MASTER

Experimental study using a dual fuel system on a heavy duty diesel engine

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Experimental study using a dual fuel system on a heavy duty diesel engine.

Part I: Direct injected diesel-butane blends
Part II: Reactivity controlled compression ignition

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Abstract

In current diesel combustion engines, legislated emission levels are generally met through after-treatment. The development of combustion technologies with lower emissions of smoke and nitric oxides can minimize the after-treatment system required, and thus reduce the related costs. Due to the increasing need for energy, but still taking the global environment into account, various alternative fuels are investigated for the conventional diesel combustion regime (CDC).

For short term applications this work investigates the impact of premixed butane-diesel blends on conventional diesel combustion. LPG (Liquefied Petroleum Gas) is for long being used as an automotive fuel in light vehicles like private cars and taxies. LPG is a low-cost waste product coming from the refining process of fossil fuels. In this work, a modified six-cylinder direct-injected 12.6 liter heavy duty DAF engine is used to investigate premixed butane-diesel blends. In the interpretation of experimental results, emphasis lies on the effect on combustion phasing, emission reduction and cost reduction which are inherent challenges to enable viable CDC combustion.

Although severe advantages in emissions and fuel cost reductions can be achieved by premixed alternative fuels, for the long term application of internal combustion engines, other combustion concepts are investigated. Premixed Charge Compression Ignition is such a combustion concept that holds the promise of combining emission levels of a spark-ignition engine with the efficiency of a compression-ignition engine. However, in this combustion concept, control of combustion phasing is lost due to early direct injection.

With the help of these early injections, this combustion concept efficiently reduces emissions of smoke and nitric oxides by increasing Combustion Delay, for instance by using a low reactive blend of different fuels. However, during highly transient engine operation it is difficult to adapt fuel reactivity fast enough. To vary the reactivity depending on load conditions and engine speed, and therefore obtaining control of combustion phasing, truly real time control is desired, in this investigation called Reactivity Controlled Compression Ignition (RCCI). Such a strategy is studied on a dedicated test cylinder of a modified six-cylinder 12.6 liter heavy duty DAF engine.

For these investigations, this test cylinder has been equipped with port fuel gasoline injection complementing the stand-alone diesel injection system, EGR circuit, and air compressor. Timing and rate of heat release can be directly controlled by varying the balance between direct injected diesel and port fuel injected gasoline, cycle by cycle. The current study presents the proof of concept of the dual fuel system. Furthermore, results of experimental investigations of different injection strategies are analyzed, mainly with respect to combustion phasing, emissions and efficiency.

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Definitions, acronyms, abbreviations

aBDC	After Bottom Dead Centre
AFR	Air Fuel ratio
aROHR	Apparent Rate of Heat Release
aTDC	After Top Dead Centre
bBDC	Before Bottom Dead Centre
BD	Burn Duration
BMEP	Brake Mean Effective Pressure
BP	Boiling Point
bTDC	Before Top Dead Centre
But15	Diesel-Butane blend with 15wt% butane
But33	Diesel-Butane blend with 33wt% butane
But50	Diesel-Butane blend with 50wt% butane
CA	Crank angle
CAX	Crank angle at which X% of the fuel is burnt
CD	Combustion Delay
CDC	Conventional Diesel Combustion
CI	Compression Ignition
CN	Cetane Number
CO	Carbon Monoxide
CR	Compression Ratio
CT	The Combustion Technology group of the TU/e
DCN	Derived Cetane Number
DI	Direct Injection
DPF	Diesel Particulate Filter
EGR	Exhaust Gas Recirculation
EOA	End of Injection (actuation)
EOI	End of Injection (fuel delivery)
EOI1	End of Injection of the first diesel injection
EOI2	End of Injection of the second diesel injection
EPA10	North American legislated emission levels for HD as of 2010
EU	European
Euro VI	European legislated emission levels for HD as of 2013
EVC	Exhaust Valve Close
EVO	Exhaust Valve Open
FMEP	Friction Mean Effective Pressure
FSN	Bosch Filter Smoke Number
Gas	Gasoline
HC	Hydrocarbons
HCCI	Homogeneous Charge Compression ignition
HDDE	Heavy Duty Diesel Engine
HDDI	Heavy Duty Direct Injection

HR	Heat Release
HRR	Heat Release Rate
IMEP	Indicated Mean Effective Pressure
ISCO	Indicated Specific Carbon Monoxide emission
ISFC	Indicated Specific Fuel Consumption
ISHC	Indicated Specific Hydrocarbon emission
ISNOx	Indicated Specific Nitric Oxides emission
ISPM	Indicated Specific Particulate Matter emission
IVC	Intake Valve Close
IVO	Intake Valve Open
LHV	Lower Heating Value
LPG	Liquefied Petroleum Gas
LTC	Low Temperature Combustion
MPRR	Maximum Pressure Rise Rate
NO_x	Nitric Oxides
O2	Oxygen
ON	Octane Number
PCCI	Premixed Charge Compression Ignition
PFI	Port Fuel Injection
PIG	Port Injected Gasoline
Pin	Intake Pressure
PM	Particulate Matter
PRF	Primary Reference Fuel
PRR	Pressure Rise Rate
RCCI	Reactivity Controlled Compression Ignition
ROHR	Rate of Heat Release
SCR	Selective Catalytic Reduction
SI	Spark Ignition
SOA	Start of Actuation
SOA1	Start of Actuation of first diesel injection
SOA2	Start of Actuation of second diesel injection
SOC	Start of Combustion
SOGI	Start of Gasoline Injection
SOI	Start of Injection (fuel delivery)
SOI1	Start of Injection of first diesel injection
SOI2	Start of Injection of second diesel injection
TDC	Top Dead Centre
Tin	Intake Temperature
TU/e	Eindhoven University of Technology
T10	Temperature at which 10% mass has vaporized
T50	Temperature at which 50% mass has vaporized
T90	Temperature at which 90% mass has vaporized
UHC	Unburned Hydrocarbons
US	United States
wt%	weight percentage

1. Introduction

Due to more stringent emission standards, engine development is forced to invest heavily in research for advanced combustion systems and exhaust after-treatment devices. The exotic materials used for after-treatment systems are pushing production costs; therefore reduction of in-cylinder emission formation is needed. Furthermore, rising fuel cost and a focus on reduction of greenhouse gases drive developments towards a better efficiency of the internal combustion engine.

Due to the increasing need for energy, but still taking the global environment into account, various alternative fuels are investigated for the conventional diesel combustion (CDC) regime. Fuel properties are of great influence on combustion and can be used as a control-parameter for optimizing combustion concepts. In this work two different investigations of fuel impact on combustion behavior are done.

1.1 Short term scenario

For short term applications this work investigates the impact of premixed butane-diesel blends on conventional diesel combustion. LPG (Liquefied Petroleum Gas) is for long being used as an automotive fuel in light vehicles like private cars and taxies. LPG is a waste product, coming from the refining process of fossil fuels, resulting in a low fuel price [1]. LPG consists of butane and propane, available in different mixing fractions generally depending on the season, for example 70 vol% of propane and 30 vol% of butane in the winter. In the Netherlands, LPG is widely available and due to tax benefits from the government it is attractively priced compared to, for example, diesel.

Recently, LPG supply systems in diesel engines have been attracting considerable attention due to the requirement for more powerful engines in for example urban buses [2]. LPG can be used in heavy-duty diesel engines combining port fuel injection of LPG and a direct injection of diesel. Many names are known for these systems, but in this report dual fuel is defined on the one hand as a direct injected premixed blend, or on the other hand as a system making use of an external port fuel injection system for in-cylinder blending. Dual fuel engines equipped with a port fuel injection (PFI) system suffer from the problems of poor brake thermal efficiency and high HC emissions, particularly at low loads [2]. According to Poonia et al [3] using hot Exhaust Gas Recirculation (EGR), resulting in a higher intake temperature, HC emissions can be lowered and brake thermal efficiency can be increased. Alternatively, in this investigation a butane-diesel pre-mixed blend is directly injected, using only one fuel system, to determine the influence of butane on conventional diesel combustion. For hardware safety reasons a diesel-butane blend is used for the experiments done in this report, rather than a diesel-LPG blend. When using real LPG, a certain amount of propane in the blend will lead to higher vapor pressures, which are hard to control on the current experimental set-up and therefore present a risk due to over pressure in the injector's fuel supply line. More details about this experiment can be found in chapter two.

Dong Jian et al [2] found, with the use of a dual fuel PFI system, a significant reduction in soot emissions, but an increase in HC, NO_x and CO emissions for the same power output. Qi Donghui et al [1] used a premixed LPG/diesel blend for a directly injected diesel engine and found longer ignition delays. This resulted in a better premixing of the fuel with air, a reduction in soot emissions, small increase in HC emissions and slightly higher fuel consumption. As a reason for lower soot production with DI premixed blends J. Cao [4] mentions better premixing of the fuel with air due to flash boiling in the injection spray when LPG is added.

The key issue for the direct injection strategy of premixed blends is finding the right diesel-butane content at different operating conditions. A low butane content may slightly reduce costs but have no effect on reducing soot emission. Higher butane content can reduce costs and emissions even further; however in-cylinder pressure can increase rapidly and damage the engine [1]. When engine load is low, on the other hand, higher butane content can influence combustion phasing, performance, emissions and pressure rise rate without damaging hardware of the engine.

Apart from the changing combustion phasing, emissions, and pressure rise rate during combustion, the change in fuel consumption plays an important role. A diesel-butane blend contains less energy per volume, resulting in higher fuel consumption in L/km [1,4]. With the right amount of butane in the blend this fuel penalty might be compensated by lower fuel costs.

The work presented concerns the description of a direct injected heavy duty diesel engine and the total performance as a function of butane content in a diesel-butane blend.

1.2 Long term scenario

In the second part of this report the long term scenario is investigated using a new combustion concept: premixed charge compression ignition (PCCI), based on fuel reactivity-controlled auto-ignition, by using a port gasoline injection system.

According to Leermakers [5] PCCI combustion is characterized by low temperature, partially premixed combustion by using early injections, large ignition delays and high percentages of EGR. The most common issues of PCCI combustion are wall-wetting, resulting in high unburned hydrocarbon (HC) emissions, lack of control of combustion phasing, resulting in high pressure rise rates, and lower efficiencies [5,6]. Tie Li and coworkers [6] found that for PCCI combustion at higher loads, the promotion of fuel-air mixing at relatively higher intake oxygen concentration is necessary. They propose to use low reactivity fuels, such as gasoline, and lower compression ratios, for example 12 instead of 17, to expand the operating load range of smokeless low temperature combustion. Similarly, Kalghatgi and coworkers [5, 7, 8, 9] showed that low reactive fuels such as gasoline elongate the mixing time of fuel with air and can be used for PCCI combustion at higher compression ratios. To achieve this kind of PCCI combustion in practice, gasoline- and diesel-like fuels were used by Kalghatgi.

In all of these investigations fuel was pre-blended before injection. Compression ignition based on fuel reactivity depends heavily on load, therefore fuel reactivity should be controlled per cycle. Reitz is one of the leading researchers in the field of dual fuel PCCI with use of a port gasoline injection system for controlling fuel reactivity cycle by cycle. Hanson and Reitz used the PFI system for gasoline injection and early in the compression stroke, direct injection of diesel fuel for in-cylinder fuel blending and combustion phasing control. The engine experiments were conducted with a conventional common rail injector demonstrating control and versatility of dual fuel PCCI combustion with the proper fuel blend, SOI and IVC timings at 9 bar indicated mean effective pressure (IMEP). Apparently, the efficiency of this combustion concept highly depends on engine load [10, 11]. For creating an effectively lower compression ratio Reitz used a machined camshaft with a later IVC timing than conventional (85 °BTDC) [12]. In the work presented here, dual fuel PCCI is investigated with varying injection timings, both for the port gasoline injection and the direct diesel injection. Several loads are investigated on the total performance of a heavy duty diesel engine (HDDE). Preliminary tests of the dual fuel system with use of a gasoline PFI system are presented in the third chapter.

According to Kokjohn one of the most crucial parameters with dual fuel PCCI, also called RCCI (Reactivity Controlled Compression Ignition), are multiple direct diesel injections. These multiple injections are timed approximately at 60 deg BTDC and 30 deg BTDC [10,11]. When using a double injection strategy for the diesel injection, the timing of the first injection (SOI1) was found to increase mixture stratification as SOI1 was retarded toward TDC. The increased local equivalence ratio raised local fuel reactivity, which then advanced combustion phasing and increased pressure rise rate (PRR) and NO_x emissions. The timing of the second diesel injection (SOI2) was also found to increase mixture stratification as SOI2 was retarded toward TDC. Similar to the SOI1 timing, the increased local equivalence ratio raised combustion temperatures and fuel reactivity resulting in an advanced combustion phasing.

From the aforementioned investigations it is not mentioned that multiple injection stratification can control combusting phasing, without decreasing compression ratio with the help of an adjusted camshaft. In the presented work multiple injection strategies will be investigated at a relatively high compression ratio of 15. For these investigations, this test cylinder has been equipped with port fuel gasoline injection complementing the stand-alone diesel injection system, EGR circuit, and air compressor. Timing and rate of heat release can be directly controlled by varying the balance between direct injected diesel and port fuel injected gasoline, cycle by cycle.

One of the most promising features of RCCI is an increase in thermal efficiency. In almost all published investigations, thermal efficiencies of over 50% are stated. Although the group of Wisconsin repeatedly speaks about high thermal efficiency, a real explanation is not given. According to Reitz and coworkers [10, 11, 13, 14] high thermal efficiencies can be achieved due to low temperature combustion in combination with highly accurate combustion timing and therefore minimized heat losses. In chapter five, results of preliminary tests of different injection strategies are analyzed, mainly with respect to combustion phasing and efficiency. In the last chapter, chapter six, conclusions are drawn and recommendations are given.

2. Experiment set-up

2.1 Introduction

As mentioned in the main introduction two combustion concepts for reducing engine out-emissions are investigated. For short term applications this work investigates the impact of premixed butane-diesel blends on conventional diesel combustion. For long term applications a heavy duty diesel engine is equipped with a port fuel injection system for investigating CDC and PCCI combustion based on fuel reactivity.

Together with Vialle Alternative Fuel Systems the effects of using a diesel-butane blend is investigated. A heavy duty engine has been adapted to inject pre-mixed diesel-butane blends directly into the cylinder. In the following chapter the experiment set-up is discussed.

A close partnership of the Combustion Technology group (CT) and DAF Trucks N.V. leads to ongoing research in the field of heavy duty diesel engines. In this work the effects of fuel reactivity on PCCI combustion, with help of a port fuel injection system on a dedicated engine test set-up, are investigated. Both PCCI and CDC are investigated at different loads with multiple mixing ratios of gasoline and diesel and multiple strategies of fuel injection. In this chapter the details of the experiment set-up and the measurement series are discussed.

As mentioned before, a dedicated engine set-up is used for the measurement series referred to as the CYCLOPS, in section 2.2 this test-rig is discussed. In section 2.3 the changes made to the experimental set-up together with the experimental procedures and the fuels used regarding the measurement series for Vialle are given. After this the experimental set-up for the measurement series for DAF Trucks are described in section 2.4. Section 2.5 describes the way raw data is analyzed and definitions are given. For more detailed information about the experiment set-up the reader is referred to Appendix A and Appendix C.

2.2 Experimental apparatus

For this investigation at Eindhoven University of Technology a six-cylinder DAF engine, referred to as CYCLOPS, is used. This engine set-up is described in detail by Leermakers and Boot [5, 15]. Below the description of Leermakers is used and extended with respect to the current set-up.

The CYCLOPS is a dedicated engine test rig, see Table 2.1, designed and built at the TU/e, based on a DAF XE 355 C engine. Cylinders 4 through 6 of this inline 6 cylinder HDDI engine operate under the stock DAF engine control unit and together with a water-cooled, eddy-current Schenck W450 dynamometer they are only used to control the crankshaft rotational speed of the test cylinder, i.e. cylinder 1.

Table 2.1 Cyclops specifications

Base Engine	6 cylinder HDDI diesel
Cylinders	1 Testcylinder
Bore [mm]	130
Stroke [mm]	158
Compression ratio	14.92 (original 17.0)
Bowl shape	M-shaped
Bowl diameter [mm]	100

When data acquisition is idle, for instance during engine warm up or in between measurement series, the CYCLOPS is only fired on the three propelling cylinders. Once warmed up and operating at the desired engine speed, combustion phenomena and emission formation can be studied in the test cylinder. Apart for the mutual cam- and crankshaft and the lubrication and coolant circuits, the test cylinder operates autonomously from the propelling cylinders. Stand alone air, EGR and fuel circuits have been designed for maximum flexibility as is discussed below.

Fed by an air compressor, the intake air pressure of the test cylinder can be boosted up to 5 bar. The pressure set point can be programmed from the engine control room and pressure is regulated by a pressure controller, which receives its input signal from a pressure sensor mounted in the intake manifold of the test cylinder. The fresh air mass flow is measured with a Micro Motion Coriolis mass flow meter.

Non-firing cylinders 2 and 3 function as EGR pump cylinders, see Figure 2.1, the purpose of which is to generate adequate EGR flow, even at 5 bar charge pressure and recirculation levels in excess of 70%. The EGR flow can be cooled both up- and downstream of the pump cylinders, down to ca. 30 °C, using a variable flow of process water as coolant medium. EGR mass flow is both measured with a Coriolis mass flow meter, and estimated from the fresh air mass flow and computations regarding the volumetric efficiency. Several surge tanks, to dampen oscillations and ensure adequate mixing of fresh air and EGR flows, and pressure relief valves, to guard for excessive pressure in the circuit, have been included in the design.

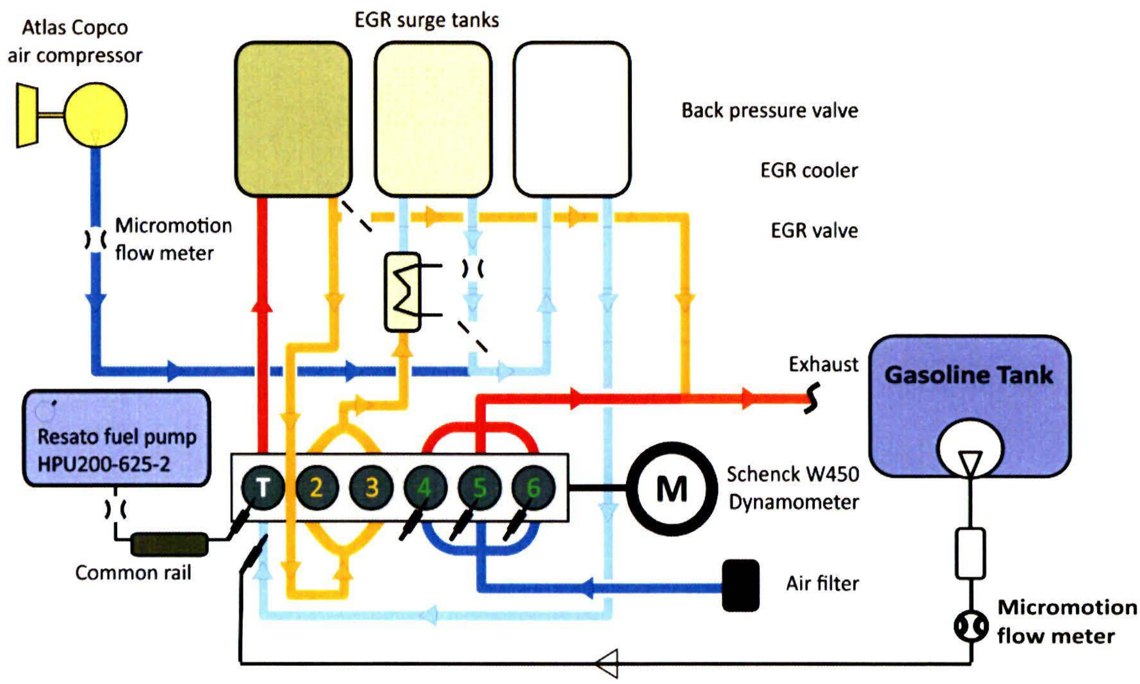


Figure 2.1 Schematic of experimental setup CYCLOPS:
modified DAF engine using separate fuel, air and EGR systems for one dedicated test cylinder

Direct injection of fuel in to cylinder 1 is provided by a double-acting air-driven Resato HPU200-625-2 pump, which can deliver a fuel pressure up to 4200 bar. An accumulator is placed near (~ 0.2 m) the fuel injector to mimic the volume of a typical common rail and dampen pressure fluctuations originating from the pump. The steady state fuel mass flow is measured with a Micromotion mass flow meter. A prototype common rail injector is used which can inject the fuel with a pressure up to 3000 bar. The nozzle used for these experiments has similar characteristics to the nozzle used in [5], which gave the best performance in the Late DI PCCI regime. This nozzle has 8 holes of 0.151 mm diameter with a cone angle of 153 degrees. The injector is controlled by a FLECS ECU where injection timing and duration can be controlled through an in-house written script.

For measuring gaseous exhaust emissions, a Horiba Mexa 7100 DEGR emission measurement system is used. Exhaust smoke level (in Filter Smoke Number or FSN units) is measured using an AVL 415 smoke-meter three times per operating point, of which the average is used. The engine is equipped with all common engine sensors, such as intake and exhaust pressures and temperatures, and oil and water temperature. These quasi steady-state engine data, together with air and fuel flows and emission levels are recorded at 20 Hz for a period of 40 seconds by means of an in house data acquisition system (TUeDACS).The average of these measurements is taken as the value for the operating point under investigation.

Finally, a SMETEC Combi crank angle resolved data acquisition system is used to record and process cylinder pressure (measured with an AVL GU12C uncooled pressure transducer), intake pressure, fuel pressure and temperature and injector current. All of these channels are logged at 0.1 °CA increments for 50 consecutive cycles, which is common practice in combustion indication. From these data the average and standard deviation of important combustion parameters, such as CA10, CA50 and IMEP are calculated online by the SMETEC software.

2.3 Short-term application

For the measurement series done for Vialle the engine set-up is adjusted to inject pre-mixed diesel-butane blends directly into the cylinder. A standard LPG tank is coupled to a high pressure pump and provides the engine with a direct injection of one of the pre-mixed blends. A purge tank is mounted to the return line of the injector to store the leak flow of the injector, as can be seen in Figure 2.2.

In the butane + diesel tank (1) a built-in fuel pump will provide fuel to the high pressure pump at about 5 bar, depending on the vapor pressure of the blend. Second, the diesel-butane blend (1) is provided to the high pressure pump (2) with a quick-connector coupling system for easy change of the different blends. After this the high pressure pump is set at 1500 bar and will provide the blend to the fuel buffer (3). The injector leak flow is captured in a surge tank (6). This tank will not reach a higher pressure than 10 bar, because this is the vapor pressure of pure butane at 90°C, which is the temperature of the cylinder head at the leak flow of the injector.

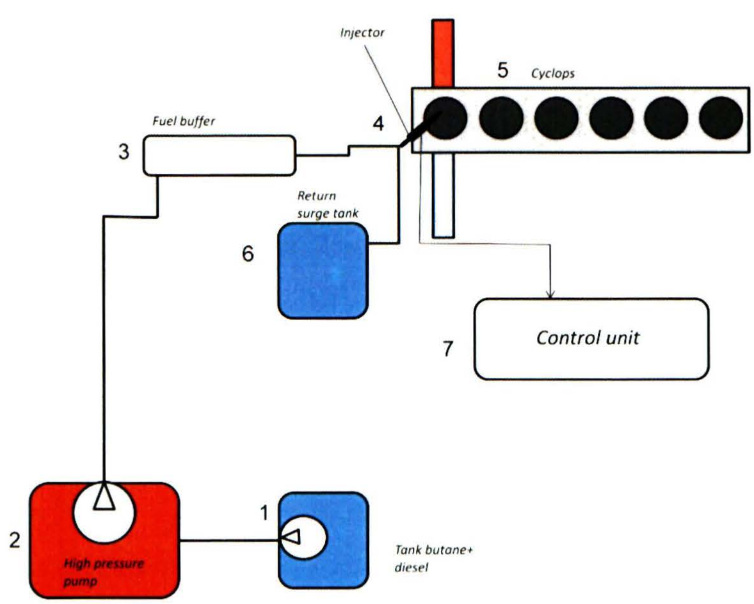


Figure 2.2 Adjustments performed for DI of diesel-butane blends. 1. LPG-tank 2. High pressure fuel pump 3. Fuel buffer 4. Injector 5. Cyclops Engine 6. Return surge tank 7. Control unit.

The most important parameters in this experiment are fuel consumption (and, related to that, combusting phasing) and emissions, because these play an important role in understanding the differences between using a two fuel blend and using regular fuel. To investigate these parameters, two different loads are studied: 6 bar IMEP and 10 bar IMEP, representing 25% and 45% of the total power, respectively. Lambda, defined as the ratio between the current air fuel ratio and the stoichiometric air fuel ratio, is kept constant at approximately 1.3 to see significant differences in soot emissions between the blends. A constant Start of Actuation (SOA) resulting in a CA50 of 5, 7.5 and 10 degree ATDC (for diesel-only) is used to reveal clear differences in combustion phasing between the blends. When start of actuation is unchanged, no adjustments have to be performed in the engine management system, which can be desirable for ease and costs. The load is maintained constant by a longer actuation duration on three injection points when more butane is added to the blend. In practice this can be realized by enlarging throttle pedal position and therefore creating a longer actuation duration.

To sum up:

- Two different loads were used: 6 bar and 10 bar IMEP
- @6 bar IMEP: Inlet pressure 1 bar, SOA -12, -9, -6 degr ATDC
- @10 bar IMEP: Inlet pressure 1.5 bar, SOA -14, -12, -9 degr ATDC
- Approximately constant Lambda 1.3 (± 8%)

<u>Fuel properties</u>

To determine the experimental conditions the energy density of the different blends is calculated. In Table 2.1 all the properties of the blends, except for the estimated cetane number (ECN), are calculated on a mass fraction base, since the content of butane in the blend is given in mass fractions.

Table 2.2 Fuel properties of diesel-butane blends

	wt% Butane	vol% Butane	LHV [MJ/kg]	LHV [MJ/L]	Density [kg/m^3]	ECN	$P_{sat\ max}$ [bar] (at 365 K) Ref [9]
Diesel	0	0	43.00	37.30	830	55.90	-
Butane	100	100	45.75	27.70	550	16.38	12.9
Blend1:But14	14	20	43.39	35.27	790	41.90	4.50
Blend2:But33	33	43	43.92	32.96	757	31.40	7.98
Blend3:But50	50	60	44.38	31.23	690	25.50	9.88

According to Kalgathgi [5] the cetane number scales with the molar fraction. The following empirical relation (2.1), as described by Kalghatgi in [5], is used to calculate the CN for a gasoline like fuel.

$$CN_{Butane} = 54.6 - 0.42 * ON_{Butane} \tag{2.1}$$

The estimated cetane number of the blends is calculated with help of the molar fractions X_i of the two fuels in the blend (2.2) and is used as a measure for the reactivity [3]. The more butane is added to the fuel the less reactive the blend becomes. Note that the energy density per unit volume decreases when the butane content is increased.

$$CN_{Blend} = X_{Fuel\,A} * CN_{Fuel\,A} + X_{Fuel\,B} * CN_{Fuel\,B} \tag{2.2}$$

To determine the vapor pressure of one of the blends in the fuel line, Raoult's law (2.3) can be used as is stated as follows: the vapor pressure of an ideal solution is dependent on the vapor pressure of each chemical component times the mole fraction of the component present in the solution, cf. [8]. In this case the vapor pressure of diesel can be neglected:

$$p_{sat}^{max} \simeq X_{but} * p_{but}^{max}(T_{max}) \tag{2.3}$$

With X_{but} representing the molar fraction of butane in the blend.

2.4 Long-term application

To control auto ignition of the injected fuel based on fuel reactivity, gasoline is added through port fuel injection, see Figure 2.1. The gasoline is provided by a standard Vialle tank with inbuilt fuel pump. A filter is placed in the fuel line and the flow through the gasoline fuel line is measured with a Micromotion mass flow meter. The injector used is a Vialle28 LPG injector placed on the intake manifold with an angle of 120 degrees resulting in an injection spray positioned on the intake valve for better vaporizing of the gasoline, see appendix A. The gasoline injector is controlled by a MoTeC M400 engine management system. With this system injection timing and duration, as is done with the direct diesel injection, can be controlled.

<u>Measurement matrix</u>

In this investigation two different combustion concepts using a port fuel injection system are investigated. First conventional diesel combustion timings (Start of injection near TDC) are used to ignite the mixture of gasoline and air. Secondly, premixed charge compression ignition is investigated by using one or two early diesel injections to control the reactivity of the fuel. RCCI combustion is highly dependent of fuel reactivity and in-cylinder conditions, characterized as the temperature, pressure and air fuel ratio. The potential of these concepts described above are investigated with respect to combustion phasing, performance and emissions.

To investigate the potential of online control of fuel reactivity, first the maximum allowable fuel fraction of gasoline via a port fuel injection is determined. This is done at low loads when running in conventional diesel combustion mode. By testing the following measurement matrix, limitations of using a port fuel injection system on a HDDE can be clarified. This matrix will be tested at 0% EGR and at 40% EGR. Start of gasoline injection is placed just after intake valve open and after exhaust valve close (i.e. 300 deg bTDC) to spray directly into the cylinder and avoid blow-through of gasoline (see appendix A.3). The measurements are performed using a geometric compression ratio of 15:1 and the following conditions:

Conventional diesel combustion

- Two loads: 4 and 6 bar IMEP.
- Two EGR levels: 0% and 40% EGR.
- Three intake pressure levels: 1, 1.25 and 1.5 bar.
- Injection sweep in CDC regime: -5,-10,-15 degr ATDC.
- Replacement of diesel fuel by port fuel injected gasoline up to 60wt%.
- Two port gasoline injection timings (SOGI): 140°CA bTDC and 300°CA bTDC

Almost all possible combinations are tested, taking the limitations mentioned above into account. Except for the gasoline injection timings and the intake pressures, only one load and one EGR point was investigated. Testing this full matrix enables the identification of the possible fuel fraction of port fuel injected gasoline in the CDC regime.

In the second part of this measurement series auto-ignition based on fuel reactivity will be tested as follows:

Reactivity Controlled Compression Ignition

- One load: 9 bar IMEP.
- One EGR level: 60% EGR.
- Three intake pressure levels: 1.0, 1.25 and 2.0 bar.
- Injection sweep with one injection in PCCI regime: -40 to -90 degr aTDC.
- Injection strategy with two direct injections at -90 until -40 degr aTDC and -25 until -5 degr aTDC with 10 and 5 degree increments respectively.
- Replacement of diesel fuel by port fuel injected gasoline up to limitations of experiment set-up.

Again all possible combinations are tested, taking the limitations mentioned above into account. Except for the two diesel injection strategy, only one intake pressure was investigated. Testing this full matrix enables the identification of the potential of port fuel injected gasoline in the RCCI regime.

2.5 Experimental procedure

Before the start of any measurement series several external systems are switched on, for example the compressor for the intake air, the Resato fuel pump and the cooling water system. After starting the engine a warm-up procedure is followed until the cooling fluid and oil temperature are 90 and 80 degrees Celsius respectively. The operating conditions are set to the desired values with an operating engine speed of 1200 RPM. When all conditions are stabilized, measurements can be performed. The direct diesel fuel injection pump is set to 1500 bar, unless otherwise stated, and fuel temperature is 30 degrees Celsius. EGR flows through an EGR cooler and is, unless otherwise stated, cooled to a temperature of ca. 300 K. The net fuel pressure of the port fuel injection system is set to ca. 3 bar by controlling the rotational speed of the fuel pump in the gasoline tank.

Depending on the measurement matrix gasoline and diesel fuel flows are both controlled to keep the load constant at a certain injection timing. Diesel fuel can be injected once or twice per compression stroke for investigating fuel controlled auto-ignition. For every combination of operating conditions under investigation, a sweep of start of actuations (SOA) of the diesel injector will be performed. For every SOA, the engine is stabilized until the standard deviation of IMEP is below 0.1 bar and all the exhaust emissions are constant. Starting from conventional CI timings, SOA is advanced at 5 degree increments, being a tradeoff between ease of measurement and accuracy, skipping SOAs at which combustion is not acceptable. Acceptable combustion in this aspect is defined by both engine hardware limitations and combustion quality targets.

Hardware limitations of the present setup define combustion as acceptable when resulting in:

- A maximum pressure rise rate (MPRR) of 45 bar/°CA, which is a common unit used with engine hardware experiments. This value was chosen for engine hardware safety reasons. Also this MPRR is coupled to maximum engine noise, which should not exceed 95 dB according to DAF Trucks N.V.
- At conventional timings, the air excess ratio λ cannot drop below 1.3 because of emissions reasons, quenching and temperatures of the combustion.
- Hydrocarbon emission not exceeding 5000 ppmC. Apart from the environmental implications of hydrocarbon emissions, they are also seen as an indicator of wall wetting occurring in the test cylinder and therefore they are limited to prevent oil dilution and liner damage.

These conditions have to be met for all investigated points. Furthermore, for operating conditions to be seen as viable, the combustion should also meet the following targets:

- NO_x emission levels below 250 ppm as target.
- CO emission levels below 4000 ppm.
- Bosch Filter Smoke Number (FSN) below 2.0, with below 0.5 as a target.
- Relative standard deviation of the Indicated Mean Effective Pressure (IMEP) below 2.5%.

Fuels

For auto-ignition control based on fuel reactivity a low reactivity fuel is necessary to extend ignition delay. For this investigation gasoline is used in combination with diesel to create the desired reactivity of a mixture. By choosing these two practical fuels, the experiments can be closely related to future applications.

In table 2.3 below one can find the specifications of the two fuels.

Table 2.3 Fuel specifications of diesel and gasoline

Properties	Diesel EN590 [Appendix B]	Gasoline RON 95
T10	210° C	65° C
T50	268.5° C	115° C
T90	333.3° C	185° C
DCN	55.9	14.7
Total aroms	30.7 wt%	35.0 wt%
Density @300 K	825 kg/m^3	825 kg/m^3
LHV	41.54 MJ/kg	44.40 MJ/kg [14]

2.6 Data analysis and definitions

As mentioned before a SMETEC Combi crank angle resolved data acquisition system is used to record and process in-cylinder pressure, intake pressure, fuel pressure and temperature and injector current. With these readings the IMEP is calculated as follows:

$$Imep = \frac{\int pdV}{\pi(\frac{B}{2})^2 * S} \qquad (2.4)$$

With B the bore and S the stroke of the XE DAF engine.

Normally when comparing emission levels and performance of a heavy duty diesel engine (HDDE) it is common practice to calculate these brake specific, i.e. with respect to the power output at the crankshaft. This power at the crankshaft is characterized by the Brake Mean Effective Pressure (BMEP). In the present test setup, measured torque is influenced by many external factors, which would make the test results not reproducible. Therefore in this case the IMEP as calculated from the in cylinder pressure signal is used.

The IMEP as calculated in equation 2.4 takes in account the pumping losses coming from an exhaust turbo compressor. In the present test set-up the intake pressure is controlled by an external compressor and pumping losses are negligible comparing to a stand-alone heavy duty engine. For this reason the gross IMEP, which ignores pumping losses, is used to calculate emission levels and fuel consumption.

To calculate the performance both flows, gasoline and diesel, are directly measured and added to one total fuel consumption. For calculating indicated efficiency the total injected mass is used, however when referring to the thermal efficiency the actual converted fuel is used. In section 4 port gasoline injection further description is given.

By analyzing the data for combustion from the data acquisition some definitions are made. For the start of combustion CA5, defined as the crank angle where 5 percent of the injected mass is burned, is used. CA5 is chosen because of a significant higher stability compared to CA2. According to the work of Leermakers [5] the difference between Start of Actuation and Start of Injection is a constant value, equal to 4 °CA.

Given the assumptions made above, the following definitions are used to characterize the combustion. Here combustion delay is used to describe the average mixing time, while ignition dwell represents the separation between injection and combustion events.

- Start of Injection = SOA + 4 °CA [°CA].
- Combustion Delay = CA50 - SOI [°CA].
- Ignition Dwell = CA5 - EOI [°CA].
- Ignition Delay = CA5 - SOI [°CA].
- Burn Duration = CA90 - CA5 [°CA].

Before every measurement a baseline measurement is performed. This is done to compare the baseline measurements and to be sure the reliability of the measurement is sufficient. When all the baseline measurements are compared a standard deviation between the measurements can be defined.

2.7 Summary

In this chapter the details on the experimental set-up and the experimental performance were given. First the experimental set-up was discussed in section 2.2. Furthermore, the experiments are divided into two parts: the short-term application and the long-term application. In the first application, experiments using diesel-butane blends are described. In the second part, the adjustments made on the engine test-rig for port gasoline injection are explained. The measurement matrix constructed for this program, which contains the different levels of different parameters under investigation, was given for both experiments. In subchapter 2.6, data-analysis and definitions were discussed.

In the following part of the report, the results are discussed. In the first part the experiment-results using diesel-butane blends are shown. In the second part, starting from Chapter 4, the experiment-results using port gasoline injection for RCCI combustion are given.

Results Part I

3. Dual fuel DI of diesel-butane blends

3.1 Introduction

As mentioned in the introduction, LPG is often used in heavy-duty diesel engines combining port fuel injection of LPG and a direct injection of diesel. Alternatively, in this investigation a butane-diesel pre-mixed blend is directly injected, using only one fuel system, to determine the influence of butane on conventional diesel combustion.

The key issue for the direct injection strategy of premixed blends is finding the right diesel-butane content at different operating conditions. Apart from the changing combustion phasing, emissions, and pressure rise rate during combustion, the change in fuel consumption is investigated.

In the following chapter the results of the experiments with direct injection of diesel-butane blends will be analyzed. First, combustion phasing will be discussed due to the large influence of butane on start of combustion and burn duration. Fuel properties have extremely large impact on combustion phasing. For simplicity of a fuel system, it can be desirable not to change injection timing when adding more butane to the blend. Naturally this can only be maintained when the difference of combustion phasing is not of great influence on engine efficiency.

Due to the decreasing energy density per unit volume when the butane content in the blend is increased, fuel consumption will rise. In the section Performance, fuel consumption and the decreasing power output when the butane content is increased will be explained with the help of the indicated mean effective pressure (IMEP). After this, the engine-out emissions as a function of butane content are shown. Finally, the maximum pressure rise rate (MPRR) is shown. This parameter is often used as a measure for engine hardware durability and engine noise. At the end of this chapter a short discussion follows.

3.2 Dual fuel direct injection of diesel-butane blends

Combustion Phasing

The fuels under investigation are characterized by the ignition delay due to the lower reactivity of butane, as described in chapter 2. Ignition delay is a very important parameter, which greatly influences burn duration, emissions and pressure rise rate. In this investigation ignition delay is defined as CA5 – SOI (Start of injection). With a longer ignition delay the time to pre-mix is elongated and with the help of local leaner mixtures emissions can be reduced. To show the shift of combustion CA5, defined as the crank angle where 5 percent of the injected mass is burned, is chosen as start of combustion. For a fixed start of actuation (SOA) one can see, in Figure 3.1a, that due to the lower reactivity of butane the ignition delay is extended, meaning a retarded start of combustion. The difference between the fuels diesel and But14 is less than 1.5 deg CA, but from But33 one can see that the lower reactivity influences the ignition delay with an extension of larger than 2 deg CA.

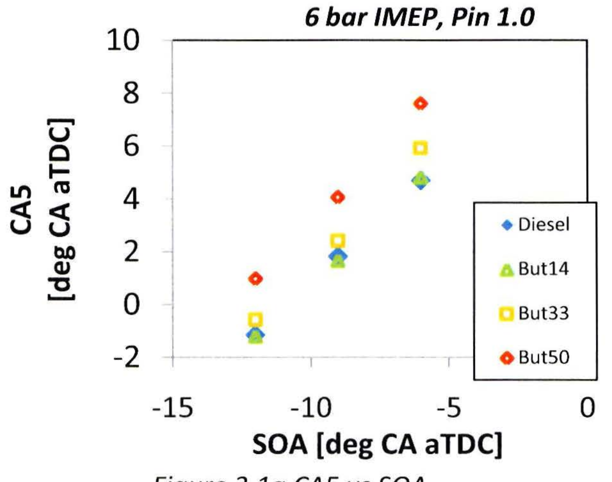


Figure 3.1a CA5 vs SOA

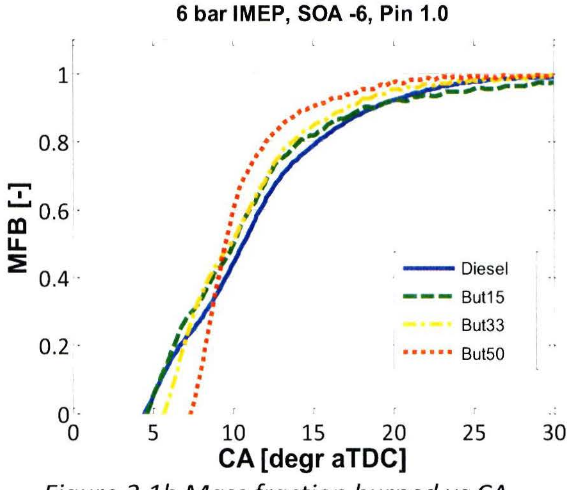


Figure 3.1b Mass fraction burned vs CA

In Figure 3.1b one can again see that due to the lower reactivity of butane, heat release starts later in the power stroke. Although combustion starts later, end of combustion is more near TDC when more butane is added to the blend. This shorter burn duration (BD) can be related to the more rapidly vaporizing butane in the blend. This also means that due to the shorter burn duration, CA50 (crank angle where 50 percent of the injected mass is burned) is placed more near TDC, resulting in higher peak temperatures (see 3.2 Emissions section).

With a shorter BD the timing of combustion can be controlled more easily, benefitting the efficiency. The down side of the shorter BD is often a higher MPRR which can result in engine hardware damage and exceeding of the engine noise limits (see 3.2 MPRR section). The burn duration calculated for the blend with the largest amount of butane is 8 degrees CA shorter compared to 15 deg CA for diesel-only.

At higher loads the difference between the blends in combustion phasing can be neglected. In these loads it is believed that pressure and temperature are dominant in combustion phasing compared to fuel composition [5, 22]. For that reason no figures of these measurements are shown here.

Performance

In spite of the benefits of shorter burn durations, using more butane in the fuel results in a fuel consumption penalty in L/km. This fuel penalty occurs because of its lower energy density. Unlike the other figures, in Figure 3.2 the actuation duration (AD) of the injector is kept constant. In this experiment one can see that the indicated mean effective pressure (IMEP) drops with a maximum of 8 percent (for But50) when injecting the same duration with every blend.

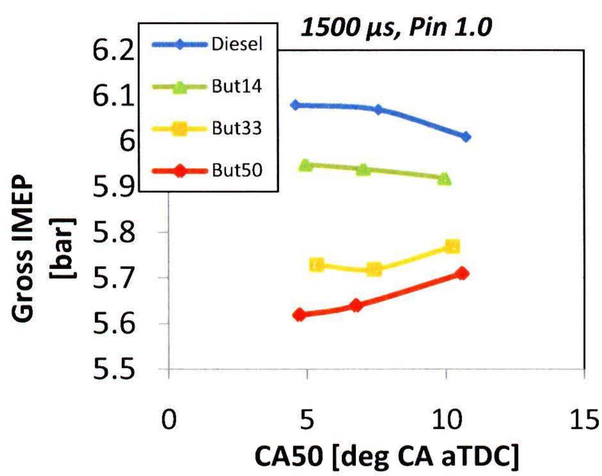


Figure 3.2 IMEP vs CA50 (constant AD)

One can see in Figure 3.3a that the fuel consumption in mass does not change much, even in favorite of the 50wt% butane fuel when it is compared to diesel. This can be explained on the one hand by the higher energy density per mass of butane and on the other hand by a shorter burn duration. Yet, due to the lower energy density per volume, the fuel consumption in liters/kWh rises when the mixing ratio is increased. Although the indicated efficiency does not change significantly when maintaining the IMEP constant, the fuel penalty in liters raises with 10 to 20 percent, which can be seen in Figure 3.3b.

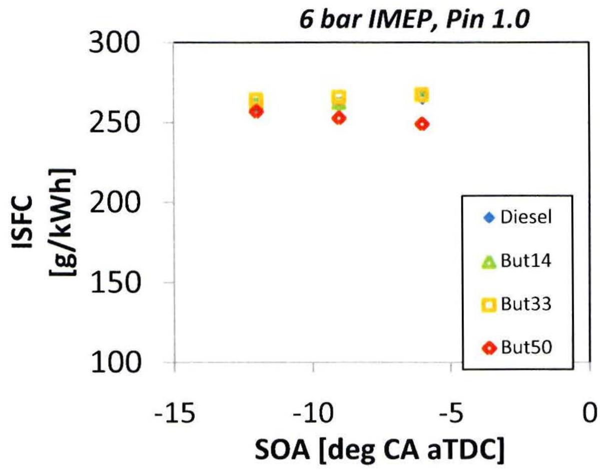


Figure 3.3a ISFC [g/kWh] vs SOA

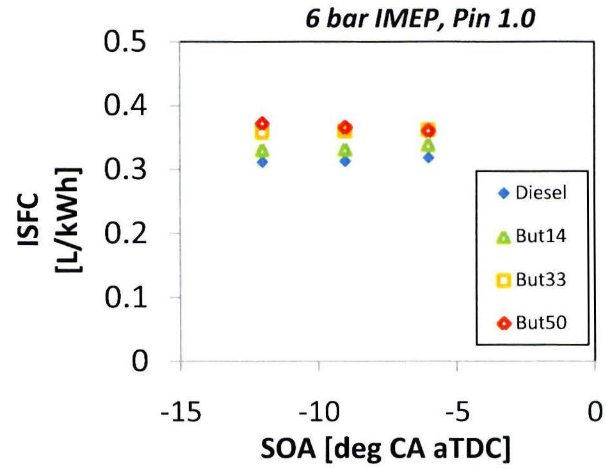


Figure 3.3b ISFC [L/kWh] vs SOA

At first sight the indicated efficiency (Figure 3.4), defined as the net power divided by the injected energy, is slightly lower when the butane content is increased. However, with the 50wt% butane blend the indicated efficiency increases again. The exact explanation of this change in efficiency is not clear. The indicated efficiency is influenced by combustion phasing, representing the timing of CA50. Although this effect is clear to see in these experiments, further research is needed to provide certainty.

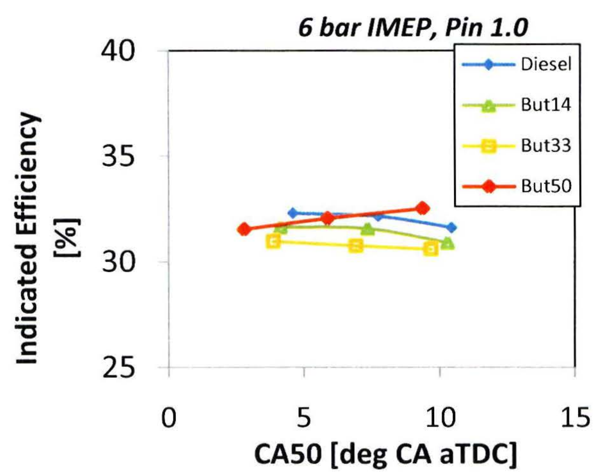


Figure 3.4 Indicated efficiency vs CA50

Emissions

The rapid vaporization of butane in the blend, as mentioned before, has the advantage of better pre-mixing with the air compared to regular diesel, certainly with an extended ignition delay. Due to this better premixing, local spots become less rich and CO emissions can be reduced. In the figures below the axes are chosen such that trends between the different blends are easily recognizable. Unlike the other blends, in Figure 3.5a it is noticed that the blend with the largest amount of butane shows a declining trend.

One possible reason for this is flash vaporization; due to the high butane fraction, the liquid spray immediately "flashes" into vapor. The result is a better premixed, more complete combustion. NO_x is formed with close-to-stoichiometric and lean combustion at high temperature. These conditions are achieved for the highest butane-fraction fuel and will, due to higher flame-temperatures, produce more NO_x. In Figure 3.5b one can see that for lower butane fractions, NO_x emissions are not changed significantly.

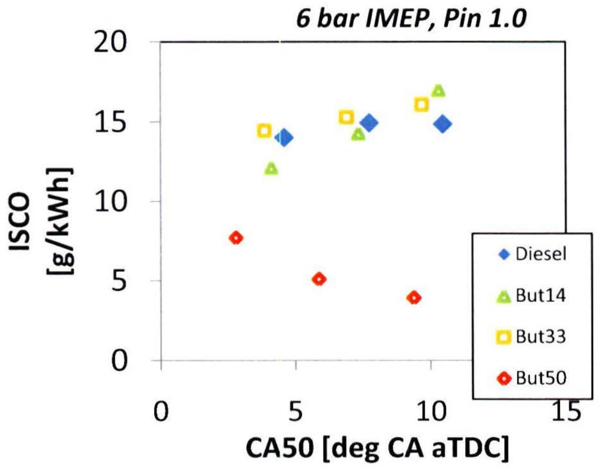


Figure 3.5a ISCO vs CA50

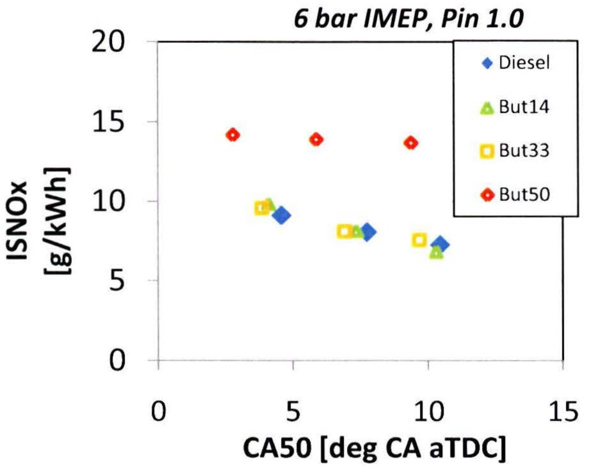


Figure 3.5b ISNOx vs CA50

Figure 3.6a shows the unburned hydrocarbon emissions. Due to more butane in the blend and a faster vaporization of the injected mass compared to diesel it is believed by [1] that more mixing gas is trapped into the crevice volume of the combustion chamber. However, this is highly unlikely because of the shortage of time when injecting directly at 5 degrees CA BTDC. A more likely reason to relate higher HC emissions are local high air fuel ratio spots due to the better pre-mixing, possibly resulting in local flame quenching ("overleaning"). Note that NO_x emissions are raised, meaning that at least locally, in the richer region, the combustion temperature becomes higher when the butane content is increased.

The air fuel ratio is proven to be dominant for producing soot emissions. In these experiments the global lambda is approximately kept constant by controlling intake pressure and actuation duration, directly related to IMEP, of the injector. This way the differences in producing soot emissions is directly related to the difference in fuel composition. In Figure 3.6b one can see that soot-emissions are greatly reduced by adding butane to the blend. This can be explained by four reasons. First, it can be related to the rapidly vaporizing fuel, creating locally less rich mixtures. Another way butane can reduce local rich spots, and probably the most dominant reason for lower soot emissions, is by its lower reactivity. Due to its lower cetane number, ignition delay is elongated and more time for the fuel to mix with the air is created. Third, it is believed that butane in the blend will promote diesel atomization by enhancing gas perturbation with fluctuating pressure in the spray field and enhancing mixing with air in-cylinder to reduce the soot formation in the diesel-fuel spray [1, 4]. Next to the higher vapor pressure, another important aspect of butane is the molecule structure.

Aliphatics, like butane, are known as less soot forming fuels. Although these four explanations are all true to some extent, further research is needed to find out which one is the most dominant in producing less soot compared to regular diesel.

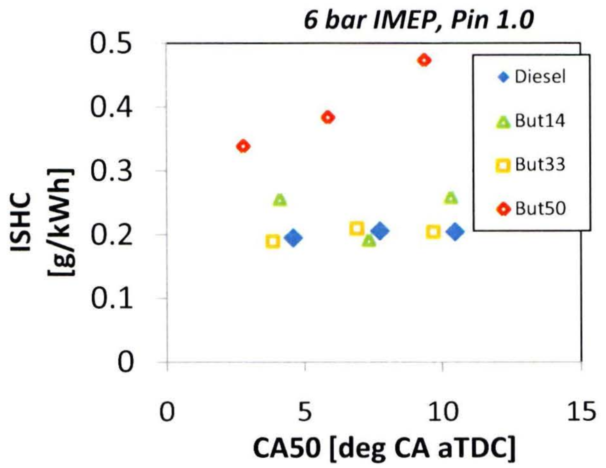


Figure 3.6a ISHC vs CA50

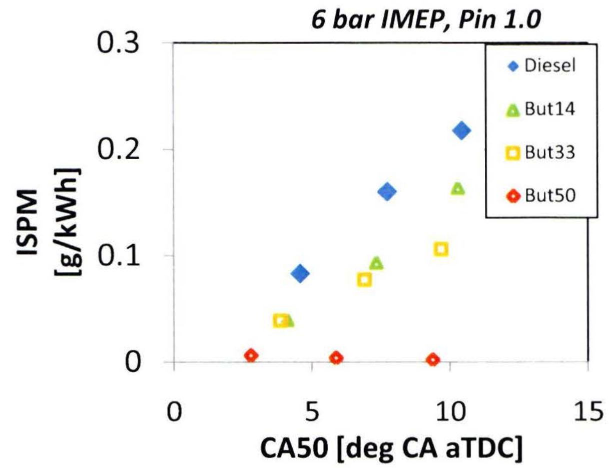


Figure 3.6b ISPM vs CA50

Although lambda is kept similar to the experiments done at low loads, due the higher intake pressure air fuel ratio at higher loads is slightly higher compared to the lower loads, meaning an overall leaner combustion. Under these conditions NO_x and soot emissions are formed without significant difference between the different blends, which we can see in Figure 3.7a and 3.7b.

Of course when it is possible to maintain the same exhaust emissions with more butane added to the blend, several possibilities to lower the costs can be created. This is further addressed in the Discussion, section 3.3.

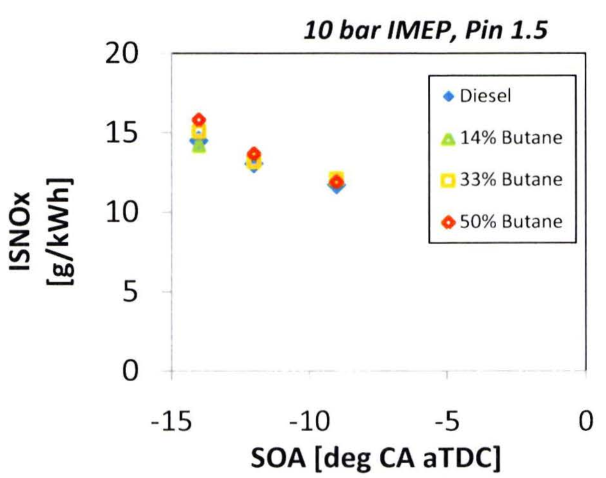


Figure 3.7a ISNOx vs SOA

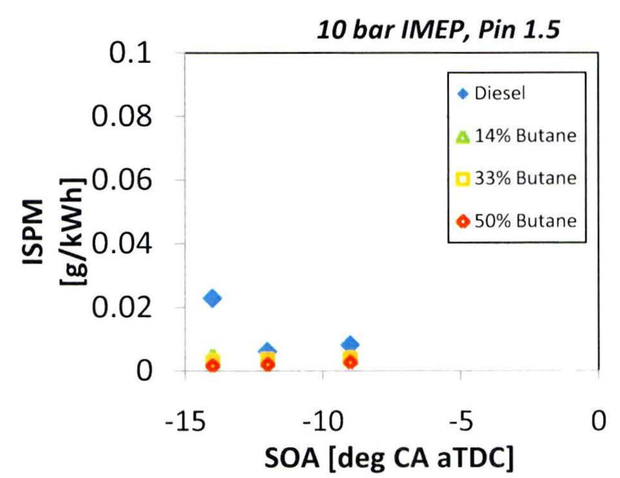


Figure 3.7b ISPM vs SOA

Maximum pressure rise rate

The maximum pressure rise rate (MPRR) is a measure for the mechanical load and the noise of an internal combustion engine. It is dominated by burn duration and therefore the combustion phasing, represented by CA50, which we can see in Figure 3.8a. One can see that the difference between the blends becomes evident at higher butane contents. The shorter burn duration allows a larger heat release in shorter time, resulting in high pressure rise rates. At the same injection timing CA50 is advanced. The shift in combustion phasing increases maximum pressure rise rate and this can exceed noise limits, which are around 25 bar per degree CA. Note that burn duration is dominant in combustion phasing compared to the larger ignition delay of a specific blend (Figure 3.8b).

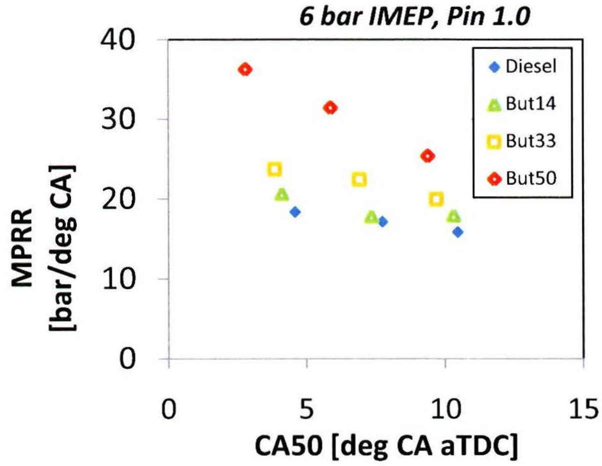


Figure 3.8a MPRR vs CA50

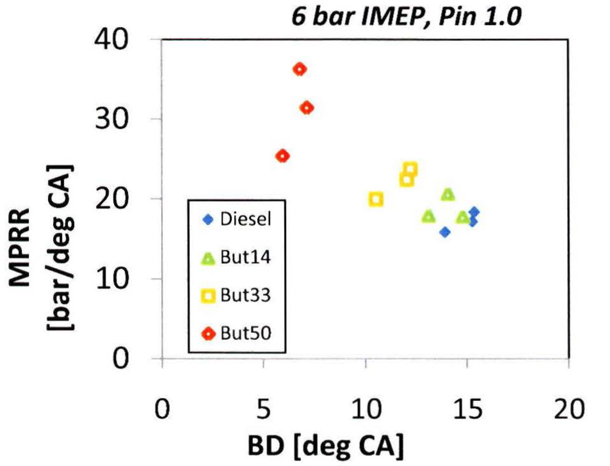


Figure 3.8b MPRR vs BD

3.3 Discussion

In the results shown above the load is kept constant using an increasingly longer actuation duration for the butane blends. Due to the lower energy density of butane the fuel consumption in liters rises with 10 to 20 percent. When it is desired to use a butane blend without changing injection timing and actuation duration, a certain amount of butane can be added. Due to a shorter burn duration and longer ignition delay, combustion phasing changes. To make optimal use of the change in combustion phasing, which can lead to a better efficiency and lower soot emissions, the amount of butane should not exceed 33wt% in the blend to compromise for the raise in CO and NO_x emissions. Next to the lower soot emissions with this butane content, engine noise and mechanical load do not exceed the limits in these experiments. When the butane fraction is increased even further, shorter burn durations will lead to high peak temperatures and will increase NO_x emissions and pressure rise rates.

A simple calculation shows that with a butane content of 33wt%, fuel savings per volume will be 17 percent with today's fuel prices of diesel and LPG (table 3.1). Fuel consumption in liters/kWh will be 10 percent more, but is easily compensated by the lower fuel price. The greatest benefit is derived in terms of soot emissions, where a decrease of 50 percent is realized without producing more of the other regulated harmful emissions.

Table 3.1 Price per liter of diesel, LPG and the tested blends, 3 nov. 2010, The Netherlands.

Diesel [€/L]	LPG [€/L]	But14 [€/L]	But33 [€/L]	But50 [€/L]
1.23	0.74	1.13	1.02	0.94

In these experiments butane is used. When LPG is used, which contains a certain amount of propane, depending on the season, it is believed that due to the even lower reactivity of propane emissions can be reduced even more. Adding propane will lead to higher ignition delays which will enhance pre-mixing. The downside of this shift in combustion is that injection timing has to be controlled when efficiency should not be harmed. Note that the energy density per volume of propane is slightly lower than that of butane, so it will increase fuel consumption to some extent.

The cetane number of LPG is estimated on 12.5 based on LPG existing of 70 vol% propane and 30 vol% butane, which is about 23% lower than that of pure butane. When assuming But33 is the best fuel in the experiments, by using LPG the results can be influenced in a positive way, but the fuel fraction of LPG in the blend will be lower. Note that these influences provide ample reason for more research concerning, for example, a higher vapor pressure in the fuel line.

3.4 Conclusions and recommendations

Butane-diesel blends with different mixing ratios were tested on a heavy duty direct injection diesel engine. From the results one can see that when the butane content is increased to a certain amount of extra butane, combustion phasing is influenced. Start of combustion is retarded when start of injection is kept constant. Ignition delay is enlarged and the time for the fuel to mix with the air is elongated. Unlike CA5, CA50 is advanced. This is hypothetically related to the more rapidly vaporizing butane in the blend, resulting in closer-to-stoichiometric conditions and therefore shorter burn duration.

When increasing the amount of butane in the blend the fuel consumption in liters will rise. Due to the lower energy density per volume of butane the fuel consumption in L/kWh raises with 10 to 20 percent, depending on the butane content. When butane content is increased to 50 percent, combustion phasing changes significantly and might improve the indicated efficiency by a shorter burn duration and a better timing of CA50.

Owing to the larger ignition delay, emissions are reduced. CO and soot emissions are decreased radically due to the better pre-mixing. NO_x and HC emissions can both slightly increase, depending on the butane content. An explanation for the higher HC emissions is, owing to the more rapidly vaporizing butane, local lean spots compared to diesel sprays. According to [1] this rapid vaporization of butane can also lead to more mixing gas trapped into the crevices of the combustion chamber. However, this is probably not true because of the lack of time to reach the crevices when injecting directly near TDC. The higher NO_x emissions can be related to the shorter burn duration, which results in high peak temperatures.

Noise and engine hardware limits can be exceeded when maximum pressure rise rate is raised. When the fuel fraction of butane is increased, MPRR is raised due to the shift in combustion phasing and can lead to higher peak temperatures. Tentatively, our results seem to indicate that the amount of butane in the blend should not exceed 33 wt% for engine emissions and hardware safety reasons.

At higher loads (10 bar IMEP) no significant difference between the blends can be seen in emissions, combustion phasing and maximum pressure rise rate. This means that for the higher load the butane content can be increased up to 50 wt% without increasing emissions and MPRR.

Results Part II

4. Initial dual fuel PFI tests

4.1 Introduction

As discussed in the introduction the results from this experimental study are divided into two parts: a short term scenario and a long term scenario. In this part the port fuel injection system for gasoline is investigated for reducing emissions in the long term. This second part of the results is again divided into two parts. In the first part, Chapter 4, different operating conditions using port gasoline injection, up to 60 wt% of the total injected fuel mass, are investigated. From these measurements the best operating conditions for running in PCCI mode are selected and used in the test matrix for Chapter 5.

In section 4.2 combustion phasing, emissions and engine performance are investigated as a function of gasoline addition. In the subsequent sections 4.3 and 4.4 different operating conditions are shown for determining the best conditions for running in RCCI mode. In the last section, section 4.5, the conclusions of using the concept with gasoline PFI for fuel reactivity controlled auto-ignition are given.

4.2 Port injection of gasoline (PIG)

Combustion phasing

Generally the combustion phasing, from a heat release point of view, can be defined with four stages [16]: First the period between the start of fuel injection into the combustion chamber and the start of combustion, defined as the ignition delay phase. In this investigation this is defined as CA5-SOI. The second phase, defined as the premixed combustion phase, combustion of the fuel, which has mixed with air to within the ignition limits during the ignition delay period, occurs rapidly in a few crank angle degrees. After this the mixing-controlled combustion phase follows. In this phase the burning rate (or heat release rate) is controlled by the rate at which mixture becomes available for burning. While several processes are involved –liquid fuel atomization, vaporization, mixing of fuel vapor with air, chemical reactions – the rate of burning is primarily controlled by the fuel vapor-air mixing process [4]. In the fourth and last phase, the so-called late combustion phase, heat release continues at a lower rate, due to decreasing pressure and temperature, into the expansion stroke.

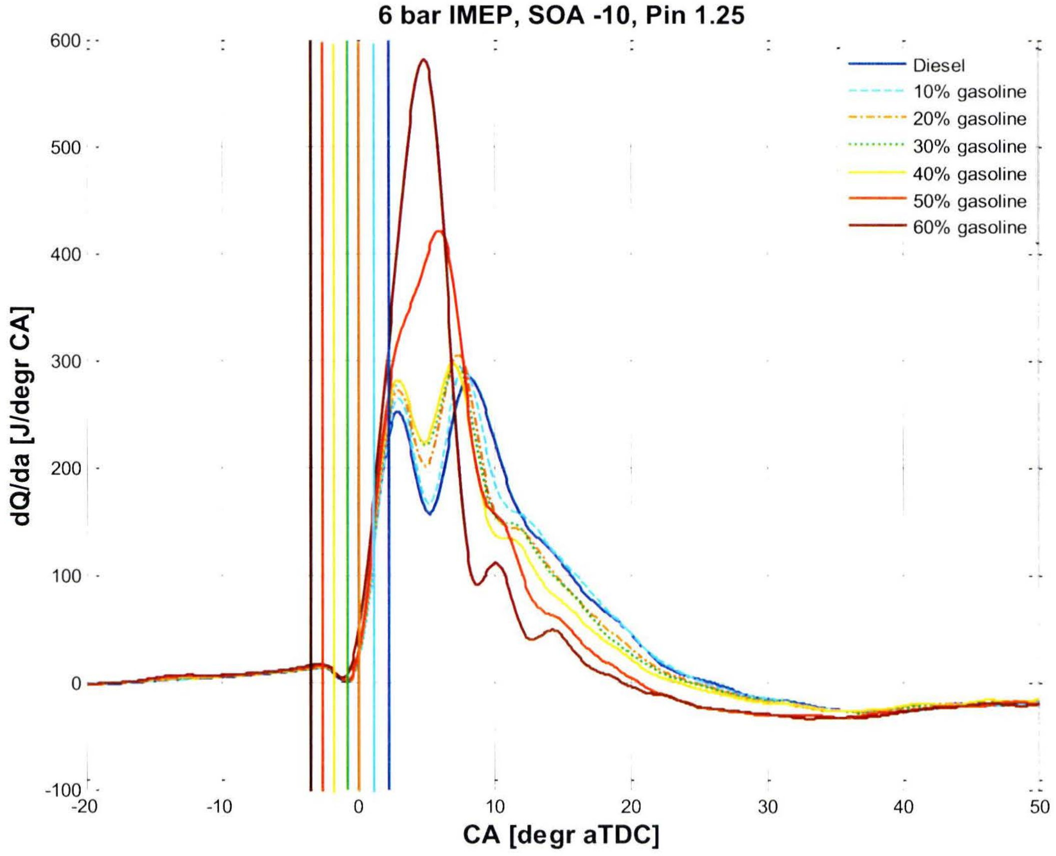


Figure 4.1 Heat release rate at 6 bar IMEP for different gasoline fuel fractions. The vertical lines represent end of diesel-injection (EOI).

In Figure 4.1 one can see that during the ignition delay phase, heat release does not start earlier when the gasoline fuel fraction is increased; it is still controlled by the start of the diesel injection. Here measurement series of 6 bar IMEP are used for indicating trends between different mixing ratios of gasoline. Apart from the heat release, the end of diesel injection (EOI) is also shown with vertical lines. In the first large peak, heat release increases when more gasoline is added through port fuel injection. In the second large peak, known as the mixing-controlled combustion phase, one can see that more heat release occurs when more fuel is injected directly into the cylinder. This is due to larger injection sprays which will combust after the premixed diesel and gasoline mixture has ignited. When more diesel is replaced by gasoline through intake port injection, more heat release occurs in the premixed combustion phase. Although the cumulative heat release (Appendix D) is not shown here, it should be mentioned that overall heat release decreases when the gasoline fuel fraction is increased, due to a lower combustion efficiency.

When the fuel fraction of gasoline exceeds 40 wt%, it is believed that a flame propagates through the gasoline mixture and the premixed phase and mixing-controlled phase are combined in one large heat release peak. For these cases, it is likely that flammability limits of gasoline are reached and right after the injection and auto-ignition of diesel, all gasoline burns rapidly. Note that auto-ignition of the gasoline–air mixture does not occur before the diesel injection.

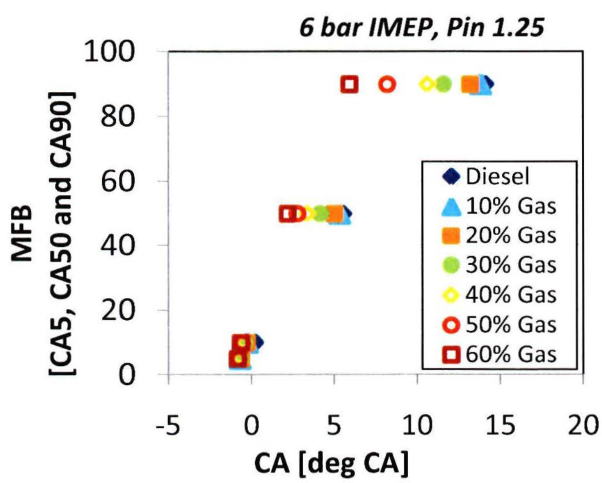


Figure 4.2a MFB vs CA

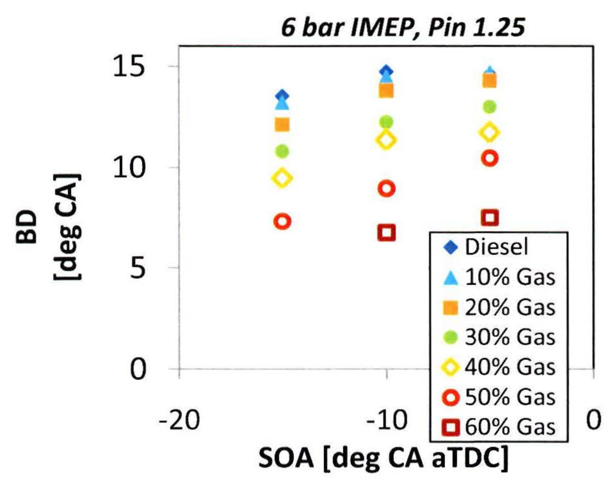


Figure 4.2b BD vs SOA

When using conventional diesel combustion timings, start of combustion, here defined as CA5, is controlled by the start of the diesel injection. Up to a gasoline fraction of 60 wt%, the gasoline injected through the intake port does not auto ignite due to compression, which we can see in Figure 4.2a. The part of the combustion phasing from SOC until CA50 shows less differences when the gasoline content is increased. In the second phase, the premixed combustion phase, flame propagation leads to high flame speeds and thus shorter burn durations. In Figure 4.2b one can see that burn duration decreases due to the larger amount of premixed fuel (gasoline). Note that the amount of gasoline injected up to 40 wt% is still small in an absolute sense, so diesel combustion on these measurement points is dominant, resulting in high temperature turbulent combustion.

Emissions

The higher gasoline fuel fraction does not only influence combustion phasing and performance; due to the leaner premixed mixture of gasoline and air, emissions change too. According to the Shell research group [29], flammability limits of gasoline are approximately twice the stoichiometric air/fuel ratio at 19.2 bar and 700 K. In the current investigation, and in particular the measurement series discussed in this chapter, the air fuel ratio of the premixed gasoline and air mixture approaches these flammability limits just before adding the diesel spray. It is therefore that certain changes are noticed when more diesel is replaced by port gasoline injection.

A start of actuation closer to TDC can lead to a SOC too far in the expansion stroke, which leads to lower temperatures. Figure 4.3a shows that CO emissions increase with these lower temperatures and initially also with the increase of gasoline addition. When the gasoline fuel fraction is icreased, up to a certain point, more incomplete combustion occurs due to lean mixtures. Again, when exceeding a certain limit of added gasoline, CO emissions decrease again due to a more close-to-stoichiometric gasoline and air mixture. At these lower temperatures less NO_x is produced which can be seen in Figure 4.3b. Slightly more NO_x is produced when the gasoline fuel fraction is increased, but this can be seen as negligible.

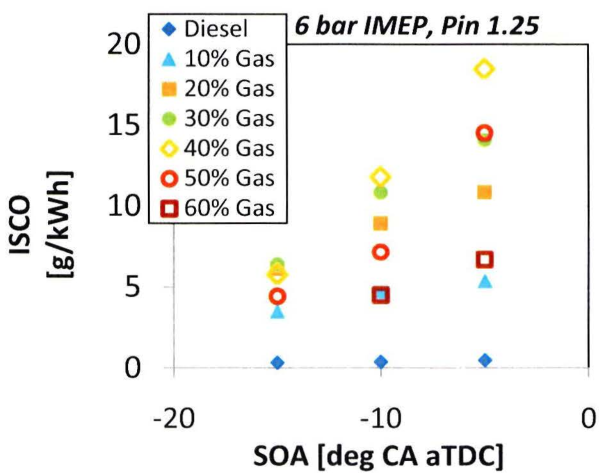


Figure 4.3a ISCO vs SOA

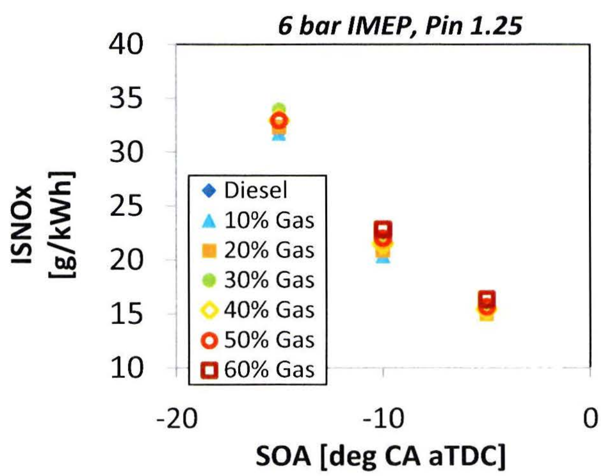


Figure 4.3b ISNO$_x$ vs SOA

In the following figure, Figure 4.3c, one can see that HC emissions increase significantly when more gasoline is added. Several reasons for this can be thought of: First, again as a result of the leaner gasoline and air mixture, incomplete combustion dominates. Second, because of the port fuel injection system and the almost full cycle to premix, unburned hydrocarbons can get stuck into inlet manfold or crevices of the combustion chamber. And third, one could say that due to the nozzle geometry of the injector, which is normally used to inject a liquid mixture of propane and butane, the gasoline is not atomized significantly and can lead to an increase of HC emissions.

Soot production is characterized by combustion of rich mixtures at low temperature and incomplete combustion. However, no significant decrease in soot production is found. Absolute values of soot emissions are very low, making it difficult to see trends. In Figure 4.3d the PM emissions are shown. In this case, at low load, incomplete combustion of gasoline, due to it's higher flammability limits, is dominant. One would excpect that when the gasoline fuel fraction is increased, more fuel is premixed and soot production is decreased.

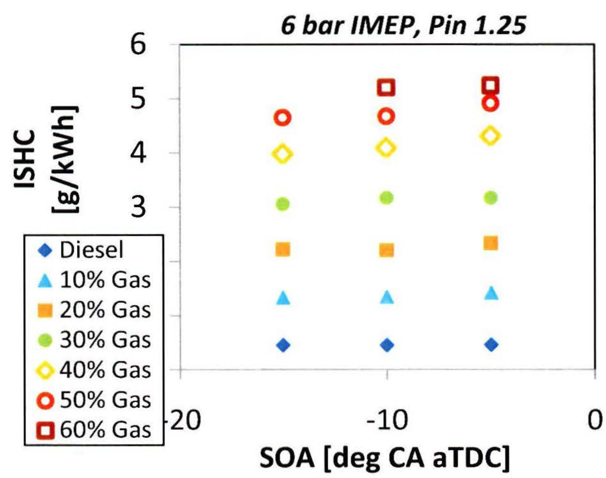


Figure 4.3c ISHC vs SOA

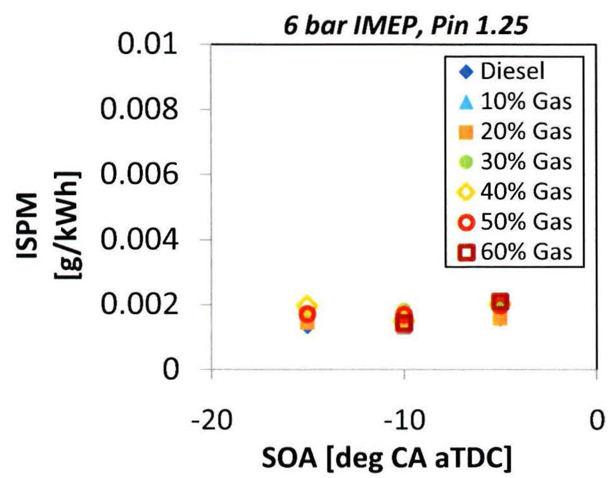
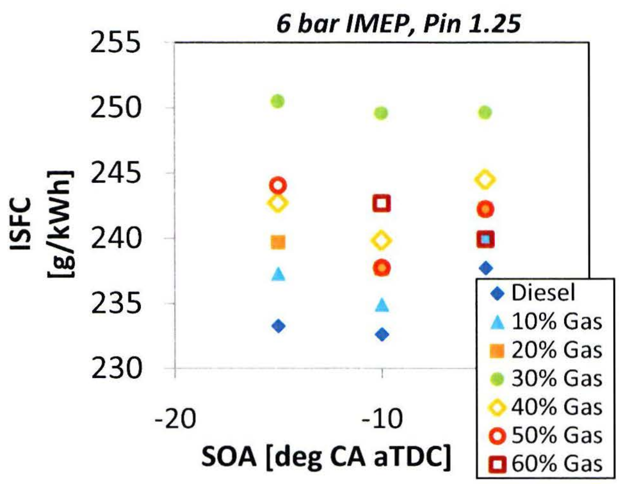


Figure 4.3d ISPM vs SOA

Performance

The reason for implementing the port gasoline injection system is to control auto-ignition based on fuel reactivity in the PCCI regime. However, when the system is used for conventional diesel combustion (CDC), and thus replacing directly injected diesel for gasoline at injection timings near TDC, several issues may cause a bad efficiency.

At 6 bar IMEP the total injected fuel mass is 0.85 g/s. With these relatively small amounts of gasoline, creating a premixed mixture before start of combustion, the in-cylinder air-fuel mixture becomes very lean, typical lambda values in the range of 20 to 6 (see Table 4.1). When this mixture becomes too lean it may not reach the lower flammability limits of gasoline, and will not burn completely. Secondly, unburned hydrocarbons, certainly with premixed air fuel mixtures, can get stuck into the crevices of the combustion chamber during compression stroke, which also negatively affects the efficiency.



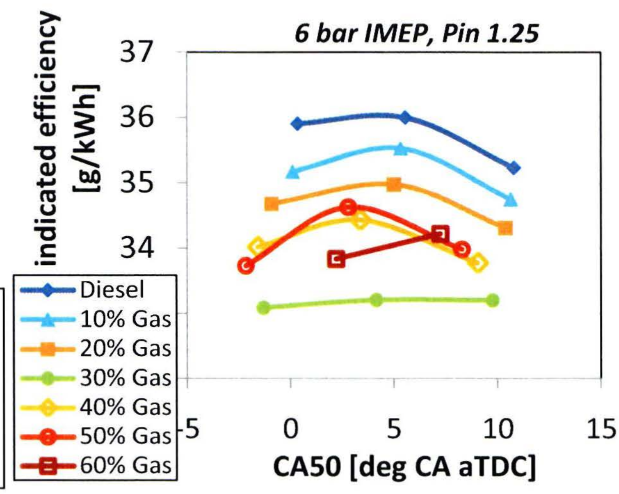


Figure 4.4a ISFC vs SOA *Figure 4.4b Thermal Efficiency vs CA50*

In Figure 4.4a it is shown that fuel consumption increases when the gasoline fuel fraction is increased. As mentioned before, incomplete combustion of the lean premixed gasoline is responsible for this. Although CA50 does not seem to shift when more gasoline is added, at first the efficiency decreases. Indicated efficiency is highly dependent of the timing of CA50, but since CA50 does not change much, the decreasing trend which can be seen in Figure 4.4b must be related to incomplete combustion of gasoline. Secondly, when more gasoline is added, the premixed gasoline reaches higher flame-speeds. In the case of 50 wt% gasoline added, auto ignition of gasoline occurs after diesel injection and efficiency increases again. Moreover indicated efficiency also depends on heat losses and thus shorter burn durations can result in higher efficiencies.

Homogenous combustible gas-air mixtures are flammable, that is, they can propagate flames freely, within a limited range of compositions [36]. The flammability limits, mentioned above, are upper and lower limits of which a combustible gasoline-air mixture can be burned. In this case, the premixed gasoline-air mixture should contain significant gasoline to stay within these limits. Here the limits of flammability are affected by the temperature, pressure, direction of flame propagation and surroundings (such as the turbulent flame of the diesel spray).

By using the flame-speed S_L, an estimation can be made for the flammability limits. When reaching flame speeds of almost zero, one could say the mixture exceeds upper- or lower flammability limits. From Heywood [16] the following equation is used for calculating an approximate flame speed.

$$S_L = S_{L,0} * (\frac{T_u}{T_0})^{\alpha} * (\frac{p}{p_0})^{\beta} \quad (4.1)$$

$$\alpha = 2.18 - 0.8(\phi - 1) \quad (4.2)$$

$$\beta = -0.16 + 0.22(\phi - 1) \quad (4.3)$$

$$S_{L,0} = B_m + B_{\phi} * (\phi - \phi_m)^2 \quad (4.4)$$

Where B_m, B_{ϕ} and ϕ_m are constants for a given fuel, in this case gasoline. The values of these constants can be found in Heywood [16]. One can notice that flame speeds increase with the addition of more gasoline. Note that this is only an approximation and that further investigation is needed for the exact determination of the flammability limits.

Table 4.1 (before diesel injection @ Pin=1.25, -20 CA degr aTDC)

Gas wt %	λ	T [K]	P [bar]	SL [cm/s]
0 %	∞	587	23.4	-
10%	21.3	587	23	-
20%	9.6	587	22.9	-
30%	6.4	587	22.8	0.3
40%	4.9	587	22.5	9.8
50%	4	587	22.3	18.4
60%	3.3	587	22.2	27.7

Port Injected Gasoline (PIG) stuck in the crevices or not playing a role in the combustion process adds to the HC emissions. Thus, it was of interest to analyze the mass of HC emitted per mass of PIG as an indicator of the role that the PIG plays in the combustion event. Plotting this ratio versus the percentage of injected gasoline provides important conclusions. In particular, when the gasoline utilization is constant the HC-trend versus the percentage of gasoline would be expected to be linear.

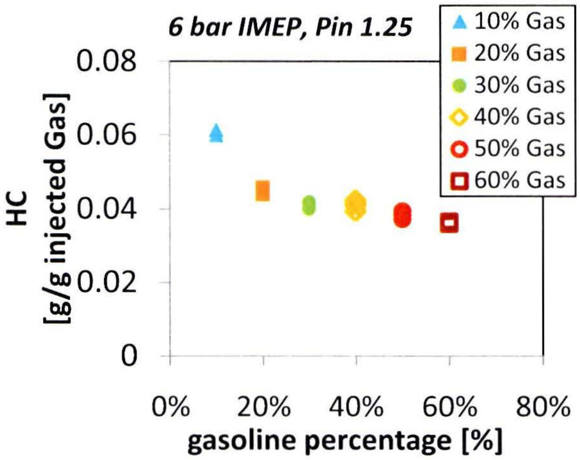


Figure 4.5a HC vs SOA

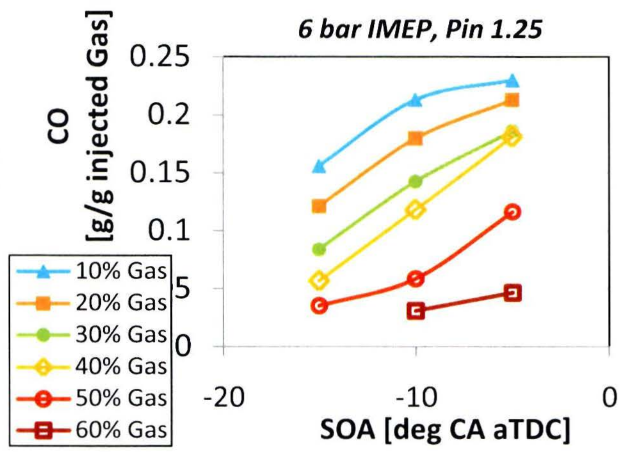


Figure 4.5b CO vs SOA

Increasing PIG levels further improves the gasoline utilization, which corresponds to the theory of a mixture being within its flammability limits. In Figure 4.5 it can be seen that the amount of unburned fuel (indicated with HC and CO emissions per injected gasoline) keeps decreasing when more diesel is replaced by gasoline. This indicates that in the combustion process more premixed gasoline is burned when the air-fuel ratio of gasoline and air is lowered. Note that this converging trend shows that a constant amount of unburned hydrocarbons gets stuck in the crevices or is blown through the engine during valve overlap. The amount of CO emissions produced by the injected gasoline keep decreasing when more gasoline is added because of higher flame speeds coming from richer mixtures, resulting in more complete combustion.

To form an energy balance, the equations 4.5 and 4.6 are used. To exclude the constant produced HC's due to engine hardware dimensions, the actual converted fuel into energy is calculated with equation 4.5. In this equation the amount of unburned fuel, divided into HCs and COs, is subtracted from the total injected mass. The fuel converted into CO is calculated by excluding the fuel part that will be converted into CO_2. From this energy balance and the results of HC and CO emissions one can conclude that within the flammability limits of gasoline, the mixture burns more complete, which results eventually in a constant production of HC and CO emissions due to engine hardware dimensions (crevices and valve overlap).

$$\eta = \frac{P}{\dot{m}_f * Q_{LHV}} \qquad (4.5)$$

$$\text{Actual converted fuel} = (\dot{m}_{gasoline} + \dot{m}_{diesel}) - \dot{m}_{HC} - \dot{m}_{CO} * \frac{M_C}{M_{CO}} * \left(1 - \frac{\Delta Hc,fuel \rightarrow CO/kgC}{\Delta Hc,fuel \rightarrow CO2/kgC}\right) \qquad (4.6)$$

With $\dot{m}_i$ representing a massflow, M_i being the molecular mass and $\Delta Hc, fuel \rightarrow CO/kgC$ being the combustion enthalpy coming from the oxidation of carbon to carbon oxide.

In Figure 4.6 the thermal efficiency is shown. It is calculated by excluding the UHC and CO emissions and thereby using the actual converted fuel. One can see that efficiency increases again when more than 30 wt% gasoline is added on a start of actuation of -5 CA deg aTDC. This can be contributed to the fact that the premixed gasoline-air mixture burns very rapidly when flame speeds are increased. However, when injecting diesel earlier, pressure and temperature may not be high enough to reach the flammability limits of the gasoline-air mixture, which can be seen in Figure 4.6.

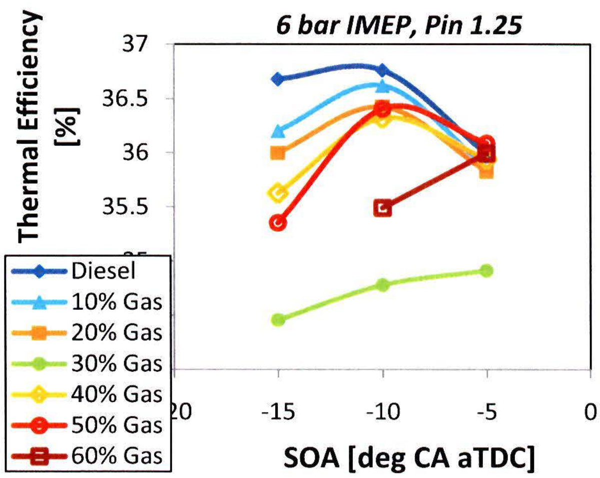


Figure 4.6 Thermal efficiency vs SOA

MPRR

In Figure 4.7a it is shown that BD is of great influence on the MPRR. In the second phase of combustion, here defined as CA90-CA50 (Figure 4.7b), a higher gasoline content results in a shorter burn duration. It is hypothesized that due to the more close-to-stoichiometric mixture of gasoline and air, compared to a rich conventional diesel-spray, combustion velocity of the complete mixture is increased, resulting in an increase of pressure rise per crank angle. Note that when even more gasoline is added, in combination with the compression ratio of 15 and no EGR, the risk exists that the premixed gasoline with air forms a mixture within the ignitibility limits for auto-ignition of gasoline itself. When this occurs, the control of SOC is lost and MPRR increases dramatically. In these measurement series the maximum pressure rise rates values are even higher than the before mentioned limit. This was not noticed during the experiments.

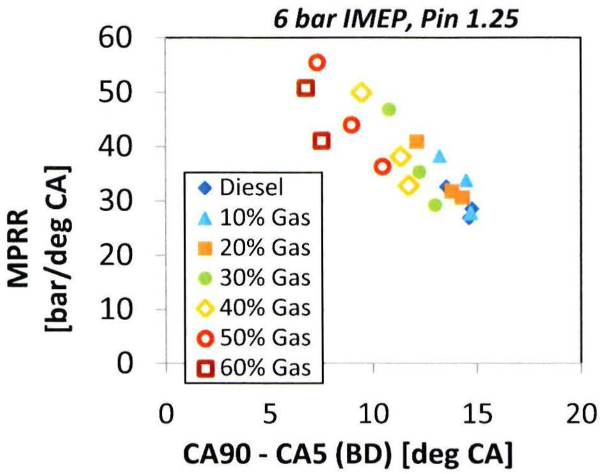


Figure 4.7a MPRR vs BD

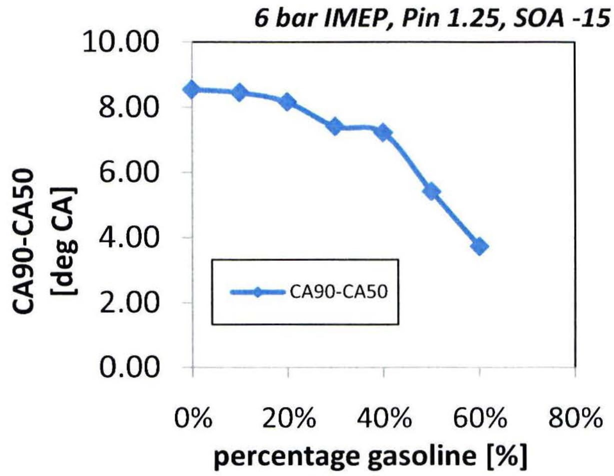


Figure 4.7b CA90-CA50 vs % gasoline

4.3 Effect of intake temperature

In the measurement series done for this subchapter it was found that the inlet valves of the test-cylinder were leaking (see Appendix A.4). Please note that the measurements (only for this section) are influenced and might differ from regular conditions.

On the present engine set-up, an EGR cooler is used to decrease the intake temperature when high percentages of EGR are used. By using the cooler, intake temperature can decrease to 305K when using up to 40% EGR. When using the same amount of un-cooled EGR, intake temperature can increase to 330K. For investigating RCCI (reactivity Controlled Compression Ignition) combustion, intake air temperature could be increased for reducing enhancing vaporization of the injected fuel. However, for hardware safety reasons of the current engine set-up EGR should be cooled, to protect fixed EGR coolers from too hot temperatures. In the following figures hot and cold EGR are compared with respect to an increase in gasoline fuel fraction at 6 bar IMEP.

Hot EGR in the intake air might decrease HC-emissions by better vaporization of the injected fuel. In this case, where all the working points are stationary, the difference in unburned hydrocarbons, see Figure 4.8a, between hot and cold EGR is shown. It is unexpected that HC-emissions increase when hot EGR is used, but it is clear that HC-production coming from the crevices is dominant compared to the HC-production coming from bad vaporization due to the use of PFI. The use of hot EGR is therefore not needed to decrease UHC on this experiment set-up.

According to previous measurement series, done by Leermakers [5], intake temperature influences combustion phasing greatly, generally resulting in a shorter ignition delay. This can lead to a shift in combustion phasing which may result in lower thermal efficiencies. In these measurement series the influence of intake air temperature on combustion phasing is not visible, which we can see in Figure 4.8b. When the gasoline content is increased, the pressure rise rate slightly increases due to the shorter burn duration of the premixed gasoline. When increasing intake temperature, the injected gasoline should vaporize faster leading to a more premixed combustion. However, this effect is not visible in the measurements and therefore the effect on maximum pressure rise rate of intake temperature with respect to adding more gasoline is negligible.

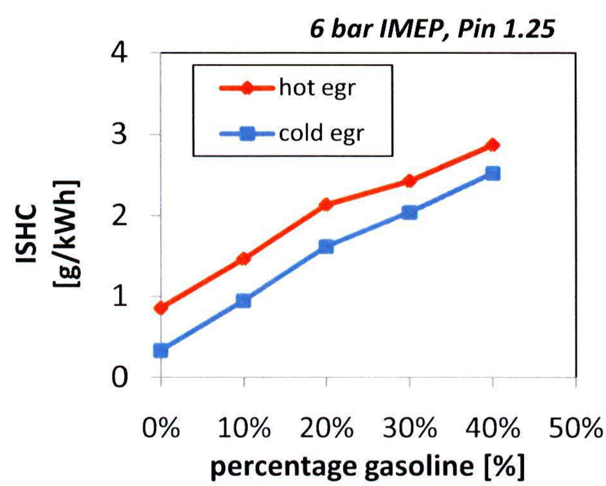


Figure 4.8a ISHC vs % gasoline

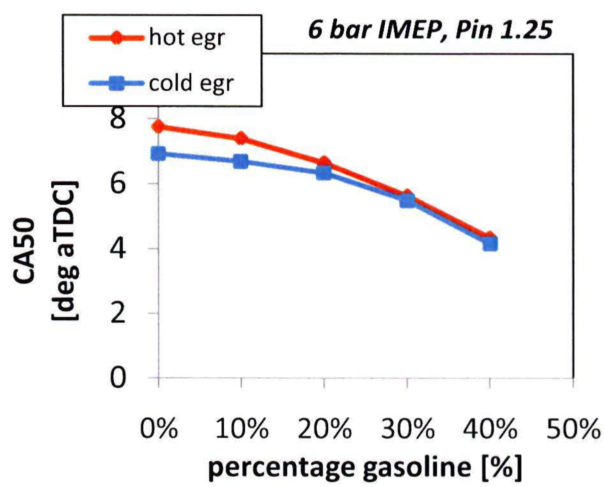


Figure 4.8b CA50 vs % gasoline

4.4 Effect of timing of port fuel injection

When using a port fuel injection system, the timing of injection may influence the vaporization of the injected fuel. Generally, in gasoline engines, fuel is sprayed onto the hot intake valve at cold starts and transient conditions [26, 27]. In measurement series for this investigation all parameters are constant when a measurement series is started. In this measurement series, two different injection timings are investigated: SOI placed on intake valve open (IVO) and timing placed on intake valve closed (IVC), with start of injection when IVO at 300°CA bTDC and IVC at 140°CA bTDC, see Figure 4.9. In the case of start of gasoline injection SOGI @ 140°CA bTDC the gasoline can vaporize almost one full cycle.

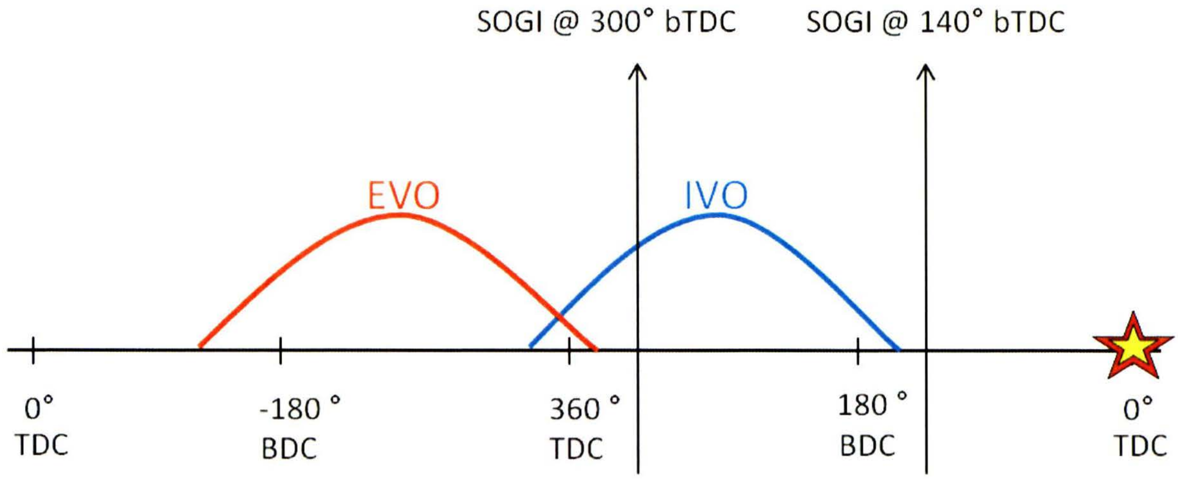


Figure 4.9 Start of gasoline injection (SOGI) @ 140°CA bTDC (gasoline ca. 3/4 cycle in intake manifold) and 300°CA bTDC (gasoline flows directly into the cylinder)

The influence of spraying on a hot valve on combustion delay is not visible, which we can see in Figure 4.10a. Combustion delay is slightly decreased when more gasoline is added. Note that the decrease in burn duration, because start of combustion relates to the diesel injection, is only visible at 6 bar IMEP, at lower loads this cannot be seen. Although influences of different injection timings of the gasoline injector are small on combustion phasing, unburned hydrocarbons are slightly decreased when injection is timed at IVO, which we can see in Figure 4.10b. It is believed that, especially at low loads, the gasoline is vaporized better after injection with the help of swirl and in-cylinder conditions. Moreover, when using the 300°CA bTDC timing injection takes place when intake valve is open and exhaust valve is close, excluding any UHC blow-through.

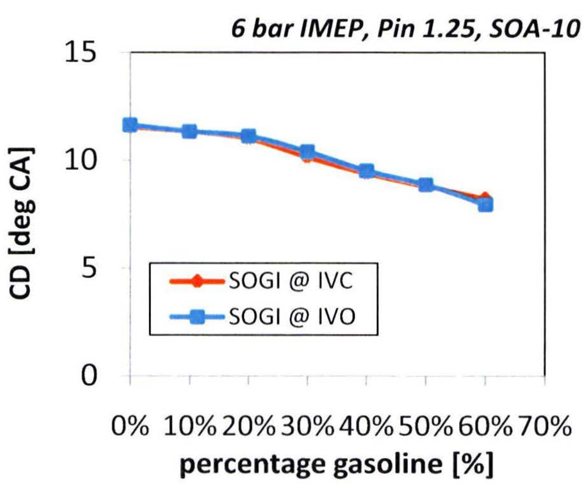


Figure 4.10a CD vs % gasoline

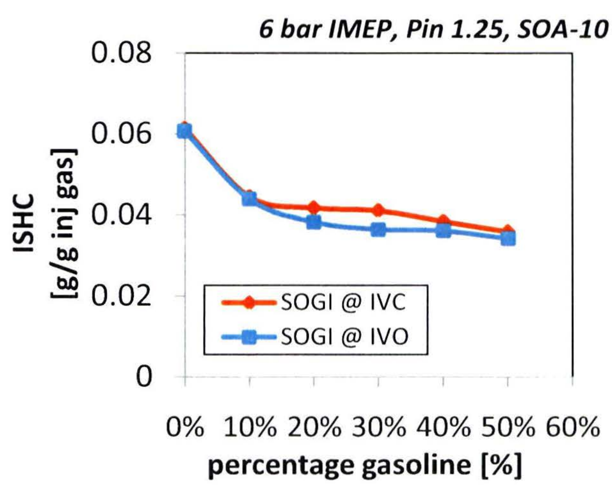


Figure 4.10b ISHC vs % gasoline

4.5 Summary and discussion

A port fuel gasoline injection system was implemented on a heavy duty DI diesel engine, to investigate the addition of port injected gasoline (PIG) on a heavy duty diesel engine.

The main advantages of this concept are longer ignition delays, thus better premixing, and control over combustion phasing for better efficiency. To be able to run in RCCI combustion, direct injection timings have to be advanced and multiple injection strategies have to be investigated.

From the results discussed above one can conclude that injecting a certain amount of gasoline influences combustion phasing. When using conventional diesel injection timings, start of combustion is still controlled by start of injection of the diesel spray. However, with an increase in the gasoline fuel fraction, the intake mixture reaches its flammability limits at a certain gasoline fuel fraction and a premixed mixture will burn with higher combustion speeds. When even more diesel is replaced by PIG, auto-ignition of the premixed gasoline and air mixture might occur right after the diesel injection.

When moderately increasing the amount of gasoline with diesel injection at conventional timings, part of the gasoline is over-mixed and too lean to be within flammability limits. Because of the incomplete combustion associated with this, HC and CO emissions are raised. It is therefore necessary that the amount of gasoline is sufficiently high; to make sure that the resulting gasoline-air mixture is within its flammability limits for complete combustion of the injected mass.

Because of a shorter burn duration when more diesel is replaced by PIG, higher pressure rise rates occur. Moreover, shorter burn durations may lead to higher peak temperatures which enhance NO_x production. However, besides the disadvantages, high thermal efficiencies can be achieved when CA50 is well positioned.

Furthermore, the influence of SOGI (Start Of Gasoline Injection) on combustion phasing and HC emissions is investigated. Generally, in gasoline engines, fuel is sprayed onto the hot intake valve at cold starts and transient conditions for better vaporizing of the fuel. In these measurement series it is found that ,especially at low loads, the gasoline is vaporized better after injection with the help of swirl and in-cylinder conditions when injected at intake valve open (IVO). Moreover, when using the 300°CA bTDC timing injection takes place when intake valve is open and exhaust valve is close, excluding any UHC blow-through.

In the next chapter PIG and an early direct diesel injection are used to establish a certain in-cylinder-mixture reactivity for control of combustion phasing.

5. Towards RCCI combustion

5.1 Introduction

As stated in the introduction of this report, RCCI combustion is characterized by early injections, long ignition delays and lean low-temperature combustion. In this chapter the shift from conventional diesel combustion towards RCCI (reactivity controlled compression ignition) is investigated. According to Hanson [10] in-cylinder fuel blending can be used to achieve fuel reactivity stratification and control of combustion phasing. Experimental tests (see section 2.4 measurement matrix) are done to investigate the potential of this new combustion concept regarding port injected gasoline addition, injection timing and a multiple injection strategy.

In the second section it is shown why the use of EGR is needed for controlling ignition delay and pressure rise rates. In section 5.3 RCCI combustion using gasoline and one diesel injection is discussed. In the following section two diesel injections are applied for better control of combustion phasing. In the section 5.5, the ratio between the two injections is varied and analyzed. Finally, the chapter ends with a comparison of these experiments and the ones performed in Wisconsin [10], and a comparison between RCCI combustion and conventional diesel combustion.

5.2 Experimental conditions

In the experiments Exhaust Gas Recirculation (EGR) is used to control the maximum pressure rise rate (MPRR) and combustion delay (CD). EGR is used to reduce the oxygen concentration and to control the rate of combustion by acting as a heat sink, especially for the higher loads (9 bar IMEP) where closer-to-stoichiometric mixtures are present. When the shift from CDC to PCCI is made, injection timings are advanced, creating a gap between injection and start of combustion (SOC). This delay is dominated by the amount of injected diesel and therefore EGR is needed as a measure to extend this gap. In Figure 5.1 one can see that MPRR does not exceed 30 bar/deg CA when using 40% of EGR. It can also be seen that when using larger amounts of gasoline, this results in shorter burn durations, which enlarges the need of EGR even further, especially compared to the measurements shown in section 4.2.

Maximum rates of pressure rise in closed chamber explosions are influenced by the composition, pressure and temperature of the fuel–air mixture (factors which determine the rate of heat release) and by the volume and shape of the enclosure, the ignition source size, energy and position, the pre-existing or combustion-created turbulence (factors which determine the amount of generated heat as well as the amount of heat losses during flame propagation) [37].

When EGR is added, combustion timings are retarded and, as a result of piston expansion, the chamber volume expands, which is expected to reduce the PRR.

$$\frac{dp}{da} = \frac{k-1}{V}\frac{dQ}{da} - k\frac{p}{V}\frac{dV}{da} \tag{5.1}$$

Herein is $\frac{dp}{da}$ the pressure rise rate, $k = c_p/c_v$, V the combustion volume, $\frac{dQ}{da}$ the rate of heat release, p the pressure and $\frac{dV}{da}$ the volume change rate [5].

The dilution of the inert gas in EGR may also slow down the chemical reaction rate, which also contributes to the reduction of PRR. In comparison with the experiments done in section 4.2 the MPRR is nearly halved, see Figure 5.1.

When injection is advanced, and therefore control on combustion phasing is lost, long CDs ensure a significant mixing time and a well-placed combustion phasing. When decreasing the amount of injected diesel, and thus local fuel sprays become leaner, the influence of EGR on combustion phasing is increased. To run in the RCCI-regime and thus when the diesel injection is advanced, high percentages of EGR and gasoline are needed, for a well positioned SOC. Because of the shift in injection timing, more time for the fuel to mix with the air results in lean low-temperature combustion. In this combustion region, high gasoline fuel fractions (up to 90 wt%) can be used.

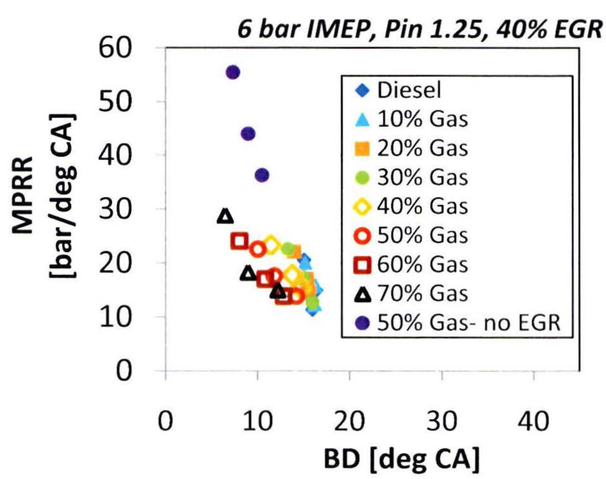


Figure 5.1 MPRR vs BD

Several experiments concerning RCCI combustion are performed in this chapter. In the following sections different strategies for RCCI combustion are investigated. First, a single diesel injection strategy with increasing gasoline fractions is investigated. After this a multiple injection strategy is briefly discussed.

5.3 RCCI combustion with one diesel-injection

In the present investigations a strategy is explored to achieve reactivity controlled compression ignition, from now on called RCCI combustion. In order to avoid high PRR, as mentioned above, an EGR flow of 60 weight percent is used. To start these experiments, a strategy of port gasoline injection and one diesel injection is used for control of combustion phasing by varying the amount of port injected gasoline and direct injected diesel. For the experiments below a load of 9 bar IMEP is used, corresponding to ca. 40% of the rated power. For injecting small amounts of diesel (at 9 bar IMEP 0.18 g/s), unless otherwise stated, an injection pressure of 1000 bar is used.

Combustion phasing

When injecting diesel fuel early in the compression stroke, the mixing time is elongated, creating a partially premixed mixture of gasoline, diesel and air. When more time is available for the diesel fuel to mix with the premixed gasoline- and air-mixture, local lambda values become higher. Figure 5.2 shows different injection timings for the main diesel injection with 90wt% of the total fuel flow coming from port injected gasoline (PIG). As SOI is advanced, it is seen that combustion phasing is retarded. This can be explained by locally leaner, and less reactive, mixtures due to higher ignition delays. It is believed that the diesel fuel, although it is partially homogeneously spread and mixed with gasoline, is used to create a locally reactive mixture and thus acts as an ignition source for the premixed gasoline-air mixture. This results in multiple ignition points and thus a rapid, one peak heat release. The heat release analysis shows that start of combustion seems to be at the same point (-10 aTDC) for every SOI but combustion is delayed when SOI is advanced. However, shorter burn durations are measured when injection is advanced, because of the more close-to-stoichiometric conditions. It is assumed that heat release calculation is not sufficiently accurate for the analysis of the first stage of combustion (CA10-CA2).

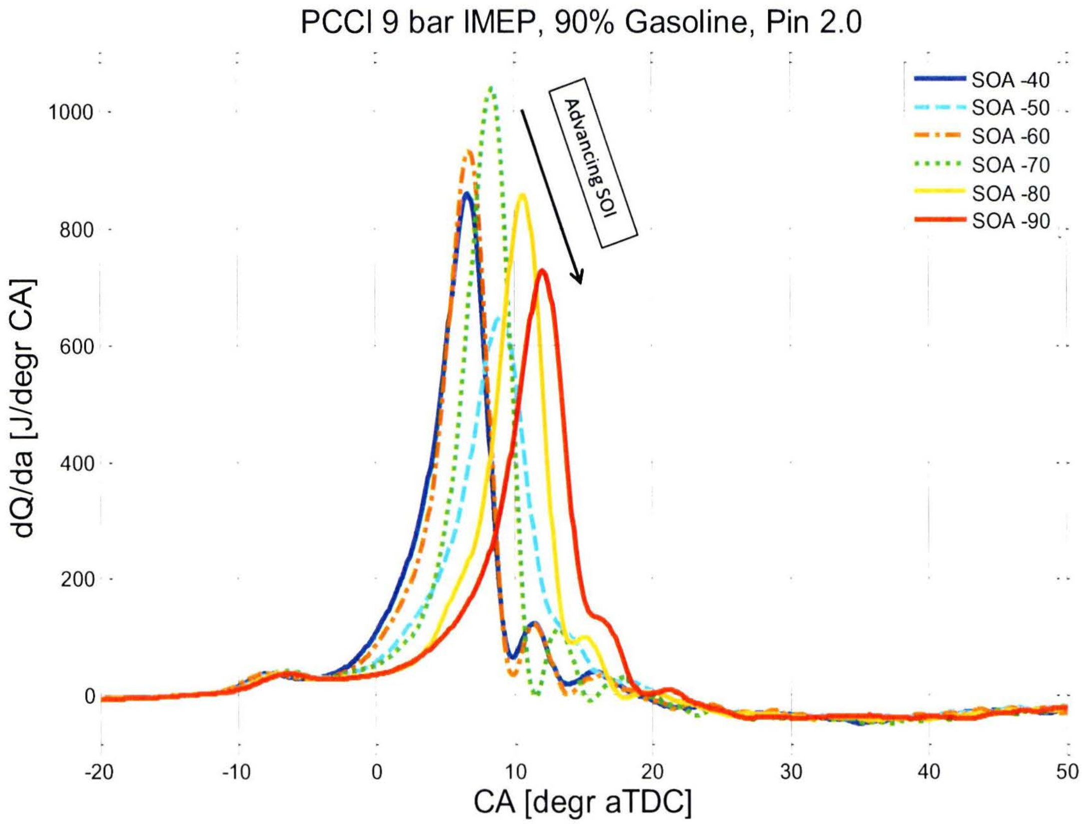


Figure 5.2 Heat release rate of different SOI1 timings. In the experiment 60% of EGR is used.

One of the most important possibilities of RCCI is the control of SOC by controlling the in-cylinder mixture's reactivity. Normally in the CDC regime start of injection is used as a control for start of combustion. In this case, when control is sought in the reactivity of the in-cylinder fuel, it is important to verify what the actual ignition trigger is. According to Reitz [38] global fuel reactivity can be controlled by adding diesel to the gasoline creating a new in-cylinder fuel blend. However, the influence of adding diesel seems to have greater effect than addition of gasoline, which indicates that local diesel might determine time of ignition dependent on injection timing and amount of fuel. To validate the effect of the diesel injection on SOC, various gasoline fuel fractions are investigated. In Figure 5.3 one can schematically see the two experiments under investigation in this chapter. In the first series, indicated with 1, total fuel flow is kept constant. In the second measurement series, indicated with 2, gasoline fuel flow is decreased and the diesel injection is kept constant. One should notice that due the original cetane numbers of 55.9 and 14.7 for diesel and gasoline respectively, the global reactivity of the mixture is not changed significantly. However, the diesel spray will ensure a higher reactivity locally in the combustion chamber.

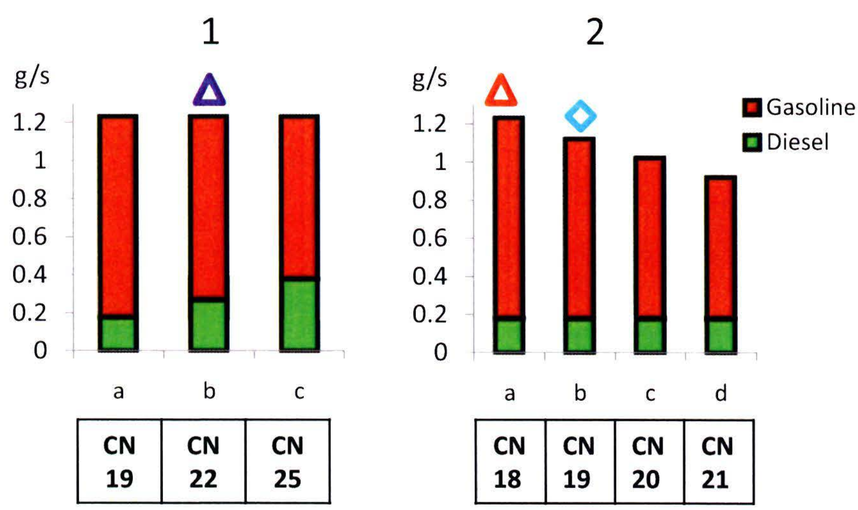


Figure 5.3 diesel and gasoline fuel fractions for two experiments. The estimated global cetane numbers are given for each mixture.

In Figure 5.4 results of measurement series number 1 and 2 combined are presented. The measurement points are shown in FIgure 5.3. It was found that when the gasoline fuel flow is increased, the change in SOC is less significant than when the diesel fuel flow was increased. This dependency on injection timing (ignition delay) and amount of fuel with respect to the available oxygen can be explained by the change in cetane number. The change in cetane number is more dependent on the diesel variation because of the high gasoline percentages in the in-cylinder blend and therefore becomes a parameter for control of combustion phasing. Therefore, increasing the diesel flow ensures higher local reactivity spots (mixture stratification) in the combustion chamber. These reactive spots may function as multiple ignition points.

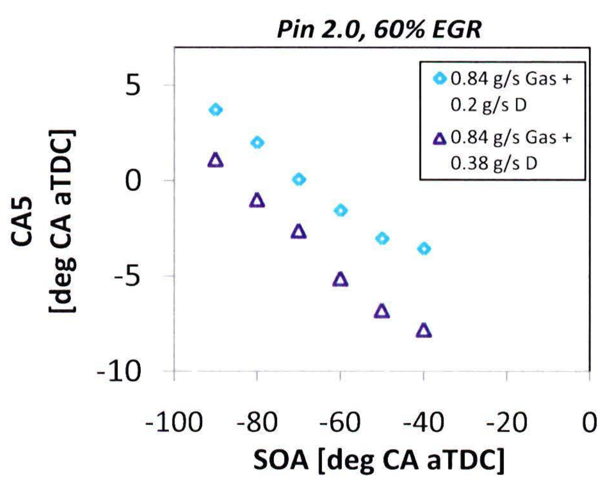


Figure 5.4a CA5 vs SOA (increasing diesel flow)

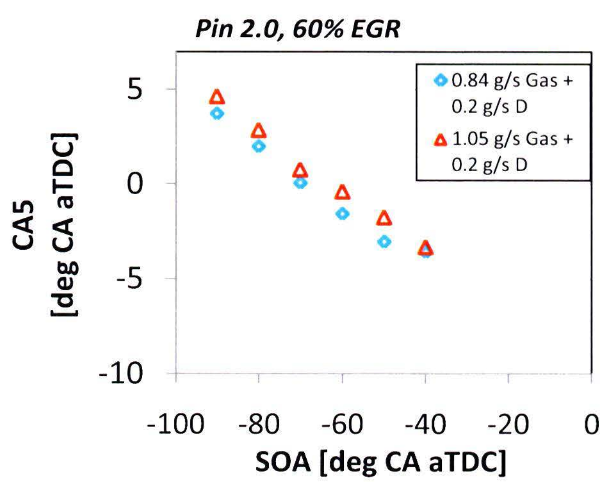


Figure 5.4b CA5 vs SOA (increasing gasoline flow)

From measurement series 1 and 2 one can conclude that the diesel injection timing and quantity are the two most important parameters to control combustion phasing. However, to investigate emissions, performance and combustion phasing at constant load, measurement series 1 is used.

In the following section the results of measurement series 1 are shown. SOI1 (main diesel injection) is held constant at -70 degr aTDC and the amount of PIG is varied from 70 until 90 wt% of the total injected fuel mass. It is expected that decreasing the amount of diesel has a similar effect on combustion phasing as advancing SOI1. Again, the local lambda value of the mixture of reactive diesel-fuel and air increases and CA5 is retarded. Figure 5.5a shows SOC, defined as CA5, plotted against SOA. As one can see, when the gasoline fuel fraction is increased, and thus the diesel fuel fraction decreases, SOC is retarded. Figure 5.5b shows that CA50 is also retarded and burn durations are extremely short compared to diesel-only conventional combustion, which are typically 16 degr CA.

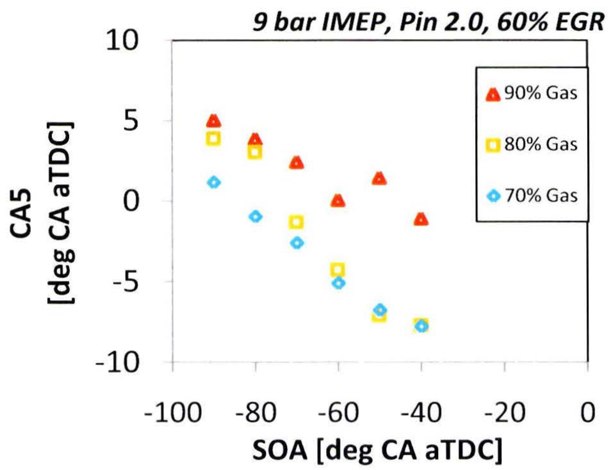


Figure 5.5a CA5 vs SOA

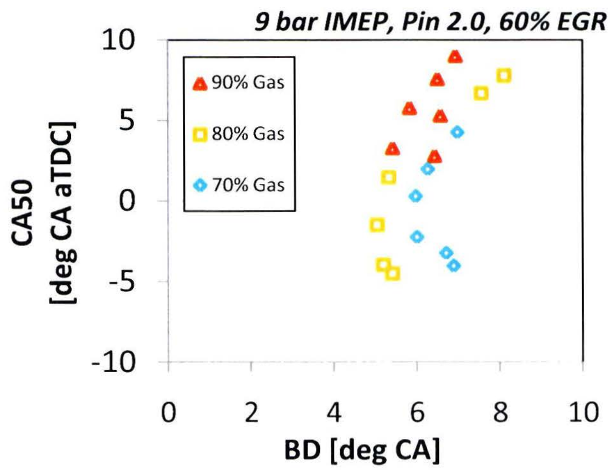


Figure 5.5b CA50 vs BD

<u>Emissions</u>

Longer ignition delays (CA5- SOI) create the possibility for partially premixed air-fuel mixtures to combust at lean and low temperature conditions. In the RCCI combustion concept, the majority of the injected fuel is premixed and thus is able to combust under such conditions. The advantage of lean low temperature combustion can be seen In Figure 5.6a. As combustion delay, here defined as CA50-SOI, increases, NO_x levels decrease dramatically. Note that NO_x emissions are reached that are below the EU VI emissions standards, which are for NO_x levels 0.4 g/kWh. However, these experiments are stationary and thus difficult to compare with transient tests for European emission standards.

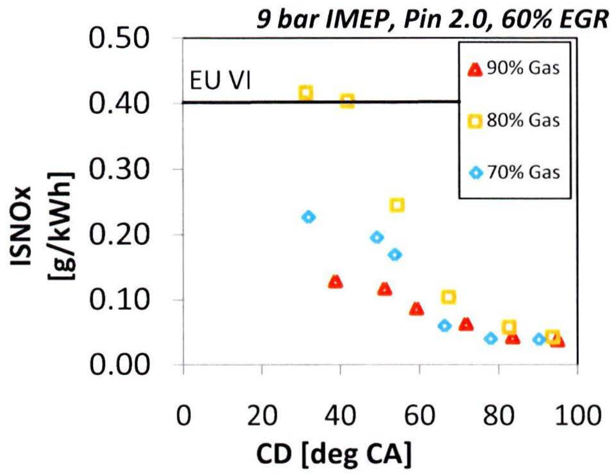


Figure 5.6a ISNOx vs CD

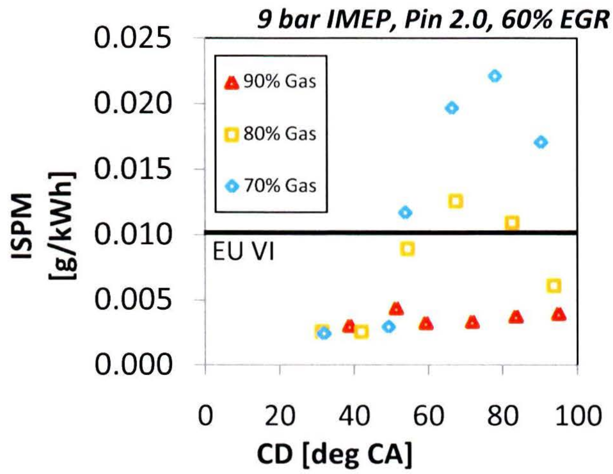


Figure 5.6b ISPM vs CD

As for the decreasing trend in NO_x emissions, similar explanations can be given for the low soot production in this combustion concept. As mentioned before, soot is produced at combustion of rich mixtures, which is typically found in diesel-jet sprays at conventional diesel combustion. Here, lean low temperature combustion produces low soot emissions (see Figure 5.6b). Especially at early diesel injections and thus long combustion delays, very low soot emissions are measured. What strikes is that in early timings of SOI1, high gasoline fuel fractions are needed for low soot production. This can be explained by the usage of more diesel at low injection pressure. It is believed that due to the relatively low injection pressure of 1000 bar, the diesel droplets need more time and higher temperatures for vaporizing. Note that absolute values of soot are very low, even when lower gasoline fuel fractions are used.

Generally when NO_x and soot emissions are decreased by lean low temperature combustion, combustion becomes incomplete, which comes with a penalty in HC and CO emissions. When diesel injections are advanced, HC-emissions are at their minimum at high gasoline fuel fractions, see Figure 5.7a. This can be explained by wall-wetting, which might increase when the amount of direct injected diesel increases. When injection timing is retarded the constant factor of UHC stuck in the crevices dominates again and will be higher when more gasoline is used compared to lower gasoline fuel fractions. Although HC-emissions are high, the early injection of diesel is needed for a large combustion delay and thus a well-placed combustion phasing. Furthermore, high levels of UHCs negatively contribute to the indicated efficiency, which is explained in the next section.

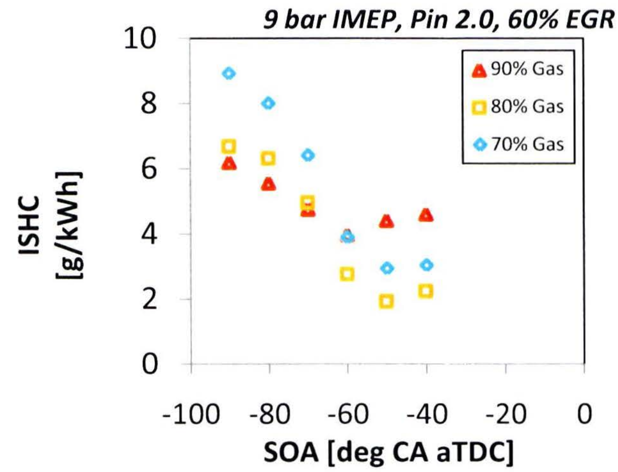


Figure 5.7a ISHC vs SOA

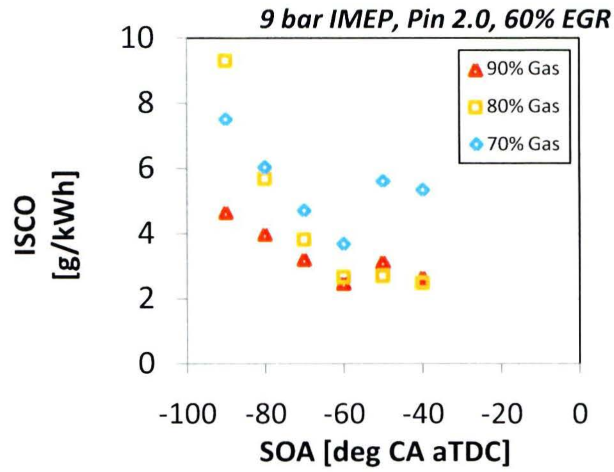


Figure 5.7b ISCO vs SOA

At lean, low temperature combustion, fuel is often only partly oxidized, resulting in higher levels of carbon monoxide. If one studies the trends in HC- and CO emissions, which are, under these conditions, often similar, little differences can be seen. The difference between trends in HC and CO emissions should be attributed to the fact that HC emissions can originate both from partly oxidized and completely unburned fuel, while CO is always a result of partial combustion. Therefore, CO-emissions, shown in Figure 5.7b, increase when injection is advanced. Due to more time for premixing the lean mixture is prone for incomplete combustion. It is believed that a significant amount of the CO emissions is produced by lean gasoline-air mixtures, since CO-emissions decrease when gasoline fuel fraction is increased. As mentioned in chapter four, the gasoline-air mixture should be within its flammability limits. The global lambda of the gasoline air mixture before the diesel injection occurs is ca 1.5. It is believed that under these conditions, this air-fuel ratio is within the flammability limits.

<u>Performance</u>

RCCI is a promising concept for increasing thermal efficiency and thereby reducing fuel consumption. Short burn durations and low temperature combustion should lead to minimized heat losses. When more diesel is replaced by PIG, fuel consumption is lowered. This can be explained by several aspects: First combustion phasing is shifted towards more efficient timings. Second, better premixing of the gasoline-air mixture ensures cold temperature combustion which minimizes heat losses through the cylinder wall. And third, burn durations may become shorter when more gasoline is used because of higher flame speeds and more close to stoichiometric mixture conditions.

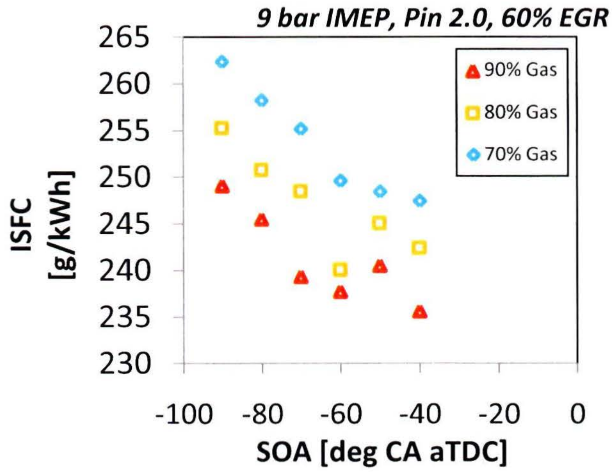


Figure 5.8a ISFC vs SOA

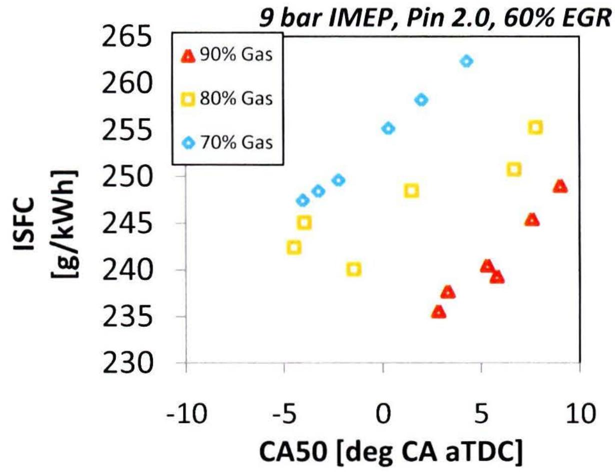


Figure 5.8b ISFC vs CA50

Figure 5.8 shows the specific fuel consumption versus SOA and CA50. As mentioned before timing is not the only aspect that can reduce fuel consumption. For example in Figure 5.8b one can see that with the same timing of CA50, more premixed gasoline results in lower fuel consumption. However, fuel consumption is decreasing, even when CA50 is positioned before TDC. One would expect that an optimal position of CA50 would lie at ca 5 deg CA aTDC.

Furthermore should be mentioned that gasoline contains slightly more energy per mass, but it is believed that a more complete combustion, instead of creating UHC and CO-emissions out of the gasoline, results in more power output per fuel mass. Again, as was explained in chapter four, the gasoline-air mixture should be within its flammability limits for ensuring complete combustion.

Figure 5.9 shows measured in-cylinder pressure and heat release data for gasoline percentages of 70, 80 and 90wt% at a constant start of injection of diesel. The additional gasoline percentage is seen to phase combustion later in the cycle by reducing the local fuel reactivity. As is shown, this later phasing of combustion drops in-cylinder pressure, and therefore temperature, which slows NO_x formation rates and lowers heat transfer. Moreover, increasing gasoline addition might decrease fuel consumption due to the lower rates of heat transfer from reduced injection-generated turbulence and soot radiation [10].

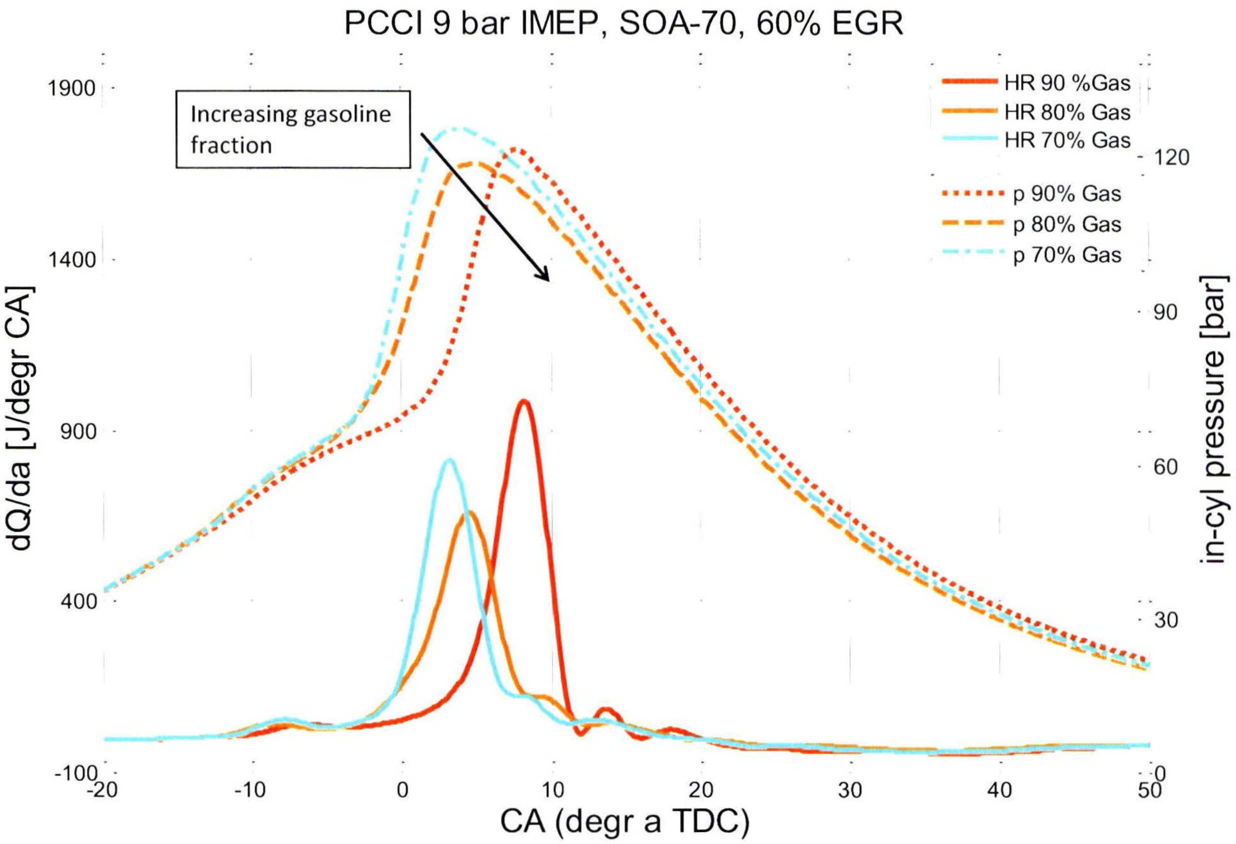


Figure 5.9 Single shot measured in-cylinder pressure and ROHR for different gasoline percentages at constant SOI1

Compared to lower gasoline fuel fractions, one would expect that the 90wt% gasoline fraction case would decrease pressure and heat release rate even further. Figure 5.9 also shows that if 90wt% is used HRR and pressure increase again. In this measurement point (SOA-70) it was found that less UHC were produced with respect to the other two mixing ratios, which could be an explanation for the higher HRR and pressure signal. Note that this higher HRR might counteract on the thermal efficiency due to higher temperature losses.

Although higher temperature losses are expected, increasing the gasoline fuel flow leads to higher thermal efficiencies, which can be seen in Figure 5.10a. This can be related to a more efficient position of CA50 and the slightly higher power density of gasoline. Moreover, the thermal efficiency is calculated by using the actually converted fuel. Due to more premixed gasoline, the local air fuel ratios are lower resulting in a more complete combustion and less UHC. When unburned fuel is taken into account, the indicated efficiency shows that the position of CA50 remains dominant, see Figure 5.10b. Again, no real explanation is found for the increasing trend in efficiency when CA50 is positioned before TDC. The indicated efficiency also shows that when sufficient gasoline is used (90 wt%) and the flammability limits are reached, less UHCs are produced, resulting in higher efficiencies.

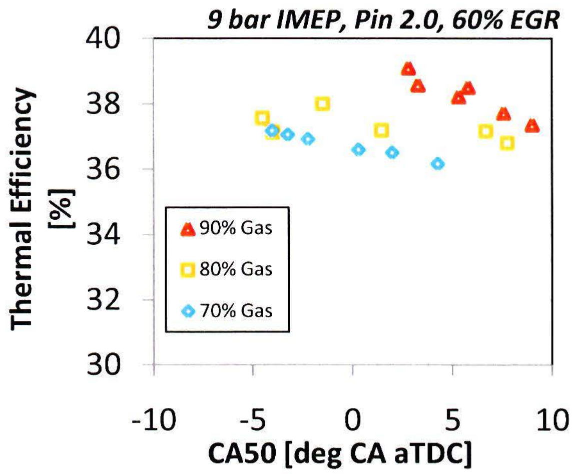


Figure 5.10a Thermal efficiency vs SOA

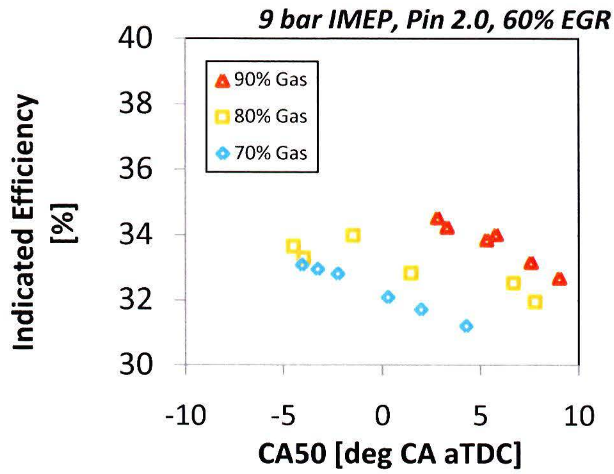


Figure 5.10b Indicated efficiency vs CA50

5.4 Effects of multiple diesel injections

As is mentioned before, the homogeneous mixed gasoline contributes to the HC-emissions because of local lean spots. Theoretically, for ensuring complete combustion, and thus reducing HC- and CO-emissions, a multiple injection strategy can be performed. For realizing the multiple injection strategy in the experiments, small amounts of diesel fuel must be injected. Because of a minimal actuation duration for the diesel injector the fuel pressure is lowered to 500 bar.

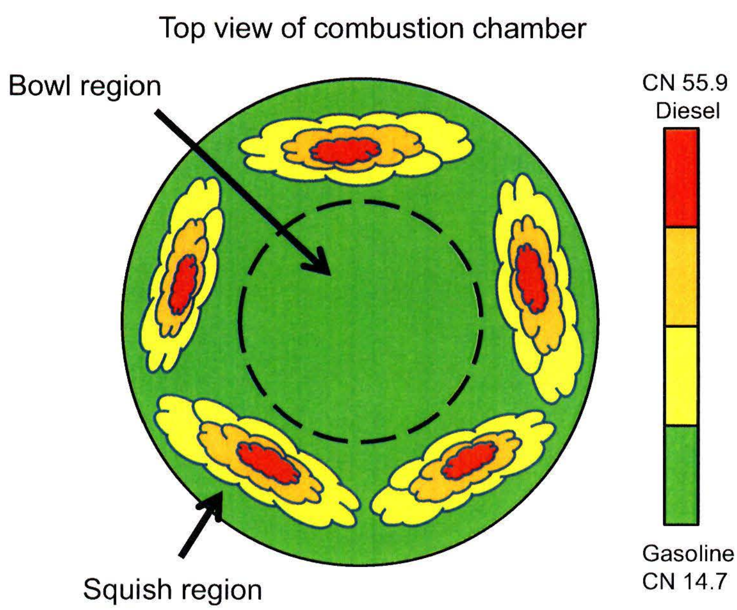


Figure 5.11 Top view of the combustion chamber. Local reactive regions function as multiple ignition points.

The purpose of the first injection is to control the fuel reactivity in the squish region of the combustion chamber. The second diesel injection is placed later in the compression stroke and covers the bowl region of the combustion chamber, see FIgure 5.11. This injection event generates a relatively high reactivity region which can act as a first ignition source [38]. However, when comparing a multiple injection strategy with a single diesel injection strategy, no improvements on engine-out emissions can be found, see Figure 5.12. HC- and CO-emissions are increased, which indicates a less complete combustion. The amount of diesel injection, at a load of 9 bar IMEP and using 80wt% of the total fuel mass on gasoline, is only 0.15 g/s when the first and second diesel injection are equally divided. This contributes to the low reactive local spots in the squish region of the combustion chamber. Moreover the lower injection pressure of 500 bar may also contribute to less vaporization of the second diesel injection and thus may lead to a less complete combustion process, which is shown in Figure 5.12. Note that CA50 for the multiple diesel injection strategy is approximately constant, even though a sweep of SOI1 here is made, shown in Figure 5.13. This indicates high control of combustion phasing due to the second diesel injection. Note that the timings of CA50 for the multiple injection strategy are slightly shifted further into the expansion stroke resulting in lower combustion temperatures. In this situation the second diesel injection should be slightly advanced for better emission performance. Although in these measurement series no increase in efficiency is measured when two diesel injections are used, this strategy can be used to improve the control of CA50.

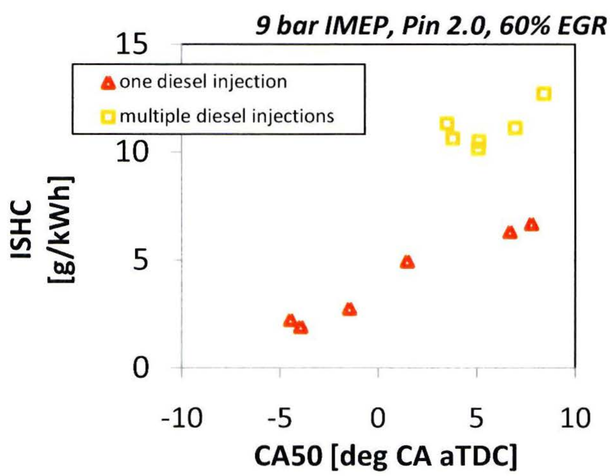


Figure 5.12a ISHC vs CA50 (constant SOA2 of -10)

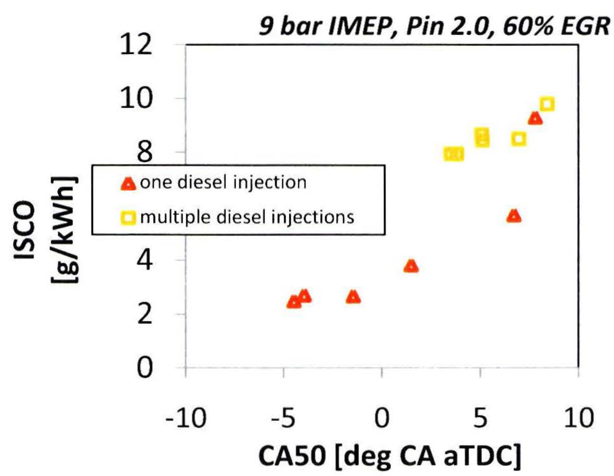


Figure 5.12b ISCO vs CA50 (constant SOA2 of -10)

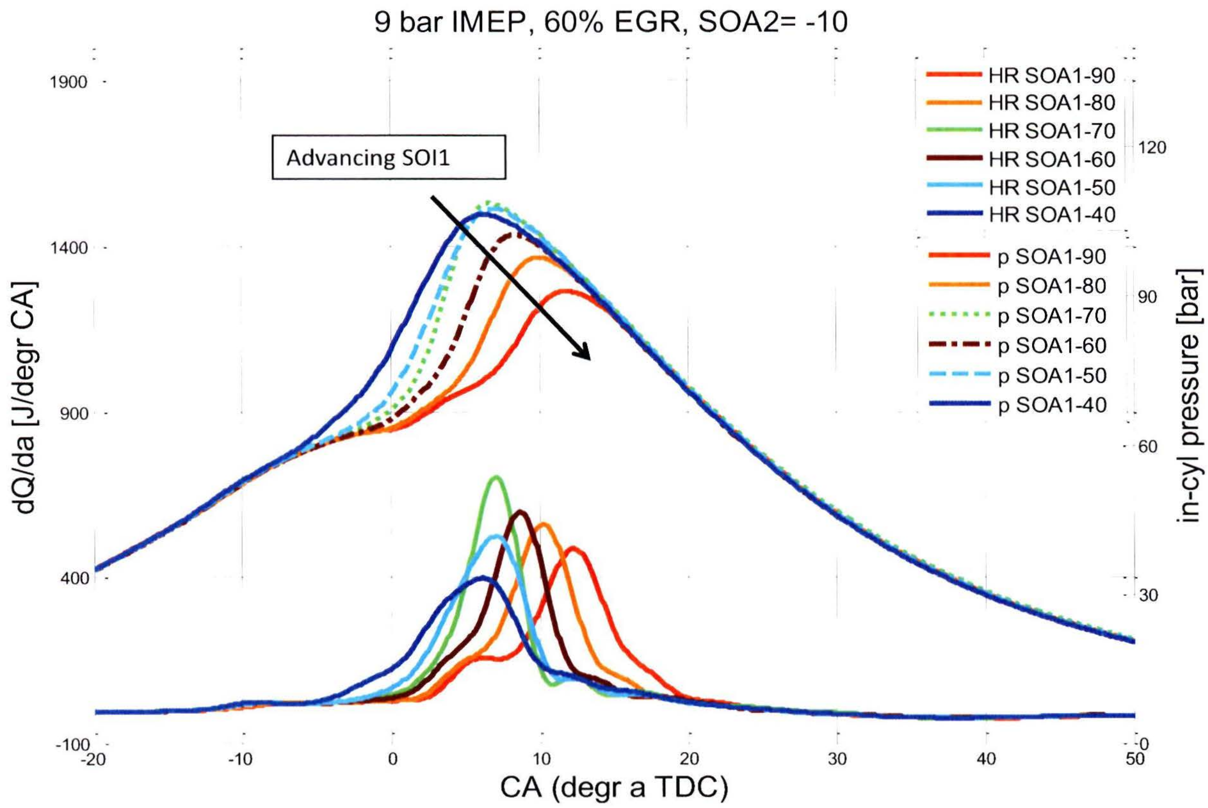


Figure 5.13 Multiple injection strategy with SOA1 sweep and a constant SOA2 at -10° CA aTDC. Diesel injections are equally divided.Gasoline percentage is 80wt% of total injected fuel mass.

As mentioned above, the control of combustion phasing is more sophisticated when using two diesel injections. By varying SOA of the first diesel injection (SOA1), the local air-fuel ratio of the premixed gasoline, the diesel and the air can be controlled. By doing so, fuel reactivity can be lowered, but fuel stratification can lead a delay in SOC. To compensate for this delay, if desired, combustion phasing can be controlled by a second diesel injection (Figure 5.13). It was found that under these conditions burn durations are slightly increased. Due to the late second diesel injection, the mixture is ignited as in the conventional diesel combustion regime. Due to this slightly elongated burn duration (ca 3 CA degr longer) pressure rise rates are decreased.

Due to the direct combustion of the second diesel injection when SOA2 is -10 CA deg aTDC, SOA2 should be advanced. By advancing the second diesel injection, PCCI like combustion occurs, resulting in more time for the diesel-fuel to mix with its surrounding. An injection sweep of -25 until -5 degr CA aTDC SOA2 is investigated, shown in Fig 5.14. Due to the conditions at these timings, the reactive diesel will combust almost immediately and ignite the remaining mixture. Moreover, it shows to be an instrument for controlling SOC, while still maintaining the benefits of low temperature combustion, since the second diesel injection does not significantly increase NO_x- or PM-emissions due to its size. Having observed this benefit for control of combustion phasing, it is of interest to investigate the division of the two diesel injections.

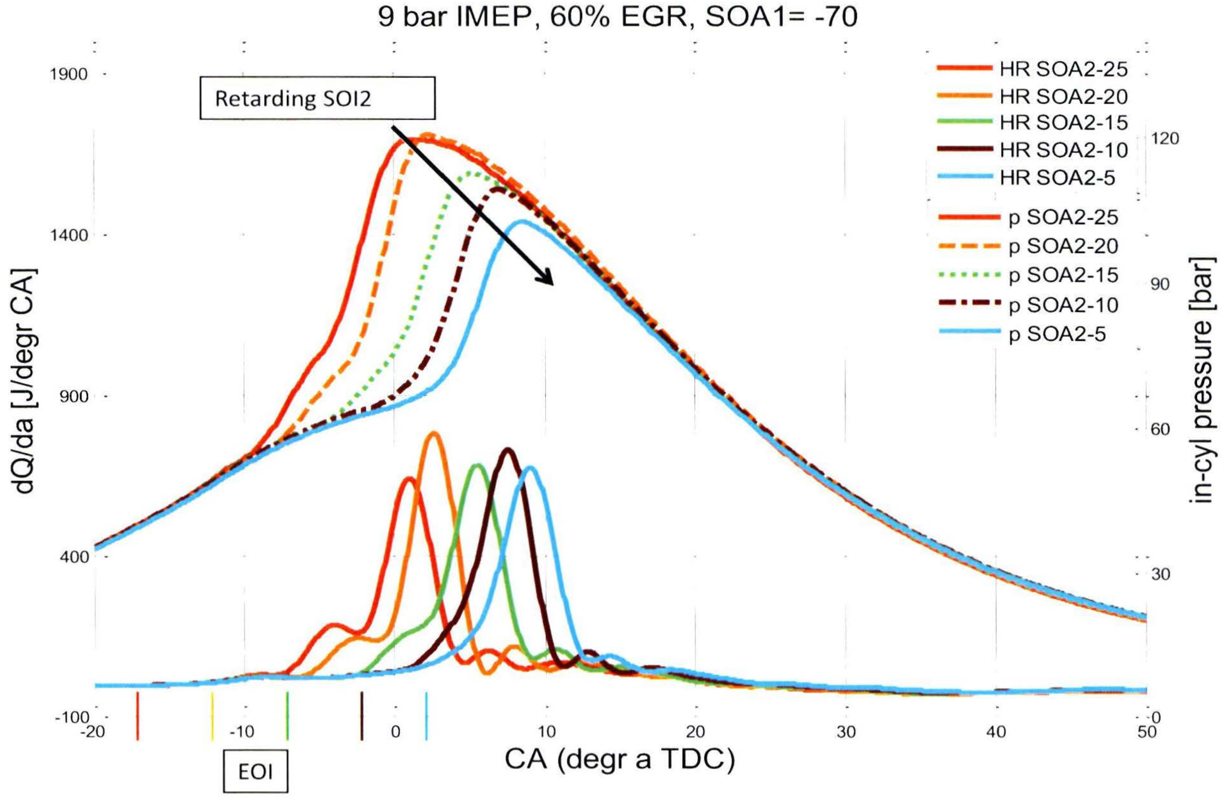


Figure 5.14 Multiple injection strategy with SOA2 sweep and a constant SOA1 at -70° CA aTDC. Diesel injections are equally divided.Gasoline percentage is 80wt% of total injected fuel mass. EOI2 is indicated.

5.5 Effects of varying injection quantity

Besides investigating the effects of SOI1 and SOI2 timings, the next parameter investigated is the fuel split between the two diesel injections. The fraction of diesel fuel injected in SOI1 was varied with 50wt% and 70wt% of the total diesel fuel injected. The total injected fuel mass contains 80 wt% of gasoline.

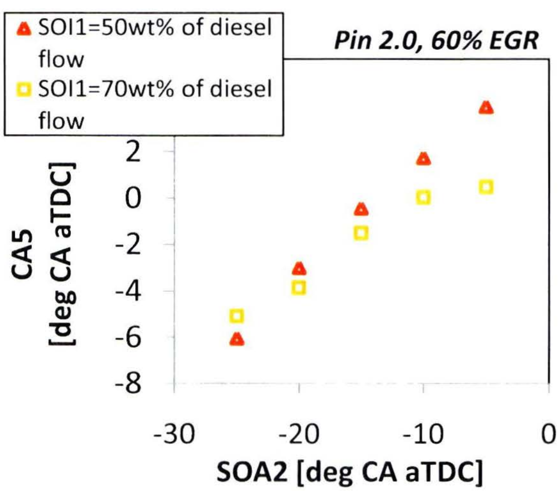


Figure 5.15a CA5 vs SOA2

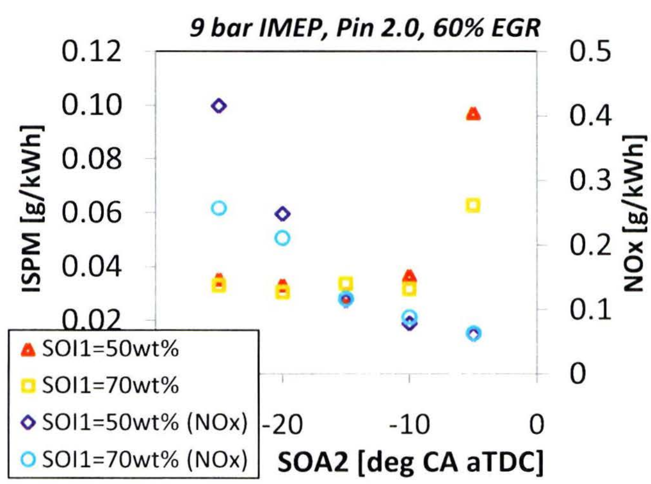


Figure 5.15b ISPM and ISNOx vs SOA2

When a small amount of diesel is used in the first injection (0.15 g/s) the reactivity of the premixed mixture is too low for auto-ignition, which can be seen in Figure 5.15a. CA5, seen as SOC, is triggered by the second injection, as SOC retards with SOA2. However, a higher percentage of diesel in the first injection leads to a shift in SOC. Figure 5.15a shows the influence of the larger quantity in the first diesel injection by a deflection of SOC, independent of SOA2. This shift can be explained by more reactive regions, due to the first injection, in the squish region of the combustion chamber, which will ignite the mixture.

It is expected that more diesel in the first diesel-injection leads to lower emissions due to a mixing time for the injected amount of fuel in SOI1. Figure 5.15b shows the measured emissions trends of PM and NO_x for the SOA2 sweep for two possible fuel quantities of SOI1. CO and HC are seen to be relatively unaffected by the first injection fuel quantity and therefore not shown here. However, NO_x- and soot-emissions have significant increases with the additional second injection fuel. Similar to the results of the SOI1 and SOI2 timing tests, when more diesel is injected earlier in the cycle fuel, reactivity decreases locally. This leads to leaner and less reactive regions and an increase in mixing time. Due to this longer mixing time for the diesel, NO_x- and soot-emissions are decreased. Note that both NO_x- and soot-emissions are extremely low and NO_x emissions even well below European emissions standards. From this section one can conclude that dividing the fuel quantities of the two diesel injections can result in a sophisticated control of combustion phasing, while still maintaining low engine out NO_x- and soot-emissions.

5.6 Comparison to Wisconsin experiments

The University of Wisconsin's Engine Research Center has published several papers describing a combustion concept very similar to the one in this report. Reitz and coworkers investigated several loads including 9 bar IMEP with a comparable engine set-up as the CYCLOPS. The Engine Research Center's test engine is a single cylinder Caterpillar SCOTE 3401E. The engine specifications are shown in Table 6.1. Under the same experimental conditions, similar results were found for NO_x- and soot-emissions. Also HC and CO emissions are comparable, however, in thermal efficiency significant differences are found. In the best measurement series Reitz reports a thermal efficiency of >55%, in contrast to this report where a maximum of 40% thermal efficiency is achieved.

Although experiments are very similar to the ones in this report, several differences can be found. Reitz uses a slightly higher compression ratio, which can benefit the thermal efficiency. It was also not completely clear how the thermal efficiency of the Caterpillar engine was calculated. It is assumed that the gross IMEP is used and UHCs are excluded. However, the calculation of thermal efficiency in this report is also done this way, which seems to rule out this argument as a root cause of the difference. The coolant temperature of the SCOTE engine was also not found. The experiments in Wisconsin may be performed at higher coolant temperatures, which can result in less heat losses through the cylinder wall.

Table 5.1 Engine specifications for the Caterpillar 3401E SCOTE engine used by the Engine Research Center at the University of Wisconsin for RCCI combustion

Test Engine	Caterpillar 3401E SCOTE
Displacement	2.44 L (single cylinder)
Bore x Stroke	137 mm x 165 mm
Engine speed	1300 rpm
Compression ratio	16:1
Piston Style	OEM bowl geometry
Fuel	Gasoline and Diesel
Injection Type	Port gasoline injection, Bosch Gen 2 common rail DI diesel, multiple diesel injection strategy
Induction	External air source
EGR	High pressure loop, cooled, 40 wt%

The amount of EGR Reitz and coworkers use is 40 wt%, compared to 60 wt% in the experiments performed in Eindhoven. An explanation for this could be the difference in fuel reactivity of the diesel fuel. The cetane-number of the US-diesel that was used by Reitz and coworkers is ca 40 compared to a CN of 55.9 of the EN590 diesel. When the reactivity is lower, and the injection timing is constant, local rich fuel spots in the combustion chamber are less reactive resulting in a shift in combustion phasing.

As mentioned in the introduction of this report, according to Reitz one of the most crucial parameters with RCCI, using port gasoline injection, are multiple direct diesel injections. These multiple injections are timed approximately at 60 deg BTDC and 30 deg BTDC [10,11]. In the experiments of Eindhoven two timing sweeps of SOI1 and SOI2 are conducted, which contain the same injection timings as the experiments performed in Wisconsin. Both investigations found that when using a double injection strategy for the diesel injection, the timing of the first injection (SOI1) was found to increase mixture stratification as SOI1 was retarded toward TDC. The increased local equivalence ratio raised combustion temperatures and local fuel reactivity, which then advanced combustion phasing and increased pressure rise rate (PRR) and NO_x emissions. The timing of the second diesel injection (SOI2) was also found to increase mixture stratification as SOI2 was retarded toward TDC. Similar to the SOI1 timing, the increased local equivalence ratio raised combustion temperatures and fuel reactivity resulting in an advanced combustion phasing.

Besides, differences in engine speed, engine geometry and valve timing, the main difference between the two investigations was found to be the fuel reactivity of the diesel fuel. Due to a lower reactivity of the US diesel, the second diesel injection experiences a longer ignition delay and therefore lower EGR rates could be used.

5.7 RCCI vs diesel-only CDC

In conventional diesel combustion, diesel jet-sprays ensure locally rich mixtures. Because of these jet-sprays, high temperature turbulent flames produce high NO_x- and soot-emissions. By enlarging the mixing time for the fuel to mix with air, lean low temperature combustion can be realized. In PCCI combustion diesel is injected early in the compression stroke for an increase in mixing time. However, combustion phasing is generally controlled with start of injection, and is therefore lost when injection is advanced. In RCCI combustion this control is found in the local reactivity of the in-cylinder blend. Besides the fuel reactivity, injection timing of the diesel injection is used to control fuel stratification and therefore combustion phasing.

To discover the full potential of RCCI combustion, the concept is compared to diesel-only conventional diesel combustion. The experiments done for this comparison are at equal loads: 9 bar IMEP. To create a similar experiment, 60 wt% of the total mass flow is EGR for both combustion concepts. However, this large amount of EGR is not necessary at CDC, therefore also experiments with 20 wt% EGR are included. The RCCI measurement series contains the best points measured with 90 wt% gasoline and one main diesel injection timed at -70 deg CA aTDC.

In Figure 5.16a the indicated efficiency, based on the total injected fuel mass, is shown. One can see that at an equal timing of CA50 the indicated efficiency is raised when the RCCI combustion concept is used. This can be related to shorter burn durations and less heat losses due to lower temperature combustion.

However, as mentioned before, the long ignition delays lead to higher HC-emissions, one the one hand because of UHC stuck in the crevices, and on the other hand because of incomplete combustion of local lean spots.

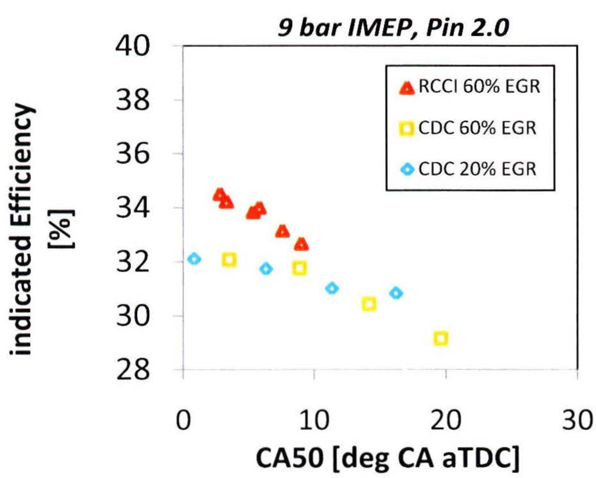


Figure 5.16a Indicated Efficiency vs CA50

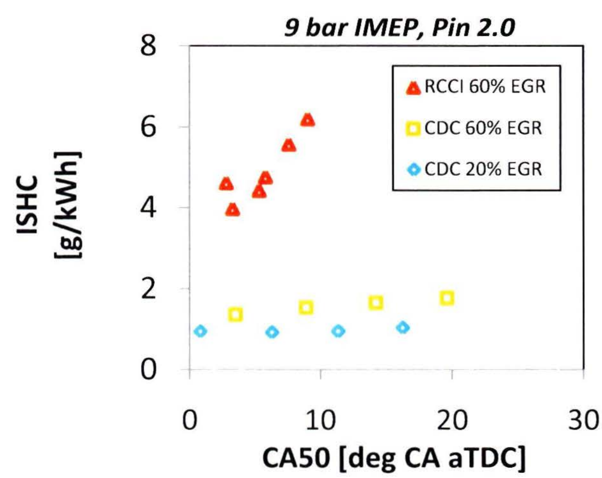


Figure 5.16b ISHC and CA50

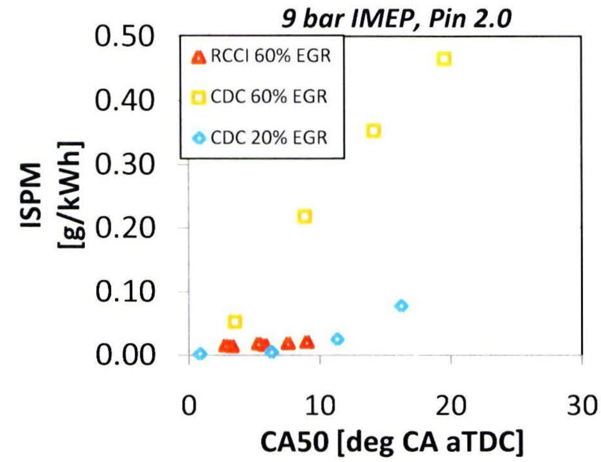


Figure 5.16c ISPM vs CA50

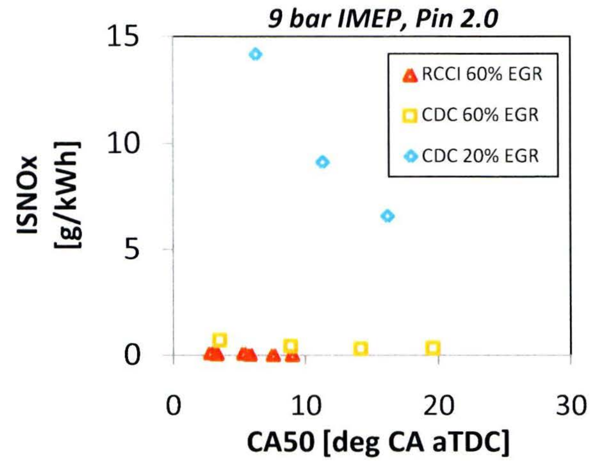


Figure 5.16d ISNOx vs CA50

Compared to CDC, due to a longer mixing time for the gasoline- and diesel-fuel to mix with air, lean and low temperature combustion will lead to significantly lower NO_x- and soot-emissions (Figure 5.16c and 5.16d). In CDC, when more EGR is used, a low temperature, rich combustion will ensure higher soot production. Higher air-fuel ratios, for lowering the soot-production, will lead to high temperature combustion and therefore higher NO_x levels. In RCCI combustion the advantage of lean and low temperature combustion is combined without losing control of combustion phasing. Next to the potential for decreasing engine out-emissions, and therefore the possibility to decrease the need for after-treatment-systems, higher efficiencies are measured.

5.8 Summary and discussion

This chapter explored the use of in-cylinder blending of diesel and gasoline to achieve equivalence ratio and reactivity stratification and to control combustion phasing and rate of heat release in a RCCI combustion strategy. In-cylinder fuel blending using port fuel injection of gasoline and early-cycle, direct injection of diesel was used for control of the before mentioned parameters.

It was found that, using a single diesel injection strategy, advancing the direct diesel injection (SOI1) leads to longer ignition delays, a retarded combustion phasing, and therefore a reduce in pressure rise rates. It was also found that when the gasoline fuel fraction was increased, the local reactivity of the in-cylinder mixture decreases and SOC was retarded. Varying the amount of diesel fuel has a greater effect on shifting combustion phasing than varying the amount of gasoline. It was found that the timing and the amount of the diesel injection are essential for control of combustion phasing.

Longer ignition delays create the possibility for partially premixed air-fuel mixtures to combust at lean and low temperature conditions. In the RCCI combustion concept, the majority of the injected fuel is premixed and thus is able to combust under such conditions. In these conditions very low NO_x and soot emissions are produced, however, incomplete combustion leads to an increase in HC- and CO-emissions.

RCCI is a promising concept for increasing thermal efficiency and thereby reducing fuel consumption. Short burn durations and low temperature combustion should lead to minimized heat losses. When more diesel is replaced by PIG, fuel consumption is lowered. Although burn durations are extremely short, a significant increase in thermal efficiency, as is seen in the Wisconsin experiments, remains unproven.

Furthermore, a multiple injection strategy was investigated for reducing UHC and improving control of combustion phasing. It was found that timing of SOI2 (late direct injection of diesel) has great influence on SOC and therefore can reduce PRR and HRR by exact control of combustion phasing. By dividing the total injected diesel fuel consequently over the two injections, the first injection can benefit from the long ignition delay and the second one can be used as a trigger for SOC. For reducing UHC, the multiple injection strategy seems to have a counter effect. From this section one can conclude that a multiple injection strategy can results in a sophisticated control of combustion phasing, while still maintaining low engine out NO_x- and soot-emissions.

Compared to diesel-only conventional diesel combustion, due to a longer mixing time for the gasoline- and diesel-fuel to mix with air, lean and low temperature combustion will lead to significantly lower NO_x- and soot-emissions. Next to the potential for decreasing engine out-emissions, and therefore the possibility to decrease the need for after-treatment-systems, higher efficiencies are measured.

6. Conclusions and recommendations

6.1 Conclusion

The first part of this investigation analyzed premixed diesel-butane blends direct injected into a heavy duty diesel engine. From the results one can see that when the butane content is increased to a certain amount of extra butane, combustion phasing is influenced. Start of combustion is retarded when start of injection is kept constant. Ignition delay is enlarged and the time for the fuel to mix with the air is elongated.

When increasing the amount of butane in the blend the fuel consumption in liters will rise. Due to the lower energy density per volume of butane the fuel consumption in L/kWh raises with 10 to 20 percent, depending on the butane content. When butane content is increased to 50 percent, combustion phasing changes significantly and might improve the indicated efficiency by a shorter burn duration and a better timing of CA50.

Owing to the larger ignition delay, emissions are reduced. CO and soot emissions are decreased radically due to the better pre-mixing. NO_x and HC emissions can both slightly increase, depending on the butane content. An explanation for the higher HC emissions is, owing to the more rapidly vaporizing butane, local lean spots compared to diesel sprays.

Noise and engine hardware limits can be exceeded when maximum pressure rise rate is raised. When the fuel fraction of butane is increased, MPRR is raised due to the shift in combustion phasing and can lead to higher peak temperatures. Tentatively, our results seem to indicate that the amount of butane in the blend should not exceed 33 wt% for engine emissions and hardware safety reasons.

The second part of this study explored the use of in-cylinder blending of two fuels of different reactivity to achieve fuel stratification and local fuel reactivity for control of combustion phasing and rate of heat release in a low temperature combustion strategy. In-cylinder fuel blending using port-injected-gasoline and both early and late injections of diesel were used as charge preparation and fuel blending strategy. Blends of commercially available gasoline and diesel fuel were used to provide the desired auto-ignition timings. Engine experiments were performed to investigate dual fuel operation on a heavy duty diesel engine both in the conventional diesel combustion (CDC) regime, and in the premixed charge compression ignition (PCCI) regime.

Firstly, the addition of port injected gasoline with conventional timings was investigated. It was found that moderately increasing the amount of gasoline with a diesel injection at conventional timings, leads to over-mixed and too lean gasoline parts in the combustion chamber. Because of the incomplete combustion associated with this, HC and CO emissions are raised. It is therefore necessary that the amount of gasoline is sufficiently high; to make sure that the resulting gasoline-air mixture is within its flammability limits for complete combustion of the injected mass.

When the diesel injection is advanced and the shift towards RCCI combustion is made, several engine parameters need to be corrected. Even when using a low reactivity fuel like gasoline, the addition of the reactive diesel requires high EGR mass flows up to 60wt% of the total intake flow, to ensure longer ignition delays. For decreasing HC and CO emissions, and thus realizing a more complete combustion, intake pressure should be low; the intake pressure should be sufficiently low to maintain the gasoline-air mixture within its flammability limits. For minimizing HC-emissions the SOGI (start of gasoline injection) should be positioned at intake valve open.

Longer ignition delays create the possibility for partially premixed air-fuel mixtures to combust at lean and low temperature conditions. In the RCCI combustion concept, the majority of the injected fuel is premixed and thus is able to combust in the previously mentioned conditions. In these conditions, very low NO_x and soot emissions are produced; however, incomplete combustion still leads to an increase in HC- and CO-emissions.

RCCI is a promising concept for increasing thermal efficiency and thereby reducing fuel consumption. Short burn durations and low temperature combustion should lead to minimized heat losses. When more diesel is replaced by PIG, fuel consumption is lowered. Although burn durations are extremely short, a significant increase in thermal efficiency with respect to conventional diesel combustion remains unproven.

Furthermore, a multiple injection strategy was investigated for reducing UHC and improving control of combustion phasing. For reducing UHC, the multiple injection strategy seems to have a counter effect. Due to very small amounts of diesel-fuel per injection, lean local spots in the combustion chamber lead to incomplete combustion. The reactive diesel of the second diesel injection will not mix with the in-cylinder blend. It was found that the timing of SOI2 (late direct injection of diesel) has great influence on SOC and therefore can reduce PRR and HRR by exact control of combustion phasing. By dividing the total injected diesel fuel consequently over the two injections, the first injection can benefit from the long ignition delay and the second one can be used as a trigger for SOC. From this report one can conclude that control of SOC should be sought in diesel injection timing and quantity for fuel stratification and local fuel reactivity. Moreover a multiple injection strategy can result in an even more sophisticated control of combustion phasing, while still maintaining low engine out NO_x- and soot-emissions.

Compared to CDC, due to a longer mixing time for the gasoline- and diesel-fuel to mix with air, lean and low temperature combustion will lead to significantly lower NO_x- and soot-emissions. In RCCI combustion the advantage of lean and low temperature combustion is combined without losing control of combustion phasing. Next to the potential for decreasing engine out-emissions, and therefore the possibility to decrease the need for after-treatment-systems, higher efficiencies, compared to conventional CI engines, are measured.

6.2 Recommendations

Part I

From the results and conclusions given in Part I, a large scope of new research activities can be formed. The first step is to use LPG instead of butane. However, this needs some extra attention for the engine set-up.

Next to changing the fuel, to provide scientific foundation, one should show the maximum allowable butane content at high loads. When air fuel ratio is monitored very carefully through the experiments, differences between the blends can be expressed at higher loads.

When it is desirable to keep injection timing and actuation duration in the engine management system unchanged, it is interesting to show the maximum allowable butane content before limits, like emissions regulations and engine efficiency, are exceeded.

One of the challenges using LPG in a combustion engine is starting the engine under cold temperature conditions. Under these conditions, LPG is hard to ignite due to the lower reactivity compared to diesel. One of the points for further research is to investigate to what extent this cold start can be an issue.

To broaden the research and to compare with direct injection, a LPG port fuel injection system can be installed for injecting LPG into the inlet manifold on a DI diesel engine. Note that with this kind of injection, LPG is pre-mixed and the direct injection of diesel is not. This will lead to ignition of the premixed LPG by a diesel injection which is likely to produce a flame front at every injected spray. Temperatures in such a flame front in combination with high air fuel ratios can lead to high NO_x emissions. Another issue with these PFI systems is high HC-emissions due to UHC stuck in the crevices because of long in-cylinder mixing times.

Finally, the good indicated efficiency and fairly low emissions of NO_x, HC and CO shown, altogether form a good reason to further explore the proposed strategy of pre-mixed direct injection of LPG and diesel.

Part II

From the results and conclusions given in part II, new research activities can be recommended.

Fuel stratification and therefore local reactivity was found to be an important parameter for controlling combustion phasing. Efficiencies and heat losses are difficult to determine from the measured data, and certainly the contribution of different factors, such as flammability limits and radiation of soot particles. Here lies a challenge to model these processes, in for example multi-zone models, to investigate the effect fuel stratification and local fuel reactivity.

According to Reitz and coworkers [40], lower EGR rates were found to be beneficial to increasing thermal efficiency. It is therefore recommended to decrease EGR rates by investigating multiple injections strategies for more control of combustion phasing. It can, for example, be investigated what the minimum amount of diesel should be in the second injection for sufficient control of combustion phasing. Furthermore, a less reactive fuel can be used in a multiple injection strategy. This less reactive fuel, for example US diesel, can be used to advance the second diesel injection and therefore create more mixing time for diesel-fuel to mix with the in-cylinder blend. The advantage is a better pre-mixed mixture, including the coverage of the bowl-region, where combustion phasing is still controllable by the second injection.

Besides mid-loads, as were presented in this report, low- and high-loads should be investigated to cover a full range of the RCCI combustion potential. Note that high loads are limited due to pressure rise rates when less EGR is used and low loads are possibly limited by a minimal actuation duration of the diesel injector. Future work should demonstrate that with an injection strategy optimized for mid-load, good results are possible for a wider range of loads.

At lower loads one could try, to benefit the efficiency, an increase in compression ratio to enable a full comparison of today's diesel engines. This higher compression ratio can be achieved by changing the geometric compression ratio to, for example, 17:1.

Finally, a first response on the results shown in this report is a comparison of the total efficiency, including combustion efficiency, thermal efficiency, heat loss efficiency and exhaust loss efficiency, with the experiments done in Wisconsin. With this comparison an explanation should be found for the difference in thermal efficiency of the experiments in Eindhoven and in Wisconsin.

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Bas van den Berge

Tilburg, June 2011

Appendix A: Experimental apparatus

A.1 Port fuel injection control

To control the Vialle28 LPG injector a Motec M400 engine management system is used. For controlling injection timing and injector actuation duration two signals are needed as an input for the Motec. The first signal is a REF signal, which provides a single pulse every rotation of the crankshaft. Second the SYNC signal is needed to provide a pulse every two engine rotations, for indicating a single stroke of a complete cycle, which is normally done by the camshaft.

The REF signal coming from the original configuration of the DAF XE engine is described in A.2. In the motec software this configuration is not programmed and cannot be used. For this reason a universal trigger wheel from Van Kronenburg is mounted on the crankshaft. This trigger wheel features 60 theeth with two missing teeth every 360 degrees. A hall sensor, also coming from van Kronenburg, is placed next to the trigger wheel to sense the pulse of the missing theeth and provide the motec an input signal. This configuration is recognized by the motec software.

For the SYNC signal the motec software needs a step signal every two crank rotations. Again the original DAF configuration cannot be used. To solve this an extra pulse coming from the Flecs engine management system, which normally provides the diesel injector a peak-hold signal, is used to trigger the SYNC signal input in the motec software. A peak hold signal is converted into a step signal of 2 ms, as it would come from a hall sensor, and is given to the motec system. This way the REF and SYNC signal easily can be synchronized with the help of software instead of hardware configuration.

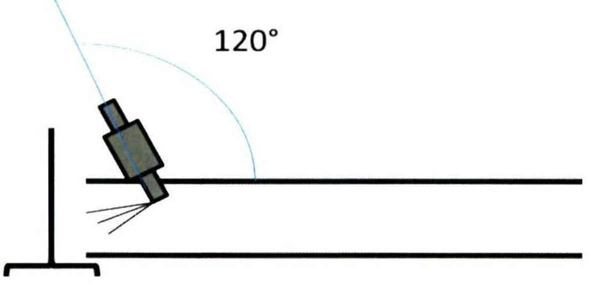

Figure A.1 injector position in inlet manifold

The port gasoline injector is placed into the inlet manifold with an angle of 120°. This done to spray directly into the cylinder, minimizing wall wetting on the side of the inlet manifold, see Figure A.1.

To validate the gasoline injector, manufactured by Vialle, different actuation durations at an a engine rotation speed of 1200 rpm are tested. The flowrate is measured by a Micro Motion Coriolis CNFM 025 flowmeter. The maximum flow needed for this experiment set-up is about 1.0 g/s @ 1200 rpm, which is easily achieved with the PFI system. The fuel pressure is not measured, but the flowrate is measured at all times.

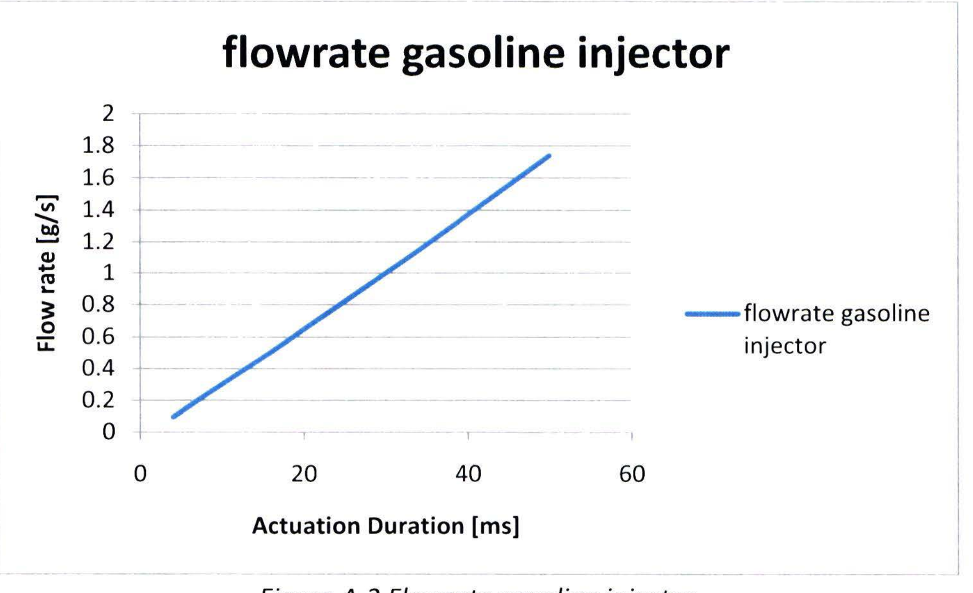

Figure A.2 Flowrate gasoline injector

A.2 DAF ECU timing configuration

The REF and SYNC signal are provided by trigger wheels in combination with Hall sensors. Each wheel has x teeth with one missing each rotation to identify reference pulse to the sensors. Slots or holes can be used instead of teeth and only require that the sensor wiring to be reversed in order to achieve the correct signal polarity.

The crank wheel is spaced to have 60 teeth around the circumference, at 6 degree intervals. Note that two teeth are removed from the crank wheel in the positions where they coincide with a CAM tooth. This leaves a crank wheel with three groups of 18 teeth. The crank wheel is aligned so that compression stroke TDC is on the 13th tooth.

The cam wheel has six identification teeth spaced at 60° intervals around the circumference; each tooth corresponds to a cylinder. An extra tooth identifies that the next tooth represents cylinder 1, this is often referred to as the plus1 tooth.

The cylinder 1 tooth on the cam wheel is positioned 34.5° or 40.5° cam deg before TDC (on cylinder 1) depending on if TDC is on the 11th or 13th tooth → MX uses 13th tooth.

The 'plus one' tooth, to show that the next tooth identifies cylinder 1, is positioned 15° before the cylinder 1 tooth, see Figure 1, below (note that Figure A.3 shows TDC for the 11th tooth, so 34.5° is shown).

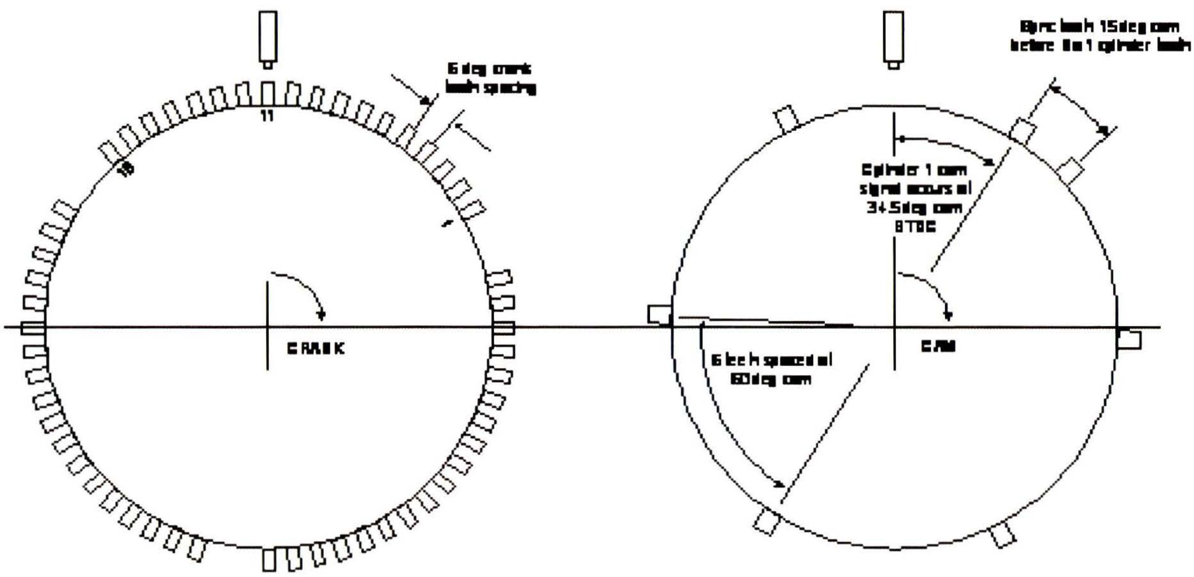

Figure A.3: Crank and cam wheel configuration for DAF XE engine

A.3 Valve Timing

Injection of port gasoline is timed at 300 °bTDC, resulting in an injection of gasoline at IVO and EVC. By using this timing no UHC are blown through the engine due to the valve overlap. In the figure below one can see the valve timings used on the CYCLOPS. There is one situation that can cause for any UHC; that is during injection of the port gasoline wall wetting in the intake manifold can occur, which can be blown into the exhaust during the valve overlap of the next cycle. It is believed that this amount of UHC is too little for influencing fuel efficiency.

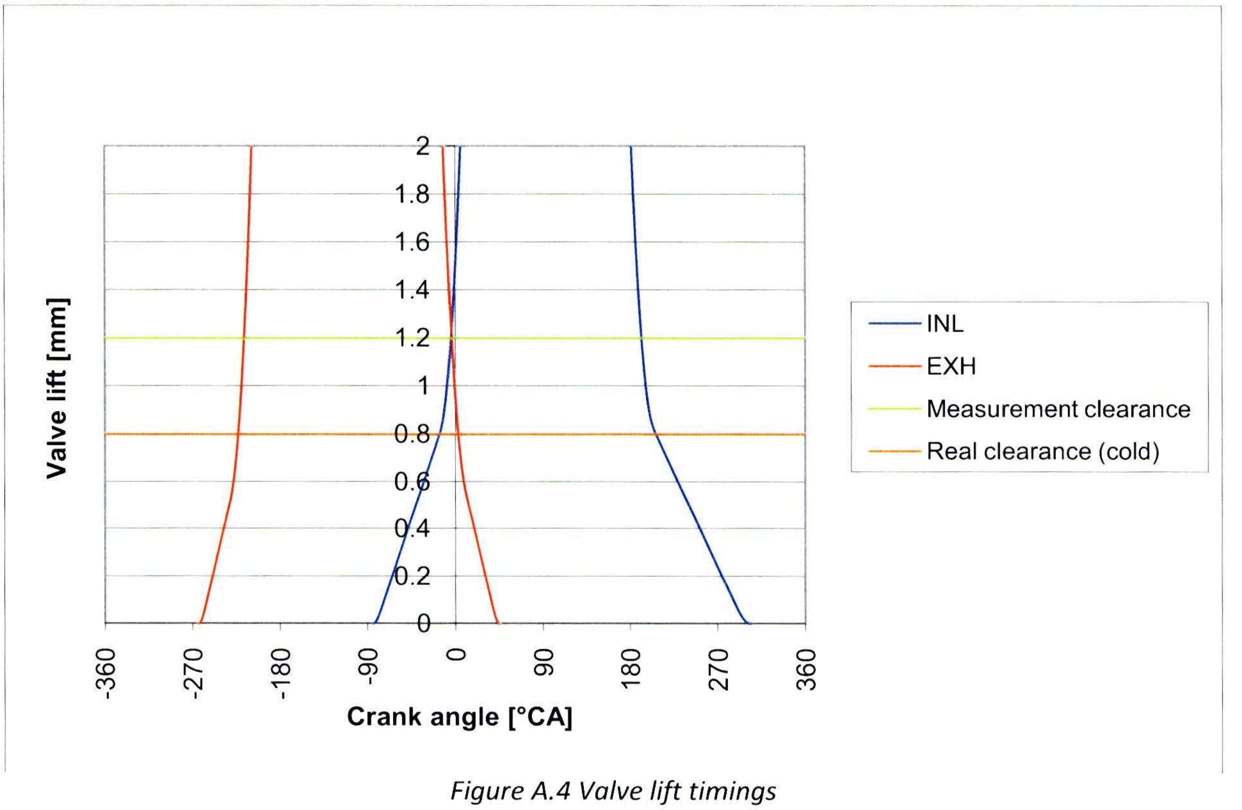

Figure A.4 Valve lift timings

	Measurement clearance		Real clearance (cold)
IO	4 °CA bTDC	-4	16 °CA bTDC
IC	12 °CA aBDC	192	27 °CA aBDC
EO	36 °CA bBDC	-216	52 °CA bBDC
EC	4 °CA bTDC	-4	14 °CA aTDC

A.4 Leaking intake valve

During the experiments losses in the indicated mean effective pressure were noticed. The losses were due to a leaking inlet valve as can be seen in Figure A.5 and A.6. The inlet valve probably encountered high local temperatures on the valve seats, which was probably caused by high percentages of EGR. High percentages of EGR can cause high soot percentages in the inlet air, which can cluster on the inlet valve en form local hot spots. The cylinder head has been revised at DAF Trucks, where 6 new intake valves were replaced.

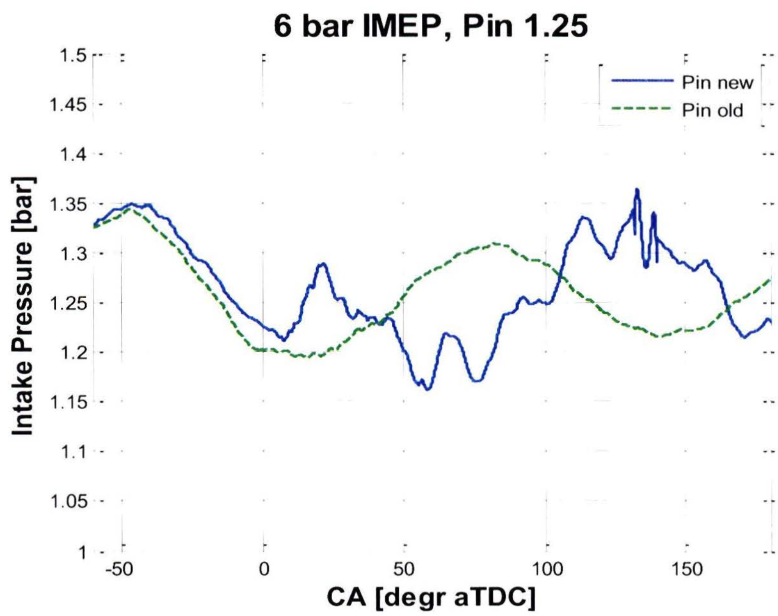

Figure A.5 Intake pressure April 2010 vs January 2011.

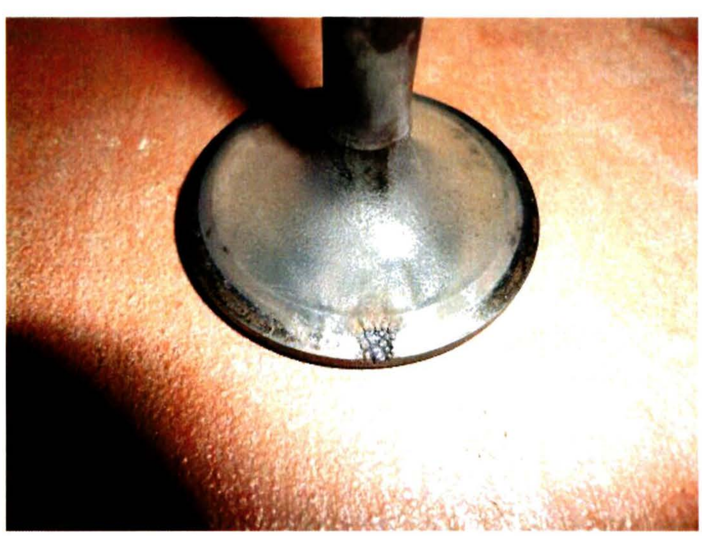

Figure A.6 Damaged intake valve

Appendix B: Fuels and calculations

B.1 Reference fuels

In Table B1 below, the test results are given for the EN590 diesel.

Table B1 EN590 diesel base fuel specifications

Component	Method	Units	Result
IBP (I123	IP 123	Deg C	170.2
10% RECOVERY (123-3)	IP 123	Deg C	210
20% RECOVERY (I123-4)	IP 123	Deg C	225.7
30% RECOVERY (I123-5)	IP 123	Deg C	240.3
40% RECOVERY (I123-6)	IP 123	Deg C	254.9
50% RECOVERY (I123-7)	IP 123	Deg C	268.5
60% RECOVERY (I123-8)	IP 123	Deg C	281.9
70% RECOVERY (I123-9)	IP 123	Deg C	296
80% RECOVERY (I123-10)	IP 123	Deg C	312.5
90% RECOVERY (I123-11)	IP 123	Deg C	333.3
95% RECOVERY (I123-12)	IP 123	Deg C	349.6
FBP (I123-13)	IP 123	Deg C	361.4
RESIDUE (I123-14)	IP 123	% Vol	1.2
RECOVERY (I123-15)	IP 123	% Vol	98.6
LOSS (I123-16)	IP 123	% Vol	0.6
RECOVERY@240C (I123-27)	IP 123	% Vol	30.3
RECOVERY@250C (I123-28)	IP 123	% Vol	37
RECOVERY@340C (I123-29)	IP 123	% Vol	93
RECOVERY@345C (I123-30)	IP 123	% Vol	
RECOVERY@350C (I123-36)	IP 123	% Vol	95.7
RECOVERY@360C (I123-92)	IP 123	% Vol	97.7
RECOVERY@370C (I123-37)	IP 123	% Vol	
Lubricity	ISO 12156	micron	229
Derived Cetane Number (I498)	IP 498		55.9
Di+Tri(+)Aroms (I391)	IP 391	% m/m	3.4
Mono Aroms (I391-1)	IP 391	% m/m	27.3
Di Aroms (I398-2)	IP 391	% m/m	3.3
Tri(+) Aroms (I391-3)	IP 391	% m/m	0.1
Total Aromatics (I391-4)	IP 391	% m/m	30.7
TOTAL AROMATICS (EN12916)	IP 391	% m/m	30.7
Gross Heat of Combustion (I12)	IP 12	MJ/kg	44.52
Gross Heat of Combustion (I12-1)	IP 12	cal(IT)/g	10630
Net Heat of Combustion (I12-2)	IP 12	MJ/kg	41.54
Net Heat of Combustion (I12-3)	IP 12	cal(IT)/g	9920

B.2 Lambda calculation of dual fuel application

C=12
O=16
H=1
N=14
θ= mass fraction diesel in fuel blend

$$AF_{dual} = \frac{m_{air}}{m_{diesel}+m_{gasoline}} \tag{1}$$

The air-excess ratio λ by definition equals the air-fuel ratio over the stoichiometric ratio. The air-to-fuel ratio AF_{dual} follows than from Eq. (1). For dual fuel operation we can write

$$\lambda_{dual} = \frac{AF_{dual}}{AF_{st.dual}} = \frac{AF_{dual}}{\theta * AF_{st.diesel} + (1-\theta) * AF_{st.gasoline}} \tag{2}$$

The stoichiometric air-to-fuel ratios for the two fuels are derived from their reaction equations. Assuming $[O_2]/[N_2]$ =3.76, the reaction equation for diesel is then

$$C_{13}H_{28} + 20\ O_2 + 75.2\ N_2 \rightarrow 14\ H_2O + 13\ CO_2 + 75.2\ N_2$$

Resulting in $AF_{st.diesel} = 14.92$

For gasoline this equation holds

$$C_8H_{18} + 12.5\ O_2 + 47\ N_2 \rightarrow 9\ H_2O + 8\ CO_2 + 47\ N_2$$

Resulting in $AF_{st.gasoline} = 15.05$

Using Eq (1) en (2) for the overall 'dual fuel mixture' of diesel and gasoline we obtain

$$AF_{st.dual} = 14.92 * \theta + 15.05 * (1-\theta) \tag{3}$$

$$\lambda_{dual} = \frac{m_{air}}{(m_{diesel}+m_{gasoline}) * [14.92\theta + 15.05(1-\theta)]} \tag{4}$$

Appendix C: Measurement procedure

C.1 Starting and stopping the engine

The procedures below are adapted from [5] and expanded.

In order to start the engine a number of steps have to be followed carefully:

- Turn on the coolant system for engine cell 6
- Turn on the power for the "day tank" diesel fuel circuit
- Engine cell 8: Fill the diesel "day tank" with at least 100 liters. Any excess fuel will be spilt off back into the main tank.
- Turn on the air treatment system (+ extra ventilation) for engine cell 6 (next to the door of engine cell 6)
- Turn on the air supply to the test cylinder (in engine cell 5)
- In engine cell 6, open all the coolant valves on the engine (*e.g.* two full turns)
  - Engine coolant, near the engines crankshaft
  - Cooled EGR
  - Dynamometer coolant
- Open all the fuel valves to and from the engine in engine cell 6
- Turn on the low pressure fuel pump (3-phase jack)
- Turn on the battery charger
- Turn on the AVL smoke meter
- Turn on three 220V sockets, two next to the resato pump and one in at the computer on the left wall,with these you:
  - Turn on the FLECS power adapter
  - Turn on the FLECS PC and press Enter
  - Turn on the power to the Heidenhain encoder
- Turn on/reset the in-cylinder pressure sensor amplifier
- Turn on the low pressure fuel pump (on the resato) for the test cylinder
- Walk around the engine to check for leakages or other defects
- Check the oil level (and color/condition) and the coolant level
- Leave the cell door open
- Turn on the room ventilation of the operating room
- Set the intake pressure to the test cylinder to 1 bar (*e.g.* at higher values the engine will not start)

- Turn on the brake power system

- Reset the system
- Turn on the contact and start the engine (*e.g.* the engine work point is set to its default idling value)
- Walk around the engine to check for leakages or other defects
- Open the air pressure to the Resato fuel pump and set the fuel pressure.
- Close the cell door.

To shut down the engine, perform the above actions in reverse order. Make sure the engine is idling for some minutes before shutdown. Don't forget to turn down the air pressure to the test cylinder.

C.2 Horiba Mexa 7100 DEGR

Typically the emission equipment is calibrated before the engine is turned on to save fuel and so the operator is not distracted.

Check the various filters of the system for excessive fouling and replace them if necessary

- Turn on the pressurized air to the Horiba emission equipment
- In the storage room for gases, open valves 6 (N2), 9 (O2), 7 (SynAir), 5 (O2), 2 (CO2), 4 (NOx), 1 (CO), 8 (H2/He) and 12 (THC).
- In measuring cell 6, open valves 12, 9, 8, 7, 6, 5, 2, 1 and 4.
- Turn on the Horiba PC, it is a Linux system
- Click "enter"
- In "menu" click "continue" and in "view" click "1-line-chart"
- In "standby" click "standby" and allow the system to heat up for 30-60 minutes
- When the system is at operating temperature, the flashing "alarm" light will turn from red to white.
- In the PC menu, click THC (*e.g.* hydrocarbon emissions)

- Click "zero", the THC emissions should go to zero, if not click "cal"
- Set "span" to lowest range, THC should rise to the level of the calibration gas, if not click "cal"
- Click "reset"

- The same procedure holds for NOx, O2, CO2 and CO, with their respective calibration values.
- When calibration is complete close valves 1, 2, 4, 9 and 12 in measuring cell 6 and the gas storage room
- The Horiba equipment is ready to be used
- When the measurements are complete press "purge" and wait for five minutes
- Press shut down
- Close valves 5 thru 8 in measuring cell 6 and the gas storage room
- Close the valve for pressurized air to the Horiba

Appendix D: Additional results

D.1 Effect of air excess ratio

The air-fuel ratio determines the oxygen available per fuel part. When the air-fuel ratio exceeds upper or lower flammability limits, fuel will burn incompletely, which will result in higher CO and HC emissions. In these measurement series the air-fuel ratio is raised by increasing intake pressure. Generally, when using only DI diesel, higher intake pressures decrease HC and CO emissions effectively [5]. However, when increasing intake pressure, combustion starts earlier due to higher pressure at the end of compression stroke. In Figure D.1a one can see that CA5, which is independent of the amount of added gasoline, is advanced due to the higher intake pressure. It is hypothesized that this also influences burn duration but, the BD, which can be seen in Figure D.1b, is dominated by higher flame speeds of the gasoline-air mixture after injection due to the lower air-excess ratio. As mentioned before, when using port gasoline injection, the premixed gasoline, representing the mixture before the diesel injection occurs, needs a certain mixture composition for ignition. When air excess ratio is lowered, the premixed gasoline-air mixture becomes richer, resulting in a more complete combustion. Figures D.1c and d show that CO- and HC-emissions decrease when intake pressure is lowered. When more than 50wt% of gasoline is added to the intake air mass, this mixture becomes sufficiently rich to form high flame speeds and burns more completely. It is therefore needed to run with lower intake pressures, to stay within the flammability limits, when RCCI combustion is realized.

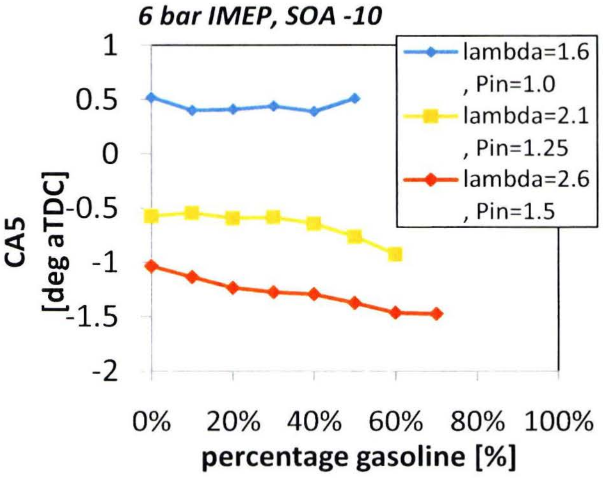

Figure D.1a CA5 vs % gasoline

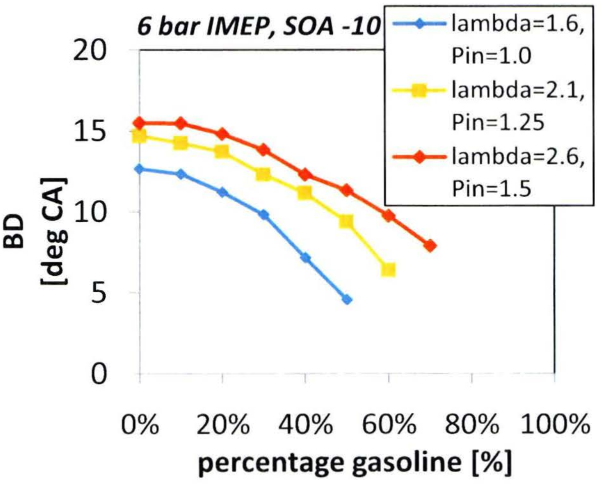

Figure D.1b BD vs % gasoline

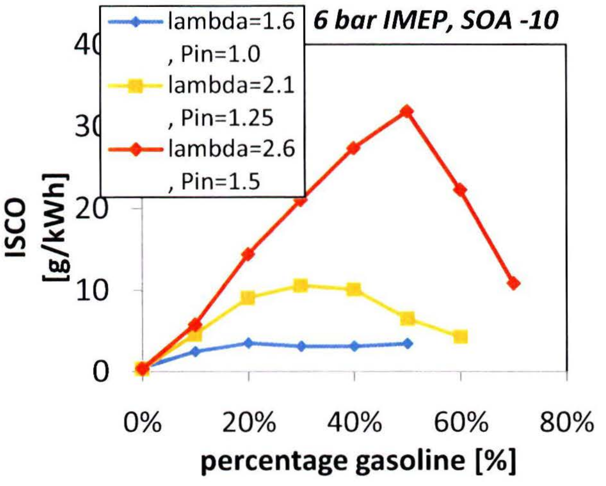

Figure D.1c ISCO vs % gasoline

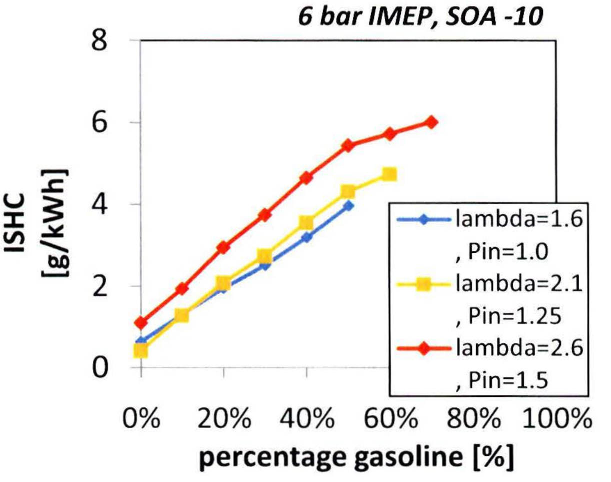

Figure D.1d ISHC vs % gasoline

D.2 Cumulative heat release

In Chapter 4 of this report, no cumulative heat release data is shown. Below one can find an additional result to show the total heat release produced. The measurement was performed at 6 bar IMEP, a constant SOA of the diesel injector and an increasing gasoline fuel fraction. The results show that when more diesel is replaced by port injected gasoline, burn durations decrease, combustion speeds increases, and a sufficiently amount of gasoline is needed to produce sufficient energy at an early stage in the combustion, for a well-positioned CA50.

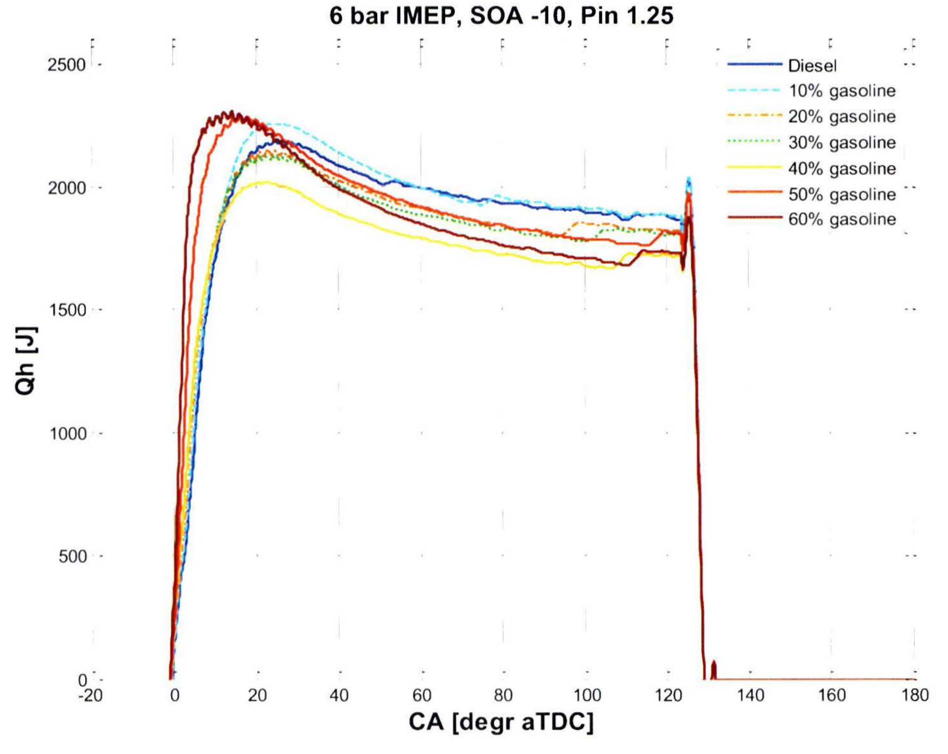

Figure D.1 Cumulative heat release for different gasoline fuel fractions at CDC timings