the constants of the quadratic trendlines when varying SOI and RP at all three test points. The trendlines have the format of equation (6-3) and (6-4) and the coefficient of determination is above 0.95 for all test points. The trendlines of the PN vs RP curve is actually overfitted as three data points are necessary for a second order polynomial fit, but as explained above, a quadratic trend seems likely based on other researchers work.

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PN emissions [#/kWh]		A1	A2	В	R ²
Point #6	SOI	4.811	5.7 ¹²	2.5 ¹³	1.00
	RP	3.7 ⁸	-6.5 ⁹	1.213	1.00
Point #2	SOI	-7.9 ¹¹	1.3 ¹²	3.4 ¹³	0.99
	RP	3.3 ⁸	-3.110	1.413	1.00
Point #3	SOI	-1.111	-2.212	1.4 ¹³	0.99
	RP	7.87	-1.910	7.8 ¹²	1.00

Table 6-4: Equation parameters and R² value of quadratic trendlines for PN emissions.

$$[PN]_{SOI} = A1 \cdot \Delta SOI^2 + A2 \cdot \Delta SOI + B \tag{6-3}$$

$$[PN]_{RP} = A1 \cdot \Delta RP^2 + A2 \cdot \Delta RP + B \tag{6-4}$$

The contour plots of the PN emissions for all tree test points and their modelled plots are presented in Figure 6-17 to Figure 6-19.



Figure 6-17: Specific PN emissions and its fitted model at point #6.

At point #6 (low load) the PN emissions increase with decreasing RP and retarding SOI. At point #2 the highest PN emissions are formed around the original SOI and low injection pressures. At high load the maximum PN emissions occur at advanced injection timing and low RP.



Figure 6-18: Specific PN emissions and its fitted model at point #2.



Specific PN emissions [#/kWh] at point #3 - raw data

Specific PN emissions [#/kWh] at point #3 - modelled

Figure 6-19: Specific PN emissions and its fitted model at point #3.

Particulate emissions are mainly formed in the diffusion combustion phase, which is taking place after the premix fraction has been burned [273]. Since the diffusion combustion phase lasts usually much longer than the premixed combustion phase, it was assumed that the diffusion combustion is proportional to the combustion

duration (CA90 – CA05). Figure 6-20 shows the combustion duration across the design of experiment and a relationship between the length of the combustion process and the PN emissions can be observed.



Figure 6-20: Comparison of combustion duration of point #6, #2 and #3.

Figure 6-21 shows an example of the HRR at point #3 when changing the RP. For all three RP the start and shape of the pilot HRR are identical. Also the ID for the main injection is constant and fuel ignites at the same SOC for all three RPs. At higher injection pressure the rise of heat release after start of main combustion is much sharper and the peak HRR is higher. This indicates that at higher pressures the initial fuel-air mixing rate is stronger due to smaller fuel droplets and better distribution of the droplets in the chamber. This favours the NO_x emissions and reduces PN. At lower RPs the HRR lasts longer causing a longer diffusion combustion phase. The longer HRR can be explained with the smaller injection rate at lower pressure meaning that more time is required to inject the same volume causing longer needle opening times. This promotes the formation of PN as towards the end of combustion fuel is injected in a cooler and oxygen leaner environment. Both conditions prohibit the breakdown and oxidation of particulates.



Figure 6-21: HRR at point #3 when varying RP.

Figure 6-22 shows an example of the HRR at point #3 when the SOI changes. The start of the HRR shifts with the change of SOI without majorly affecting the shape of the curve. Although the shape of HRR does not change, the cylinder pressure and temperature increase with advanced injection and the formation of NO_x and PN changes.



Figure 6-22: HRR at point #3 when varying SOI.

With having the linear NO_x and quadratic PN model it is possible to determine the most efficient operating point across the design space. A MATLAB programme was used to scan across the design space to find the respective lowest NO_x and PN emissions at various operating conditions. The minima of NO_x and PN form the green NO_x -PN trade-off curve illustrated in Figure 6-23.



Figure 6-23: NO_x-PN trade-off curve of SME - point #2.

When the injection was advanced, the NO_x emissions increased and PN emissions decreased. Retarding the injection decreased NO_x , but PN increased instead. The RP for all operating points on the green optimisation curve were very close to its original high pressure value. Reducing the RP has a too strong penalty in the PN emissions due to the quadratic trend, while the benefits of reduced NO_x are rather small at lower injection pressures. Thus changing the SOI is the main parameter when optimising fuel injection.

The mineral diesel emission benchmark at standard ECU calibration is represented by the black marker. The red point represents the original SME emissions with the higher NO_x output, but significantly lower PN. The blue marker shows a potential new operating point when changing the ECU calibration. At this point, NO_x emissions can be reduced to levels below the B0 benchmark without high penalties in PN. Figure 6-24 compares the standard calibration of B0 and SME with the newly applied injection strategy for SME. In terms of NO_x , the emissions can be reduced from 5 % above the benchmark to more than 7 % below the benchmark, while PN only increases from 26.6 % to 30.2 %, which is still showing a reduction in particulates of almost 70 %. The increase in fuel consumption is only marginally higher increasing from 14.1 % to 15.5 % above the B0 diesel benchmark.



Figure 6-24: Comparison of NO_x, PN and BSFC of B0, original SME and new SME operation at point #2.

Figure 6-25 presents the HRRs of B0 and SME B100 at standard engine calibration and SME B100 at the 3 CAD retarded injection representing the newly proposed operating point. The start of the pilot injection is very similar although the start of pilot injection is also retarded by 3 CAD for new SME (green curve). At the start of the main combustion the retarded fuel injection of SME is very clear and the HRR curve is shifted further into the expansion stroke.



Figure 6-25: Comparison of HRR of B0, original SME and new SME operation at point #2.

In Figure 6-26 the optimisation curve of point #6 (low engine load) is presented. The shape of the curve is similar to Figure 6-23 and the NO_x emissions also decreased with retarded injection timing while PN emissions increased. The new operation point was set to 2.6 CAD after the original SOI and a new RP of 11 bar below its original value. The small reduction in RP is not causing a significant effect in the exhaust emissions and the slight reduction has been neglected.



Figure 6-26: NO_x-PN trade-off curve of SME - point #6.

At this newly proposed SME operating point (blue marker), the NO_x emissions were reduced by nearly 10 % from being almost 5 % above the B0 benchmark to being 5 % below (see Figure 6-27). The PN emissions were 70 % lower than B0 diesel fuel and increase to be still 52.3 % below the B0 benchmark. This new control strategy also leads to a small increase in BSFC from 13.9 % to 14.8 % higher consumption compared to B0 diesel fuel.



Figure 6-27: Comparison of NO_x, PN and BSFC of B0, original SME and new SME operation at point #6.

Figure 6-28 illustrates the optimisation curve of point #3 (high engine load) and here a more distinct curvature with a local minimum at +2.8 CAD SOI is visible. This minimum in the NO_x -PN trade-off is caused by the trend of PN emissions at high engine loads presented in Figure 6-19.





At this new SME running point, the NO_x emissions are reduced by 7.5 % to meet the benchmark (see Figure 6-29). The PN emissions have been further reduced from around 80 % to 85 % of the particulate number with mineral diesel. However, the



new fuel consumption is about 1.5 % higher with the new strategy compared to the original calibration.

Figure 6-29: Comparison of NO_x, PN and BSFC of B0, original SME and new SME operation at point #3.

It can be concluded that injection strategy optimisation for SME B100 is very promising and the NO_x emissions can be reduced below the set benchmark for all test points. In return the particulate emissions only increased slightly and were still far below the benchmark. At high engine load, the NO_x and PN emission could even be reduced both simultaneously. Due to the reductions in emissions the BSFC increased for all new optimisation strategies by about 1 - 1.5 %. Since the emissions could be reduced below the B0 benchmark for SME, it is expected that PME as well as UCOME can achieve the same target due to emitting lower emissions than SME in the first place. Similar reductions of NO_x and PN and increase in BSFC are expected with these fuels.

6.3 Engine optimisation of HVO B100

The optimisation tests for HVO B100 have been carried similar to the optimisation of SME B100. The same test points #6, #2 and #3 as in Figure 6-8 were chosen with the same test matrix shown in Figure 6-9.

The NO_x emissions with varying SOI and RP are presented in Figure 6-30 and Figure 6-31, respectively, for all test points.



Figure 6-30: Specific NO_x emissions with varying start of injection.

Similar to SME, the NO_x emissions also reduced for HVO when retarding the main injection timing and the same reasons discussed thoroughly in section 6.2.1 are responsible for the trend with HVO fuel. Trendlines were fitted through the data points and a linear correlation between NO_x emissions and SOI was established.



Figure 6-31: Specific NO_x emissions with varying rail pressure.

Similar to SME combustion, the reduction of RP reduced the NO_x emissions using HVO fuel. Linear trendlines were used to represent this trend with acceptable R² values above 0.8 for all three test points. Figure 6-32 shows the contour plot of the NO_x emissions at high load (point #3) in dependence on SOI and RP. The trend in NO_x is very similar to SME in Figure 6-14 and a linear model was chosen to represent the NO_x emissions.

Specific NOx emissions [g/kWh] at point #3 - raw data

Specific NOx emissions [g/kWh] at point #3 - modelled



Figure 6-32: Specific NO_x emissions and its fitted model at point #3.

The models for point #2 and #6 are similar and have therefore not been plotted.

The PN emissions of HVO are presented in Figure 6-33 and Figure 6-34 when changing SOI and RP, respectively.

Similar to SME fuel combustion, the PN emissions did not follow a linear trend, but a second order polynomial equation was selected. At low and medium load (point #6 and #2) the PN emissions increased with retarded SOI, while it decreased at high engine load (point #3). This is in agreement with the findings in section 6.2.2 showing that the trend of PN shifts with different loads.



Figure 6-33: Specific PN emissions with varying start of injection.

When reducing RP the PN emissions increase in a quadratic order. As discussed in section 6.2.2 the SMD increased with lower injection pressures, which makes it more difficult for the particulates to evaporate and oxidise. The slope of the established trendlines were higher at low load meaning that at low engine load the effect of changing RP is stronger than at high load. At all three test points, the RP was reduced by a constant pressure of maximum 300 bar. While at low load, the RP changes from 1150 bar to 850 bar, equivalent to a drop of 26 %, at high load the rail pressure drops from 1600 bar to 1300 bar resulting in a 19 % decrease only.



Figure 6-34: Specific PN emissions with varying rail pressure.

The quadratic trendlines have been used to build a PN model with varying SOI and RP. The PN emissions at point #6 are presented in Figure 6-35 and show a very similar behaviour to SME at the same load condition.



Figure 6-35: Specific PN emissions and its fitted model at point #6.

At point #2 the highest PN emissions are formed around the original SOI and low injection pressures (Figure 6-36). At high load, the maximum PN emissions occur at advanced injection timing and low RP (Figure 6-37).



PN emissions [#/kWh] at point #2 - modelled

Figure 6-36: Specific PN emissions and its fitted model at point #2.

It has been discussed in section 6.2.2, that the PN emissions are mainly formed in the diffusion combustion phase, which is fuel richer and cooler than the premixed combustion phase. The combustion duration is approximately proportional to the diffusion combustion length and the same correlation between combustion duration and PN emissions illustrated in Figure 6-20 is valid for the combustion of HVO. The reason for the changes in combustion duration towards advanced injection at higher engine load cannot be explained at this stage.



Figure 6-37: Specific PN emissions and its fitted model at point #3.

In Figure 6-38 the NO_x -PN trade-off curve for HVO at low load is shown. At advanced fuel injection, the PN emissions reduced while NO_x emissions increased. When retarding the injection, the NO_x emissions reduced and PN increased slowly until the NO_x values reached approximately 10 g/kWh and then the PN emissions started to increase significantly.



Figure 6-38: NOx-PN trade-off curve of HVO - point #6.

The HVO combustion at low load can be optimised by leaving the RP unchanged and retarding the SOI by 0.9 CAD. At this point, the PN and NO_x emissions with HVO are below the benchmark B0 diesel.





Figure 6-39 shows the comparison of the original and newly proposed HVO strategy with B0 diesel fuel. While the NO_x emissions were reduced by approximately 3 %, the PN emissions increased by 20 % to 95.9 % of the B0 benchmark value. As the engine aftertreatment system can deal with a small increase in NO_x emissions,

keeping the original injection strategy is beneficial as the increase in PN emissions is high by only gaining small reductions in NO_x emissions. When advancing the injection timing, the savings in PN can increase even more and investigations have to be made how much margin is available in the aftertreatment system to deal with the increase in NO_x . Advancing the injection by 3 CAD, for example, the resulting NO_x emissions would be 11 % higher than the B0 benchmark, but the PN emissions would only be 34 % of the benchmark and fuel savings of about 2 % can be made. Engine manufacturer might want to consider how to deal with the higher NO_x emissions, but potentially reduce the size of the particulate aftertreatment solution, such as diesel particulate filter (DPF).



Figure 6-40: NOx-PN trade-off curve of HVO - point #2.

Figure 6-40 shows the NO_x-PN trade-off curve of HVO at point #2. Similarly to point #6 the option of high PN savings and reasonable NO_x emissions are possible or the PN benchmark emissions can be met when lower NO_x emissions are favoured. When advancing the SOI by 3 CAD, the increase in NO_x emissions compared to the benchmark is in the range of 2.3 % above B0, while the PN emissions can be reduced by more than 50 % of the B0 emissions.



Figure 6-41: Comparison of NO_x, PN and BSFC of B0, original HVO and new HVO operation at point#2.

Figure 6-41 shows the comparison of emissions and fuel consumption when retarding the injection by 2 CAD. It results in a reduction of NO_x emissions of more than 7 % compared to B0, while PN emissions are slightly below the B0 diesel fuel benchmark. The savings in fuel consumption is 1.7 %, while it could improve to 3.5 % below benchmark, when advancing the injection and allow higher NO_x emissions.



Figure 6-42: NOx-PN trade-off curve of HVO - point #3.

At high engine load (illustrated in Figure 6-42), the NO_x -PN curve of HVO is similar to the curve of SME at high load and has a very distinctive minimum of NO_x and PN at the same time. Although with the original ECU calibration, the NO_x and PN emissions of HVO are both already below the benchmark the emissions can be further reduced when retarding the SOI by 2.7 CAD and leaving the RP unchanged. The new emission levels are illustrated in Figure 6-43. While the NO_x emissions can be reduced from 99.5 % to 90.1 % of the benchmark, the PN emissions can decrease at the same time from 79.4 % to 64 % of the B0 benchmark. The accounted fuel consumption will only increase by 1.2 % to 98.1 % of the B0 benchmark and is therefore still less than fuel consumption of mineral diesel.



Figure 6-43: Comparison of NOx, PN and BSFC of B0, original HVO and new HVO operation at point#3.

It can be concluded that optimising the injection strategy for HVO B100 shows similar results to the SME optimisation. For all test points, NO_x and PN can be reduced below the B0 diesel benchmark. At low and medium engine load (point #6 and #2) the ideal operating point is difficult to set and subjected to the requirements of the engine aftertreatment system and which type of emissions are preferably reduced. When higher NO_x emissions are acceptable, it is advisable to advance the injection and benefit from high PN reductions. However, when retarding the injection the NO_x emissions can be reduced by about 10 % without increasing PN above the B0 benchmark. At high engine load, the NO_x and PN emission can even be reduced both simultaneously and the fuel consumption with HVO will still be lower than B0.

Chapter 7. Conclusions and future work

7.1 Conclusions

The main aim of this research was to investigate the spray and combustion characteristics of various diesel-like biofuels, their impact on emissions and performance as well as the potential optimisation of the injection strategy in HD diesel engines. The conclusions of this work are drawn in relation to the project objectives as outlined in section 1.3:

1. Conduct a review of various literature and research articles regarding the feasibility of various biofuels for commercial use, fuel injection and combustion of different diesel-like biofuels, and the impact of certain fuel properties on the emission formation.

Based on the review of published literature it can be concluded:

- Modern fuel injection systems are designed for the optimum fuel injection of mineral diesel. When injecting biofuels with different fuel properties the spray characteristics, such as spray penetration distance, spray cone angle, droplet size distribution and fuel evaporation rate will be affected resulting in changes in the combustion process and emission formation of diesel engines
- The combustion process in a diesel engine consists mainly of two combustion phases, namely premixed and diffusion controlled combustion. Changes in the spray characteristics, cetane number (CN), fuel density and lower heating value (LHV) will affect the fuel-air mixing process and potentially shift the ratio of premixed to diffusion combustion. If the ignition delay (ID), which is mainly influenced by cylinder temperature and CN, is prolonged a larger quantity of fuel mixes with the surrounding air before the self-ignition point is reached. A long ID results in a strong premixed combustion phase with a sharp increase in heat release rate (HRR) and higher engine noise and NO_x emissions.
- Particulate matter (PM) is mainly formed in the diffusion combustion phase in locally fuel-rich zones and low air-fuel ratios. The absence of aromatics and oxygen bonded in some fuels will reduce PM emissions significantly.

- A very promising biofuel is biodiesel which is produced by a mechanism called transesterfication when vegetable oils or animal fats react with methanol and convert into fatty acid methyl esters (FAMEs). These FAMEs can be used in diesel engines without major engine modifications. Biodiesel fuels have higher fuel densities and viscosities than mineral diesel fuel, and have poor oxidation stability and cold-start properties. Also, the production price of biodiesel is usually higher than the price of mineral diesel and therefore, biodiesel cannot be clustered as a serious competitor to mineral diesel yet.
- The price of biodiesels and their fuel properties are mainly dependent on the oil feedstock. Some biodiesel are considered to be more sustainable than others based on their life-cycle analysis and whether they are food-competitive. Most researchers agree that biodiesel combustion causes higher NO_x emissions due to oxygen bonded in the fuel, but significant reductions in PM, CO and THC emissions due to the lower aromatics content and higher oxygen content.
- Based on various criteria SME, PME, UCOME and HVO have been chosen for this project as they are believed to be very feasible fuels for future diesel alternatives. SME is the most commonly produced biodiesel in the world and already commercially available despite it being food-competitive. PME has a high saturation level and its high CN promises very low exhaust gas emissions. UCOME is made from waste products and therefore highly sustainable and cheap. HVO is a new generation fuel and a strong prospects for commercial use due to excellent combustion properties and large-scale production potential.

2. Set up a state-of-the-art HD diesel engine test bed for combustion analysis and exhaust emission measurements of various biofuels.

A state-of-the-art diesel engine test bed was built with the following components:

• A diesel engine test rig was built using a Cummins Euro V heavy-duty engine and a steady-state EC dynamometer to control engine speed and torque separately within an accuracy of less than ±5 rpm and ±5 Nm, respectively.

- A fuel supply system was designed to allow the use of various biofuels and to measure the fuel mass flow irrespective of the fuel density. The fuel system enables the quick change of fuel type without any fuel contamination due to sufficient drain valves and closed-loop systems.
- Two emission analysers, a Horiba gas analyser and a Horiba particle counter, were connected to the engine exhaust system to measure real-time exhaust emissions according to the current Euro VI emission regulations.
- An AVL pressure transducer and crank angle encoder was mounted onto the engine for combustion analysis. The TDC offset and drift compensation was carried out by using the motored curve and intake pressure correction, respectively.
- A comprehensive LabVIEW data acquisition and engine control system was developed to record all required engine data accurately. A MATLAB program was developed for post-processing the test data.
- 3. Set up a comprehensive constant volume vessel (CVV) test rig including a high speed camera system as well as a high pressure fuel system to investigate the macroscopic spray characteristics of various biofuels.

A CVV has been built with the following components:

- A CVV was commissioned for the use of various biofuels with different injectors and nozzle types. The vessel was designed to allow potential fuel combustion in the future at pressures and temperatures of up to 100 bar and 700 °C, respectively. The four quartz windows provide good visual accessibility and allow very sophisticated optical diagnostics in the future.
- A high pressure fuel system was designed for a variety of biofuels and consists of an air-driven high-pressure pump and standard common-rail system for accurate fuel injection between 500 bar and 1800 bar. The injection system is controlled by a solenoid triggering system with potential upgrade to piezo-based injection.
- The fuel system enables users to change the fuel type quickly without any fuel contamination due to sufficient drain valves and closed-loop systems.

- A high speed CCD camera system was used to record spray images with a high resolution and frame rate providing accurate and clear spray images.
- A camera and injector trigger system as well as a DAQ module was designed with LabVIEW by the author for accurately injection control. A MATLAB program was developed for post-processing the test data.
- An injector calibration rig was built to determine the injected fuel quantity per stroke for various fuel types, injector nozzles and injection conditions.
- 4. Investigate and analyse engine performance and exhaust emission (CO, NOx, THC and PN) using various biofuels in a representative engine test cycle.

The ESC tests using various biofuels have drawn the following conclusions:

- The BSFC benchmark of the ESC using mineral diesel (B0) was 216.1 g/kWh. The fuel consumption increased with increasing engine speed and decreasing engine load.
- The specific NO_x emission benchmark of the ESC cycle using mineral diesel (B0) was 8.76 g/kWh. The emissions increased with decreasing engine load.
- The specific PN emission benchmark of the ESC cycle using mineral diesel (B0) was 6.19x10¹³ #/kWh. The emissions increased with increasing engine speed and engine load due to shorter retention time at high speeds and lower air to fuel ratios at high loads.
- The specific CO and THC emission benchmarks of the ESC cycle using mineral diesel (B0) were 1.18 g/kWh and 0.06 g/kWh, respectively.
- The maximum engine power drops by approximately 9 % when using biodiesel, while with HVO only a negligible power loss was observed. The power loss increases linearly with increasing biodiesel blend ratio. The reason for the power drop is the lower volumetric LHV as the engine management system calculates the injection quantity based on standard diesel density and LHV.
- The BSFC is lowered by 2.3 % when using HVO fuel and approximately 1 % when using biodiesel. The BSFC changes linearly with the blend ratio and the LHV is responsible for the difference.

- HVO produces slightly higher NO_x than mineral diesel (0.4 % increase) due to better fuel-air mixing process concluded in the spray tests. The NO_x emissions of the different FAMEs are strongly dependent on the CN. In the increase in NO_x for PME, UCOME and SME is 0.9 %, 4 % and 5.7 % while the CN is 64.9, 58.4 and 53.7, respectively. The higher the CN the shorter the ID and the lower the NO_x formation. No linear trend between NO_x emissions and blend ratio was observed, but changes in the injection strategy seem to be responsible for the fluctuations.
- The PN emissions with HVO fuel were 14 % below the benchmark, while the PN emissions for FAME were reduced by more than 75 %. Main reasons for the reduction are the absence of aromatics and the oxygen content in biodiesel. The PN emission decrease with increasing blend ratio, but similar to the NO_x trend, fluctuations are observed due to changes in the injection strategy.
- The CO emissions are 28.6 % lower using HVO and 51.8 %, 45.2 % and 44.9 % lower using PME, UCOME and SME, respectively. The absence of aromatics and the oxygen content of biodiesel are responsible for the reduction. The CN might also effect the CO formation positively as PME shows the highest reductions followed by UCOME and SME.
- The THC emissions with HVO fuel were 60.6 % below the benchmark, while the THC emissions for FAME were reduced by more than 95 %. The lower boiling point of HVO and biodiesel (due to the lower aromatic content) will most likely be the predominant reason for the large reductions.
- The mechanical efficiency increased with the use of biodiesel from 77 % to 80 % due to the higher viscosity and better lubrication properties reducing the friction losses in the engine.
- HVO fuel produces higher peak cylinder pressures and HRR compared to mineral diesel, which concludes the better fuel-air mixing process and higher NOx emissions. The effect of the higher CN is not visible at higher engine load with pilot injection. The pilot combustion heats up the combustion chamber and the ID of the main injection is very short for all fuels.
- A relationship between the peak HRR and NO_x concentration can be established showing that the higher the peak HRR the higher the NO_x

concentration. This relationship is constant for all fuels showing that increase in NO_x emissions with biofuels is not driven by the thermal NO_x mechanism, but by the prompt NO_x mechanism due to the formation of hydrocarbon radicals at mainly unsaturated molecules. The unsaturation level is highest for SME, followed by UCOME and PME and the NO_x emissions follow the same trend.

5. Investigate the macroscopic spray characteristics of various biofuels using optical diagnostic tools.

The investigations of the macroscopic spray characteristics have drawn the following conclusions:

- Neat PME and UCOME fuel showed very poor spray characteristics at ambient fuel temperature. The spray penetration was much shorter due to the high fuel viscosity. The injector calibration also revealed that the injection quantity was much lower for both fuels as well. The spray images show that the fuel injection breaks up earlier from the nozzle and the injection duration is therefore reduced.
- At fuel temperature of 80 °C and hot chamber conditions, all fuels show more similar spray characteristics. High density fuels such as FAMEs have slightly longer spray penetrations than B0 fuel, and HVO has a shorter penetration curve due to the very low fuel density.
- The Hiroyasu and Arai penetration model is valid in its general assumptions but with higher injection pressures and chamber densities the constant parameters change and different fuel densities have a significant effect on the penetration, too. A new spray model was developed based on that of Hiroyasu & Arai. It is valid for diesel-like alternative fuels, such as FAME and HVO, at chamber densities of about 60 – 70 kg/m3 and injection pressures of 600 to 1800 bar. It also includes an exponential density factor of 0.3094.
- The spray cone angle is very constant during the injection event of each fuel. The spray cone angle of HVO is slightly larger at 600 and 1800 bar compared to the other fuels due to low viscosity and density. At 80 °C, the high

viscosity FAMEs show very similar cone angle to B0, but the angle just after SOI is rising slowly showing a jet-shaped spray pattern until the spray breakup point due to high viscosities. At cold injection conditions, the cone angle is much lower for PME and UCOME due to the higher fuel viscosity. The cone angle increases with higher fuel temperature, higher injection pressure, and higher chamber density.

- The initial injection velocity is higher for HVO and the liquid core breakup occurs earlier. The fuel spray of biodiesel breaks up slightly later than mineral diesel fuel. The injection velocity increases with increasing rail pressure and decreasing chamber density.
- At a fuel temperature of 25 °C, HVO and B0 have the highest spray surface area followed by SME, UCOME and PME due to their lower viscosity and higher spray cone angle. At 80 °C, the spray area is identical for all fuels at the same injection conditions. However, with different fuel densities and heating values the injection timing on an engine will change to achieve the same power output.
- HVO has the best preconditions for a strong air-fuel premixing process resulting in high premix combustion and NO_x formation, but the high cetane number of the fuel causes earlier ignition countering this effect. This explains the slightly higher NO_x emissions with HVO in the engine test results despite the higher cetane number.

6. Carry out potential engine power restoration and engine recalibration using few fuel representatives.

The engine restoration and optimisation tests have drawn the following conclusions:

- The FQS provides very accurate results when measuring fuel density and LHV of various fuels, which are the two main parameters for engine power restoration. Also CN can be measured within 10 % accuracy. The biofuel content measurement has to be improved, but it offers a good tool to record biofuel history in vehicles for warranty purposes.
- More than 95 % of the original power output can be restored when increasing the torque limiter by 15 % to compensate for the 11 % higher miscalculated

engine torque. Another safety interlock seems to prohibit full power recovery. Lifting the torque table is not a safe and best-practise solution, but it shows that full power can theoretically restored with acceptable BSFC and NO_x and PN emissions.

- When restoring engine power, the BSFC increases by not more than 1.6 % while NO_x emissions are reduced below or just above the B0 diesel benchmark. Restoring power prolongs the injection duration, but the initial premixed combustion remains the same, where NO_x is primarily formed. Due to the longer injection in the expansion stroke, the PN emissions increase by 100 % or more due to the long combustion duration and reduced air to fuel ratio.
- When optimising fuel injection for SME and HVO, a linear increase in NO_x emissions from retarded to advanced injection is observed due to increasing cylinder pressure and temperature. Also the ID reduces when retarded injection as the piston is closer to TDC where cylinder temperature and pressure are higher.
- NO_x emissions also increase linearly with increasing RP using HVO and SME fuel. The higher RP results in a higher injection rate and finer fuel droplets improving the fuel-air mixing process. Thus higher premixed combustion and higher HRR are expected causing NO_x emissions to form.
- PN emissions follow a second order polynomial trend when changing SOI and RP. At low engine load, the PN increases with retarded injection while the opposite effect is observed at high engine load. The increase in PN emissions is proportional to the combustion duration.
- For SME and HVO, accurate NO_x and PN emission models with varying SOI and RP have been established based on the trends described above. With the models, NO_x PN trade-off curves were derived to show the optimum operating points.
- At all three SME test points, the NO_x emissions can be reduced below the B0 diesel benchmark when retarding the injection by approximately 2.5 2.9 CAD. When injection retards, the PN emissions increase but are still more than 50 % below the B0 diesel benchmark. The BSFC increases by max 1.5

%. It is not beneficial to reduce the RP as the penalty in PN emissions is much greater than the effect of reduced NO_x emissions.

- At high engine load, NO_x and PN can be reduced at the same time due to a distinct minimum in the NO_x PN trade-off curve at 2.9 CAD retarded injection.
- When using HVO fuel, NO_x and PN emissions can also be reduced below the B0 benchmark, when retarding the injection by 0.9, 2.0 and 2.7 CAD at low, medium and high engine load, respectively. The BSFC would increase by approximately 1 %, but still be just below the benchmark of B0 mineral diesel fuel. However, at these injection conditions the reduction in NO_x emissions is relatively small, while the increase in PN emissions is large. Since the NO_x emissions were always either below the B0 benchmark or slightly above, it could be considered to accept higher NO_x emissions when advancing the injection timing and benefit from much higher reduction in PN emissions. If NO_x emissions of 10 % above the B0 benchmark are acceptable, the PN emissions can be reduced to be more than 60 % below the B0 benchmark at low engine load for example. This would potentially enable engine manufacturers to significantly downsize the aftertreatment system by installing smaller DPFs. Also the BSFC would between 2 % and 4 % lower than the B0 diesel benchmark.
- At high engine load, a distinct minimum in the NO_x PN trade-off curve offers similar to SME an operating point where both emissions, NO_x and PN, can be reduced at the same time when retarding the injection by 2.7 CAD. The NO_x and PN emissions would be 10% and 36 % below the benchmark, respectively. The BSFC would increase slightly, but still be below the B0 benchmark.

7. Propose an optimised engine control strategy using biofuels to improve engine performance and minimise engine exhaust gas emissions.

The holistic investigation of various biofuels showed that many fuel parameters are responsible for the engine performance and emission formation. Even within the EN590 diesel standard, the limits are wide and small differences in fuel properties can be enough to exceed the very tight emission standards. On top of that, the number of different biofuels is increasing and with it are the differences in fuel properties. Nowadays more flexible control strategies are necessary for modern diesel engines to allow clean and efficient fuel combustion for a large variety of fuels. Especially trucks travelling wide distances across continents have to deal with different types of fuels from different countries. Although general approaches can be adapted which directs engine manufacturers to improve combustion by increasing rail pressure, adding EGR and multiple injection events, some more real-time fuel-related information are required to apply a more flexible control strategy. Continental developed flexible fuel quality sensors to measure some basic fuel parameters of diesel-like fuels with an instant response to potentially feed the information to the engine management system. The basics of the FQS are described in section 3.2.3 and the most important fuel parameters for engine restoration and recalibration can be measured within acceptable accuracy for a huge variety if fuel types and blends. Figure 7-1 illustrates the basic working principle of the FQS.



Figure 7-1: Schematic of proposed FQS integration for automatic torque restoration.

The sensor is small and can be easily fitted to an existing fuel line without restricting the flow or affecting the engine operation in any other way. As the sensor is still on prototype stage, no final statement about the costs have been made, but it should not exceed the price of about $20 \in$. The sensor requires a 12 V power supply, which is easily available on heavy-duty vehicles, and the output signals can be feed into the ECU via an interface box. The sensor solution would allow engine manufacturers to supply engines with a flexible control strategy by using fuel density and LHV parameters as variable inputs into the ECU to determine the correct fuel quantity per stroke to match the original engine certifications with a wide range of fuels. Especially for heavy-duty vehicles exposed to various different fuels this sensor technology can provide a time- and cost effective solution. In a second stage aromatics content and CN could also be fed into the engine management system to

apply more sophisticated injection control by flexibly changing injection timing or EGR settings to improve the NO_x -PN trade-off.

Another approach towards cleaner and more efficient fuel combustion is the wider use of HVO fuel. HVO has proved to be an excellent fuel with outstanding combustion characteristics and a high sustainability index. Although biodiesel produce very low exhaust emissions compared to mineral diesel, the poor oxidation stability and cold-start ability are major challenge on the route for large-scale commercial applications. HVO uses the same feedstock as biodiesel and any vegetable oil or animal fat can be used in a refining process with a with very competitive production costs. HVO offers good cold-start and oxidation stability and most fuel parameters are similar to mineral diesel except fuel density and cetane number. Since HVO production is a chemical reaction process the fuel properties are not affected by the oil feedstock used. Thus, HVO can be produced with very reliable and constant fuel parameters as long as process conditions, such as catalyst, temperature and pressure remain constant. This is a strong advantage compared to mineral diesel and biodiesel production, and engine manufacturers are able to optimise the injection strategy based on very consistent fuel parameters. The author of this work believes that HVO will become the strongest competitor to mineral diesel in the transportation sector.

7.2 Recommendations for future work

Engine test bed:

- Investigations into the cold-start behaviour of various biofuels will provide valuable addition to the current work. Also, the test bench can be upgraded to a Euro VI engine as the Cummins ISB version has a similar engine block and flywheel to the current Euro V engine. When replacing the engine with a Euro VI version, the effect of EGR and VGT (variable-geometry turbocharger) can be further investigated and compared to the Euro V test data.
- The investigation of various biofuels in transient test modes has been less investigated than steady-state operation. Carrying out transient tests in combination with variable valve timing (VVT) and electric turbocharger assist technology area research areas worth pursuing.
- In Chapter 6 the effect of SOI and RP on engine performance and emissions has been investigated. The effect of dwell times, post injection and additional pilot injection can be further investigated with the current equipment. Furthermore, the shift of combustion duration and PN emissions with increasing engine load could not be explained by the tests carried out in this work. Pursuing the effect of reduced PN emissions at high engine load would be beneficial to understand and explain the distinctive emission minima in Figure 6-28 and Figure 6-42.
- Chapter 4 revealed that significant differences in emissions are present when increasing biodiesel blend ratio. The reasons for the strong fluctuations could not be fully explained at this stage, but further investigation of where the fluctuations come from will provide valuable knowledge.

Constant volume vessel:

• The CVV was designed to also investigate biofuel combustion. The formation of NO_x emissions with biodiesel is not fully investigated yet. As mentioned in Chapter 4 the additional NO_x formation with biodiesel might well be caused by the prompt NO_x mechanism and the optical vessel offers the opportunity to conduct further studies on the emission formation in biofuel combustion.

- The vessel also offers the possibility to study the microscopic and near-field spray characteristics of various biofuels by using tools such as Mie scattering, Schlieren technology, Laser Induced Fluorescence (LIF), Particle Image Velocimetry (PIV).
- Currently only standard passenger car fuel injectors can be mounted onto the CVV. Designing a new injector holder for heavy-duty injectors will allow to test the same injector as used on the engine test bed, which would make both test rigs more transferable.
- At the start of this project it was planned to carry out combustion analysis of various biofuels using the two-colour method. Due to delays of the vessel commissioning it was not possible to pursue these tests. However, the operating principles and data acquisition programmes have been developed and tests can be carried out as soon as the vessel design is completed. These tests will provide information about the local flame temperature and soot formation zones of various fuels by using the thermal radiation of the soot particles and comparing the corresponding RGB values of the obtained combustion images with the radiation of a black body. A more detailed description can be found in Appendix C.

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APPENDIX A

	Method Unit		Diesel Diesel		Diesel Diesel Dies		Diesel HVO		FAME	FAME	FAME	
Sample Name			BO	HVO B10	SME B10	PME B10	UCOME B10	HVO B100	SME B100	PME B100	UCOME B100	
Visual Rating			<u>clear</u>	<u>clear</u>	<u>clear</u>	<u>clear</u>	<u>clear</u>	<u>clear</u>	turbid with few	<u>clear</u>	<u>clear</u>	
Density @ 15 °C	DIN EN ISO 12185	kg/m³	840,4	834,2	844,7	843,7	844,1	780,1	<u>885,2</u>	876,5	879,5	
Colour	Farbe (visuell)		amber (orange)	amber (orange)	amber (orange)	amber (orange)	rot	amber (orange)	amber (orange)	amber (orange)	rot	
Content C	NPA-051	Mass%	86,27	86,06	85,32	85,31	85,56	84,68	77,33	77,02	76,7	
Content H	NPA-051	Mass%	<u>13,49</u>	<u>13,63</u>	<u>13,35</u>	<u>13,38</u>	<u>13,42</u>	<u>15,02</u>	<u>12,04</u>	<u>12,4</u>	12,29	
Content O	NPA-051	Mass%	0,24	0,31	<u>1,33</u>	<u>1,31</u>	<u>1,02</u>	0,3	10,63	<u>10,58</u>	<u>11,01</u>	
Sum C+H+O	NPA-051	Mass%	100	<u>100</u>	100	100	<u>100</u>	<u>100</u>	<u>100</u>	100	100	
Viscosity @ 40 °C	DIN EN ISO 3104	mm²/s	2,82	2,83	<u>2,91</u>	2,94	2,93	<u>3,02</u>	4,18	4,55	4,35	
Mono-Aromatics	DIN EN 12916	Mass%	24,2	22	21,5	21,5	22					
Di-Aromatics	DIN EN 12916	Mass%	3	<u>2,7</u>	<u>2,6</u>	<u>2,6</u>	2,5					
Tri+-Aromatics	DIN EN 12916	Mass%	<u>0,3</u>	<u>0,3</u>	<u>0,2</u>	<u>0,3</u>	0,2					
Poly-Aromatics	DIN EN 12916	Mass%	3,3	3	2,8	<u>2,9</u>	2,7					
Total Aromatics	DIN EN 12916	Mass%	27,5	<u>25</u>	24,3	24,4	24,7					
Heating value Ho	DIN 51900	kJ/kg	45715	<u>45902</u>	<u>45119</u>	<u>45122</u>	45153	47088	39708	39879	39816	
Heating value Hu	DIN 51900	kJ/kg	42853	43010	42286	42283	42305	43902	37230	37320	37200	
Heating value Hu Vol.	DIN 51900	kJ/l	<u>36014</u>	<u>35879</u>	<u>35719</u>	<u>35674</u>	35708	34248	32956	32710	32718	
HFRR (WS 1.4)	DIN EN ISO 12156-1	μm	<u>427</u>	<u>392</u>	<u>166</u>	217	162					
Oxidation stability Temperature	DIN EN 15751	°C	<u>110</u>	110	<u>110</u>	<u>110</u>	110					
Oxidation stability auto eval	DIN EN 15751	h	<u>36,2</u>	<u>26,3</u>	<u>17,7</u>	<u>10,7</u>	32,5					
Oxidation stability manual eval.	DIN EN 15751	h	<u>36,3</u>	<u>23</u>	<u>17,3</u>	<u>9,7</u>	31,9					
Water content	DIN EN ISO 12937	mg/kg	<u>71</u>	<u>52</u>	<u>72</u>	<u>76</u>	50					
Water content %	DIN EN ISO 12937	Mass%	<u>0,007</u>	<u>0,005</u>	<u>0,007</u>	<u>0,008</u>	0,005					
Dist. IBP	DIN EN ISO 3405	°C	<u>175,7</u>	<u>177,9</u>	<u>178</u>	<u>177,6</u>	175					
Dist.(Temp. 5%)	DIN EN ISO 3405	°C	<u>191,1</u>	<u>193,6</u>	<u>192,4</u>	<u>192,9</u>	192,5					
Dist.(Temp. 10%)	DIN EN ISO 3405	°C	<u>199,8</u>	204,4	201.7	201,1	201,1					
Dist.(Temp. 15%)	DIN EN ISO 3405	°C	<u>208,2</u>	213,6	211,7	211,8	210,5					
Dist.(Temp. 20%)	DIN EN ISO 3405	°C	216,9	223,7	222,3	222,6	221,9					
Dist.(Temp. 30%)	DIN EN ISO 3405	°C	<u>238,1</u>	<u>245,4</u>	247,8	248,2	246,7					
Dist.(Temp. 40%)	DIN EN ISO 3405	°C	<u>259,8</u>	<u>264,8</u>	271,3	271	269,7					
Dist.(Temp. 50%)	DIN EN ISO 3405	°C	278	279,3	<u>288,1</u>	<u>288,3</u>	286,8					
Dist.(Temp. 60%)	DIN EN ISO 3405	°C	<u>292,2</u>	<u>290,9</u>	<u>302,8</u>	<u>302,7</u>	301,4					
Dist.(Temp. 70%)	DIN EN ISO 3405	°C	<u>306,3</u>	<u>302,7</u>	315,3	315,5	314,1					
Dist.(Temp. 80%)	DIN EN ISO 3405	°C	321,3	316,6	<u>327,6</u>	<u>327,4</u>	326,5					
Dist.(Temp. 85%)	DIN EN ISO 3405	°C	<u>330,4</u>	325,7	<u>334</u>	<u>333,7</u>	333					
Dist.(Temp. 90%)	DIN EN ISO 3405	°C	341,1	337,4	341,4	341,1	340,8					
Dist.(Temp. 95%)	DIN EN ISO 3405	°C	356	353,6	353,6	352,4	352,4					
Dist. FBP	DIN EN ISO 3405	°C	364,8	364,4	<u>362,4</u>	361,6	361					
Dist. Recovery	DIN EN ISO 3405	Vol%	<u>98,1</u>	<u>98,2</u>	<u>98,3</u>	<u>98,2</u>	98,1					
Dist. Residue	DIN EN ISO 3405	Vol%	<u>1,4</u>	<u>1,4</u>	<u>0,3</u>	<u>1,4</u>	1,7					
Dist. Loss	DIN EN ISO 3405	Vol%	0,5	<u>0,4</u>	<u>1,4</u>	0,4	0,3					
Dist.(Residue + Loss)	DIN EN ISO 3405	Vol%	1,9	<u>1,8</u>	1,7	<u>1,8</u>	2					
Dist. (evap. 250°C)	DIN EN ISO 3405	Vol%	<u>35,7</u>	32,5	31,1	<u>31,1</u>	31,5					
Dist. (evap. 350°C)	DIN EN ISO 3405	Vol%	93,7	94,4	94,2	<u>94,5</u>	94,5					



Date & Time:	Operator:				Fuel:			Blend Ratio:		Comments:		Humidity:	%	Cell Temp	°°
// : Test Run: 1/3	(Protocol) (Emissions) (In-cylinder Pressure)			Ref Diesel B0 Gas oil SME PM HVO WVO			B10 B50	B20 B100	entilation valves V2+V3 and drain valve V6 closed?						
Index	Ref. Speed [rpm]	Ref. Torque [Nm]	Actual Speed [rpm]	Actual Torque [Nm]	Consumption [kg/h]	IMEP [bar]	Pressure [bar]	Manifold Temp. [°C]	Temp. [°C]	HC [ppm]	NOx [ppm]	CO [ppm]	CO2 [ppm]	O2 [ppm]	PM-# [-]
1	1490	730													
2	1855	380													
3	1855	565													
4	1490	385													
5	1490	575													
6	1490	200													
7	1855	730													
8	1855	200													
9	2220	650													
10	2220	180													
11	2220	495		1						1					
12	2220	335													

Date & Time:	Operator:				Fuel:			Blend Ratio:		Comments:		Humidity:	%	Cell Temp	:°C
// : Test Run: 2 / 3			_(Protocol) _(Emissions) _(In-cylinder)	Pressure)	RefDieselBO	PM HVO	Gas oil	B10 B50	B20B100						
Index	Ref. Speed [rpm]	Ref. Torque [Nm]	Actual Speed [rpm]	Actual Torque [Nm]	Consumption [kg/h]	IMEP [bar]	Pressure [bar]	Manifold Temp. [°C]	Temp. [°C]	HC [ppm]	NOx [ppm]	CO [ppm]	CO2 [ppm]	O2 [ppm]	PM-# [-]
1 2	1490 1855	730 380													
3	1855	565													
5	1490	575													
6	1490 1855	200 730													
8	1855	200													
9 10	2220 2220	650 180													
11 12	2220 2220	495 335													

Date & Time:	Operator:				Fuel:			Blend Ratio: Comm				Humidity:	%	Cell Temp	:°C
// : Test Run: 3 / 3			(Protocol) (Emissions) (In-cylinder)	Pressure)	Ref Diesel BO	РМ ПНVС	Gas oil D 🔲 WVO	B10 B50	B20 B100						
Index	Ref. Speed [rpm]	Ref. Torque [Nm]	Actual Speed [rpm]	Actual Torque [Nm]	Consumption [kg/h]	IMEP [bar]	Pressure [bar]	Manifold Temp. [°C]	Temp. [°C]	HC [ppm]	NOx [ppm]	CO [ppm]	CO2 [ppm]	O2 [ppm]	PM-# [-]
1	1490	730													
2	1855	380													
3	1855	565													
4	1490	385													
5	1490	575													
6	1490	200													
7	1855	730													
8	1855	200													
9	2220	650													
10	2220	180													
11	2220	495													
12	2220	335													

Combustion image processing

To gain an understanding of the combustion behaviour in CI engines, combustion analysis based on recorded in-cylinder pressure data during each cycle is a very reliable and cost effective method to get information about indicated power output, HRR and combustion efficiency. However, to get more detailed knowledge of combustion process, such as flame temperatures, flame propagation and soot formation zones more sophisticated methods must be used in modern engine research, such as optical diagnostic tools. Although those techniques are very expensive, without the insight into the combustion process no further development and improvement seem to be possible in DI engines [274].

The two-colour method has been used to gain information about local flame temperatures and soot concentration. The combustion of diesel and diesel-like fuels causes high luminosity as a result of the thermal radiation of the soot particles at very high temperatures. In order to use the high luminosity, the two-colour method has been developed to measure diesel and biodiesel combustion temperatures [275, 276].

The soot concentration was represented as the KL number, which is proportional to the soot density [277]. A blackbody furnace in the range of 1400 K to 2400 K was used to calibrate the colour camera. Images at each temperature were taken and the correspondent RGB values of the calibration images were measured using MATLAB. It is very important that during the diesel combustion tests the camera settings, such as aperture and focal length are identical to the calibration settings as the brightness of the pictures changes and the RGB values would not represent the correct flame temperature and KL number anymore.

A more detailed explanation of the principle and background information can be found in the work of Zhao and Ladommatos [277]. The KL factor is an indication for the soot particle concentration consisting of the absorption coefficient K, which is proportional to the number density of soot and the geometric thickness of the flame L along the optical axis. If knowing the flame temperature the KL number can be calculated using equation (C-1):

$$KL = -\lambda^{\alpha} \ln \left[1 - \left(\frac{e^{C2/(\lambda T)} - 1}{e^{C2/(\lambda T_a)} - 1} \right) \right]$$
(C-1)

with C_2 as the second Planck's constant, λ the wavelength and T_a the apparent temperature of a block body at the same luminosity. The value of α depends on the optical and physical

properties of the soot in the flame. The flame temperature can be obtained by equalising the equation for the two specific wavelengths, λ_1 and λ_2 :

$$\left[1 - \left(\frac{e^{C_2/(\lambda_1 T)} - 1}{e^{C_2/(\lambda_1 T_{a1})} - 1}\right)\right]^{\lambda_1^{\alpha_1}} = \left[1 - \left(\frac{e^{C_2/(\lambda_2 T)} - 1}{e^{C_2/(\lambda_2 T_{a2})} - 1}\right)\right]^{\lambda_2^{\alpha_2}}$$
(C-2)

The flame temperature can be calculated if Ta_1 and Ta_2 at λ_1 and λ_2 are known. Both temperatures can be measured at their wavelength (usually in the visible spectrum), which is the reason for stating it two-colour method.

With MATLAB the images were converted to 512x512 pixels and 256 grey levels. The grey scale of these pictures were measured and averaged at each wavelength to obtain the apparent temperature at various crank angles and an apparent temperature versus grey scale correlation was established. With knowing the apparent temperature at each pixel for two different wavelengths the flame temperature and soot concentration (KL factor) was calculated for each pixel according to equation (C-1) and (C-2). A jet map and a hot map for soot and flame temperature, respectively, have been applied at the images in MATLAB to visualise the results.



Figure C-1: Example of two-colour method processing