ABSTRACT

Early Direct Injection Premixed Charge Compression Ignition (EDI PCCI) is a widely researched combustion concept, which promises soot and CO$_2$ emission levels of a spark-ignition (SI) and compression-ignition (CI) engine, respectively. Application of this concept to a conventional CI engine using a conventional CI fuel faces a number of challenges. First, EDI has the intrinsic risk of wall-wetting, i.e. collision of fuel against the combustion chamber periphery. Second, engine operation in the EDI regime is difficult to control as auto-ignition timing is largely decoupled from fuel injection timing.

In dual-mode PCCI engines (i.e. conventional DI at high loads) wall-wetting should be prevented by selecting appropriate (most favorable) operating conditions (EGR level, intake temperature, injection timing-strategy etc.) rather than by redesign of the engine (combustion chamber shape, injector replacement etc.). This paper presents the effects of EGR concentration, intake temperature, intake pressure, injection timing, injection pressure and fuel temperature on engine performance and emission behavior in EDI PCCI mode. In addition, several minor adjustments to the conventional injector nozzle are investigated. Wall-wetting and engine performance are characterized by the measured emissions (smoke and unburned hydrocarbons) and in-cylinder pressure (CA50 and IMEP). The main contribution of this paper is to investigate the cumulative effects on engine performance and emissions, unburnt hydrocarbons (HC) in particular, of various known measures designed to address wall-wetting.

All experiments have been performed at low load (~ 3-4 bar IMEP) and at an engine speed of 1200 RPM, using a modified 6-cylinder 12.6 liter heavy-duty DI DAF XE 355 C engine. Experiments are conducted in one dedicated cylinder, which is equipped with a stand-alone fuel injection system, EGR circuit and air compressor, fuelled with commercial diesel fuel (EN590).

INTRODUCTION

A lot of research effort is being invested to develop new combustion concepts which capture the advantageous efficiency of the CI and low emissions of the SI engine [1-9]. A shared fundamental characteristic of many of these concepts is that a more or less premixed fuel/air charge is brought to auto-ignition. As a result, both fuel and (flame) temperature are well dispersed in the combustion chamber, ideally resulting in near-zero levels of both soot and NOx respectively. Collectively, such concepts are referred to as Premixed Charge Compression Ignition.

In order to achieve the desired degree of premixing, it is necessary to separate (in time) the injection and combustion event, at least to a large extent. In other words, the ignition delay should be longer than the injection duration.
In literature, various methods are proposed to stretch the ignition delay.

EARLY DIRECT INJECTION PCCI [1,2,3] – Early direct injection (EDI) PCCI involves a relatively early injection timing, typically when the piston has reached roughly two-thirds of its upward movement in the compression stroke. The prevailing temperature and pressure is at that time too low to initiate auto-ignition chemistry, but, at least to a certain extent, high enough to facilitate the evaporation process. Accordingly, time is created for mixing prior to auto-ignition in order to achieve PCCI combustion.

When utilizing stock HDDI hardware, it is difficult to achieve EDI PCCI, whilst maintaining low soot and unburned hydrocarbon (HC) emissions. This can largely be attributed to wall-wetting. Apart from poor emissions, there is also the tendency of the relatively reactive diesel fuel to prematurely (i.e. well before TDC) auto-ignite, resulting in poor thermal efficiency and the risk of knock. Lastly, the absence of a clear dependence of the combustion timing on injection timing makes controlling EDI PCCI particularly challenging.

EDI PCCI will most likely be used initially in dual-mode engines [1], i.e. in combination with conventional DI combustion at high load. Dual-mode operation will entail running the engine under EDI conditions at low loads. For cold start and higher loads (upper one-third to one-half of the operating range) conventional DI combustion takes over. Meeting legislated targets for soot and NOx is becoming increasingly reliant on sophisticated aftertreatment. Because these aftertreatment systems are most often catalytic in nature, they become ineffective when – at low engine loads or during cold start - exhaust temperatures remain below the catalyst light-off temperature. Conversely, at high load there is ample thermal energy to facilitate catalytic conversion of harmful emissions. EDI PCCI promises to be a viable short term solution for meeting legislated NOx and soot targets in the low engine load regime. A more extensive overview on the drawbacks of EDI, along with various approaches to (partially) overcome these, is presented in [1].

LATE DIRECT INJECTION PCCI [3,4,5,9] – Late direct injection (LDI) involves a relatively late injection, typically from around 10 crank angles before TDC (or later). This late timing is combined with high EGR rates (> 25 wt-%) to prolong the ignition delay. Compared to EDI PCCI, the ignition delay is now markedly shorter, but sufficiently long (i.e. longer than the injection duration) to avoid fuel-rich areas where soot is formed [3]. Meanwhile, NOx is kept low by means of heavy EGR and the aforementioned premixing. Drawbacks of this concept concern the high injection pressure required to keep injection durations short [3], necessary modifications to the combustion chamber, high HC emissions and limited operating range [3,5,9]. When injection is lasting into the expansion stroke, such combustion has been referred to as Modulated Kinetics (MK). Especially with MK an additional drawback is the need to accelerate mixing times (e.g. through modification of valve/piston geometry [5] and/or high swirl [9]).

FUEL INDUCED PCCI [6,7,8] – The above mentioned strategies necessary to achieve PCCI result in large part from the low volatility and high CN of present commercial diesel fuel. It is known from literature that a form of PCCI can be realized in unmodified HDDI diesel engines under more or less conventional operating conditions, when a less reactive fuel is used. Earlier work [6, 7] showed near-zero soot and NOx emissions from a HDDI diesel engine, in an operating point corresponding to highway cruising, using moderate EGR (< 25 wt-%) and conventional fuel injection equipment. The fuel was a blend of commercial diesel and a low reactive cyclic oxygenate (i.e. cyclohexanone). The favorable results were attributed to the long ignition delays incurred and resulting higher degree of premixing. Kalghatgi [8] performed an extensive study on fuel requirements for PCCI engines and reported that, for injection timings that went up towards TDC, volatile, low reactive fuels such as gasoline are more appropriate for this combustion concept than diesel fuel.

This paper initially focuses on individual measures for minimizing wall-wetting and optimizing IMEP in EDI PCCI mode at low load (~ 4 bar IMEP), performed with a conventional HDDI diesel fuel injection nozzle. Subsequently, these measures will be utilized simultaneously to ascertain the cumulative potential. In a third step, the additional benefits of smaller nozzle holes and a narrower cone angle are investigated. For all nozzles and with all measures utilized, the injection timing is optimized.

EXPERIMENTAL APPARATUS

A dedicated engine test rig, referred to as CYCLOPS, has been designed and built at the TU/e. The engine is a 6-cylinder 12.6 liter DI DAF XE 355 C heavy-duty Diesel engine. However, several modifications have been made to allow for investigation of PCCI combustion.

CONCEPT – Essentially, the flywheel side of the DAF engine, comprising cylinders 1 thru 3, has been left in its original state. Operating under the stock DAF engine control unit (ECU), this flank is merely exploited as a means to control the crankshaft rotational speed of the test cylinder (cylinder 6) under both firing and motored operation (Figure 1). Engine speed and torque are controlled by a water-cooled, eddy-current Schenck W450 dynamometer.

Throughout engine start, warm-up, and whenever data-acquisition is idle, the CYCLOPS runs solely on the three propelling cylinders. Once warmed up and operating at the desired engine speed, combustion phenomena and emission formation are studied in the test cylinder 6. The test cylinder has been designed to operate independently.
from the propulsion cylinders with respect to all operating conditions except engine speed. To this end, dedicated air, EGR and fuel delivery circuits have been installed as will be explained below. Note that the compression ratio has been lowered from the stock value of 16 to 11.88 by means of a 3.4 mm thicker head gasket. At the stock value for the compression ratio, the auto-ignition process occurs prematurely, far ahead of TDC, resulting in poor thermal efficiency [1]. Lowering the compression ratio by means of a thicker gasket will alter the chamber characteristics, e.g. spray interaction with bowl and squish flow. These effects, though not investigated, are not believed to significantly influence the qualitative response of the engine to changing operating conditions.

AIR – Fed by an Atlas Copco air compressor, the intake air pressure can be boosted to pressures up to 7 bar. For this application, however, the limit is set at 5 bar to prevent the end-of-compression pressure of the test cylinder from reaching the maximum allowable in-cylinder pressure. A surge tank has been mounted downstream of the compressor to muffle any pressure oscillations originating from the compressor. The intake pressure is regulated by a PMA KS 40-1 pressure controller, which receives its input signal from a pressure sensor mounted in the intake manifold of the test cylinder. The set point can be programmed from the engine control room.

EGR – Cylinders 4 thru 6 constitute the considerably more complex, non-stock section of the engine (Figure 1). Combustion phenomena and emission formation will be studied exclusively in the test cylinder. Non-firing cylinders 4 and 5 function as EGR pumps, the purpose of this intricate system is to decouple EGR production from test cylinder back pressure and to dampen pressure fluctuations originating from the test cylinder.

Table 1 CYCLOPS specifications

<table>
<thead>
<tr>
<th>Geometric engine data</th>
</tr>
</thead>
<tbody>
<tr>
<td>Base engine</td>
</tr>
<tr>
<td>Cylinders</td>
</tr>
<tr>
<td>Bore [mm]</td>
</tr>
<tr>
<td>Stroke [mm]</td>
</tr>
<tr>
<td>Compression ratio [-]</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Valve data</th>
</tr>
</thead>
<tbody>
<tr>
<td>Intake valve open</td>
</tr>
<tr>
<td>Intake valve close</td>
</tr>
<tr>
<td>Exhaust valve open</td>
</tr>
<tr>
<td>Exhaust valve close</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Fuel injection system</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type</td>
</tr>
<tr>
<td>No. of holes</td>
</tr>
<tr>
<td>Nom. hole diameter [mm]</td>
</tr>
<tr>
<td>Cone angle [degrees]</td>
</tr>
</tbody>
</table>

Apart from the mutual cam- and crankshaft, the left engine bank operates autonomously from the right flank. The stand-alone air intake, EGR circuit, and fuel injection equipment have been optimized for maximum flexibility as will be discussed in detail in paragraphs to come. Note that several surge tanks and pressure relief valves have been included in the design, to dampen pressure oscillations and to guard for excessive pressure in the EGR circuit, respectively. Note that the catalytic particle filter (cDPF) in Figure 1 has been removed from the setup in order to supply untreated EGR gas into the test cylinder for the experiments presented in this work.
FUEL INJECTION SYSTEM – The selected double-acting air-driven Resato HPU200-625-2 fuel pump can deliver a fuel pressure up to 4000 bar. Fuel pressure, herein, is regulated by adjusting the air pressure with a Resato R03-04C high precision pressure regulator. To dampen the pressure fluctuations originating from the pump, a Resato HPA-1 accumulator is mounted downstream of the pump. Corresponding values for the accumulator internal volume and maximum allowable pressure are 1 dm³ and 4000 bar, respectively.

Additionally, a second temperature-controlled (up to 250 °C) buffer is placed near (~ 0.2 m) the fuel injector to neutralize any pressure waves present in the relatively long (~ 2 m) fuel line connecting the pump with the injector and to heat up the fuel to the desired temperature. Manufactured in-house, this accumulator can sustain similar pressures as the HPA-1. The internal volume of approximately 0.114 dm³ is significantly smaller to mimic the volume of a typical common rail. Properties of the utilized commercial diesel fuel are listed in Table 2.

EMISSIONS – To determine fuel, intake air and EGR flow rates, Micro Motion ® mass flow meters were used. For measuring gaseous exhaust emissions a Horiba ® MEXA 7100 wet HC and heated dry NOX analyzer were used. Exhaust smoke level (in Filter Smoke Number or FSN units) was measured using an AVL® 415S smoke-meter. Note that all experiments presented in this paper were conducted without any form of aftertreatment. The accuracy of this equipment is discussed in [6].

CRANK ANGLE RESOLVED DATA ACQUISITION – In-cylinder pressure was measured at 0.1 °CA intervals with the SMETEC system. The accuracy of this equipment is discussed in [1].

TIME RESOLVED DATA ACQUISITION – Quasi-steady engine data (i.e. in a given work point), such as air/EGR/fuel flows, intake/exhaust pressures/temperatures and emissions were recorded at 20 Hz for a period of 40 s via an in-house data-acquisition system (TUeDACS). The accuracy of this equipment is discussed in [1].

EXPERIMENTAL PROCEDURE

MEASURES UNDER INVESTIGATION – In literature [1,2], various measures are described to minimize wall-wetting and optimize IMEP in EDI PCCI mode at low load (~ 4 bar IMEP), most of which will be investigated here. In this work, a division will be made into “software”-based strategies, involving tuning of engine operating conditions:

- optimizing injection timing;
- raising intake pressure;
- raising fuel injection pressure;
- raising intake temperature (via hot EGR [1]);
- raising fuel temperature;

and “hardware”-bases strategies, involving modifications of the engine hardware:

- decreasing nozzle hole diameter;
- decreasing nozzle hole cone angle.

ENGINE OPERATING CONDITIONS – All measures presented above will be studied by means of parameter sweeps at 1200 RPM and at a load/IMEP of approximately 4 bar. The measuring plan consists of three main parts. In each part, fuelling is set such that IMEP is 4 bar at a SOA of -15 ° CA aTDC and zero EGR. Fuelling is subsequently kept constant throughout the performed sweeps. In the fourth part, Siebers’ engineering correlation for liquid core penetration [11,13] is used to estimate the degree of wall-wetting in the various cases.

- Part 1: Individual assessment of “software”-strategies (see Table 3)
- Part 2: Cumulative assessment of “software”-strategies (see Table 4)
- Part 3: Part 2 repeated for two modified nozzles (see Table 5)
- Part 4: Spray modeling to estimate effects of above measures on wall-wetting

Table 2 Fuel (EN590) properties

<table>
<thead>
<tr>
<th>Parameter</th>
<th>ASTM Test Method</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density (kg/l) @15°C</td>
<td>D 4052</td>
<td>0.8272</td>
</tr>
<tr>
<td>Cetane Index</td>
<td>D 4737</td>
<td>56.2</td>
</tr>
<tr>
<td>Viscosity (mm²/s) @ 40°C</td>
<td>D 445</td>
<td>2.69</td>
</tr>
<tr>
<td>Distillation</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Recovered at 250 °C %(v/v)</td>
<td>D 86</td>
<td>30</td>
</tr>
<tr>
<td>Recovered at 350 °C %(v/v)</td>
<td>D 86</td>
<td>98</td>
</tr>
<tr>
<td>95 % Recovered (°C)</td>
<td>D 86</td>
<td>335</td>
</tr>
<tr>
<td>Sulphur (mg/kg)</td>
<td>D 2622</td>
<td>35</td>
</tr>
<tr>
<td>Flash point (°C)</td>
<td>D 93</td>
<td>75.5</td>
</tr>
<tr>
<td>Poly-aromatics %(m/m)</td>
<td>391 (IP Test)</td>
<td>&lt; 11</td>
</tr>
<tr>
<td>Lower heating value (MJ/kg)</td>
<td></td>
<td>43.3</td>
</tr>
</tbody>
</table>

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Author:Gilligan-SID:13440-GUID:37752499-131.155.56.78
Table 3  Operating conditions for Part 1: Investigation of “software measures” (individual). Black boxes indicate the applied range or values used in the various parameter sweeps.

<table>
<thead>
<tr>
<th>SOA</th>
<th>Intake temperature °CA aTDC</th>
<th>Intake pressure bar (abs.)</th>
<th>Injection pressure bar (abs.)</th>
<th>Fuel temperature °C</th>
<th>EGR wt-%</th>
</tr>
</thead>
<tbody>
<tr>
<td>Baseline injection timing sweep</td>
<td>-5:5:-60</td>
<td>303</td>
<td>1.25</td>
<td>750</td>
<td>30</td>
</tr>
<tr>
<td>Intake temperature</td>
<td>-50</td>
<td>303</td>
<td>1.25</td>
<td>750</td>
<td>30</td>
</tr>
<tr>
<td>Intake pressure</td>
<td>-50</td>
<td>303</td>
<td>1.25</td>
<td>750</td>
<td>30</td>
</tr>
<tr>
<td>Injection pressure</td>
<td>-50</td>
<td>303</td>
<td>1.5</td>
<td>750, 1100, 1500</td>
<td>30</td>
</tr>
<tr>
<td>Fuel temperature</td>
<td>-50:5:-65</td>
<td>303</td>
<td>1.5</td>
<td>1100</td>
<td>30, 100</td>
</tr>
</tbody>
</table>

Table 4  Operating conditions for Part 2: Investigation of “software measures” (cumulative)

| Nozzle 1 (normal) Ø=190 μm θ=154° | -50, -30, -15 | 353 (max) | 1.25 | 750 | 100 (max) | 70 |

Table 5  Operating conditions for Part 3: Investigation of “hardware measures”

| Nozzle 2 Ø=161 μm θ=150° | -50, -30, -15 | 353 (max) | 1.25 | 1600 (max) | 100 (max) | 70 |
| Nozzle 3 Ø=161 μm θ=100° | -50, -30, -15 | 353 (max) | 1.25 | 1600 (max) | 100 (max) | 75 |

RESULTS & DISCUSSION - PART 1 OF 4

BASELINE INJECTION TIMING SWEEP – As indicated in Table 3, the timing of injector actuation is advanced from -5:5:-60 °CA aTDC at two levels of EGR. It should be noted that, at a given engine speed, analysis of logged injector current and injection pressure data shows a constant 3 °CA lag between the SOA and SOI. Plotted in Figure 2 thru Figure 7, is the response of combustion phasing, engine performance and emissions. Note that the baseline experiments were carried out with nozzle 2 (Table 5) instead of nozzle 1 (Table 4), because the latter nozzle was not available yet. Note that in this study all parameters related to engine power (i.e. ISFC, NOx and HC) are indicated values and not brake-specific.

Combustion Phasing

As could be expected, CA50 is initially seen to advance with SOA up to approximately 30 °CA aTDC, from which point a further advancement of SOA has a neutral effect on CA50. The change in slope around 30 °CA aTDC marks the border between conventional DI diesel and EDI PCCI combustion. A further advancement of SOA, deeper into the EDI PCCI regime, appears to lead to a retardation of CA50. The effect of EGR on CA50 is advancing in the EDI PCCI regime compared to the (expected) retarding effect in the conventional DI combustion mode. The intake temperature for the 40 % EGR case was slightly higher at 40-45 °C. These results suggest that the advancing effect of a higher intake temperature is dominant over the retarding effect of a lower oxygen concentration when operating in EDI PCCI mode. The opposite appears to be the case for conventional DI combustion. This behavior correlates well with the observations made in an earlier work [1].

![Figure 2 Baseline injection timing sweep: CA50](image)

1 Carried out with Nozzle 2 (Table 5)
Performance

At a given fuelling level, engine performance is characterized by IMEP (Figure 3) and ISFC (Figure 4). Both parameters are coupled and show similar trends with advancing SOA. Both peak around -15 °CA, indicating an optimum performance, corresponding to a CA50 around TDC. Later or earlier combustion phasing has a negative effect on thermal efficiency. A slight stabilization, however, is visible when entering the EDI PCCI regime. Here fuel consumption and engine power initially remain unaffected by an advancing SOA. At earlier timings still, both eventually start to deteriorate again.

Emissions

It is clear that the impact of injection timing and EGR on emission behavior is more complex. Starting with NOx (Figure 5), the characteristic rise within the conventional DI regime is clearly visible.

As could be expected, the lower flame temperatures intrinsic to the utilization of EGR, lead to markedly lower NOx, especially at conventional timings.
Both HC (Figure 6) and smoke (Figure 7) reach minimum values around the same SOA where peak values are registered for NOx. Another trend of interest is the flattening out and even reversal for the zero EGR case of smoke emissions at very early timings, which has been observed elsewhere [10] as well. Decrease of soot forming with increase of EGR is explained in [10] by the reduced combustion temperature of the rich mixture formed on the fuel wall-film.

Baseline

Based on the results of the baseline experiments, two workpoints (Table 6) are selected to serve as benchmarks for conventional HDDI diesel and EDI PCCI combustion. The motivation for the selected conventional and EDI PCCI workpoints is based on minimal fuel consumption and acceptable NOx (i.e. ~ EPA 2010 (0.27 g/kWh) and < EURO VI (0.4 g/kWh)), respectively.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Unit</th>
<th>Normal DI</th>
<th>EDI PCCI</th>
</tr>
</thead>
<tbody>
<tr>
<td>SOA</td>
<td>°CA aTDC</td>
<td>-15</td>
<td>-55</td>
</tr>
<tr>
<td>EGR</td>
<td>wt-%</td>
<td>0</td>
<td>40</td>
</tr>
<tr>
<td>Pfuel</td>
<td>bar</td>
<td>750</td>
<td>750</td>
</tr>
<tr>
<td>λ</td>
<td></td>
<td>4.2</td>
<td>3.1</td>
</tr>
<tr>
<td>CA50</td>
<td>°CA aTDC</td>
<td>-0.7</td>
<td>-6.7 (-6)</td>
</tr>
<tr>
<td>IMEP</td>
<td>bar</td>
<td>4</td>
<td>3.3 (-18 %)</td>
</tr>
<tr>
<td>ISFC</td>
<td>g/kWh</td>
<td>214</td>
<td>260 (+21 %)</td>
</tr>
<tr>
<td>NOx</td>
<td>g/kWh</td>
<td>15.1</td>
<td>0.20 (-99 %)</td>
</tr>
<tr>
<td>HC</td>
<td>g/kWh</td>
<td>0.12</td>
<td>4.76 (+3867 %)</td>
</tr>
<tr>
<td>Smoke</td>
<td>FSN</td>
<td>0.03</td>
<td>1.44 (+4700 %)</td>
</tr>
</tbody>
</table>

Table 6 Normal DI and EDI PCCI baseline conditions (using nozzle 2), performance and emissions. Values in brackets are relative to the normal baseline case.

It is clear that the transition from conventional DI to EDI PCCI comes paired with substantial reduction in NOx, but also leads to an unacceptable increase in soot/HC emissions and fuel consumption. The latter increases are likely due to wall-wetting and premature auto-ignition, respectively. The goal of the measures presented in the following subsections is to improve smoke/HC emissions and fuel economy, whilst maintaining low NOx.

INTAKE TEMPERATURE – Intake temperature (Tin), as measured in the intake manifold, was gradually increased by adding more uncooled EGR to the intake. Due to the intricate system of surge tanks (Figure 1), it is quite time consuming to realize an increase in Tin from 300 to 350 K. The EGR ratio was initially set to approximately 60-65 wt-% and subsequently raised further to 65-70 wt-% in order to reach the desired maximum Tin of 340-350 K. Measurements were performed per 5 K Tin increase. It should be noted that, contrary to the baseline measurements, these experiments were carried out with nozzle 1 (see footnote Table 3). Listed in (Table 7) are the results related to combustion phasing, performance and emissions at the minimum and maximum values for Tin. A complete overview of all data points is presented in Appendix A. Values in brackets are relative to the Low Tin case to single out the influence of the measure under investigation.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Unit</th>
<th>Low Tin</th>
<th>High Tin</th>
</tr>
</thead>
<tbody>
<tr>
<td>EGR</td>
<td>wt-%</td>
<td>60</td>
<td>70</td>
</tr>
<tr>
<td>Tin</td>
<td>K</td>
<td>300</td>
<td>350</td>
</tr>
<tr>
<td>λ</td>
<td></td>
<td>2.6</td>
<td>1.9</td>
</tr>
<tr>
<td>CA50</td>
<td>°CA aTDC</td>
<td>-1</td>
<td>-7.6 (-6.6 °CA)</td>
</tr>
<tr>
<td>IMEP</td>
<td>bar</td>
<td>3.06</td>
<td>3.14 (+3 %)</td>
</tr>
<tr>
<td>ISFC</td>
<td>g/kWh</td>
<td>296</td>
<td>284 (-4 %)</td>
</tr>
<tr>
<td>NOx</td>
<td>g/kWh</td>
<td>0.04</td>
<td>0.27 (+575 %)</td>
</tr>
<tr>
<td>HC</td>
<td>g/kWh</td>
<td>7.5</td>
<td>5.2 (-31 %)</td>
</tr>
<tr>
<td>Smoke</td>
<td>FSN</td>
<td>1.32</td>
<td>0.32 (-71 %)</td>
</tr>
</tbody>
</table>

The impact on combustion phasing is quite clear. Even though the increase in EGR leads to a lower oxygen concentration (λ), the corresponding increase in Tin leads to advancement in CA50. The dominance of temperature over oxygen concentration on the CA50 in the EDI PCCI regime has been observed in earlier work [1] as well. This effect is believed to be even more noticeable as λ→1. As discussed earlier, the principal motivation for elevating Tin is to reduce wall-wetting. From Table 7 becomes clear that a hotter Tin leads to improved IMEP and ISFC, a less favorable CA50 notwithstanding. HC and smoke emissions are improved as well. These trends, which are in agreement with earlier work [1], support the assumption that a higher Tin leads to less wall-wetting. NOx increases rapidly with an elevation in intake temperature, even though this increase is realized by an increase in EGR. Like CA50, NOx appears to be more sensitive to temperature than the prevailing oxygen concentration. It should be noted that although the NOx is substantially higher for the high Tin case, the value is still very low (i.e. ~ EPA 2010 (0.27 g/kWh) and < EURO VI (0.4 g/kWh)).

INTAKE PRESSURE – Intake pressure (Pin) was incrementally increased via the external Atlas Copco compressor (Figure 1) from 1 to 2 bar (absolute). In accordance with Table 3, Tin and the EGR ratio were kept constant at 300 K and 60 wt-% respectively. Listed in Table 8 are the results related to combustion phasing, performance and emissions at the minimum and maximum values for Pin. A complete overview of all data points is presented in Appendix B. Values in brackets are
relative to the Low Pin case to single out the influence of the measure under investigation.

**Table 8** Impact of Pin on performance and emissions in the EDI PCCI regime (see Appendix B).

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Unit</th>
<th>Low Pin</th>
<th>High Pin</th>
</tr>
</thead>
<tbody>
<tr>
<td>EGR</td>
<td>wt-%</td>
<td>62</td>
<td>55</td>
</tr>
<tr>
<td>Pin</td>
<td>bar</td>
<td>1.0</td>
<td>2.0</td>
</tr>
<tr>
<td>λ</td>
<td>-</td>
<td>2.0</td>
<td>3.8</td>
</tr>
<tr>
<td>CA50</td>
<td>°CA aTDC</td>
<td>3.2</td>
<td>-4.8 (+9.8 °CA)</td>
</tr>
<tr>
<td>IMEP</td>
<td>bar</td>
<td>2.76</td>
<td>3.72 (+35 %)</td>
</tr>
<tr>
<td>ISFC</td>
<td>g/kWh</td>
<td>342</td>
<td>248 (-27 %)</td>
</tr>
<tr>
<td>NOx</td>
<td>g/kWh</td>
<td>0.03</td>
<td>0.07 (+133 %)</td>
</tr>
<tr>
<td>HC</td>
<td>g/kWh</td>
<td>9.1</td>
<td>6.5 (-29 %)</td>
</tr>
<tr>
<td>Smoke</td>
<td>FSN</td>
<td>1.6</td>
<td>0.52 (-68 %)</td>
</tr>
</tbody>
</table>

It is evident that an increase in Pin leads to a noticeable advancement of the CA50. Moreover, all performance indicators and emissions have a clear benefit from the leaner mixture, except for NOx. These results are likely attributable to reduced wall-wetting (due to the higher gas density) and improved combustion efficiency (due to the leaner mixture). NOx emissions are higher, but still considered extremely low. It should be noted that exhaust back pressure was kept constant at near-atmospheric pressure regardless of the utilized boost pressure. In reality, the IMEP and ISFC will worsen somewhat as a consequence of the resulting higher back pressure when supplying the higher Pin via a turbocharger.

**Table 9** Impact of Pfuel on performance and emissions in the EDI PCCI regime (see Appendix C).

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Unit</th>
<th>Low Pfuel</th>
<th>High Pfuel</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pfuel</td>
<td>bar</td>
<td>750</td>
<td>1500</td>
</tr>
<tr>
<td>CA50</td>
<td>°CA aTDC</td>
<td>-5.1</td>
<td>-5.5 (-0.4 °CA)</td>
</tr>
<tr>
<td>IMEP</td>
<td>bar</td>
<td>3.00</td>
<td>3.23 (+8 %)</td>
</tr>
<tr>
<td>ISFC</td>
<td>g/kWh</td>
<td>280</td>
<td>240 (-17 %)</td>
</tr>
<tr>
<td>NOx</td>
<td>g/kWh</td>
<td>&lt;0.01</td>
<td>&lt;0.01</td>
</tr>
<tr>
<td>HC</td>
<td>g/kWh</td>
<td>9.1</td>
<td>7.3 (-20 %)</td>
</tr>
<tr>
<td>Smoke</td>
<td>FSN</td>
<td>1.17</td>
<td>0.9 (-23 %)</td>
</tr>
</tbody>
</table>

**FUEL INJECTION PRESSURE** – Fuel injection pressure (Pfuel) was incrementally increased via the external Resato air-driven fuel pump (Figure 1) from 750 to 1500 bar (absolute). In accordance with Table 3, Pin, Tin and the EGR ratio were kept constant at 1.5 bar, 300 K and 0 wt-% respectively. Listed in Table 9 are the results related to combustion phasing, performance and emissions at the minimum and maximum values for Pfuel. Values in brackets are relative to the Low Pfuel case to single out the influence of the measure under investigation. A complete overview of all data points is presented in Appendix C. An increase in Pfuel from 750 to 1500 bar results in a slight advancement of the CA50. This effect on CA50 in EDI PCCI mode, likely attributable to improved mixing, is also found in literature [2]. It is clear that IMEP and ISFC benefit from the higher fuel pressure, although this is not straightforward when considering the advanced CA50. It is assumed here that the higher pressure, and expected improved mixing, leads to less wall-wetting, and therefore improved combustion efficiency. This is supported by the fact that both smoke and HC emissions show a large improvement at the higher fuel pressure. Reduced smoke emissions in the EDI PCCI regime as a result of higher Pfuel have been reported by others as well [2].

**Table 10** Impact of Tfuel on performance and emissions in the EDI PCCI regime (see Appendix D).

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Unit</th>
<th>Low Tfuel</th>
<th>High Tfuel</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tfuel</td>
<td>K</td>
<td>303</td>
<td>373</td>
</tr>
<tr>
<td>IMEP</td>
<td>bar</td>
<td>3.00</td>
<td>3.00 (-)</td>
</tr>
<tr>
<td>CA50</td>
<td>°CA aTDC</td>
<td>-5.2</td>
<td>-5.7 (-0.5 °CA)</td>
</tr>
<tr>
<td>ISFC</td>
<td>g/kWh</td>
<td>265</td>
<td>251 (-5 %)</td>
</tr>
<tr>
<td>NOx</td>
<td>g/kWh</td>
<td>0.00</td>
<td>0.00 (-)</td>
</tr>
<tr>
<td>HC</td>
<td>g/kWh</td>
<td>9.4</td>
<td>7.5 (-20 %)</td>
</tr>
<tr>
<td>Smoke</td>
<td>FSN</td>
<td>1.02</td>
<td>0.59 (-42 %)</td>
</tr>
</tbody>
</table>

The results indicate that both fuel economy and emissions improve when the fuel temperature is raised in the EDI PCCI regime. The advanced CA50, improved
ISFC and considerably lower HC and smoke emissions support the hypothesis that a higher fuel temperature can reduce wall-wetting. This is particularly evident at very early timings for the SOA (Appendix D).

SUMMARY – In this Section, the impact of injection timing, intake temperature, intake pressure, fuel pressure and fuel temperature on engine performance and emission formation has been investigated for EDI PCCI operating conditions. In Table 11, a ranking is presented of these “software” measures with respect to engine performance and emissions. Note that this ranking is purely indicative as some boundary conditions were different for the various measures. It is interesting to note that all measures, while having a neutral to positive effect on performance, smoke and HC emissions, have a neutral to negative effect on NOx. This is in line with the well-known Diesel Dilemma, which typically holds for conventional HDDI diesel combustion. It is likely that this dilemma holds for EDI PCCI combustion as well when wall-wetting occurs. A penalty in NOx in the EDI PCCI regime, however, is acceptable as the values are typically far below legislated targets.

<table>
<thead>
<tr>
<th>CA50</th>
<th>IMEP</th>
<th>ISFC</th>
<th>NOx</th>
<th>HC</th>
<th>Smoke</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tin↑</td>
<td>←</td>
<td>0</td>
<td>0</td>
<td>---</td>
<td>+++</td>
</tr>
<tr>
<td>Pin↑</td>
<td>←</td>
<td>+++</td>
<td>+++</td>
<td>---</td>
<td>+++</td>
</tr>
<tr>
<td>Pfuel↑</td>
<td>←</td>
<td>+</td>
<td>++</td>
<td>0</td>
<td>++</td>
</tr>
<tr>
<td>Tfuel↑</td>
<td>←</td>
<td>0</td>
<td>+</td>
<td>0</td>
<td>++</td>
</tr>
</tbody>
</table>

← = advancement  | → = retardation
± 0-5 % = 0   | ± 5-10 % = ± | ± 10-25 % = ± ± ±
+ > 25 % = ± ± ±

NB a “+” symbol corresponds to an improvement in …

Another general observation is that all measures lead to an advancement of the CA50. This could be expected given the fact that the measures under investigation are aimed at improving the mixing process. Of all measures, boosting the intake pressure appears to yield the best results, although these will likely worsen somewhat when boosting is achieved via an engine-driven compressor or turbocharger, instead of via an external compressor (Figure 1). A similar argument holds for increasing the injection pressure. Moreover, both measures involve improving already complex and expensive technologies. Conversely, increasing the intake temperature (via uncooled EGR) and/or fuel temperature (although less effective in the applied range) can be regarded as more or less cost-neutral and low-tech solutions.

RESULTS & DISCUSSION - PART 2 OF 4

In this Section, the cumulative impact of the measures discussed in the previous Section is examined (Table 4). Maximum values for the intake temperature (353 K) and fuel temperature (373 K) are selected. Unfortunately, a higher injection pressure than 750 bar was not possible due to a malfunctioning at higher pressures in the fuel injector. This was repaired for the later experiments. In addition, although higher pressures appear favorable, the intake pressure was set to 1.25 bar (absolute), as it is assumed that higher boost pressure can not be achieved using conventional engine hardware. At the maximum values for Tin and Tfuel, a wide SOA timing sweep is performed (Table 12) to determine the optimal injection timing. The detailed results are presented in Appendix E.

From Table 12 it becomes clear that the combination of a high Tin and high Tfuel leads to a considerable reduction in HC emissions and smoke. This reduction, however, comes at a price. IMEP and ISFC both worsen, likely due to the further advancement of CA50. NOx emissions suffer as well, which may be attributed to a large extent to the increase in Tin.

Because the operating conditions have changed significantly by introducing heavy uncooled EGR and hot fuel into the combustion chamber, it is interesting to perform a SOA timing optimization. As indicated in Table 12, the SOA is retarded from -50, to -30 and ultimately to -15 °CA aTDC. The effect on engine performance is clear. IMEP and ISFC approach their respective conventional HDDI baseline values for the best SOA timing of -15 °CA aTDC. NOx emissions are over 95 % lower. HC and smoke emissions, though still significantly higher than the conventional HDDI values, are but a fraction of the values measured in the EDI PCCI baseline work point.

It is striking that an equal SOA and IMEP/ISFC notwithstanding, the emission behavior is completely different. The emission characteristics in the -15 °CA aTDC EDI PCCI work point are more reminiscent of LDI PCCI behavior. It is known from literature [3,5] that EDI PCCI is typically realized at a SOA of -10 CA aTDC or even later and require high injection pressures and modifications to the combustion chamber geometry (e.g. piston bowl, high swirl) in order to realize near-zero soot emissions and low NOx. In the experiments presented here, the combustion chamber geometry has not been modified and a relatively low injection pressure was used. The combination of a lower compression ratio of 12 and heavy EGR apparently retard the onset of combustion such that LDI PCCI is achieved at a relatively early SOA.

From Table 12 it becomes clear that at these operation conditions, given the smooth evolution of engine performance and emissions when retarding the SOA (Appendix E), no clear distinction between the EDI and LDI PCCI regimes can be made. From a control perspective, it is interesting to note that CA50 can be shifted via the SOA in the investigated SOA range.
Table 12 Cumulative “software” measures optimization with a conventional nozzle. Values in brackets are relative to the normal DI baseline case (see Appendix E).

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Unit</th>
<th>Nozzle 2 Normal DI baseline</th>
<th>Nozzle 2 EDI PCCI baseline</th>
<th>Nozzle 1 Cumulative 1</th>
<th>Nozzle 1 Cumulative 2</th>
<th>Nozzle 1 Cumulative 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>SOA</td>
<td>°CA aTDC</td>
<td>-15</td>
<td>-55</td>
<td>-50</td>
<td>-30</td>
<td>-15</td>
</tr>
<tr>
<td>EGR</td>
<td>wt-%</td>
<td>0</td>
<td>40</td>
<td>70</td>
<td>70</td>
<td>70</td>
</tr>
<tr>
<td>Tin</td>
<td>K</td>
<td>303</td>
<td>303</td>
<td>353</td>
<td>353</td>
<td>353</td>
</tr>
<tr>
<td>Tfuel</td>
<td>K</td>
<td>303</td>
<td>303</td>
<td>373</td>
<td>373</td>
<td>373</td>
</tr>
<tr>
<td>Pin</td>
<td>bar</td>
<td>1.25</td>
<td>1.25</td>
<td>1.25</td>
<td>1.25</td>
<td>1.25</td>
</tr>
<tr>
<td>Pfuel</td>
<td>bar</td>
<td>750</td>
<td>750</td>
<td>750</td>
<td>750</td>
<td>750</td>
</tr>
<tr>
<td>λ</td>
<td></td>
<td>4.2</td>
<td>3.1</td>
<td>2.0</td>
<td>1.9</td>
<td>1.9</td>
</tr>
<tr>
<td>CA50</td>
<td>°CA aTDC</td>
<td>-0.7</td>
<td>-6.7</td>
<td>-10</td>
<td>-8.9</td>
<td>+ 0.8 (+ 1.5)</td>
</tr>
<tr>
<td>IMEP</td>
<td>bar</td>
<td>4</td>
<td>3.3</td>
<td>3.0</td>
<td>3.4</td>
<td>3.9 (- 3 %)</td>
</tr>
<tr>
<td>ISFC</td>
<td>g/kWh</td>
<td>214</td>
<td>260</td>
<td>270</td>
<td>233</td>
<td>214 (-)</td>
</tr>
<tr>
<td>NOx</td>
<td>g/kWh</td>
<td>15.1</td>
<td>0.20</td>
<td>1.3</td>
<td>1.6</td>
<td>0.65 (- 96 %)</td>
</tr>
<tr>
<td>HC</td>
<td>g/kWh</td>
<td>0.12</td>
<td>4.76</td>
<td>3.35</td>
<td>1.42</td>
<td>0.64 (+ 433 %)</td>
</tr>
<tr>
<td>Smoke</td>
<td>FSN</td>
<td>0.03</td>
<td>1.44</td>
<td>0.15</td>
<td>0.11</td>
<td>0.09 (+ 200 %)</td>
</tr>
</tbody>
</table>

Table 13 Cumulative “software” measures optimization with smaller nozzle holes (nozzle 2). Values in brackets are relative to the normal DI baseline case (see Appendix F1 and F2).

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Unit</th>
<th>Pfuel Sweep 850 bar</th>
<th>Pfuel Sweep 1200 bar</th>
<th>SOA Sweep 1600 bar @ -50</th>
<th>SOA Sweep 1600 bar @ -30</th>
<th>SOA Sweep 1600 bar @ -15</th>
</tr>
</thead>
<tbody>
<tr>
<td>SOA</td>
<td>°CA aTDC</td>
<td>-50</td>
<td>-50</td>
<td>-50</td>
<td>-30</td>
<td>-15</td>
</tr>
<tr>
<td>EGR</td>
<td>wt-%</td>
<td>70</td>
<td>70</td>
<td>70</td>
<td>70</td>
<td>70</td>
</tr>
<tr>
<td>Tin</td>
<td>K</td>
<td>353</td>
<td>353</td>
<td>353</td>
<td>353</td>
<td>353</td>
</tr>
<tr>
<td>Tfuel</td>
<td>K</td>
<td>360</td>
<td>360</td>
<td>360</td>
<td>360</td>
<td>360</td>
</tr>
<tr>
<td>Pin</td>
<td>bar</td>
<td>1.25</td>
<td>1.25</td>
<td>1.25</td>
<td>1.25</td>
<td>1.25</td>
</tr>
<tr>
<td>λ</td>
<td></td>
<td>1.7</td>
<td>1.7</td>
<td>1.7</td>
<td>1.7</td>
<td>1.7</td>
</tr>
<tr>
<td>CA50</td>
<td>°CA aTDC</td>
<td>-10.8</td>
<td>-11.2</td>
<td>-10.2</td>
<td>-8.7</td>
<td>-1 (- 0.3)</td>
</tr>
<tr>
<td>IMEP</td>
<td>bar</td>
<td>3.6</td>
<td>3.5</td>
<td>3.6</td>
<td>3.81</td>
<td>4.1 (+ 3 %)</td>
</tr>
<tr>
<td>ISFC</td>
<td>g/kWh</td>
<td>252</td>
<td>254</td>
<td>255</td>
<td>246</td>
<td>217 (+ 1 %)</td>
</tr>
<tr>
<td>NOx</td>
<td>g/kWh</td>
<td>1.4</td>
<td>1.8</td>
<td>1.0</td>
<td>0.96</td>
<td>0.65 (- 96 %)</td>
</tr>
<tr>
<td>HC</td>
<td>g/kWh</td>
<td>1.5</td>
<td>1.3</td>
<td>1.1</td>
<td>0.42</td>
<td>0.66 (+ 450 %)</td>
</tr>
<tr>
<td>Smoke</td>
<td>FSN</td>
<td>0.15</td>
<td>0.07</td>
<td>0.04</td>
<td>0.03</td>
<td>0.03 (-)</td>
</tr>
</tbody>
</table>

RESULTS & DISCUSSION - PART 3 OF 4

In this Section, the effect of the cumulative “software” measures in combination with two “hardware” measures will be discussed. As described in Table 5, experiments with a second nozzle with smaller nozzles holes and a third nozzle with both smaller nozzles holes and a narrower cone angle are performed.

SMALLER NOZZLE HOLES – First, the combination of a higher Pfuel with a high Tfuel and Tin is studied (Table 13, Appendix F) at a constant EDI PCCI SOA timing of -50 °CA aTDC. From Table 13 becomes clear that while IMEP and CA50 remain more or less unaffected by an increase in Pfuel, HC emissions and in particular smoke are noticeably decreased. Likely, this is attributable to a reduction in wall-wetting as a result of improved mixing. Next, at the highest value for Pfuel, the SOA is retarded from -50 to -30 and ultimately to -15 °CA aTDC. As was...
the case for the previous (conventional) nozzle (Table 12). ISFC and IMEP approach their normal DI baseline values, while NOx is reduced by over 95%. Smoke and HC emissions, while both significantly higher than the conventional baseline values, are substantially lower than what was the case in the EDI PCCI baseline work point (column 4, Table 12).

The influence of smaller nozzle holes can be studied when comparing the 5th column in Table 12 with the 3rd column in Table 13, as in both cases the Pfuel is around 800 bar. In agreement with literature [2], there is a clear improvement in IMEP and ISFC, but the largest effect (> 50 % decrease) is measured for HC emissions. It is assumed that the smaller nozzle holes improve the vaporization process and accordingly lead to a reduction in wall-wetting. Smoke and NOx are more or less unaffected by the change of the nozzle. Once again, from a control perspective, it is interesting to note that CA50 can be shifted via the SOA in the investigated SOA range.

NARROWER CONE ANGLE – It is known from literature [2] that a narrower cone angle can reduce the occurrence of wall-wetting, because the spray is directed downwards rather than sideways, as is typically the case for conventional diesel nozzles. Details on the utilized nozzles can be found in Table 5. The impact of the cone angle can be studied by comparing the 5th column in Table 13 with the 5th column in Table 14. While the CA50 remains constant, IMEP and ISFC both improve. It is assumed that this is the result of a reduction in wall-wetting. This assumption is supported by the reduction in HC emissions of over 30%.

NOx is little affected by the narrower cone angle, but smoke emissions show a sharp increase in excess of 300 %. A phenomenon known as pool-fire, which is a diffusion-like combustion on the piston surface, might be responsible for this. Described in [2], pool-fire can occur when a narrow cone angle is used under EDI PCCI conditions. Characteristic for pool-fire is a steady increase in smoke and simultaneous decrease in HC emissions. The latter can be explained by the relatively high temperature in the characteristic diffusion-like flames [2]. NOx emissions should reportedly suffer as well due to the aforementioned higher flame temperature [2]. NOx emissions, however, as mentioned earlier, appear to be left unaffected by the narrower cone angle in the experiments presented in this work. This discrepancy may be explained by the marginally higher EGR (i.e. 75 vs. 70 wt-%) used in the narrow cone angle experiments, which counteracts (and might balance) the above mentioned effect.

The results at retarded SOA timings show similar distinctions between the two nozzle types. For this columns 5 thru 7 (nozzle 2, Table 13) have to be compared to columns 5 thru 7 (nozzle 3, Table 14). IMEP and ISFC are slightly improved, while NOx remains more or less unchanged. Smoke emissions are substantially higher, while HC emissions are significantly lower. Accordingly, the results suggest that the pool-fire phenomenon is present in a wide range of injection timings. As was the case for the previous nozzles, CA50 and SOA appear to be coupled in the investigated SOA range. More detailed information on these measurements is presented in Appendix G.

### Table 14 Cumulative “software” measures optimization with smaller nozzle holes and narrow cone angle (nozzle 3).

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Unit</th>
<th>Normal DI baseline</th>
<th>EDI PCCI baseline</th>
<th>Nozzle 3 Cumulative 1</th>
<th>Nozzle 3 Cumulative 2</th>
<th>Nozzle 3 Cumulative 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>SOA</td>
<td>°CA aTDC</td>
<td>-15</td>
<td>-55</td>
<td>-50</td>
<td>-30</td>
<td>-15</td>
</tr>
<tr>
<td>EGR</td>
<td>wt-%</td>
<td>0</td>
<td>40</td>
<td>75</td>
<td>75</td>
<td>75</td>
</tr>
<tr>
<td>Tin</td>
<td>K</td>
<td>303</td>
<td>303</td>
<td>353</td>
<td>353</td>
<td>353</td>
</tr>
<tr>
<td>Tfuel</td>
<td>K</td>
<td>303</td>
<td>303</td>
<td>373</td>
<td>373</td>
<td>373</td>
</tr>
<tr>
<td>λ</td>
<td></td>
<td>4.2</td>
<td>3.1</td>
<td>1.7</td>
<td>1.7</td>
<td>1.7</td>
</tr>
<tr>
<td>CA50</td>
<td>°CA aTDC</td>
<td>-0.7</td>
<td>-6.7</td>
<td>-10.5</td>
<td>-7.5</td>
<td>-0.5 (- 0.2)</td>
</tr>
<tr>
<td>IMEP</td>
<td>Bar</td>
<td>4</td>
<td>3.3</td>
<td>3.73</td>
<td>4.06</td>
<td>4.13 (+ 3 %)</td>
</tr>
<tr>
<td>ISFC</td>
<td>g/kWh</td>
<td>214</td>
<td>260</td>
<td>235</td>
<td>215</td>
<td>211 (- 1 %)</td>
</tr>
<tr>
<td>NOx</td>
<td>g/kWh</td>
<td>15.1</td>
<td>0.20</td>
<td>0.95</td>
<td>0.45</td>
<td>0.46 (- 97 %)</td>
</tr>
<tr>
<td>HC</td>
<td>g/kWh</td>
<td>0.12</td>
<td>4.76</td>
<td>0.75</td>
<td>0.46</td>
<td>0.38 (+ 217 %)</td>
</tr>
<tr>
<td>Smoke</td>
<td>FSN</td>
<td>0.03</td>
<td>1.44</td>
<td>1.76</td>
<td>0.19</td>
<td>0.20 (+ 533 %)</td>
</tr>
</tbody>
</table>
### Table 15 Modeled liquid phase penetration for various operating conditions in EDI PCCI regime (SOA = -50 °CA aTDC)

<table>
<thead>
<tr>
<th>Reduction method</th>
<th>Intake temperature</th>
<th>Intake pressure</th>
<th>Injection pressure</th>
<th>Fuel temperature</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>303 K</td>
<td>353 K</td>
<td>1 bar</td>
<td>2 bar</td>
</tr>
<tr>
<td>Injection pressure [bar]</td>
<td>770</td>
<td>770</td>
<td>800</td>
<td>800</td>
</tr>
<tr>
<td>Fuel temperature [K]</td>
<td>303</td>
<td>303</td>
<td>303</td>
<td>303</td>
</tr>
<tr>
<td>In-cylinder temperature [K]</td>
<td>505.8</td>
<td>584.7</td>
<td>505.8</td>
<td>507.3</td>
</tr>
<tr>
<td>In-cylinder density [kg/m³]</td>
<td>5.27</td>
<td>4.52</td>
<td>4.24</td>
<td>8.48</td>
</tr>
<tr>
<td>LL calculated with Engineering Correlation from Siebers [mm] (d nozzle = 0.190 mm) [11]</td>
<td>335.0</td>
<td>131.2</td>
<td>380.0</td>
<td>241.7</td>
</tr>
<tr>
<td>LL calculated with Engineering Correlation from Siebers [mm] (d nozzle = 0.161 mm) [11]</td>
<td>283.9</td>
<td>111.2</td>
<td>322.0</td>
<td>204.8</td>
</tr>
</tbody>
</table>

**RESULTS & DISCUSSION – PART 4 OF 4**

It is evident from the previous Sections that HC emissions are reduced for all investigated measures aimed at curbing wall-wetting. In order to make a theoretical assessment of the hypothesis that a certain measure actually reduces wall-wetting, an estimate of the degree of wall-wetting has to be made. This is realized by implementing a well-established model for wall-wetting, namely the Liquid Length (LL) engineering correlation developed by Siebers and co-workers [11,13] in the late nineties. n-Dodecane is selected as model fuel as its boiling point corresponds to the lower limit of the diesel boiling range. Accordingly, for given operating conditions, the modeled liquid phase penetration of this compound is indicative of the minimum liquid phase penetration of diesel fuel. The properties for n-dodecane are retrieved from the DIPPR database [12].

From Table 15 it becomes clear that most investigated measures (i.e. intake temperature, intake pressure, fuel temperature and nozzle diameter) also result in a lower predicted LL value, suggesting that HC emissions may be a good indicator for the degree of wall-wetting in the investigated EDI PCCI regime.

As acknowledged by the authors [11], the model is insensitive to a change in fuel injection pressure as this parameter does not influence liquid phase penetration in the mixing-limited vaporization regime [11]. In the experiments, however, the measured decline in HC emissions suggests a decrease in LL with increasing injection pressure (Table 9). A possible explanation for this discrepancy could be related to the relatively low gas temperature used in the experiments. As the gas temperature approaches the boiling range of a fuel, the assumption of mixing-limited vaporization is no longer valid and is replaced (gradually) by a regime wherein vaporization is limited by interphase transport [13]. The importance of fuel injection pressure, and in particular its impact on droplet size, in this regime is subject of current research.

**CONCLUSIONS**

In order to reduce the occurrence of wall-wetting under early direct injection (EDI) PCCI conditions, a number of measures have been investigated:

- Higher intake temperature (via hot EGR)
- Higher intake pressure (via external boosting)
- Higher fuel pressure (via external fuel pump)
- Higher fuel temperature (via heated common rail)
- Smaller nozzle holes
- Smaller nozzle holes + narrower cone angle

All experiments have been performed at low load (~ 3-4 bar IMEP) and at an engine speed of 1200 RPM, using a modified 6-cylinder 12.6 liter heavy-duty DI DAF XE 355
C engine. Experiments are conducted in one dedicated cylinder, which is equipped with a stand-alone fuel injection system, EGR circuit and air compressor, fuelled with commercial diesel fuel (EN590).

In the EDI PCCI regime, measured HC emissions, an indicator of wall-wetting, suggest that wall-wetting is indeed (partially) suppressed for all investigated measures. Spray modeling results retrieved from the Siebers Liquid Length (LL) engineering correlation [11,13] show that for most measures the LL, a measure for wall-wetting, is indeed reduced. Fuel injection pressure appears to be an exception. Not accounted for in the utilized model, higher fuel pressures obviously reduce HC emissions in the engine experiments. Other (CFD) models which do take the fuel pressure into account are currently subject of investigation.

The best overall results achieved in the EDI PCCI regime are presented in Table 16 (column 5). In this work point the fuel and intake temperature, as well as the fuel pressure, are set to their respective maximum values. In addition, a narrow cone angle nozzle with smaller holes (nozzle 3) is used. By comparing column 4 (baseline EDI PCCI) and 5 (best EDI PCCI) in Table 16, the significant improvement in HC emissions and engine performance becomes evident, indicating that wall-wetting is indeed suppressed to a large extent. Although HC emissions can originate from other sources than solely wall-wetting, the extreme values and the observed correlation with various wall-wetting measures suggest that wall-wetting is responsible for the bulk of the HC emissions.

Compared to conventional diesel combustion (Table 16, column 5 vs. 3), NOx emissions are reduced to extremely low values without a penalty in fuel economy. Unfortunately both smoke and HC emissions are significantly higher. The relatively high smoke values may be (partially) attributed to a phenomenon known as pool-fire (i.e. diffusion-like combustion on the piston surface) which is a direct consequence of injection via a narrow cone angle nozzle [2], while the high HC emissions are likely the result of a remaining wall-wetting effect.

In an attempt to curb wall-wetting and retard CA50 more towards TDC, the injection timing was retarded further from -30 to -15 °CA aTDC and the nozzle was replaced by a conventional cone angle variant (nozzle 2). In doing so, a transition was made from an EDI PCCI (Table 16, column 5) to a late direct injection (LDI) PCCI timing (Table 16, column 6).

To confirm that the LDI PCCI work point in Table 16 indeed qualifies as PCCI, a final brief heat-release analysis is made. Herein, CA10 is taken as the start of combustion (SOC). From Table 16 becomes clear that, contrary to the normal DI baseline case, the CA10 occurs several crank angles later than of the end of injection (EOI). As discussed in the Introduction, the main qualifier for PCCI is that the fuel injection and heat release events are separated in time. Accordingly, the investigated LDI work point in Table 16 may be referred to as LDI PCCI.

Under the utilized conditions of a low compression ratio of 12 and heavy EGR (70 wt-%), retarding the injection timing and replacing the nozzle resulted in a further reduction of smoke, with an acceptable increase in HC and NOx emissions. Fuel economy and IMEP are comparable. The available data suggests that while acceptable PCCI operation is possible in the EDI regime, comparable fuel economy and near-EURO VI-like smoke and NOx emissions can be realized without smoke/NOx aftertreatment in LDI PCCI mode. HC emissions are too high, but could be dealt with by mounting an oxidation catalyst in the exhaust system. Overall it seems that when a conventional diesel fuel is considered, LDI PCCI appears to be more promising than EDI PCCI.

Lastly, it should be noted from a control perspective that the combustion phasing is sensitive to the injection timing and other operating conditions even in the EDI PCCI regime. For LDI PCCI, this sensitivity is even stronger.
Table 16 Best results.

Note that parameters relative to engine power (i.e. g/kWh) are indicated not brake specific, as they are based on IMEP.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Unit</th>
<th>Normal DI baseline Nozzle 2 Ø=161 μm θ=150°</th>
<th>EDI PCCI baseline Nozzle 2 Ø=161 μm θ=150°</th>
<th>Best EDI PCCI Nozzle 3 Ø=161 μm θ=100°</th>
<th>Best LDI PCCI Nozzle 2 Ø=161 μm θ=150°</th>
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</thead>
<tbody>
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<td>SOA</td>
<td>°CA aTDC</td>
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<td>-30</td>
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<td>°CA aTDC</td>
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<td>-</td>
<td>-</td>
<td>-11</td>
</tr>
<tr>
<td>EOI</td>
<td>°CA aTDC</td>
<td>-1</td>
<td>-</td>
<td>-</td>
<td>-6</td>
</tr>
<tr>
<td>EGR</td>
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<td>40</td>
<td>75</td>
<td>70</td>
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<tr>
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<td>1.25</td>
<td>1.25</td>
<td>1.25</td>
</tr>
<tr>
<td>Pfuel</td>
<td>bar</td>
<td>750</td>
<td>750</td>
<td>1600</td>
<td>1600</td>
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<tr>
<td>Tin</td>
<td>K</td>
<td>303</td>
<td>303</td>
<td>353</td>
<td>353</td>
</tr>
<tr>
<td>Tfuel</td>
<td>K</td>
<td>303</td>
<td>303</td>
<td>373</td>
<td>360</td>
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<tr>
<td>λ</td>
<td>-</td>
<td>4.2</td>
<td>3.1</td>
<td>1.7</td>
<td>1.7</td>
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<tr>
<td>SOC/CA10</td>
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<td>-2.9</td>
<td>-</td>
<td>-</td>
<td>-3.4</td>
</tr>
<tr>
<td>CA50</td>
<td>°CA aTDC</td>
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<tr>
<td>IMEP</td>
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<td>3.3</td>
<td>4.06</td>
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<td>ISFC</td>
<td>g/kWh</td>
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<td>260</td>
<td>215</td>
<td>217</td>
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<tr>
<td>NOx</td>
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<td>1.44</td>
<td>0.19</td>
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12. DIPPR® Database, Brigham Young University, England (http://dippr.byu.edu/).

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DEFINITIONS, ACRONYMS & ABBREVIATIONS

<table>
<thead>
<tr>
<th>AFR</th>
<th>Air Fuel Ratio</th>
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<td>After Bottom Dead Center</td>
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<tr>
<td>aTDC</td>
<td>After Top Dead Center</td>
</tr>
<tr>
<td>bBDC</td>
<td>Before Bottom Dead Center</td>
</tr>
<tr>
<td>bTDC</td>
<td>Before Top Dead Center</td>
</tr>
<tr>
<td>CA10</td>
<td>Crank Angle at which 10% of the fuel is burnt</td>
</tr>
<tr>
<td>CA50</td>
<td>Crank Angle at which 50% of the fuel is burnt</td>
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<tr>
<td>cDPF</td>
<td>Catalytic Diesel Particulate Filter</td>
</tr>
<tr>
<td>CA</td>
<td>Crank Angle</td>
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<tr>
<td>CI</td>
<td>Compression Ignition</td>
</tr>
<tr>
<td>d</td>
<td>diameter</td>
</tr>
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<td>Early Direct Injection</td>
</tr>
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<td>EGR</td>
<td>Exhaust Gas Recirculation</td>
</tr>
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<td>EN590</td>
<td>Baseline commercial diesel fuel</td>
</tr>
<tr>
<td>EOI</td>
<td>End of injection (fuel delivery)</td>
</tr>
<tr>
<td>FIE</td>
<td>Fuel Injection Equipment</td>
</tr>
<tr>
<td>FSN</td>
<td>Filter Smoke Number</td>
</tr>
<tr>
<td>HDDI</td>
<td>Heavy Duty Direct Injection</td>
</tr>
<tr>
<td>IMEP</td>
<td>Indicated Mean Effective Pressure</td>
</tr>
<tr>
<td>ISFC</td>
<td>Indicated Specific Fuel Consumption</td>
</tr>
<tr>
<td>LDI</td>
<td>Late Direct Injection</td>
</tr>
<tr>
<td>LL</td>
<td>Liquid Length</td>
</tr>
<tr>
<td>p</td>
<td>Pressure</td>
</tr>
<tr>
<td>PAH</td>
<td>Poly Aromatic Hydrocarbons</td>
</tr>
<tr>
<td>PCCI</td>
<td>Premixed Charge Compression Ignition</td>
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<td>Pfuel</td>
<td>Fuel injection pressure</td>
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<tr>
<td>Pin</td>
<td>Intake pressure</td>
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</table>
APPENDIX A: INTAKE TEMP. (TABLE 7)

Figure 8 Impact intake temperature on CA50

Figure 9 Impact intake temperature on IMEP

Figure 10 Impact intake temperature on ISFC

Figure 11 Impact intake temperature on NOx

Figure 12 Impact intake temperature on HC

Figure 13 Impact intake temperature on smoke

1 Odd trend is the result of a slight adjustment in EGR level made between two workpoints to correct for a small drift in EGR level.
APPENDIX B: INTAKE PRESSURE (TABLE 8)

Figure 14  Impact intake pressure on CA50

Figure 15  Impact intake pressure on IMEP

Figure 16  Impact intake pressure on ISFC

Figure 17  Impact intake pressure on NOx

Figure 18  Impact intake pressure on HC

Figure 19  Impact intake pressure on smoke

In Appendices B, C and D the two measuring points at a given condition are repetitions.
APPENDIX C: FUEL PRESSURE (TABLE 9)

Figure 20  Impact fuel pressure on CA50

Figure 21  Impact fuel pressure on IMEP

Figure 22  Impact fuel pressure on ISFC

Figure 23  Impact fuel pressure on NOx

Figure 24  Impact fuel pressure on HC

Figure 25  Impact fuel pressure on smoke
APPENDIX D: FUEL TEMP. (TABLE 10)

Figure 26 Impact fuel temperature on CA50

Figure 27 Impact fuel temperature on IMEP

Figure 28 Impact fuel temperature on ISFC

Figure 29 Impact fuel temperature on NOx

Figure 30 Impact fuel temperature on HC

Figure 31 Impact fuel temperature on smoke
APPENDIX E: CUMULATIVE SOFTWARE MEASURES (TABLE 12)

Figure 32  Impact of operation conditions on CA50

Figure 33  Impact of operation conditions on ISFC and NOx

Figure 34  Impact of operation conditions on smoke and HC

APPENDIX F1: NOZZLE DIAMETER (TABLE 13)

Figure 35  Impact of operation conditions on CA50

Figure 36  Impact of operation conditions on ISFC and NOx

Figure 37  Impact of operation conditions on smoke and HC
APPENDIX F2: NOZZLE DIAMETER (TABLE 13)  

Figure 38  Impact of operation conditions on CA50

Figure 39  Impact of operation conditions on ISFC and NOx

Figure 40  Impact of operation conditions on smoke and HC

APPENDIX G: CONE ANGLE (TABLE 13)

Figure 41  Impact of operation conditions on CA50

Figure 42  Impact of operation conditions on ISFC and NOx

Figure 43  Impact of operation conditions on smoke and HC