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Improvement of Combustion Process and Exhaust Emissions with Premixed Charge Compression Ignition

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Abstract: Premixed Charge Compression Ignition (PCCI) is a combustion concept that is characterized by low temperature, partially premixed combustion using early injections, large ignition delays and high percentages of Exhaust Gas Recirculation. This review paper discusses the premixed charge compression ignition (PCCI) and the way to reduce the emissions. A promising low-emission combustion concept is used by the premixed charge compression ignition (PCCI). Air and exhaust gas before auto-ignition, the soot and NOx emissions are lower than for conventional diesel combustion by partially mixing the fuel. Fuel lean premixed combustion has potentials to achieve high efficiency and low emissions. The ignitability of lean mixture, flame stability and controlling in this combustion are significant issues to be addressed. The main issues for lean burn intermittent combustion engines are (i) the mixture preparation for lean combustion requires expensive or premium technology and (ii) achieving this combustion over a wide range of load and speed is difficult for a smooth-running engine.

Keywords: Premixed charge compression ignition (PCCI), emissions, hydrocarbon (HC), carbon moNOxide **(CO),** particular matter (PM)

1. Introduction

1.1 Injection Timing to Reduce NOx Emission in Premixed Control Compression Ignition

Premixed Charge Compression Ignition (PCCI) is a combustion concept that is characterized by low temperature, partially premixed combustion using early injections, large ignition delays and high percentages of Exhaust Gas Recirculation (EGR) [1]. The concentration or temperature can be controlled several ways: High compression ratio, preheating of induction gases, forced induction, retaining or reintroducing exhaust gases [2]. Several studies [3-4] have shown that the PCCI combustion strategy can be a practical solution in diesel engines to satisfy stringent future emission regulations since the strategy achieves a good soot and NOx trade-off with an acceptable impact on fuel economy and carbon moNOxide emissions. However, the key parameters that determine the soot and NOx trade-off have not been completely identified. The objectives of this study were to achieve low emission of PCCI operation and to identify key parameters that enable the simultaneous reduction of soot, NOx, and carbon moNOxide emissions. There were six nozzles with different spray angles ranging from 50 to 154 degrees were investigated to identify the effects of the spray targeting on emissions. The results obtained from the six different spray angles were evaluated for PCCI. PCCI combustion was

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differentiated from standard diesel combustion by noting its combustion characteristics. In this study, late injection or MK combustion, which is a well-established low emissions strategy, is not considered since it has similar combustion characteristics to PCCI due to its long ignition delay. Thus, MK combustion may be considered as a special case of PCCI combustion [5,6]. It was difficult to evaluate the characteristics of MK combustion with different spray angles since the stability of MK combustion was strongly affected by only slight changes of the injection timing and EGR.

1.2 Pressure and Heat Release with Dual-Fuel Stratified PCCI to Reduce Emission in Diesel Engine

Premixes Charge Compression Ignition (PCCI) is a combustion concept that is characterized by low temperature, partially premixed combustion using early injections, large ignition delays and high percentages of Exhaust Gas Recirculation (EGR) [5]. Premixed Charge Compression Ignition (PCCI) has become the priority of engine research around the world. The advantages of PCCI are high level of thermal efficiency and very low NOx and soot emission. The combustion process is characterized by chemical kinetics as PCCI involves highly dilute mixture combustion [7,8]. Thus, the burning rate and ignition control timing is more difficult in PCCI system than in a regular diesel engine. The commercialization of PCCI is the development of a method for controlling combustion process**.**

1.3 Effect of Compression Ratio on Partially Premixed Charge Compression Ignition Engine

A value that represents the ratio of the volume of its combustion chamber from its largest capacity to its smallest capacity is known as the compression ratio of an internal-combustion engine or external combustion engine. It is a fundamental specification for many common combustion engines. In a piston engine, the compression ratio is the ratio between the volume of the cylinder and combustion chamber when the piston is at the bottom of its stroke, and the volume of the combustion chamber when the piston is at the top of its stroke. A movable plunger at the top of the cylinder head can changed the geometric compression ratio. This system is used in diesel model aircraft engines. By closing the intake valve either very late or very early with variable valve actuation (variable valve timing that enables the Miller cycle), the effective compression ratio can be reduced from the geometric ratio [9,10]. Both approaches require energy to achieve fast response. Additionally, the implementation is expensive but effective.

Figure 1 shows the illustration of the compression ratio. A high compression ratio is desirable because it allows an engine to extract more mechanical energy from a given mass of air-fuel mixture due to its higher thermal efficiency. High compression occurs because internal combustion engines are heat engines, and higher efficiency is created because higher compression ratios permit the same combustion temperature to be reached with less fuel, while giving a longer expansion cycle to create more mechanical power output and lowers the exhaust temperature [11-13]. Diesel engines actually have a higher peak combustion temperature than petrol engines, but the greater expansion means they reject less heat in their cooler exhaust. Higher compression ratios will however make gasoline engines subject to engine knocking if lower octane-rated fuel is used, which is also known as detonation. This can reduce efficiency or damage the engine if knock sensors are not present to retard the timing. On the other hand, diesel engines operate on the principle of compression ignition, so that a fuel which resists auto ignition will cause late ignition, which will also lead to engine knock. When the compression ratio is decreased from the conventional cases the ID is extended which enable complete injection of all the fuel to be accomplished prior to ignition. This condition is conducive for premixed combustion and a reduced maximum in-cylinder temperature at TDC.

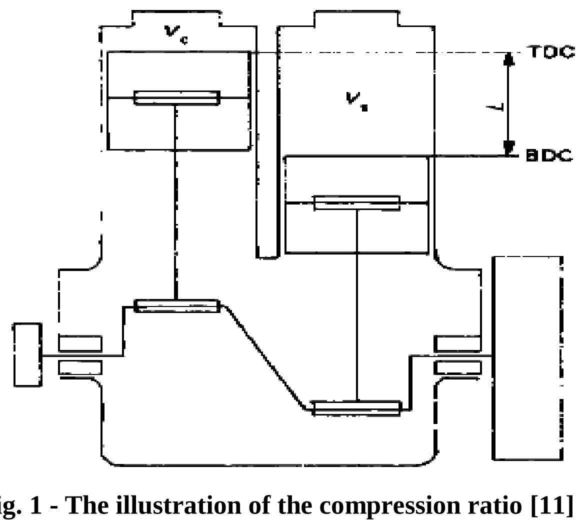

Fig. 1 - The illustration of the compression ratio [11]

HCCI engine has higher thermal efficiency at compression ratio of 10.7. There are no significant effects of NOx and smoke emission. The compression ratio is increased by increasing in CO and HC emission [6,8]. Reduction of the compression ratio from 18:1 to 16:1 was one of the strategies employed in the second generation of MK combustion which made it possible to extend the low temperature and premixed combustion to higher load conditions [7,10]. The compression ratio is increased by increasing in CO and HC emission. Bahattin Celik, (2008) [7] reported that the gasoline engine was used with higher compression ratio; the ethanol fuel was used to improve its performance and reduce emissions. The compression ratio 6/1 was used initially at various ratios of tested fuels E25, E50, E75, and E100. The experimental result indicates that E50 has given 29% of engine power compared to E0 with increasing in compression ratios. The emission characteristics specific fuel consumption was also reduced in this stage [9].

2. Experimental Setup

2.1 Injection Timing to Reduce NOx Emission in Premixed Control Compression Ignition

The present test engine consists of a Hydra single cylinder engine from Ricardo Research and a single cylinder version of a production Fiat cylinder head which was designed for a 2.4L five-cylinder engine. The cylinder head is prepared with double over-head camshafts and two exhaust ports and two intake ports, which consist of a helical port and a directed port to control the swirl ratio. The original Fiat piston was replaced with an open-crater-type bowl piston which was designed using CFD combined with a genetic algorithm (GA) optimization [10, 13] to realize the NADI concept. The engine specifications are shown in Figure 2. [11]

Engine Type	4 valve DI diesel
Bore x Stroke	82.0 x 90.4 mm
Compression Ratio	16.0:1
Displacement	477 cm^3
Piston Geometry	Open-Crater-bowl
Intake Ports	1 swirl, 1 tumble
Swirl ratio (at IVC)	1.83
Bowl diameter	48mm
IVO	10° BTDC
IVC	38° ABDC
EVO	38° BBDC
EVC	8.5° ATDC

Fig. 2 - The engine specifications [11]

The injection system used for this study was a Bosch common-rail injection system. The specifications of the common rail injection system are stated in Figure 3. [12,14]. The nozzles have eight holes except for the nozzles with narrow spray angle (50 and 85 degree) which had 6 holes due to machining restrictions. The difference in the number of nozzle holes affects the level of emissions [12,14]. However, the focus of the present study was to evaluate the effect of spray targeting in qualitative manner and emission control. Intake air was supplied from pressurized building air to simulate a turbocharger. The flow rate was measured with a critical orifice system. EGR flow was driven with the pressure difference between the intake surge tanks and the exhaust and was controlled by regulating the openings of the valves.

Injector type	Electro-hydraulically controlled injector	
Nozzle type	Dual guided VCO	
Flow Number	400cm^3/30s @100bar	
Number of nozzle holes	6	8
Hole diameter	0.154mm	0.133mm
Spray included angle	50°, 85°	120°, 130° 140°, 154°

Fig. 3 - Common rail system specifications [12,14]

The emission data recorded during the experiments included gaseous and particulate emissions. The gaseous emissions, including NO, NOx, CO and Exhaust CO_2, were measured with a Nicolet Rega model 7000, FTIR emissions analyser. Intake CO_2 was assessed with a Horiba PIR- 9000 infrared gas analyser. Exhaust smoke levels were sampled with a Bosch RTT100 instrument. The instrument measures visual opacity and converts it into mass concentration in mg/m^3 through the use of Internal tables. The mass concentration was converted into specific soot emission in g/kW-hr. Measurement of the concentration ranges from 10 to 1966 mg/m^3 with a 1 mg/m^3 resolution. The lower bound of the measuring range, 10 mg/m^3 mass concentration corresponds to around 0.05 g/kW-hr at the operating conditions in this study. Special care is required when the soot level achieved in PCCI combustion are compared to the lower bound of the unit. However, this does not affect the results of this study, since the main objective of the study is to investigate optimum ways of the injection timing to achieve low emissions of PCCI combustion in qualitative manner [12]. The quantitative values achieved in the study are less importance since they depend strongly on the operating conditions, including equivalence ratio, injection pressure, EGR, and piston geometry. Cylinder pressure was measured with a Kistler 6125A piezo-electric pressure transducer and the measured signal was amplified and converted into a voltage with a Kistler 5010 charge amplifier. The output signal from the charge amplifier was sampled with a National Instruments AT-MIO-16E-1 data acquisition board every quarter degree crank angle. The pressure traces were averaged over 300 cycles to compensate for cycle-by-cycle variations. The IMEP includes pumping loss during the gas exchange process by considering the whole engine cycle. Heat loss was not considered and the specific heat was assumed to be constant with the value of 1.33, when the heat release rate was evaluated.

2.2 Pressure and Heat Release with Dual-Fuel Stratified PCCI to Reduce Emission in Diesel Engine

The experiment conducted was to investigate on the effects of operating conditions on PCCI combustion. As part of an NCM funded program [13], the impact of fuel composition will be investigated for the use in a dual mode PCCI engine; PCCI at low loads and conventional diesel combustion at higher loads. In this topic the details of this experimental study are discussed. The measurements contained in this study were all performed on a dedicated engine test rig, referred to as Cyclops. In section 2.2.1, the test rig is discussed including the changes made to it for this measurement series, for instance new fuel injection and emission measurement equipment. In the next section the experimental procedure in which all experiments are performed are discussed and the operating conditions which are chosen to be kept constant overall operating points are given here. The measurement matrix constructed for this program, which contains the different levels of parameters under investigation is given in section 2.2.7. In the first part of this study the effects of these parameters are varied while using a conventional diesel fuel. Meanwhile, the second part of this study the influence of fuel composition on PCCI combustion will be investigated. The fuels selected for these experiments, their specifications and the reasons they are chosen for are discussed further in section 2.2.8. This topic is concluded with the definitions used and some comments on the analysis used for all results.

2.2.1 Experimental Apparatus

Frijters et al., has described the CYCLOPS engine platform used for the experiments in detail [14]. Below the setup is briefly presented, together with the changes made for the current measurements.

2.2.2 Test Engine

The CYCLOPS is a dedicated engine test rig as in Figure 4, which is 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. When data acquisition is idle, for instance during engine warm up or in between measurements, 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. For the purpose of the current investigations, to limit end of compression temperature in order to avoid premature auto ignition, the geometric compression ratio of the test cylinder has been lowered. Ideally, this would have been done by implementing variable valve actuation, or by modifying the piston bowl geometry. For this preliminary investigation here, the compression ratio has been reduced from the stock value of 17 to 12 by means of a thicker head gasket. The compression ratio has been computed using the CAD models of the piston and cylinder head, taking the geometry of the valve seats and the space below the top spring into account. Apart for the mutual cam- and crankshaft, plus the lubrication and coolant circuits, the test cylinder operates autonomously from the propelling cylinders. Standalone air, EGR and fuel circuits have been designed for maximum flexibility as will be discussed below.

Base engine	6 cylinder HDDI diesel
Cylinders	1 isolated for combustion
Bore [mm]	130
Stroke [mm]	158
Compression ratio [-]	variable, original 17.0
Bowl shape	M-shaped
Bowl diameter [mm]	100

Fig. 4 - Cyclops specification

2.2.3 Air and EGR Circuits

Fed by an Atlas Copco air compressor, the intake air pressure of the test cylinder can be increases 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 in the intake manifold of the test cylinder [15]. 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 as shown in Figure 5, where the purpose, 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. It is measured using a Micro Motion Coriolis mass flow meter. However, during the preparation for this measurement series, unrealistic values were observed for the EGR mass flow. As the result, this device has been removed from the setup. In this series, EGR mass flow is estimated from the fresh air mass flow and computations regarding the volumetric efficiency. A few surge tanks, to dampen oscillations and ensure adequate mixing of fresh air and EGR flows, and pressure relief valves which is used to guard for excessive pressure in the circuit have been included in the design.

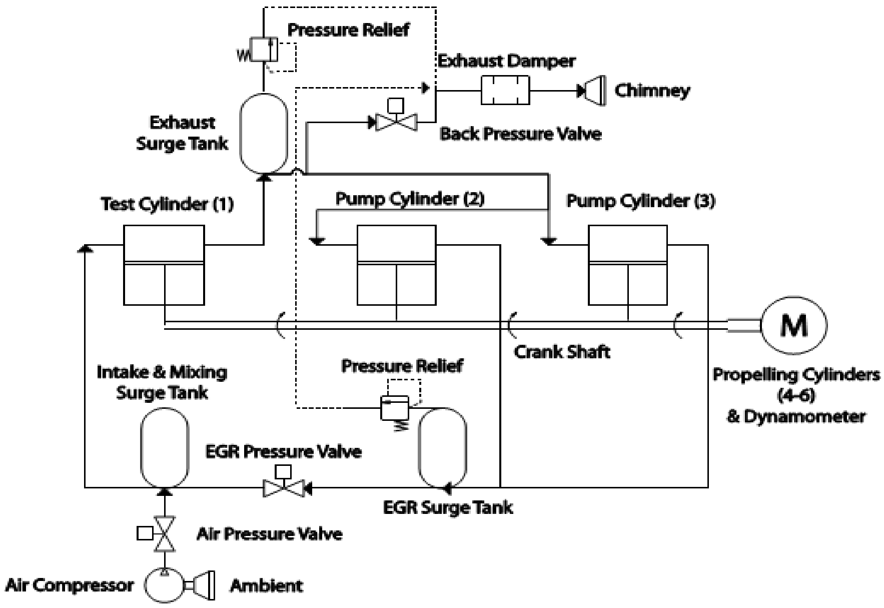

Fig. 5 - Test section of the Cyclops engine test rig [15]

2.2.4 Fuel Injection Equipment

To be able to evaluate the effect of high fuel injection pressure, fuel to cylinder 1 is prepared by a double-acting air-driven Resato HPU200-625-2 pump which can deliver a fuel pressure up to 4200 bar. To make dampen pressure fluctuations originating from the pump, two accumulators are mounted downstream of the pump. Between these two accumulators, the second, placed near (~ 0.2 m) the fuel injector is to mimic the volume of a typical common rail and can be temperature controlled (up to 250 °C), to heat the fuel to the certain temperature. For these experiments a new prototype common rail injector is used, which is able to inject the fuel with a pressure up to 3000 bar. This spray nozzle has 8 holes of 0.151 mm diameter with a cone angle of 153 degrees. With the given high injection pressures which are possible with this setup, it is assumed that this nozzle also will perform well at conventional diesel combustion. The steady state fuel mass flow is measured using a Micro Motion mass flow meter, while the time resolved pressure and temperature are recorded in the fuel line approximately 50 mm from the injector with a Kistler pressure sensor. In addition, the injector actuation current is measured with a clamp meter. Both of these signals are used to estimate the intrinsic lag between start of electric actuation (SOA) and the start of fuel injection (SOI).

2.2.5 Data Acquisition

Several parameters, which are used to monitor the test setup and to characterize combustion behaviour, are recorded using different data acquisition systems. For measuring gaseous exhaust emissions a Horiba Mexa 7100 DEGR emission measurement system is used. This system also can compute the in-cylinder lambda value from the five measured gases (CO, CO2, O2, THC and NOX). The exhaust smoke level (in Filter Smoke Number or FSN units) was measured using an AVL 415 smoke-meter three times per operating point, of which the average is used. The engine is provided with all common engine sensors, such as intake pressure and exhaust pressures, and its temperatures, and oil and water temperature. These quasi steady-state engine data, both 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) [15]. The average of these measurements is recorded as the value for the operating point under investigation. The TUeDACS system is also used to online monitor the correct operation of the test setup. 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.

2.2.6 Experimental Procedure

Prior to a measurement series, the engine speed is set to desired value and the engine is warmed up until the lubrication oil and coolant fluid are 90 and 80 °C respectively. All operating conditions are set to the certain value and the test cylinder is fired at a conventional timing to increase temperature of combustion chamber and exhaust. Measurements are only taken after the CO2 content of the exhaust flow has stabilized. All tests will be conducted at 1200 rpm similar to the engine speed of a typical road transport vehicle at highway cruising speed. This also is close to the B speed in the European Stationary Cycle (ESC). Fuel injection pressure is set to 100 bar with an uncooled common rail, resulting in a fuel temperature near the injector of 30 °C. The state EGR mass flow is heavily cooled using cold process water, resulting in an EGR temperature of ca. 300 K. A single injection strategy is used, and the injector actuation duration is kept constant, except for n-heptane because of its lower density. For each fuel and actuation duration, the average fuel mass flow is used with the resulting IMEP to compute the Indicated Specific Fuel Consumption (ISFC). By doing this, ease of measurement is greatly increased compared to keeping IMEP constant and varying the injected fuel mass accordingly. For each combination of operating conditions under investigation, a sweep of start of actuations (SOA) of the injector will be performed. For every SOA, the engine is stabilized for at least 60 seconds and until the standard deviation of IMEP is below 0.1 bar. Starting from conventional CI timings, SOA is advanced at 5 degree increments, being a trade-off 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.

2.2.6.1 Measurement Matrix

As discussed in the introduction, in the first part of this study the effects of several parameters on PCCI combustion in general and pressure rise rate in particular are investigated. Auto-ignition chemistry is mainly governed by chemical kinetics, therefore control of the combustion phasing is highly dependent on fuel type and consequently on in cylinder conditions [15]. These in cylinder conditions, characterized as the temperature (T) – pressure (p) –equivalence ratio (φ) history in this series are influenced using different levels of the following parameters: Fueling rate / EGR level / Intake pressure / Intake temperature.

2.2.6.2 Fuel Selection

For the short term scenario, fuels are selected that both reflect short term changes in diesel fuel composition and which are expected to be compatible with current conventional CI diesel engines. The European diesel fuel selected (EN590) reflects the current trend in Europe towards lower density, low-Sulphur, low poly-aromatic diesel fuels with relatively high cetane numbers and a lower final boiling point temperature. In the United States, until now, diesel fuels are characterized by a significantly lower cetane number. Testing such a fuel can give an idea of the effects of a somewhat lower CN and might show whether such a cheap, readily available distillate fuel is a good PCCI fuel or not. In 2009, the European Commission has decided in Directive 2009/28/EC that the share of energy from renewable sources in the transport sector must amount for at least 10% of final energy consumption in the sector by 2020. For diesel, the most likely scenario is therefore the increased blending of biodiesel with petroleum derived distillate diesel.

A second reason for selecting biodiesel is the fact that it is composed of a different kind of molecules than conventional distillate diesel. Research in the US by Oak Ridge National Laboratory [16] has shown that biodiesel demonstrated a different auto-ignition behaviour and a sharper pressure rise rate (following ignition) than would be expected from a low-aromatic distillate fuel with the same cetane number. Although the EC directive sets a minimum bio fuel content, but there is no upper limit on the maximum bio fuel percentage. Apart from that, of course any fuel blend should fulfil the fuel characteristics for diesel fuel as laid down in other regulations (such as EN590). Given that the base fuel already has about 5% percent RME, and in view of the above a test blend with an added RME content of 30% is used. With current diesel fuels, because of their high upper boiling point temperature, injection of fuel early in the compression stroke might result in wall-wetting, because due to the lower gas temperature at these timings vaporization is slower or incomplete. A comparison of the HC emissions from n-heptane with those from regular diesel should give an indication of the potential gains that could result from lowering the boiling point range of current diesel since its Cetane Number is similar. As n-heptane is often used as a single component fuel for diesel spray and combustion modelling, comparing these two fuels can tell if this assumption is valid for PCCI combustion.

2.2.7 Data Analysis and Definitions

When comparing emission levels and fuel consumption for heavy duty engines, 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 reproducibility of results very bad. Therefore in this case the IMEP as calculated from the in cylinder pressure signal is used [16]. Two definitions for IMEP can be used. The first one, referred to as net IMEP takes the integrated work for a full 720 degree combustion cycle, divided by the cylinder displacement. The difference between the net IMEP and the BMEP is defined as the Friction Mean Effective Pressure (FMEP). BMEP being lower than IMEP net, the latter gives a too optimistic picture. In the present test setup, where air pressure is generated by an external compressor and exhaust gas back pressure, it is hard to keep constant, as the net IMEP is directly influenced by both the intake and exhaust pressure. As the intake pressure is increased the net IMEP directly benefits from this. In a standalone HD engine, where an exhaust gas turbo charger is used to generate intake air pressure, the energy to charge this pressure comes from the hot exhaust gas, but with a penalty in exhaust gas back pressure. The gross IMEP differs from the net IMEP through ignoring these pumping losses (or gains) by only using the compression and expansion cycles, therewith enabling a more fair comparison of different parameters. Therefore in all results presented, the gross IMEP has been used to calculate indicated fuel consumption and emissions. At the relatively early injection timings and low loads under investigation, the crank angle at which 10% of the injected mass is burned, CA10, is used as the definition for start of combustion (SOC) because of its considerably higher stability compared to CA5 and CA2. Analysis of logged injector actuation current and injection pressure data show a constant 4 °CA lag between start of actuation (SOA) and start of injection (SOI). Given the assumptions made above the following definitions are used to characterize the combustion.

2.3 Effect of Compression Ratio on Partially Premixed Charge Compression Ignition Engine

The experiment was conducted in 4- stroke, single cylinder, Compression Ignition engine, modified into PCCI mode of operation as shown in Figure 6. The manifold injection method is used for premixing of fuel-air mixture. The fuel vaporiser is connected with intake manifold for vaporising the incoming fuel. The modified engine was tested at different compression ratio (19.5:1) in premixed charge compression ignition mode [6].

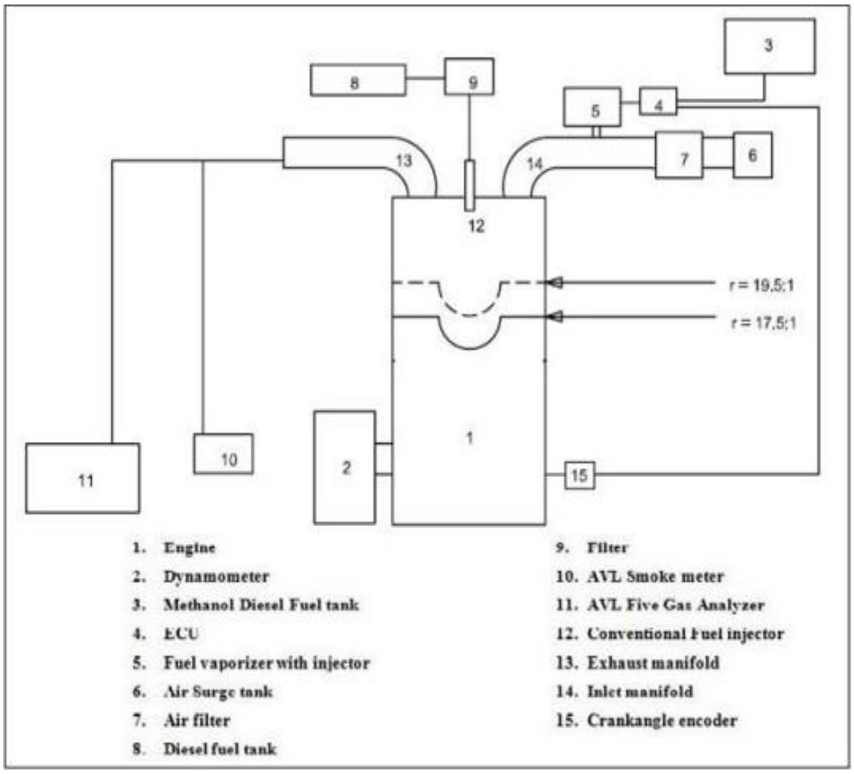

Fig. 6 - Schematic diagram of experimental setup[6]

The different piston geometry was achieved by decreasing the clearance volume of piston bowl in top of the piston. The methanol blends were injected in inlet manifold at suction stroke. The injection pressure and timing were controlled by Electronic Controlled Unit. For the preparation of homogeneous mixture condition, the air is supplied at high velocity and latent heat of vaporization absorbed by incoming fuel droplets [8]. The engine was initially run by diesel fuel for warm-up condition, then the vapour induction of methanol-diesel blends were switched to PCCI mode. The engine was completely operated with methanol-diesel vapour induction and readings were taken up to 75% load condition [17]. The PCCI engine was difficult to run at full load condition because of knocking, incomplete combustion can take place. The output readings of performance, combustion, and emission characteristics were measured using AVL Five Gas Analyzer, AVL smoke meter and AVL software systems. Figure 6 shows the schematic diagram of the experimental setup. The experiment are using the apparatus such as engine, dynamometer, methanol diesel fuel tank, KCU, fuel vaporizer with injector, air surge tank, air filter, diesel fuel tank, filter, AVL smoke meter, AVL five gas analyser, conventional fuel injector, exhaust manifold, inlet manifold, and crank angle encoder.

Figure 7 shows the value of the bore, stroke, compression ratio, nozzle opening pressure, injection timing, and rated speed.

Cooling system	Air Cooled
Stroke	110 mm
Bore	87.5 mm
Compression ratio	17.5:1, 19.5:1
Nozzle opening pressure	220 bar
Injection Timing	23 deg bTDC
Rated speed	1500 rpm

Fig. 7 - Engine specification[6]

3.0 Results and Discussion

3.1 Injection Timing to Reduce Emission in Premixed Control Compression Ignition

Injection timing is widely accepted parameter to control engine-out emissions. The present six different spray angle nozzles were investigated to evaluate the effects of injection timing on engine-out emissions in each case. The injection timings were swept from a timing as early as it could be achieved without damaging the engine to around TDC. The results shown in this section belong to the same data set explained previously.

3.1.1 Soot Emission

Soot emissions for the six different spray angle nozzles are indicated in Figure 8 with respect to the injection timing. Soot emissions in the PCCI combustion regime (i.e. when the SOI is earlier than -25 degree ATDC), are seen to depend strongly on the spray angle, in addition to the injection timing. It is hard to describe a trend regarding the influence of spray angle since each spray angle shows a different behaviour. For example, the 120 degree spray angle case shows peak soot emission at SOI -30 degrees ATDC, while the soot emissions for the 140 degree spray angle start to increase when the injection is retarded beyond that SOI. The 85 degree spray angle case does not show significant variation in soot emissions. The results imply that a new approach is needed to analyse the emissions in the PCCI regime. The combustion details and its products in the PCCI regime should be determined by the equivalence ratio distribution within the premixed mixture, which may not be directly linked to the injection timing. The equivalence ratio distribution results from an interaction between the spray and in-cylinder flow before the combustion starts. Thus, the results will be interpreted with respect to the spray targeting, which links the spray interaction with the injection timing and spray angles. On the other hand, the soot emissions in the standard diesel combustion regime increase with different slopes depending on the spray angles as the injection timing is retarded.

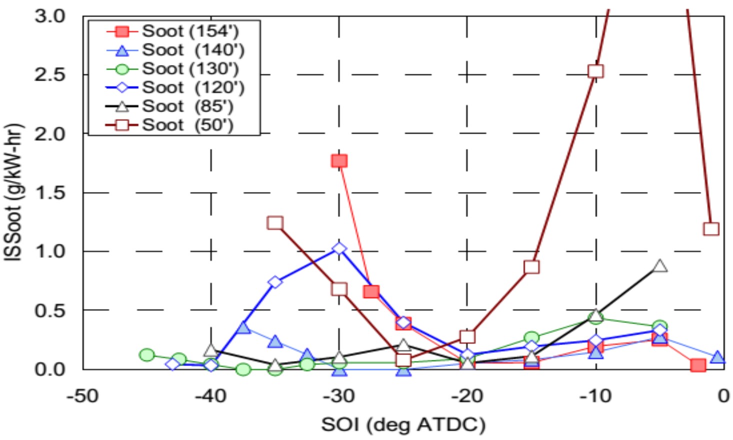

Fig. 8 - Soot emission with respect to injection timing [17]

The trend within the standard diesel combustion regime is consistent with the results of Siebers & Higgins [17] who investigated emissions in a constant volume chamber. According to their results the higher temperatures and pressures that accompany retarded injection result in shorter flame lift-off lengths and increased soot emissions due to the resulting increased equivalence ratios within the spray. The results for the 50 degree spray angle nozzle indicate much higher soot emissions than the others since the sprays for the 50 degree nozzle travel less distance before they collide with the piston bowl which is closer to the nozzle. Spray impingement near the nozzle exit prohibits spray plumes from entraining air by shortening the spray travel distance and those results in rich mixtures. The results are also consistent with the longer burn duration of the 50 degree spray angle, which indicates that less fuel is burnt in the premixed combustion regime due to lack of entrained air.

3.1.2 Spray Targeting to Minimize Emissions

As discussed in the previous section, the injection timing is not the only factor that explains the emissions in the PCCI combustion regime. When the fuel is injected into the cylinder, it reacts with the flow field and responds to the properties of the surrounding air. As far as the flow field is concerned, the injection timing is not a controlling parameter [18]. However, spray targeting is a key linkage between the injection timing and the controlling parameters like the flow field and air properties. Nevertheless, analysis of spray targeting is complicated since it requires the injection timing to be combined with the spray angle and the geometries of the piston and engine. As discussed previously, emissions in the standard diesel combustion regime depend strongly on the injection timing and only the 50-degree spray angle nozzle showed significant deviations from the others due to its extraordinarily short spray travel length.

However, the emissions in the PCCI combustion regime could not be explained by the injection timing alone since the sprays in PCCI combustion are believed to interact with the flow details, such as the swirl and squish flow due to the long ignition delays. Figure 9 show how the emissions in the PCCI combustion regime are influenced by the location where the corresponding sprays are targeted. Soot emissions for three selected three spray angles (50, 85, and 120 degrees) are plotted normal to the spray targeting point on the piston bowl surface to help reveal the effects of spray targeting on emissions. The three spray angle nozzles shown are three representative cases for which the sprays target the inner, bottom, and outer surface of the piston bowl. Sprays for the 50 degree nozzle hit the inner surface, while the sprays for the 85 and 120 degree cases impinge on the bottom and outer surfaces respectively. The relative magnitudes of the emissions are indicated by the distance perpendicular to the piston bowl surface, and the corresponding injection timings are shown at the locations where the centers of the spray plumes are aligned so as to hit at these timings. The spray targeting results reveal two “sweet spots” in Figure 9 when the soot emissions are considered. The soot emissions record much lower values when the sprays are targeted toward the bottom of the bowl and at the edge near the squish volume. The inner surface which the 50 degree spray angle nozzle targets shows poorer soot emissions. The side wall of the bowl shows poor soot except for when the spray targets near the edge. The emission trends in the standard diesel combustion regime indicated by the dotted lines are not explained by the spray targeting since in that case the emissions are determined by air entrainment in the spray and mixing processes after the ignition instead of by spray targeting and pre-ignition mixing. Though the differences regarding the targeting are very interesting, it is hard to explain how the targeting actually influences emissions.

According to Miles et al. [19], the combustion details and its products in cases of early injection like in the PCCI regime are affected strongly by the swirl and squish flows. It is a great challenge to predict the details of the spray, swirl flow, squish flow and their interactions for each specific cases. Spray targeting at the piston bowl edge (e.g., the optimum targeting for the 120 degree spray angle nozzle) is of special interest since it provides better carbon moNOxide emissions in addition to an excellent soot and NOx trade-off, and the location is very close to the squish flow. The squish flow heads toward the spray plumes and transports fuel-air mixture to the center of the piston bowl where the mixture rarely locates itself without the help of the squish flow. Consequently, the squish flow enhances the fuel-air mixing and prevents local rich spots from being formed by spreading the mixture out through the piston bowl. Once the spray is targeted above the top of the piston by advancing the fuel injection timing, the engine torque drops suddenly since the fuel hits the top of the piston and moves to the cylinder liner, leading to wasted fuel and diluted engine oil. These limit the use of advanced injection timing, especially for the cases of the 120 degree and higher spray included angles. Soot and emission for three spray angle nozzles (130, 140, 154 degrees) are compared with the 120 degree spray angle to verify the advantage of targeting at the bowl edge. The emissions show their optimum spots around the edge of the piston bowl as in the case of the 120 degree spray angle. However, the emissions for the other spray angles are much less sensitive to the spray targeting than the 120 degree spray angle case, and the optimum spots move down deeper into the piston. The optimum spots for the 140 and 154 degree spray angles are placed lower than the spots for the 120 and 130 spray angles. Two factors might cause the optimum spots to move down. First, the upper part of the spray plume periphery tends to be directed at the squish area on the top of the piston when the targeting approaches the bowl edge. The targeting shown in the plots indicates the location at which the centerline of the spray hits the piston bowl.

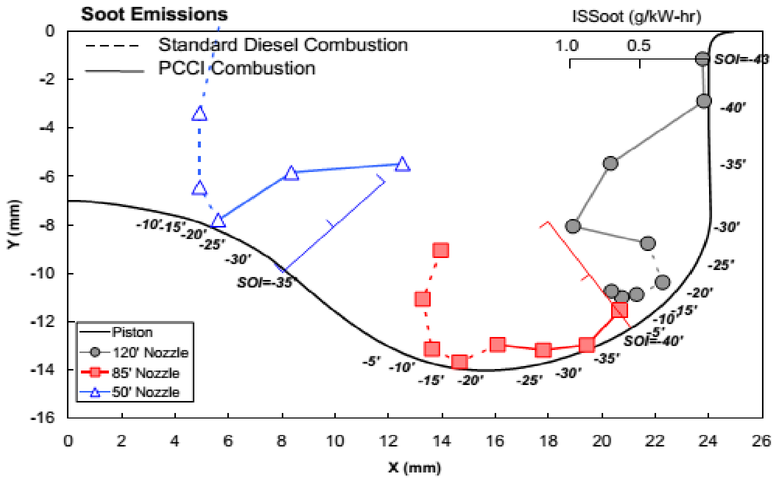

Fig. 9 - Effects of spray targeting on soot emission

A nozzle with a wider spray angle transfers more momentum into the radial direction and forces more fuel to flow into the squish region, where the fuel does not burn properly. Therefore, the radial momentum determines the upper boundary of the optimum spot. It is clear that the soot and emission with a wider spray angle start to increase at locations closer to the edge. Next, the axial component of momentum of the spray also affects the location of the optimum spot. More axial momentum transports more fuel to the bottom of the piston bowl which is far from the squish flow and gives less chance for the spray to interact with the squish flow. Therefore, the lower boundary of the optimum spot can be determined by the axial momentum and the optimum spots for a nozzle with narrower spray angle might be closer to the bowl edge. However, both theories are hard to prove with the results of this study alone, and are not sufficient to explain the big jumps of soot and emission seen with the 120-degree spray angle as the targeting moves down the bowl edge. There are complex interactions between the swirl flow, the squish flow, and the spray, as discussed in the previous section, and analysis of these flows is beyond the scope of this study. The spray targeting and squish flow interaction for nozzles with narrower spray angle than 120 degrees may suggest interesting results since they provide more room for advancing the injection timing, and consequently more mixing time which is favourable to PCCI combustion. However, the results may not be easily predicted since they are strongly affected by the spray targeting in PCCI combustion as discussed previously. The details will be explored in future studies.

3.2 Pressure and Heat Release with Dual-Fuel Stratified PCCI to Reduce Emission in Diesel Engine

3.2.1 EGR Level

Adding EGR is the classic route to reduce NOx emissions from conventional diesel combustion through charge dilution and lowering the adiabatic flame temperature. In the next subsection, first the impact on combustion phasing and performance is discussed.

3.2.2 Combustion Phasing and Performance

From Figure 10, where timing sweeps are shown for different EGR levels, the effect of EGR on combustion timing can be seen. As can be expected, a higher EGR level gives a longer ignition delay and slower combustion resulting in a significantly later CA50. One can also see that not for all timings and EGR levels the results are plotted. This is because at these combinations unacceptable combustion occurs in this case, because of too high pressure rise rates, as discussed in the experimental procedure. For a constant fuelling rate, the results are still under investigation. IMEP and ISFC, in [g/kWh] are directly coupled, as one can see from the following definition, where Qm fuel is the fuel mass flow in [g/h], and PIMEP, the power output in [kW] as computed from the IMEP.

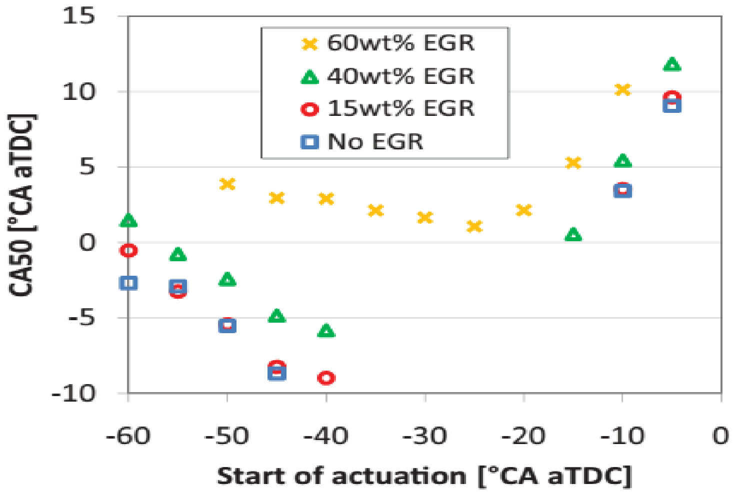

Fig. 10 - Impact of EGR on CA50 (0.53 g/s fuel≈4 bar IMEP, 1.25 pin) [20]

3.2.2.1 HC Emissions

In the CDC combustion regime, HC emissions are insignificant due to a high localized temperature during the combustion and the proximity of oxygen enabling their oxidation [20] even at low loads. In more premixed combustion mode, EGR allows the combustion process to be controlled by reducing the local temperature and by decreasing the oxygen partial pressure in the cylinder. When NOx and particle emissions are lowered, complete post oxidation of HC is then difficult to achieve. As discussed in Heywood [21], there are a number of possible sources of engine-out HC emissions. For gasoline SI engines, operating near stoichiometric equivalence ratios, trapping of fuel in crevice volumes is one of the major sources of HC emissions. For CI engines, this source is particularly of importance when early direct injection strategies are adopted, since in-cylinder pressures are relatively low and there is significant time available for fuel spray dispersion towards these crevices.

Another potential mechanism for HC emissions formation is through quenching. Quenching of oxidation reactions can occur by excessively low temperature zones by heat transfer between the in-cylinder charge and combustion chamber walls, or by natural thermal in-homogeneities as a result of mixing in the bulk gases. Over mixing to local equivalence ratios below the lean combustion limit of the mixture can also lead to regions which do not permit complete combustion on relevant engine time scales. This is mainly thought to occur at low loads, particularly for conditions where the ignition delays is long which allows for a long mixing time. Finally, wall wetting can be a source of HC emissions in direct injection engines. Depending on combustion chamber geometry and injection timing, excessive spray impingement on cylinder walls results in the formation of liquid films which may not allow complete evaporation and oxidation of the fuel. Any remaining unburned hydrocarbons within the combustion chamber formed as a result of incomplete oxidation reactions can subsequently be emitted into the exhaust gas. While HC emissions can consist of both uncombusted and partly oxidized fuel, CO emissions are always the results of partly combustion. This suggests that these CO emissions also result from the extended ignition delay encountered in low-temperature systems. This increased mixing time results in over mixed conditions, where combustion temperatures are too low for the full oxidation of CO to be completed on engine time scales.

For HC emissions, maximum cycle temperature is seen as a dominant and first order parameter. As emission levels for heavy duty applications are legislated in [g/kWh] unit, it is common to use this unit when comparing parameters. This unit can be seen to exist from the emission mass flow and the average power output. One should thus notice that for a constant emission mass flow, a decrease in power gives an increase in the indicated specific emission. Secondly, the emission mass flow can be defined from the volumetric content, as measured by the gas analyzer and the exhaust gas mass flow rate. When using EGR, a portion of this exhaust gas flow is recirculate to the intake. Therefore, the exhaust gas mass flow is defined as the fresh air mass flow plus the injected fuel mass flow. Given a constant pressure and temperature in the intake plenum, and thus an almost constant total mass flow rate. Increasing the EGR level will result in a decrease in fresh air mass flow. For a constant volumetric emission percentage, this gives a lower emission mass flow. The same effect occurs when decreasing intake pressure, where for the same volumetric emission percentage and output gross IMEP, the lower mass flow gives a lower emission in g/kWh. For the points under investigation, at a first glance it seems that for both the CDC and the PCCI regime an increase in EGR level leads to an increase in both HC and CO emissions for constant start of actuation, see Figure 11 in the CDC regime, where CA50 is close to the start of actuation and EID is thus short. It is indeed seen that more EGR increases HC emissions, although for a constant CA50 this effect is smaller than for constant SOA.

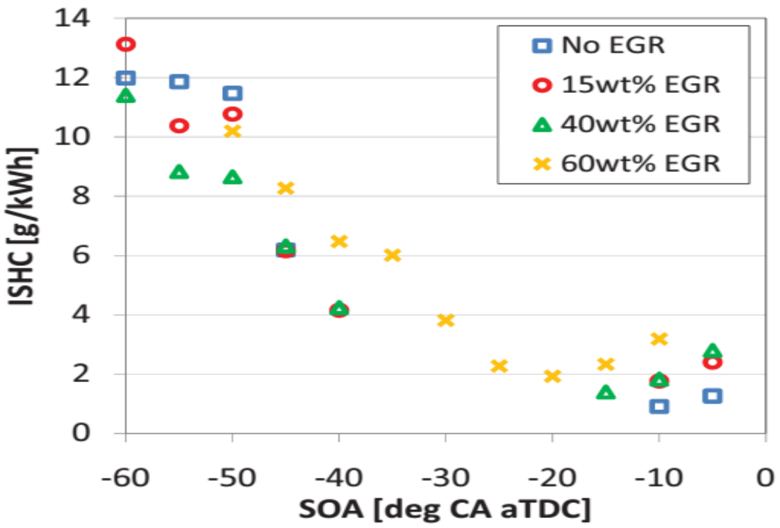

Fig. 11 - ISHC vs. SOA. Impact of EGR on HC emissions (0.53 g/s fuel≈4bar IMEP, 1.25bar Pin) [20]

3.2.2.2 NOx Emissions

As in other LTC concepts, EGR is used to lower the adiabatic flame temperature of the combustion. This brings NOx down, but with a penalty in UHC and CO emissions. Because of the higher equivalence ratio and lower combustion temperature, more soot is formed and less is combusted respectively. As with PCCI combustion the combustion temperature is also lowered by increased premixing, less EGR should be necessary to bring NOx down to acceptable levels. Yet, in the PCCI regime EGR is also used to lower combustion speed and thus limit the maximum pressure rise rate to acceptable levels. In the CDC regime, NOx levels in Figure 12 are seen to be effectively reduced by adding EGR as was expected. For longer ignition delays in the PCCI regime, the effect of EGR gets smaller, and as the charge gets more and more premixed, combustion temperatures are predominantly lowered by the leaner local mixture strength. For an even longer mixture time, when the mixture is near homogeneous, the direct effect of EGR is negligible and combustion temperatures are effectively lowered by the local equivalence ratio.

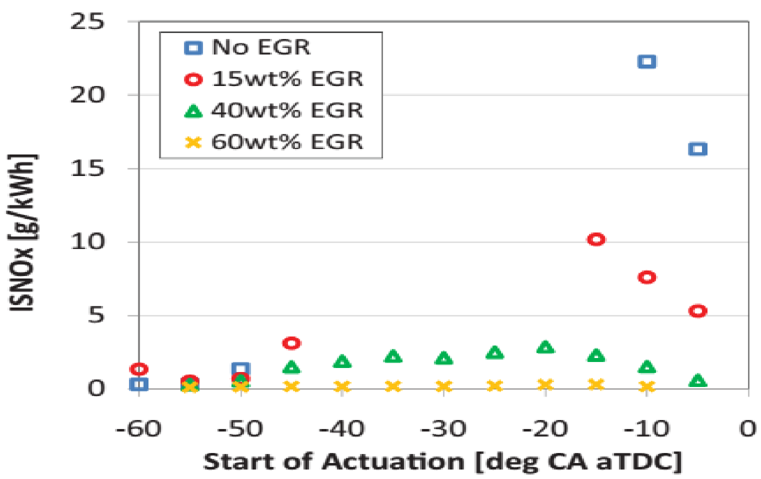

Fig. 12 - ISNOx vs. SOA. Impact of EGR on NOx emissions (0.53 g/s ≈ 4 bar IMEP, 1.05 bar Pin) 3.2.2.3 PM emissions [20]

For particulate emissions, in conventional diesel combustion, an increased EGR level lowers the equivalence ratio and therefore more soot is formed. As start of actuation is advanced into the PCCI regime, and mixture time is increased, no direct effect of EGR can be seen. Here the PM emission is predominantly caused by the longer mixing time and the lower combustion temperatures and soot oxidation rates associated therewith. When start of actuation is even further advanced, wall wetting is believed to occur and indicated specific PM emission increase. For all points under investigation the absolute PM levels are quite low, because of the relatively low loads under investigation and the injection pressure and equipment used. These soot levels are orders of magnitude lower than the CO and HC emissions and are therefore believed to originate from different mechanisms. Although these mechanisms, for instance wall wetting, can also have an effect on CO and HC emissions, however for those, they are not seen as dominant.

3.2.3 Intake Pressure

3.2.3.1 HC Emissions

A higher intake pressure seems to effectively reduce both HC emissions as shown in Figure 16, like a lower EGR level has shown to do in section 3.2.1. Figure 13 shows that HC emissions again correlate nearly perfectly linear correlation with EID. The higher intake pressure here only has a direct influence on this EID. This CO/HC trend with EID was also seen with EGR and was attributed to less complete combustion due to lower local temperature as a result of a more premixed charge. Following the same reasoning when investigating the influence of a higher charge pressure, the direct effect of a higher total charge mass and a leaner global mixture, one would expect even lower combustion temperatures, and thus higher HC emissions for the same mixing time. From these results, where a linear relation with EID is visible, it seems that the pressure does not have a direct effect, but indirectly through the elongated EID.

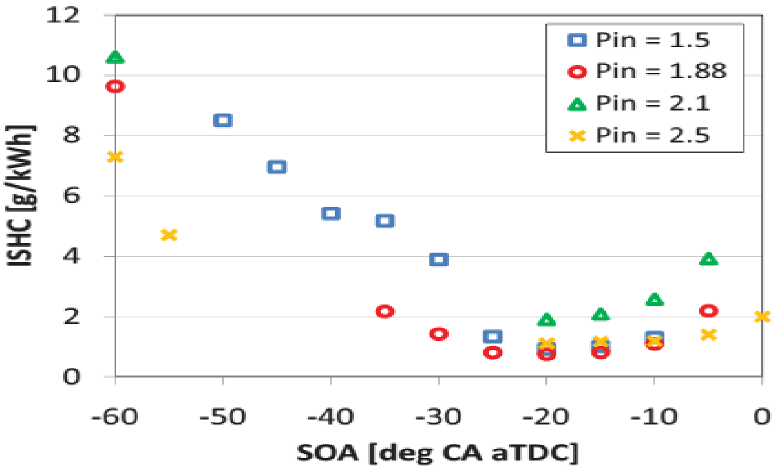

Fig. 13 - ISHC vs. SOA. Impact of intake pressure on HC emission (0.97 g/s fuel ≈ 8 bar IMEP, 55% EGR) [20]

3.2.4 Intake Temperature

A higher intake temperature largely shows the same effects as a higher intake pressure has done in section 3.2.2. Combustion is advanced significantly for the points under investigation, resulting in CA50 even before TDC, as in Figure 14, which results in unacceptably high pressure rise rates. Load and fuel consumption parameterized by IMEP and ISFC, are not directly affected through the higher intake temperature. Indirectly this is affected in two ways. Firstly, as the higher temperature advances the combustion to too early timings, this results in higher fuel consumption. For the same CA50 however, the higher temperature level has a lower fuel consumption caused by the shorter EID and more complete combustion associated with that. The principal motivation for elevating intake temperature has been reducing wall wetting.

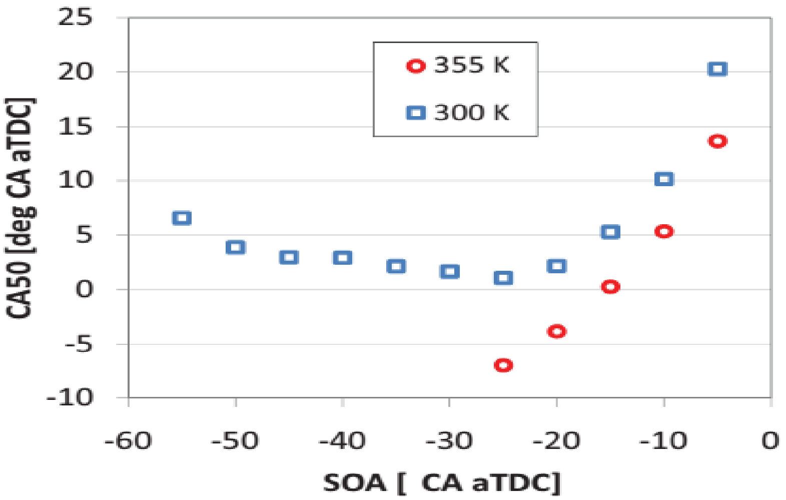

Fig. 14 - CA50 vs. SOA. Impact of intake pressure on combusting phasing and emissions (0.53 g/s fuel ≈ 4 bar IMEP, 1.25 bar Pin, 60% EGR) [20]

Again it seems that the varied parameter has a direct influence on HC and CO emissions. From the results, it can clearly be seen that CA50 is advanced by the higher starting temperature, and EID is thus shortened. HC and CO emissions benefit from the much shorter mixing time and higher combustion temperatures. For the points under investigation, no clear indication of (a reduction of) wall wetting can be seen and most CO and HC emission are thought to be formed by cold zones in the combustion chamber. If wall wetting is present, its contribution is expected to be insignificant compared to that of the cold zones. The slightly shorter EID, together with a higher starting temperature, has been seen to increase the NOx emission very significantly even at the high EGR rates used. It is therefore of great importance that the EGR used is cooled. Also for particulate emissions, elevating intake temperature shows the same trends as elevating intake pressure did. For very short EIDs in the CDC regime, no significant effect of the higher temperature can be seen. It can only be noted that for the lower temperature, the ignition delay, and thus mixing time is more easily increased. These significantly lower the PM emissions.

3.3 Effect of Compression Ratio on Partially Premixed Charge Compression Ignition Engine

Figure 15 shows the relationships between the unburned hydrocarbon and the percentage of load in the variation of unburned hydrocarbons. Load is the weight of the mixture and the Unburned Hydrocarbon (UHC) found in gas phase of diesel exhaust are a mixture of many hydrocarbon species derived from diesel fuel and from lubricating oil [22]. UHC is measured with the Flame Ionization Detectors (FID). The exhaust gas is burned in a carbon free flame, normally hydrogen–helium environment i.e. 40% hydrogen and 60% helium and fed between two electrodes with an applied voltage. The UHC in the exhaust results in ionized carbon and free electrons. The carbon ions are positive and are subsequently pulled to the negative electrode.

The resulting current is proportional to the number of carbon atoms in the UHC. The calibration of FID should be accomplished with a zero UHC gas for example nitrogen gas and a gas containing a known UHC concentration on a regular basis. In neat diesel as the percentage of load increase the unburned hydrocarbon decreases. Same also goes to the other type of diesels, that are the diesel PCCI and methanol-diesel PCCI. The reason why the unburned hydrocarbon decreases as the percentage of load increase, is because in the PCCI engine operation, the mixture of incoming fuel was fully premixed of fuel and air. The combustion can takes place by low temperature, and then it forms unburned hydrocarbons. The oxidation process cannot be completed at low temperature condition because in low temperature combustion the fuel might have burn incompletely. It was necessary to completely burn the fuel to reduce this emission. To reduce this unburned hydrocarbon emissions, the exhaust gas can be recycled.

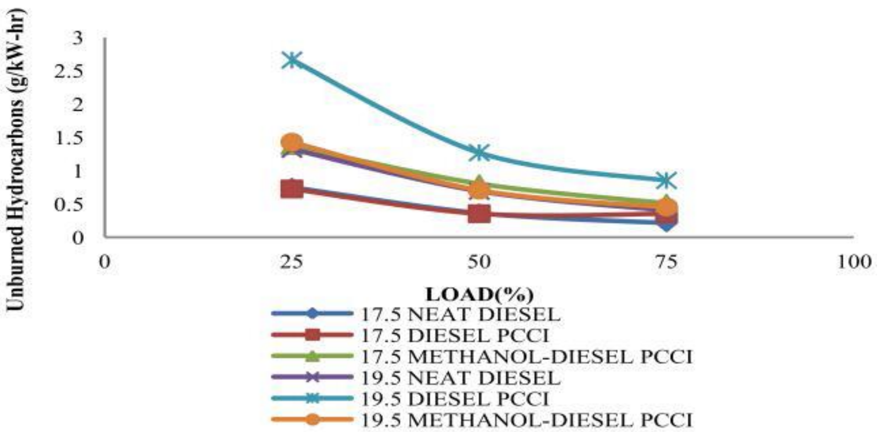

Fig. 15 - Variation of unburned hydrocarbons with load [22]

3. Conclusion

The experimental investigations show the injection timing to reduce NOx emission in premixed control compression ignition, the pressure and heat release with dual-fuel stratified PCCI to reduce emission in diesel engine and the effect of compression ratio on partially premixed charge compression ignition engine. Based on the results, the NOx and particulate matter produce low but high production of HC and CO in PCCI engine.

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