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Experimental study on the impact of operating conditions on PCCI combustion

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Abstract: In a short-term scenario, using near-standard components and conventional fuels, PCCI combustion relies on a smart choice of operating conditions. Here, the effects of operating conditions on ignition delay, available mixing time, combustion phasing and emissions are investigated. In the PCCI regime, NO_x and smoke have been shown to be efficiently reduced with elongated mixing time. For viable PCCI combustion, one would require a Combustion Delay (CD) which is long enough to bring both NO_x and smoke levels down to acceptable values. For the completeness of combustion, the resulting unburned hydrocarbon and carbon monoxide emissions, as well as the associated fuel consumption; mixing time should, however, be as short as possible. Most parameters strongly correlate with combustion delay, independent of how this is achieved. Lastly, the best points experienced for a number of cases are given.

Keywords: PCCI; premixed charge compression ignition; diesel; operating conditions; EGR; exhaust gas recirculation; intake pressure; intake temperature; combustion delay; mixing time.

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1 Introduction

Current diesel combustion technology results in NO_x and soot levels much higher than the limits to be imposed by the forthcoming emissions legislation (EURO VI, US 2013). Hence, after-treatment systems based generally on Selective Catalytic Reduction (SCR, Held et al., 1990) and Diesel Particulate Filtration (DPF, Johnson, 2010) technologies have to be used for the reduction of NO_x and soot, respectively. The development of combustion technologies with intrinsically lower NO_x/soot emissions can minimise the after-treatment system requirements and thus, reduce the related costs.

Combustion temperature and thus the NO_x level, can be lowered by running lean, pre-mixed fuel or by using Exhaust Gas Recirculation (EGR) to change the trapped mass composition (Heywood, 1988). Smoke levels are generally reduced by promoting mixing of fuel and air before combustion starts. Premixed Charge Compression Ignition (PCCI) combustion is characterised by using EGR and early fuel injection, compared to that of conventional diesel combustion, to enable premixing of fuel and air and to lower the combustion temperature. As this (partially) premixed charge is brought to auto-ignition, low NO_x and soot levels are experienced (Kalghatgi et al., 2007; Shimazaki et al., 2007; Lu et al., 2008; Bression et al., 2008).

Conventional Heavy Duty Direct Injection (HDDI) hardware and diesel fuels present a number of challenges to PCCI. The high boiling range of current diesel fuels has the intrinsic risk of wall-wetting when injecting early (Chiara and Canova, 2009; Andre et al., 2009; Sakai et al., 2005). These fuels have cetane numbers of 40–60 and are very prone to auto-ignition (Kalghatgi, 2005). To allow better mixing of fuel and air before heat release occurs, this auto-ignition needs to be delayed to reduce smoke. One of the problems of a more pre-mixed combustion could be a high heat release rate, which can lead to unacceptably high Pressure Rise Rates (PRRs) and noise (Kalghatgi et al., 2007).

In a short-term scenario, using near-standard components and conventional fuels, the challenges posed above should be met with a smart choice of operating conditions, but still in a limited operating range. This can, however, well be in line with common strategies that are explored to meet the Euro VI and EPA13 directives. These require effective combinations, both in costs and CO_2 emission, of different Low Temperature Combustion (LTC) concepts, advanced fuel- and air-delivery circuits and emission after-treatment systems. Where earlier research at the Eindhoven University of Technology has focused on the effects of different parameters on wall-wetting (Boot et al., 2009a; 2009b), here the effects of operating conditions on ignition delay, available mixing time, combustion phasing and emissions are investigated.

In the PCCI regime, NO_x and smoke will be shown to be efficiently reduced with elongated mixing time. For viable PCCI combustion, one will require a Combustion Delay (CD) that is long enough to bring both NO_x and smoke levels down to acceptable values. For the completeness of combustion, the resulting unburned hydrocarbon and carbon monoxide emissions, as well as the associated fuel consumption; mixing time should, however, be as short as possible. Most parameters will be shown to strongly correlate with combustion delay, independent of how this is achieved. Lastly, in Section 3.5, the best points experienced for a number of cases are given.

2 Experiment setup

2.1 Experimental apparatus

The Cyclops is a dedicated engine test rig, see Table 1 and Leermakers et al. (2011), based on a DAF XE 355 C engine. Cylinders 4 through 6 operate under the stock DAF engine control unit and together with a water-cooled, eddy-current Schenck W450 dynamometer, they are merely used to control the crankshaft rotational speed of the test cylinder, i.e., cylinder 1.

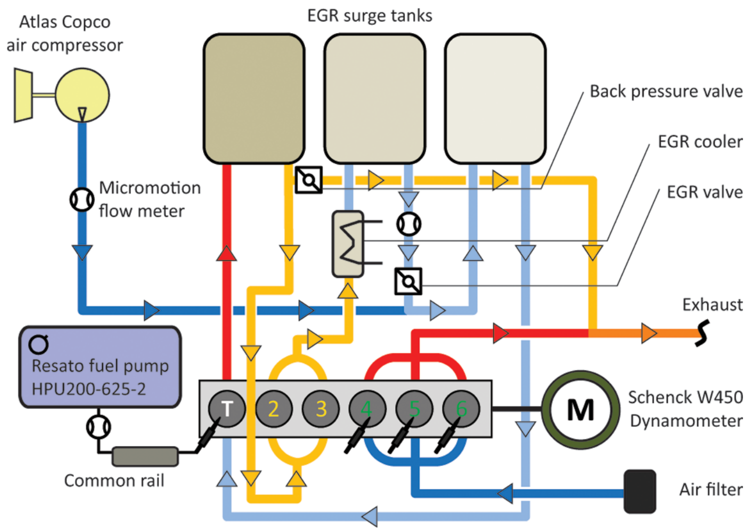
Table 1 Cyclops specifications

Base engine	*6 cylinder HDDI diesel*
Bore [mm]	130
Stroke [mm]	158
Compression ratio [–]	Variable, here 12:1

Combustion phenomena and emission formation can be studied in this test cylinder. Apart from the mutual cam- and crankshaft and the lubrication and coolant circuits, the test cylinder operates autonomously from the propelling cylinders. Stand alone air, EGR and fuel circuits have been designed for maximum flexibility, as will be discussed below.

Fed by an Atlas Copco air compressor, the intake air pressure of the test cylinder can be boosted up to 5 bar. Non-firing cylinders 2 and 3 function as EGR pump cylinders (see Fig. 1), the purpose of which is to generate adequate EGR flow, even at 5 bar charge pressure and recirculation levels in excess of 70%. The EGR flow can be cooled both upstream and downstream of the pump cylinders and several surge tanks and pressure relief valves have been included in the design.

Figure 1 Cyclops engine test rig schematic (see online version for colours)



Fuel to cylinder 1 is provided by a double-acting air-driven Resato HPU200-625-2 pump, which can deliver a fuel pressure up to 4200 bar. An accumulator is placed near (≈0.2 m) the fuel injector to mimic the volume of a typical common rail. Like for fresh air and EGR, the steady state fuel mass flow is measured with a Micro Motion mass flow meter. The prototype common rail injector used is equipped with a nozzle that has the same dimensions as the one used in the study by Boot et al. (2009a), which gave the best performance in the Late DI PCCI regime. This nozzle has 8 holes of 0.151 mm diameter, with a cone angle of 153 degrees.

For measuring gaseous exhaust emissions a Horiba Mexa 7100 DEGR emission measurement system is used and the exhaust smoke level (in Filter Smoke Number or FSN units) was measured using an AVL 415 smoke meter. All quasi steady-state engine data, as also intake and exhaust pressures and oil and water temperatures, are recorded by means of an in-house data acquisition system.

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.

2.2 *Experimental procedure*

In all experiments described here, a single injection strategy is used. For each target load, the injection duration is determined, which is necessary in the conventional combustion regime. This injection duration is then kept constant, while varying the start of actuation and other operating conditions. 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).

For every combination of operating conditions under investigation, a sweep of Start of Actuations (SOA) of the injector is performed. Starting from conventional CI timings, SOA is advanced at 5-degree increment steps, skipping SOAs at which combustion is not acceptable. Acceptable combustion in this aspect is defined by both engine hardware limitations and combustion quality targets.

For more information on the design of the experimental apparatus and measurement procedure, the reader is referred to Leermakers et al. (2011).

2.3 Data analysis and definitions

When comparing emission levels and fuel consumption for heavy duty engines, it is common practice to calculate the brake specifics, i.e., with respect to the power output at the crankshaft. In this test setup, also the three propelling cylinders (i.e., cylinders 4 through 6) are connected to the engine brake, so brake specific values cannot be given. Therefore, in this case, the IMEP as calculated from the in-cylinder pressure signal is used. To be able to evaluate the combustion performance also at different intake pressures and varying exhaust back pressures in all results presented, the gross IMEP has been used to calculate the indicated fuel consumption and emissions.

CA10 is used to indicate Start Of Combustion (SOC), because of its considerably higher stability compared to CA5. Analysis of the logged injector actuation current and injection pressure data furthermore shows a constant 4°CA lag between SOA and Start Of Injection (SOI) at the engine speed set in this study (1,200 rpm). Given the assumptions made above, the following definitions are used to characterise combustion. Here, CD is used to describe the average mixing time, while ignition dwell represents the separation between injection and combustion events.

- Start of combustion (SOC) = CA10
- Start of injection (SOI) = SOA + 4°CA
- Combustion delay (CD) = CA50 – SOI
- Ignition dwell (IDw) = SOC – EOI
- Ignition delay (IDy) = SOC – SOI

Specific emissions are computed from their respective concentrations by using their molar weights. Only NO_x is treated as NO_2 in line with European legislation. Smoke emission is computed from the measured FSN, using an empirical correlation.

2.4 Measurement matrix

All tests are done at 1,200 rpm, representative of the engine speed of a typical road transport vehicle at highway cruising. This is also near the B speed in the European Stationary Cycle (ESC). Fuel injection pressure is set to 1,500 bar and fuel temperature near the injector is 30°C. Unless stated otherwise, the EGR flow is heavily cooled using cold process water, resulting in an EGR temperature of approximately 300 K.

To limit the end of compression temperature to avoid premature auto ignition, the geometric compression ratio of the test cylinder has been lowered by means of thicker head gaskets to a value of 12:1. For the short-term scenario, a standard European diesel fuel is used, of which the properties are known.

Auto-ignition chemistry is mainly governed by chemical kinetics. Therefore, control of combustion phasing is highly dependent on in cylinder conditions, see Boot et al. (2009b). These in-cylinder conditions, characterised as the temperature (T)–pressure (P)–equivalence ratio (φ) history in this series, are influenced using the following parameters:

- four injector actuation durations: 850–1150–1550–2100 μs, targeting net IMEPs of 6–7.5–10–12.5 bar
- four EGR levels: 0–15–40–60 wt%
- six intake pressure levels: 1.05–1.25–1.5–1.88–2.1–2.5 bar
- Intake temperature using both heavily cooled EGR, resulting in 300 K maximum and uncooled EGR, resulting in 370 K.

Except for the intake temperature, which is only tested at two intake pressures and three loads, all possible combinations are tested, taking engine hardware limitations and combustion quality targets into account. Testing this full matrix enables the identification of feasible PCCI operating ranges.

3 Results

In this section, the results of the decoupling of injection and combustion are illustrated for the PCCI concept, first for an initial timing sweep. Then for one parameter, i.e., the EGR level, the effects on combustion phasing, performance, pressure rise rate and emissions are discussed in detail, with a discussion on the respective origins of the effects. After that, for intake pressure and temperature respectively, all relevant results are shown one by one. The best points experienced for a number of loads are given to stress the practical relevance of the measurements presented. To conclude, a summary is given of the relevant effects of different parameters on PCCI combustion.

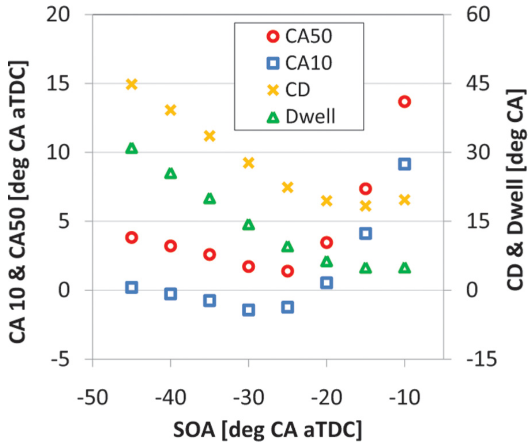
3.1 Injection timing

PCCI combustion is characterised by decoupling of combustion and injection, by using EGR and early fuel injection compared to that of conventional diesel combustion, to enable premixing of fuel and air and to lower combustion temperature. From an initial timing sweep (see Fig. 2), one can see that as SOA is advanced from late timings, at first combustion delay and ignition dwell remain near constant as injection and combustion are coupled. However, as SOA is advanced beyond –20°CA aTDC, CD and Ignition dwell are significantly elongated.

For the operating conditions shown in Figure 2, with a low end of compression temperature and heavy EGR, for all SOA values the ignition dwell is seen to be positive and injection and combustion are thus separated. Apart from the operating conditions, this separation can also be explained by the fact that CA10 is used as start of combustion and not CA0. In this work, therefore the absolute value of the ignition dwell is not used to define PCCI combustion, but the change in dwell and combustion delay. Where these parameters are significantly longer than at conventional timings, SOAs have an elongated mixing time and are considered to represent PCCI combustion.

From Figure 3, the effects of this elongated mixing time can be seen. Decoupling at timings before –20°CA aTDC results in significantly lower NO_x levels and in smoke emissions approaching zero. Fuel consumption is seen to be the lowest at a timing of –25°CA aTDC, which is already defined as PCCI combustion. As timing is further advanced and ignition dwell is large, both wall wetting (Boot et al., 2009b), overmixing and overleaning Kook et al. (2005) give rise to higher HC and CO emissions and therefore, fuel consumption significantly increases. At even earlier timings (< – 45°CA aTDC) presumed wall wetting, or incomplete combustion of the soot formed earlier in the stroke, results in smoke emissions increasing again.

Figure 2 Combustion phasing for an initial timing sweep. Operating conditions: 10 bar target IMEP, 60 wt% EGR, 1.5 bar pin (see online version for colours)



3.2 EGR level

Adding EGR is the classic way to reduce NO_x emissions from conventional diesel combustion, through charge dilution and lowering of the adiabatic flame temperature. In the following subsections, the effects of EGR on combustion phasing, performance and emissions are shown in detail for PCCI combustion.

3.2.1 Combustion phasing and performance

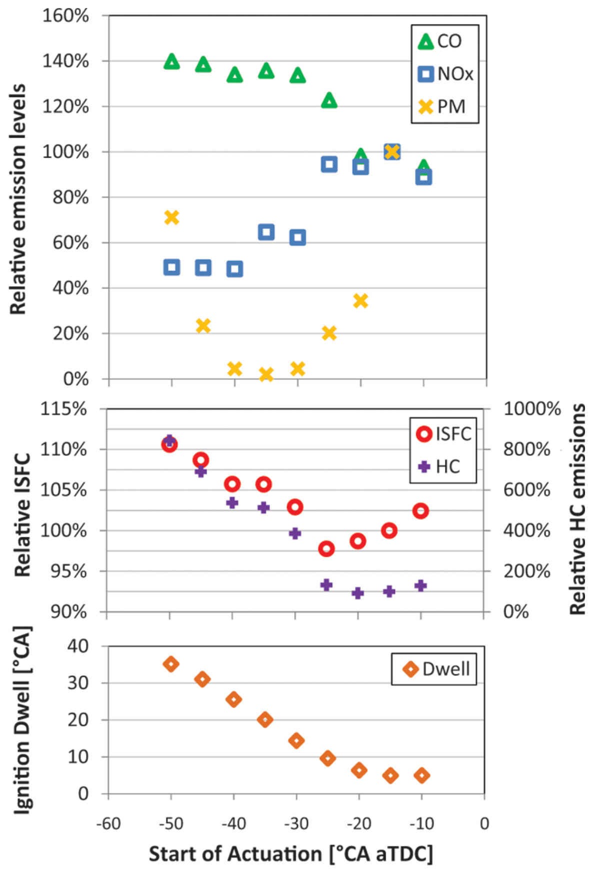
From Figure 4(a), where timing sweeps are shown for different EGR levels, the effect of EGR on combustion timing can be seen. As can be expected, a higher EGR level gives a longer ignition delay and slower combustion, resulting in a significantly later CA50. Results are not plotted for all timings and EGR levels.

For a constant fuelling rate, as in the results under investigation, IMEP characterises performance and efficiency. Figure 4(b) shows that for a constant SOA, a higher EGR level results in a lower IMEP. While it seems that EGR has a direct negative effect on efficiency, it will be shown that this is not the case, but that it is an indirect effect resulting from increased combustion delay.

In Figure 5(a), one can see a distinction between two regimes. The nearly horizontal lines, at about 5 bar IMEP, represent Conventional Diesel Combustion (CDC). For this regime, EGR only affects combustion phasing and for a constant CA50, EGR is not seen to have a significant effect on efficiency.

The second regime, with earlier injection timings, does not show any correlation between IMEP and CA50. However, if these points are plotted against CD, in Figure 5(b), a decrease in load is seen with a longer CD. For these points with an increased mixing time, this effect is dominant over the direct effect of EGR. This can be attributed to more incomplete combustion, as will be shown later in this section.

Figure 3 Impact of injection timing on combustion: a) relative CO, NO_x and PM emissions vs. SOA; b) relative fuel consumption and HC emissions vs. SOA and c) ignition dwell vs. SOA. Operating conditions: 10 bar target IMEP, 60 wt% EGR, 1.5 bar pin. For relative values, reference 100% is taken at SOI=–15° aTDC; this is because at these combinations, unacceptable combustion occurs–in this case because of too high pressure rise rates (see online version for colours)

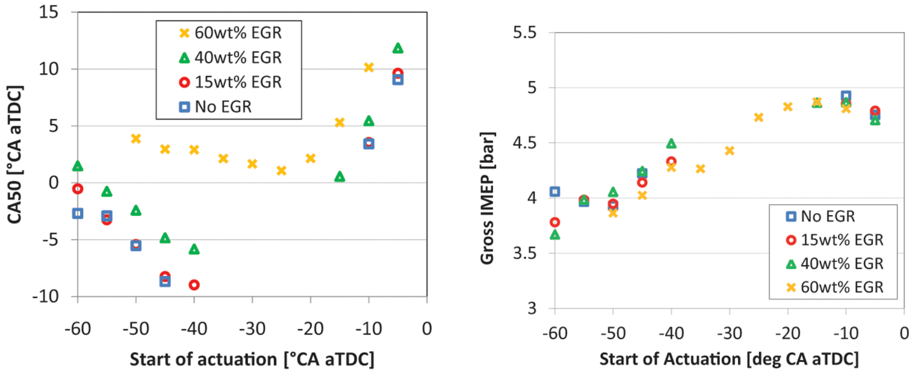


3.2.2 HC and CO emissions

In the CDC combustion regime, HC and CO emissions are insignificant due to high localised temperatures during the combustion and the proximity of oxygen, enabling full oxidation (Bression et al., 2008), even at low loads. EGR allows the combustion process to be controlled by reducing the local temperature and by decreasing the oxygen partial pressure in the cylinder. While NO_x emissions can be lowered like this, complete post oxidation of HC and CO can become more difficult to achieve.

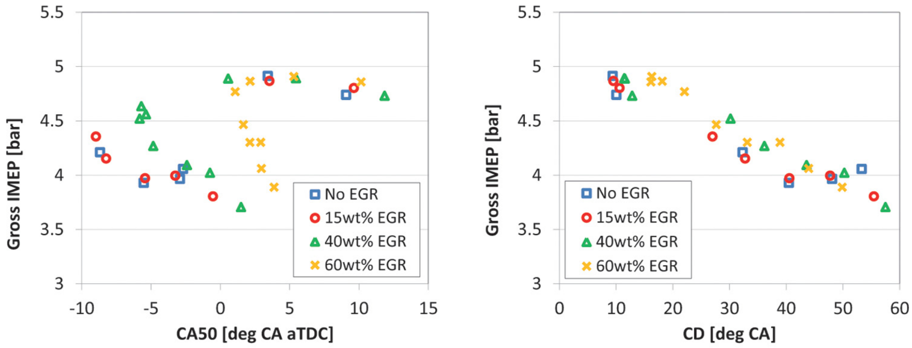
As discussed by Heywood (1988) and summarised by Kashdan et al. (2007) and Colban et al. (2007), there are a number of possible sources of engine-out HC emissions. For gasoline SI engines, trapping of fuel in crevice volumes is one of the major sources of HC emissions. For CI engines, this source is 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.

Figure 4 Impact of EGR on combustion phasing and performance. Operating conditions: 5 bar target IMEP, 1.25 bar pin (see online version for colours)



(a) CA50 vs. SOA. (b) IMEP vs. SOA

Figure 5 Impact of EGR on IMEP, plotted vs. CA50 and CD. Operating conditions: 5 bar target IMEP, 1.25 bar pin (see online version for colours)



(a) IMEP vs CA50 (b) IMEP vs CD

Another potential mechanism for HC emissions formation is through flame quenching. Quenching of oxidation reactions may occur due to excessively low temperature zones caused by heat transfer to combustion chamber walls, or by natural thermal inhomogeneities 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 that do not permit complete combustion on relevant engine time scales. This is mainly thought to occur at low loads, particularly for conditions where ignition delay is long, which allows for a long mixing time.

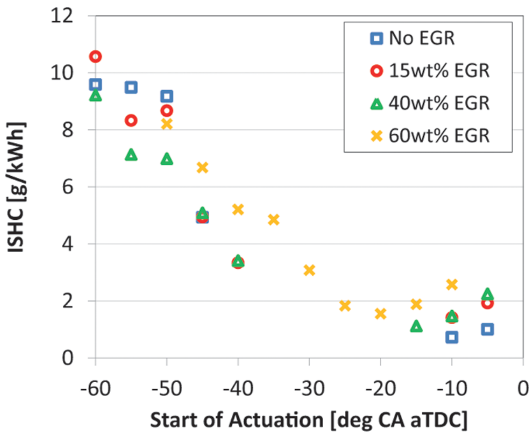
Finally, wall wetting can be a source of HC emissions in direct injection engines. Depending on the combustion chamber geometry and injection timing, spray impingement on cylinder walls results in the formation of liquid films, which give incomplete evaporation and oxidation of the fuel. Any remaining unburned hydrocarbons within the combustion chamber can subsequently be emitted into the exhaust.

Kook et al. (2005) also discusses a number of possible causes of CO emissions. While HC emissions can consist of both uncombusted and partially oxidised fuel, CO emissions are always a result of incomplete combustion. This suggests that these CO emissions are also a result of 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 full oxidation of CO to be completed on engine time scales.

Considering the above, for both HC (Kashdan et al., 2007) and CO emissions (Kook et al., 2005), the maximum cycle temperature is found to be a dominant and first order parameter.

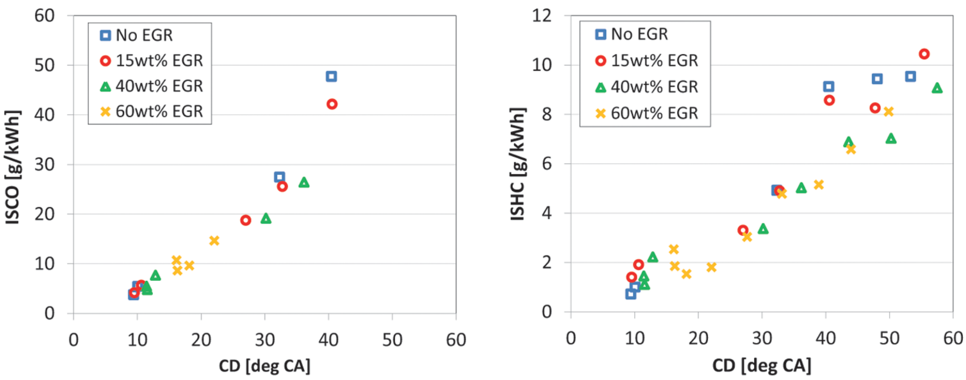
For the points under investigation here, at first it seems that for both the CDC and the PCCI regime, an increase in EGR level leads to an increase in HC emissions for constant start of actuation, see Figure 6. In the CDC regime, where CA50 is close to the start of actuation and CD is thus short, it is indeed observed that more EGR increases HC emissions. The effects on CO emissions (not shown here) are the same as for HC.

Figure 6 Impact of EGR on unburned hydrocarbon emissions, as a function of SOA. Operating conditions: 5 bar target IMEP, 1.25 bar pin (see online version for colours)



EGR both increases the available mixing time and lowers the combustion temperature, through changing the heat capacity of the charge. For earlier injection timings, while entering the PCCI regime, the latter effect has a predominant effect on HC and CO emissions. These show a clear correlation with CD (see Fig. 7), which indicates that for these low loads in the PCCI regime, HC and CO emissions are predominantly caused by more and larger cold zones which cause flame quenching.

Figure 7 Impact of EGR on ISCO and ISHC, as a function of CD. Operating conditions: 5 bar target IMEP, 1.25 bar pin (see online version for colours)



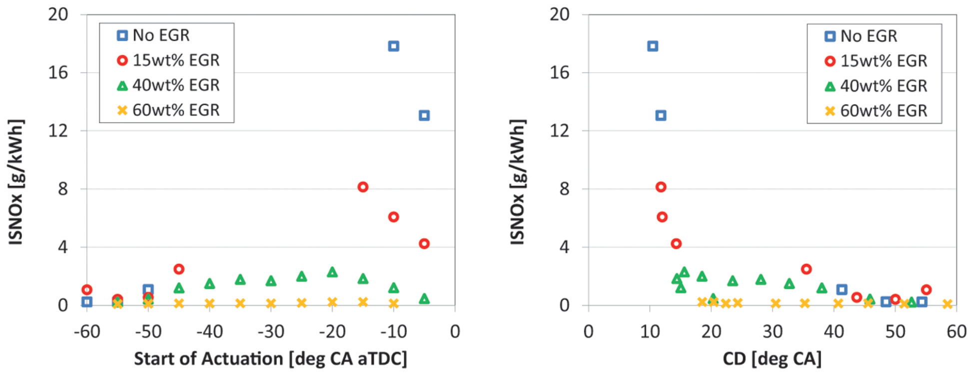
(a) ISCO vs. CD (b) ISHC vs. CD

3.2.3 NO_x emissions

As with other LTC concepts, EGR is used to lower the adiabatic flame temperature of the combustion. This brings NO_x 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. Because the combustion temperature is also lowered by increased premixing with PCCI combustion, less EGR should be necessary to bring NO_x down to acceptable levels.

In the CDC regime, NO_x levels (Fig. 8) are seen to be effectively reduced by adding EGR, as expected. For longer ignition delays, in the PCCI regime, the effect of EGR becomes smaller and as the charge becomes increasingly premixed, combustion temperatures are predominantly lowered by the leaner local mixture strength.

Figure 8 Impact of EGR on NO_x emissions, as a function of SOA and CD. Operating conditions: 5 bar target IMEP, 1.05 bar pin (see online version for colours)

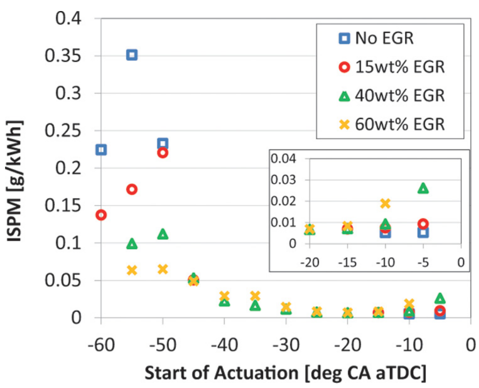


(a) ISNOx emissions vs. SOA (b) ISNOx emissions vs. CD

3.2.4 Smoke emissions

For particulate emission in conventional diesel combustion, an increased EGR level lowers the air excess ratio and therefore, more soot is emitted, see zoomed plot in Figure 9.

Figure 9 Impact of EGR on smoke emissions, as a function of SOA. Zoomed plot to illustrate differences in PM emissions for conventional timings. Operating conditions: 5 bar target IMEP, 1.05 bar pin (see online version for colours)



As start of actuation is advanced into the PCCI regime and mixture time is increased, no direct effect of EGR can be seen. Here, smoke emission is predominantly caused by the longer mixing time and the lower combustion temperatures and soot oxidation rates associated therewith. When injection timing is even further advanced, wall wetting is believed to occur and smoke emission increases. For all points under investigation, the absolute smoke levels are quite low, because of the relatively low loads and the injection pressure and equipment used.

3.3 Air intake pressure

3.3.1 Combustion phasing and performance

A lower air intake pressure significantly retards combustion phasing, resulting in a longer CD (Fig. 10). This is necessary to phase combustion correctly at early timings, because when CA50 is advanced before TDC, pressure rise rates become unacceptable.

Figure 10 Impact of intake pressure on CD, as a function of SOA. Operating conditions: 10 bar target IMEP, 55 wt% EGR (see online version for colours)

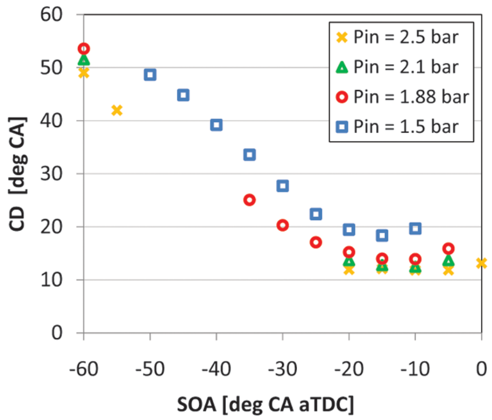
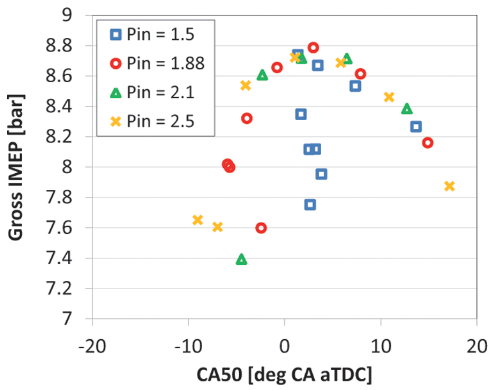


Figure 11 Impact of intake pressure on IMEP, as a function of CA50. Operating conditions: 10 bar target IMEP, 55 wt% EGR (see online version for colours)

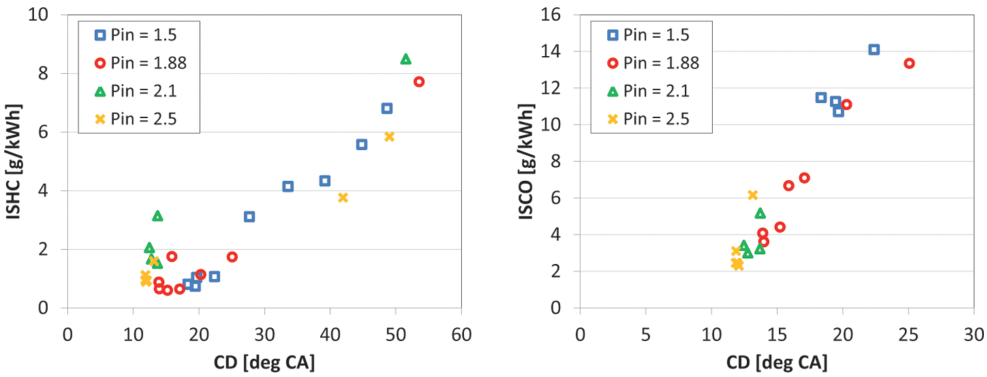


Considering that all comparisons are based on the gross IMEP, one would not expect intake pressure to make a big difference to IMEP. This is also shown in Figure 11, where CA50 is shown to be the dominant parameter. For longer combustion delays, IMEP again correlates with this CD (not shown here).

3.3.2 HC and CO emissions

Figure 12 shows that HC and CO emissions are again shown to be nearly perfectly linear with CD. The higher intake pressure here only shows a direct influence on this CD. This CO/HC trend with CD 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. Because the lower intake pressures have longer combustion delays, they also result in higher HC and CO emissions. Another important effect on increasing intake pressure is the increase in in-cylinder gas density, which can improve mixing and reduce spray penetration and thus, wall-wetting. This effect might be present, but shows largely indirectly through more mixing, or shows insignificantly compared to the effect seen before, also for EGR.

Figure 12 Impact of intake pressure on HC and CO emissions, as a function of CD. Operating conditions: 10 bar target IMEP, 55 wt% EGR (see online version for colours)



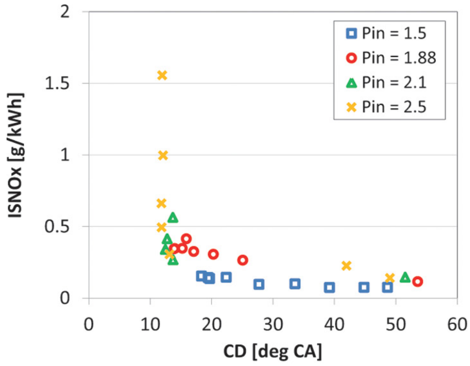
(a) ISHC emissions vs. CD (b) ISCO emissions vs. CD

3.3.3 NO_x and smoke emissions

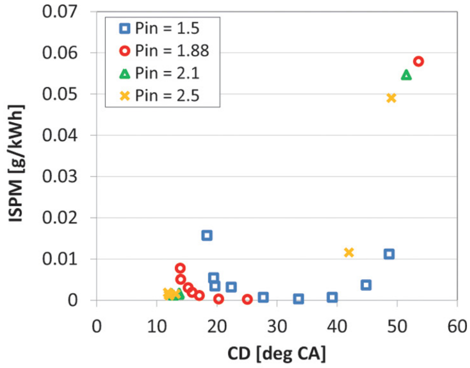
As ignition delay is increased into the PCCI regime, even slightly, NO_x levels are greatly decreased, because of a more premixed charge, see Figure 13(a). Only small differences can be seen between the different pressure levels, attributed to their respective oxygen concentrations.

Smoke emissions for both CDC and slightly premixed CI from Figure 13(b) can be seen to depend on the intake pressure and thus, the global equivalence ratio. One can clearly see that premixing greatly reduces smoke emissions levels and even at a CD of 30 deg CA, smoke emissions approach zero for all pressure levels. As CD is further increased, even lower combustion temperatures and the resultant lower soot oxidation (and possibly wall wetting) result in considerable smoke levels. It is, therefore, desirable to have the CD not excessively long, while still achieving the desired NO_x and smoke levels.

Figure 13 Impact of intake pressure on NO_x and smoke emissions. Operating conditions: 10 bar target IMEP, 55 wt% EGR (see online version for colours)



(a) NO_x emissions vs. CD



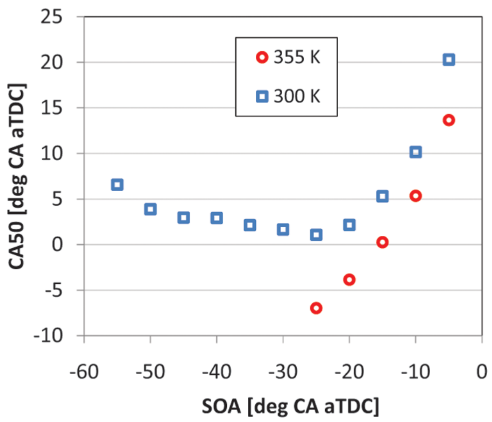
(b) Smoke emissions vs. CD

3.4 Air intake temperature

A higher intake temperature shows by and large the same effects as a higher intake pressure. Combustion is advanced significantly for the points under investigation, resulting in CA50 even before TDC (see Fig. 14), which results in unacceptably high pressure rise rates.

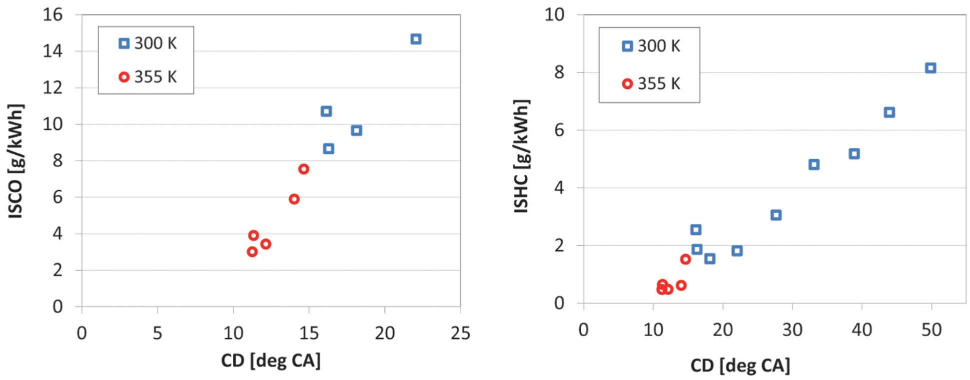
Load and fuel consumption, parameterised by IMEP and ISFC, are not directly affected by higher intake temperature. Indirectly, though, they are affected in two ways. First, as the higher temperature advances combustion to early timings, this results in higher fuel consumption. For the same CA50, however, a higher temperature level has a lower fuel consumption, caused by the shorter CD and more complete combustion associated with it. For compactness, graphs have not been included here, as they show exactly the same trends as presented above for intake pressure.

Figure 14 Impact of intake temperature on combustion phasing, as a function of SOA. Operating conditions: 5 bar target IMEP, 60 wt% EGR, 1.25 bar pin (see online version for colours)



The principal motivation for elevating intake temperature has been to reduce wall wetting. From the result, however, it can clearly be seen that CA50 is advanced by higher starting temperature and CD is thus shortened. HC and CO emissions benefit from the much shorter mixing time and higher combustion temperature, see Figure 15. For the points under investigation, no clear indication of (a reduction of) wall wetting can be seen and most CO and HC emissions are thought to be formed by cold zones in the combustion chamber. If wall wetting would be present, one could expect a sharp rise in UHC emission disproportional to CO emissions.

Figure 15 Impact of intake temperature on HC and CO emissions, versus CD. Operating conditions: 5 bar target IMEP, 60 wt% EGR, 1.25 bar pin (see online version for colours)



(a) ISCO emissions vs. CD (b) ISHC emissions vs. CD

A shorter CD, even if it is only slightly shorter, together with a higher starting temperature increases NO_x emission very significantly (not shown here), even at the high EGR rates used. It is, therefore, of great importance that the EGR gases used are cooled. Further, for particulate emissions, elevating intake temperature shows the same trends as elevating intake pressure did. For very short CDs, in the CDC regime, no significant effect of a higher temperature on the PM can be seen. It can only be noted that for a lower temperature, the ignition delay and thus the mixing time, is more easily increased. This significantly lowers the PM emissions.

3.5 Best points

All best points given below are for a compression ratio of 12:1. Even when NO_x and smoke limits are met for the best points below, an oxygen catalyst may have to be used to meet Euro VI limits on carbon monoxide and unburned hydrocarbon emissions. For the purpose of comparison, best points are defined in three ways:

Best conventional diesel combustion (CDC) point

In this conventional scenario, no emission levels are imposed. Best in this case is defined as the operating point where the Indicated Specific Fuel Consumption (ISFC), as based on the Gross IMEP, is the lowest.

Best point with after treatment

Assuming SCR efficiency to be a minimum of 90%, maximum NO_x levels may be 4 g/kWh to meet Euro VI levels post SCR. Furthermore, smoke emission may be 0.1 g/kWh if a passively regenerated DPF is used to further reduce smoke to Euro VI levels post DPF. The point with lowest ISFC, while still meeting these two targets, is taken as the best point for this case.

Best direct Euro VI point

This case is thought to be of importance during cold starts, for instance, where after treatment systems are not effective. The point with lowest ISFC where Euro VI levels (0.4 g/kWh for NO_x and 0.01 g/kWh for smoke) are met is declared to be the best point.

3.5.1 5 bar IMEP

In Table 2, the best points are given for the 5 bar IMEP case. For all points under investigation, smoke is not the limiting factor in meeting Euro VI levels, but NO_x is. Furthermore, the best ISFC is experienced when CA50 comes slightly after TDC. For the points where EGR is used, these gases are cooled to ambient conditions.

At this compression ratio, very high pressure rise rates are experienced for the CDC best point. Furthermore, one can note that the best ISFC is experienced when no EGR is used. Therefore, this is really conventional diesel combustion, although the ignition dwell is positive. This is attributed to the short injection duration for this load.

When using after treatment systems to meet Euro VI legislation limits, a medium EGR level of 40% is seen to give the lowest fuel consumption. This classic route to low temperature combustion uses the same injection timing and suffers a 1.2% penalty in fuel consumption compared to the best point where no NO_x limits have to be met.

As engine out NO_x levels have to be further reduced to directly meet Euro VI, even more EGR is used and the injection is somewhat advanced. CD is then only slightly longer, but long enough to meet the targets. Further, MPRR is effectively lowered. Notwithstanding the somewhat higher ISCO and ISHC, fuel consumption is nearly the same as with the higher NO_x limit. This implies a more efficient combustion, likely attributable to a more premixed, more constant volume combustion process.

Table 2 Diesel best points – 5 bar IMEP (standard deviation between brackets)

		CDC	*SCR + DPF*	*Euro VI*
Conditions	SOI [°CA aTDC]	–10	–10	–15
	Pin [bar]	1.24	1.27	1.28
	Tin [K]	298	303	304
	EGR [wt %]	no EGR	40%	61 %
Heat release	CA10 [°CA aTDC]	1.58	3.91	0.19
	CA50 [°CA aTDC]	3.42 (0.24)	5.45 (0.18)	5.3 (0.2)
	Ign dwell [°CA]	2.46	4.79	6.07
	Ign delay [°CA]	7.58	9.91	11.19
	CD [°CA]	9.42	11.45	16.3
	MPRR [bar/°CA]	43.9 (9.9)	26.1 (5.9)	14.0 (4.1)
Performance	Gross IMEP [bar]	4.93 (0.08)	4.86 (0.09)	4.88 (0.08)
	ISFC [g/kWh]	185.8	188.1	188.1
Emissions	Lambda [–]	3.83	2.44	1.70
	ISPM [g/kWh]	0.0001	0.0003	0.0001
	ISCO [g/kWh]	3.76	5.50	8.72
	ISHC [g/kWh]	0.72	1.46	1.88
	ISNO*x* [g/kWh]	16.42	2.61	0.16

3.5.2 10 bar IMEP

Like for the lower load, three 10-bar best points are established for 0, 40 and 60% EGR respectively, see Table 3. For the CDC best point, MPRR is excessively high again. In this case, because of the longer injection duration, ignition dwell is negative and thus, a large overlap between injection and combustion exists.

For the best point where after treatment is necessary, dwell is still negative, but combustion and ignition delay have already somewhat increased. NO_x is greatly decreased and even the HC levels drop. Although CO does not significantly differ, ISFC is still somewhat higher. For this viewpoint, MPRR still is too high for commercial use.

The best point, where Euro VI is directly met, uses an unconventional early injection timing and has a significant decoupling from injection and combustion. The maximum pressure rise rate has been further reduced, although it is still only just acceptable. Although HC is still low, CO is almost doubled and fuel consumption is 2.8% worse compared to the conventional regime. One should also note that for these operating conditions, the lambda is 1.15, which means stoichiometric combustion is approached.

4 Conclusion

In the PCCI regime, which is defined here as combustion after an increased mixing time with respect to conventional diesel combustion, NO_x and smoke have been shown to be efficiently reduced with this elongated mixing time. The lower combustion temperatures associated with a better mixed charge, though they help reduce NO_x, result in lower combustion efficiency, which can be seen in higher CO and HC emissions and fuel consumption. For viable PCCI combustion, one thus would require a CD which is long enough to bring both NO_x and smoke levels down to acceptable values. However, for the completeness of combustion, the resulting HC and CO emissions and the associated fuel consumption, mixing time should, however, be as short as possible.

Table 3 Best points – 10 bar IMEP (standard deviation between brackets)

		CDC	*SCR + DPF*	*Euro VI*
Conditions	SOI [°CA aTDC]	–10	–10	–25
	Pin [bar]	1.89	1.89	1.53
	Tin [K]	298	298	299
	EGR [wt %]	no EGR	38%	58%
Heat release	CA10 [°CA aTDC]	–0.15	1.57	–1.23
	CA50 [°CA aTDC]	5.75 (0.25)	6.32 (0.24)	1.39 (0.19)
	Ign dwell [°CA]	–4.31	–2.59	9.61
	Ign delay [°CA]	5.85	7.57	19.77
	CD [°CA]	11.75	12.32	22.39
	MPRR [bar/°CA]	41.1 (8.0)	26.8 (6.1)	19.9 (1.5)
Performance	Gross IMEP [bar]	8.93 (0.09)	8.82 (0.08)	8.70 (0.11)
	ISFC [g/kWh]	187.0	189.6	192.3
Emissions	Lambda [–]	3.23	2.05	1.15
	ISPM [g/kWh]	0.0003	0.0005	0.0032
	ISCO [g/kWh]	1.78	1.93	14.16
	ISHC [g/kWh]	1.25	0.63	1.07
	$ISNO_x$ [g/kWh]	17.46	2.10	0.14

For most parameters, a strong correlation with the combustion delay exists, independent of how this CD is achieved. The mixing time can be increased by increasing the EGR level, lowering intake pressure or temperature and advancing injection timing. While injection timing and intake pressure are easy to set in a modern heavy duty engine, in-vehicle-use issues are associated with lowering intake temperatures, especially at high levels of EGR, because of the large coolers necessary. While the desired CD may be relatively easy to achieve by different parameters, to meet the maximum pressure rise rates and to have as low as possible fuel consumption, combustion phasing has to be optimised.

Even when using a low compression ratio, while using a fuel as reactive as the current EN590 diesel, not every CD can be chosen, especially at higher loads. Even at low loads, very high EGR levels and low intake pressure and temperature are necessary to allow combustion to be phased correctly. The best points have shown that it is possible to achieve Euro VI NO_x and smoke levels with a fuel penalty of less than 3% compared to the best conventional diesel point. The combustion efficiency associated with these Euro VI points is only slightly worse, but at higher loads the, pressure rise rate reaches the limit.

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Nomenclature

aTDC	After top dead centre
CA	Crank angle
CAXX	Crank angle at which XX% of the fuel is burnt
CDC	Conventional diesel combustion
CI	Compression ignition
CO	Carbon monoxide
DI	Direct injection
DPF	Diesel particulate filter
EGR	Exhaust gas recirculation
CD	Combustion delay
EOI	End of injection (fuel delivery)
EPA13	North American legislated emission levels for HD as of 2013
Euro VI	European legislated emission levels for HD as of 2013
FSN	Bosch filter smoke number
HC	Hydrocarbons
HDDI	Heavy duty direct injection
IMEP	Indicated mean effective pressure
ISFC	Indicated specific fuel consumption
ISXX	Indicated specific XX emissions
LTC	Low temperature combustion
MPRR	Maximum pressure rise rate

Nomenclature (continued)

NO_x	Nitric oxides
PCCI	Premixed charge compression ignition
Pin	Intake pressure
PM	Particulate matter
PRR	Pressure rise rate
SCR	Selective catalytic reduction
SI	Spark ignition
SOA	Start of injection (actuation)
SOI	Start of injection (fuel delivery)
TDC	Top dead centre
Tin	Intake temperature
UHC	Unburned hydrocarbons