Low Temperature Combustion Optimization and Cycle-by-Cycle Variability Through Injection Optimization and Gas-to-Liquid Fuel-Blend Ratio

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The advent of common rail technology alongside powerful control systems capable of delivering multiple accurate fuel charges during a single engine cycle has revolutionized the level of control possible in diesel combustion. This technology has opened a new path enabling low-temperature combustion (LTC) to become a viable combustion strategy. The aim of the research work presented within this paper is the understanding of how various engine parameters of LTC optimize the combustion both in terms of emissions and in terms of fuel efficiency. The work continues with an investigation of in-cylinder pressure and IMEP cycle-by-cycle variation. Attention will be given to how repeatability changes throughout the combustion cycle, identifying which parts within the cycle are least likely to follow the mean trend and why. Experiments were conducted on a single-cylinder 510cc boosted diesel engine. LTC was affected over varying rail pressure and combustion phasing. Single and split injection regimes of varying dwell-times were investigated. All injection conditions were phased across several crank-angles to demonstrate the interaction between emissions and efficiency. These tests were then repeated with blends of 30% and 50% gas-to-liquid (GTL)-diesel blends in order to determine whether there is any change in the trends of repeatability and variance with increasing GTL blend ratio. The experiments were evaluated in terms of emissions, fuel efficiency, and cyclic behavior. Specific attention was given to how the NO_x–PM trade-off changes through increased injection complexity and increasing GTL blend ratio. The cyclic behavior was analyzed in terms of in-cylinder pressure standard deviation. This gives a behavior profile of the repeatability of in-cylinder pressure in comparison to the mean. Each condition was then compared to the behavior of equivalent injection conditions in conventional diesel combustion. Short-dwell split injection was shown to be beneficial for LTC, while NO_x was shown to be reduced by the substitution of GTL in the fuel. In-cylinder pressure cyclic behavior was also shown to be comparable or superior to conventional combustion in every case examined. GTL improved this further, but not in proportion to its blend ratio. [DOI: 10.1115/1.4024090]

Introduction

Traditionally compression-ignition combustion has been regarded as significantly more stable than spark-ignition, and as such it is not common to investigate trends in cyclic variability. Even so, cyclic trends are usually investigated in terms of cycle-by-cycle IMEP variance. Following work presented earlier [1] an investigation into the cyclic behavior of in-cylinder pressure during low-temperature combustion in diesel and increasing blends of gas-to-liquid and diesel fuel will be investigated.

Low-temperature combustion enables low to medium load engine conditions to be run with near-zero nitrous oxide (NO_x) and particulate matter (PM) emissions [2,3]. In LTC, the adiabatic flame temperature generally does not exceed 1800 K and so the Zel'Dovich mechanism (thermal mechanism) for NO_x production does not contribute to the emission generation as it is understood to have a very high activation energy (946 kJ/mol) [4,5].

LTC is inherently a complex combustion method to implement; it necessitates both low temperature and a low air-to-fuel ratio. The easiest way to attain both these conditions is with high levels of exhaust-gas recirculation (EGR) so as to drastically limit the amount of oxygen available to the fuel. Typical EGR rates necessary to attain LTC are between 55 and 65% by volume [3,6,7]

Barring the possibility of being able to accurately monitor the temperature of the combustion, denoting a combustion point as LTC has no rigid definition. Several markers denote LTC, however, their presence is not always observed; the most readily apparent marker is the existence of a cool-flame reaction in the heat release. The cool-flame reaction manifests as a low-peak rise in heat release at the start of combustion with a small plateau or depression before rising again to the full amplitude of heat release in the main power-producing combustion. It is not always apparent, particularly at low load conditions, and consequently cannot be insisted upon for every test-point. Another method to ensure LTC conditions is to progress along a series of tests, with a single variable increasing towards a condition correlated with LTC, typically EGR (as increased EGR increases the equivalence ratio) and observe a reversal in smoke emission trend. To wit, for a given test-point, increasing EGR will make FSN emissions increase progressively, until a break-point after which FSN will decrease dramatically. This represents a transition on the ϕ-T map "over" the high-smoke island and into the LTC region [7–9].

Experimental Methodology

Experiments were carried out on an externally boosted AVL single-cylinder research engine. Details are described in Table 1.

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Table 1 Engine specification

Bore × stroke (mm × mm)	85 × 90
Swept volume (cm^3)	510.7
Chamber geometry	Re-entrant bowl
Compression ratio	17.1:1
Swirl ratio	1.78
Induction	External forced aspiration
Number of injector holes	5
Injection angle	19 deg below fire deck
Nozzle hole diameter (mm)	0.18

The engine was controlled with an ETAS engine controller using INCA software to enable user-defined injection regimes. Tests were performed on user-specified single, and 50–50 split injections, in order to optimize conditions for low-temperature combustion.

The fuelling system for the engine included an automotive belt-driven fuel pump providing fuel at up to 1400 bar to a high-pressure common rail. The high-pressure rail was of an equivalent length and volume to that used in a four-cylinder automotive size diesel engine. The fuel was supplied to the injector through a thick-walled fuel line of approximately 40 cm length with an internal diameter of 2.75 mm. The only difference from a multicylinder engine was that the common rail supplied only this single line; all other fittings and controls were the same as for a multicylinder engine.

Fuel line and in-cylinder pressure (ICP) were all recorded over 200 cycles at 0.5 deg crank angle (CA) resolution. Preliminary tests (not shown) verified that sample sizes greater than 200-cycles yielded negligible increases in both the mean and the standard deviation (SD) of these parameters. The in-cylinder pressure was measured using a water-cooled flush-mounted piezoelectric pressure transducer (AVL QC 34-C). The pressure in the fuel line connecting the common rail to the injector was measured using a high-speed transducer (AVL SL 31-D2000), installed in the fuel line approximately 10 cm upstream from the inlet to the injector. High-frequency measurements of the pressure in the common rail were not recorded as part of the work reported here.

The in-cylinder pressure is used to calculate the indicated mean effective pressure (IMEP) and the heat release rate. This work reports the net IMEP, which includes the work done during the exhaust and intake strokes as well as the compression and power strokes. The heat release rate is calculated using the equation [4]

$$\frac{dQ_n}{d\theta} = \frac{\gamma}{\gamma - 1} p \frac{dV}{d\theta} + \frac{1}{\gamma - 1} V \frac{dp}{d\theta}$$

Where p is the cylinder pressure, V is the volume, θ is the crank angle, and γ is the specific heat ratio (presumed to be constant).

Injector opening current was also recorded along with high-speed data in order to tabulate the injection regime with the in-cylinder pressure.

Fuel consumption (FC) was measured using an AVL 733 Dynamic Fuel Meter; its values stripped of statistical irregularities via Chauvenet's criterion [10] and averaged over a 2 min period per setting, following a 5 min stabilizing time. Chauvenet's criterion was employed as the samples tended to have one or two excessively large spikes, often more than 20 times the value of the mean. The method removes any values that diverge from the mean by more than a factor of 2.81 times the standard deviation (for tests containing between 50 and 100 samples). As an indication, between zero and two values per test-point were removed from each data set out of a total of circa 90 points per sample.

The fuel injection event was controlled by the engine control system mentioned above. This system allowed fine control of the timing and quantity of up to four injections per cycle. The timing is set in terms of crank-angle phasing, and is not modified by the controller. However, the injection quantity command (in mg) is converted into an injection opening duration (pulse-width), based on a performance map from the engine supplier. This map adjusts the commanded duration on the basis of speed and fuel rail pressure only; it does not account for injection timing. In the case of multiple injections that are commanded to occur too closely together, the controller will reduce the injection duration to ensure that the minimum required injector dwell time (0.2 ms) is maintained.

Table 2 Set engine conditions

Engine speed (RPM)	1500
Engine load (bar IMEP-net)	~3
Intake pressure (bar)	1.20–1.21
Injection pressure (bar)	600/700

EGR is supplied through a cooled and heated pipeline drawing from downstream of the exhaust resonance tank and feeding in just upstream of the intake manifold. The EGR pipeline is externally controlled and has both adjustable back-pressure and an adjustable choke.

Previous work performed in [1,11–13] demonstrates that large injection events set in motion a stationary wave within the fuel line. Consequently, any subsequent injection events occur at a fuel pressure that may differ greatly from that demanded. This effect is of no consequence in single-injection tests; however, its existence necessitates an adjustment of the pulse-width in the second half of split-injection regimes in order to ensure the correct quantity of fuel is delivered. Set engine conditions are shown in Table 2.

Emissions are measured by a HORIBA MEXA7100HEGR analyzer drawing a sample downstream of the exhaust resonation tank via a filtered and conditioned pipeline. Measured were nitrous oxides (NO_x), carbon monoxide (CO), total hydrocarbons (tHC) and carbon dioxide (CO2). The latter is also measured via a separate channel drawing from the intake manifold in order to establish an accurate measure of the EGR rate. Particulate matter (PM) is measured with an AVL415 opacimeter and graded in the FSN scale. For each test-point four samples of PM were drawn and averaged over a two minute period.

Previous experimentation carried out by Cong and Sarangi [1–3] as well as the present authors provided the basis for the test-points examined. All test-points represent combustion regimes which are bordering on the minimum amount of EGR ensuring LTC conditions. This was done as limiting EGR is considered key to enabling LTC to become a commercially viable combustion strategy. The test strings are selected to portray a range of combustion phasing of a given fuelling quantity outputting a net of approximately 3 bar IMEP at 1500 rpm with an intake pressure of 1.20–1.21 bar. This is typical of low-load operating conditions and a prime example of engine conditions which commercially could be converted to run in LTC. It also represents an in-depth investigation into low-load LTC to correlate with work performed by Sarangi [3] at high load.

Three injection strategies are examined: Single injection, split injection with 7 deg dwell and split injection with 10 deg dwell. Earlier testing showed that dwell-times of 6 deg or shorter were not sufficiently long enough for the ECU to control and dwell times over 11 deg resulted in excessively high tHC and CO emissions coupled with low fuel efficiency, typical of high amounts of unburned fuel in the exhaust as the latter part of the split injection did not oxidize sufficiently. All tests were performed at 600 and 700 bar rail pressure. EGR rate is between 59–60% for the single-injection tests and between 52–53% for the split injection tests. For each test-string five test-points are examined, distinguished by their 50% burn point (CA50) and phased in roughly 2 deg CA increments. Adjustment of CA50 was achieved by shifting the entirety of the injection regime without adjusting the relative timings and durations of individual injection events.

In general terms, across the test strings, the earliest phased tests will demonstrate zero or near-zero PM but relatively high NO_x and the latest tests will border on exiting the LTC boundary by

Table 3 Fuel characteristics

Analysis type	Diesel	G30	G50
Density at 15 °C (kg/l)	**0.8312**	**0.8178**	**0.8101**
Cetane number (CN)	**59.6**	**68.7**	**68.5**
Calculated cetane index (CN)	**57.5**	**66.2**	**71.6**
Evaporated vol. at 250 °C (%)	24.3	22.1	18.9
Evaporated vol. at 350 °C (%)	97.0	96.2	96.2
Initial boiling point (°C)	169.7	183.2	193.8
50% recovery point (°C)	280.7	285.8	288.2
Final boiling point (°C)	354.1	355.4	354.4
PMC flash point (°C)	70	75	79
Kinematic viscosity at 40 °C (cST)	3.035	3.199	3.269

having PM emissions that ramp up sharply coupled to slightly lower NO_x.

In the latter tests, GTL-diesel blends were used. These blends were made a few days before experimentation. Blending was done volumetrically with an accuracy of circa 1% to ensure a near-perfect 30% and 50% by volume blend.

Whenever a switch in fuel type was performed, the engine was run for half an hour at approximately 4 bar load and 1800 rpm to ensure any remaining fuel was flushed out. The fuel can used during this time was then removed and disposed of as the fuel return-line would by then have diluted the blend with the residual fuel left over from the previous running. At that point, the system was flushed again and a new can of identical blend was connected and experiments run.

When performing tests of fuel blends, the fuelling system was short-circuited to bypass the fuel conditioning unit. Regardless, fuel temperature was measured immediately prior to entry into the low-pressure pump (a point which would naturally be downstream of the fuel conditioning unit) and was found to be approximately 2 °C higher than the conditioning unit's output, with a minor fluctuation of ±1.5 °C (whereas the conditioning unit usually outputs ±1.0 °C). This bypass was necessary as there may have been a lubricity issue with the fuel conditioner unit running the blends.

Physical characteristics of the pump-grade diesel, 30% GTL-diesel blend and 50% GTL-diesel blend are tabulated in Table 3: Fuel Characteristics. Cetane Number is the result of combustion in a Cooperative Fuel Research™ Engine as per ISO-5165. Cetane Index is a calculated estimation of Cetane using physical characteristics of the fuel.

LTC Emissions

In the interest of brevity, the most representative of test-strings have been tabulated: diesel, G30 and G50 single and 7 deg dwell split-injection (see Figs. 1 and 2). Several trends are immediately apparent: firstly, with the exception of the 7 deg split diesel tests, most PM emissions do not seem to follow the customary trend of reduction with higher injection pressure. This is likely due to the low load of the engine condition, as similar tests performed at 6 bar IMEP by Cong et al. [2,6] have shown to respond better to rail pressure.

PM also appears to ramp-up faster for increased GTL blend ratio, but has a lower minimum; while the diesel test-strings appear to plateau at a minimum of ~0.03 FSN, both the G30 and G50 tests observe 0.00 FSN at the earliest points.

Both of these effects are assumed to be due to the fuel's physical characteristics. GTL is slightly less dense than diesel, and therefore its spray atomizes better, reducing local rich pockets which correlate with high PM emissions. As the CA50 phasing gets later, the effect of higher Cetane index overtakes that of the lower density: a shorter ignition delay (and therefore less mixing time) increases PM as a higher percentage of the fuel is burnt in mixing-controlled combustion instead of premixed combustion, which is a more smoky combustion.

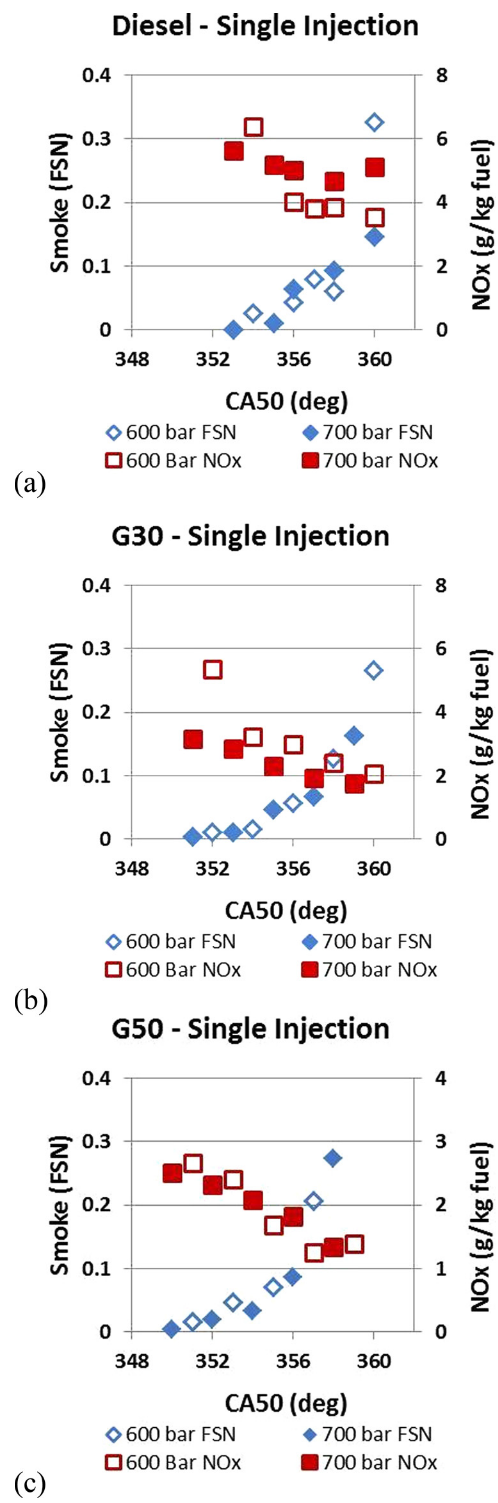
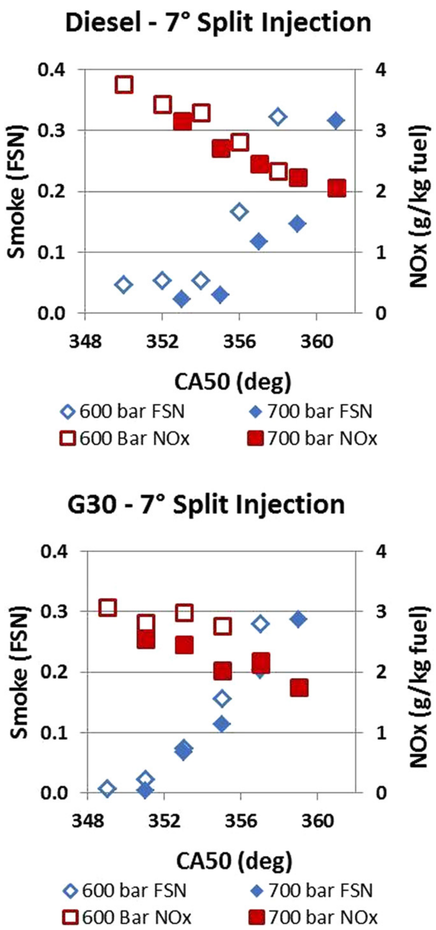


Fig. 1 NO_x and smoke emissions in single injection against CA50 phasing for diesel, G30 and G50

While PM is not significantly improved by the switch to split-injection regime NO_x is notably reduced (note that the NO_x scale for the single injection with diesel and G30 is doubled). For single injection LTC, NO_x is improved both for the G30 and the G50 fuel. Diesel single injection exhibits NO_x emissions roughly between 6 and 3 g/kg of fuel, while G30 and G50 range from 4 to 2 and 3 to 1 g/kg of fuel, respectively. The GTL blends have a shorter ignition delay thanks to their higher Cetane number and consequently have a slightly longer



(a)

(b)

Fig. 2 NO_x and smoke emissions in 7 deg split injection against CA50 phasing for diesel G30 and G50

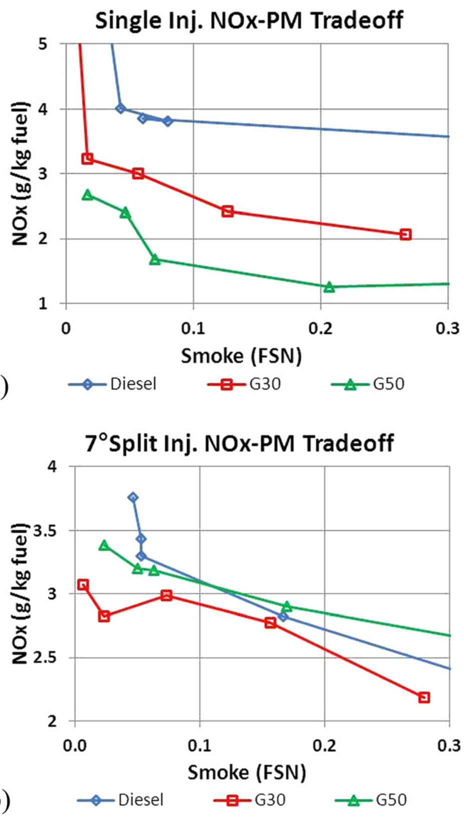


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Fig. 3 NO_x–PM tradeoff for 600 bar single and 7 deg split injection LTC (note different *y*-axis scale)

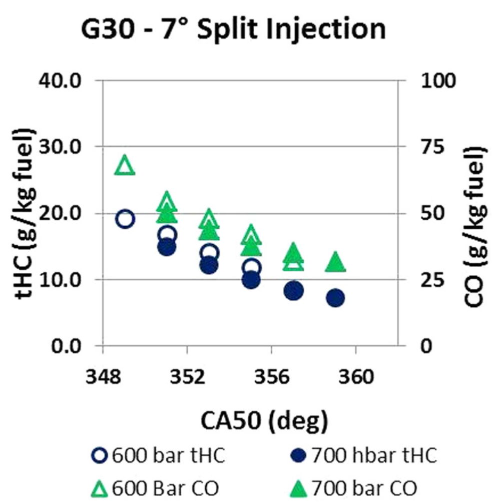

Fig. 4 tHC and CO emissions in 7 deg split injection with G30 across CA50 phasing

combustion duration, which leads to lower peak pressure and peak temperatures during combustion. Lower peak temperature inhibits NO_x formation via the Zel'Dovich mechanism. This makes the NO_x–PM trade-off significantly in GTL's favor, especially in the single-injection tests as can be seen in Fig. 3. Both cases of split injection did not show as clear an improvement in NO_x–PM trade-off as single injection with increasing GTL blend ratio.

There is some improvement between the total NO_x emissions of the G50 and the G30 blend, but it is not proportional to the amount of GTL. This suggests that whatever beneficial effect the GTL was having may be reaching a point of diminishing returns. NO_x also appears to be mostly unaffected by the rail pressures investigated (600 and 700 bar).

It is not clear why the improvement in NO_x with increase in GTL ratio is so much larger in single injection tests than in split injection tests. A possible explanation would involve spray over-penetration; being less dense, GTL blends naturally tend to penetrate less than pure diesel. Splitting an injection is known to minimize spray penetration (and possible wall-wetting), therefore it stands to reason that the single injection LTC benefits the most, in NO_x terms, from the switch to GTL blends.

CO and tHC emissions were very similar throughout all tests, reducing with later phasing of CA50. They follow NO_x trends, improving slightly with increasing GTL percentage and improving significantly transitioning from single to split injection. The single exception is the 10 deg split tests which had universally worse tHC and CO in particular as an amount of unburnt fuel was being exhausted.

Earlier CA50 tests observed increases in both emissions: in order to obtain earlier CA50, progressively more advance had to be demanded. Consequently, spray over-penetration (and consequent wall-wetting) may have been the determining factor in the increased emissions. This is further evidenced by reduced fuel efficiency: Early CA50 phasing exhibits indicated specific fuel consumption (ISFC) of 190–200 g/kWh while CA50 tests closer to top-dead center (TDC) improve to circa 180 g/kWh.

This deterioration in combustion efficiency could be partially mitigated with an optimized fuel delivery system like a shallower angle injector or a higher swirl ratio. It would also be possible to

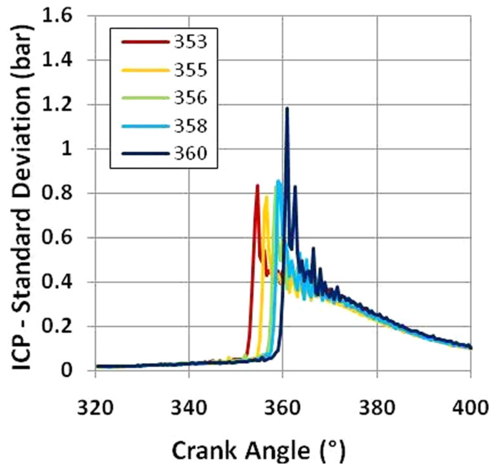

Fig. 5 ICP-SD for single injection at 700 bar rail pressure with diesel across CA50 phasing

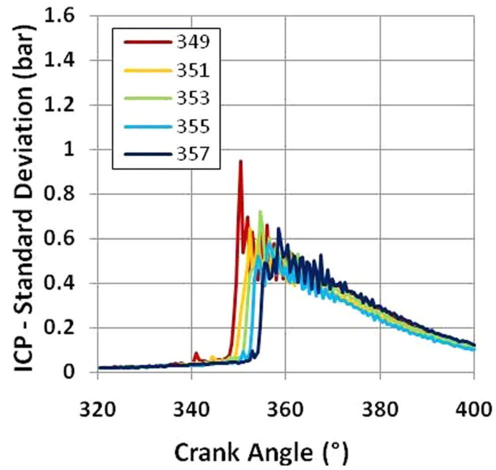

Fig. 7 ICP-SD for 7 deg split injection at 600 bar rail pressure with G30 across CA50 phasing

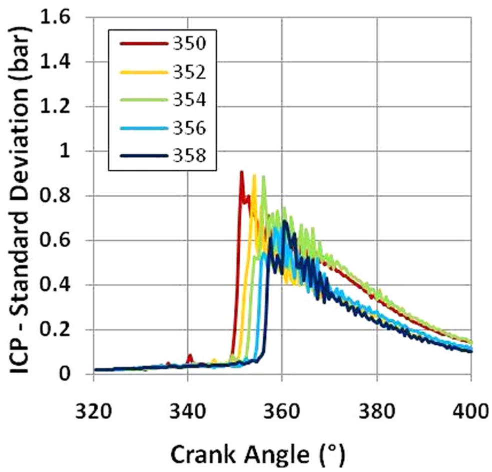

Fig. 6 ICP-SD for 7 deg split injection at 600 bar rail pressure with diesel across CA50 phasing

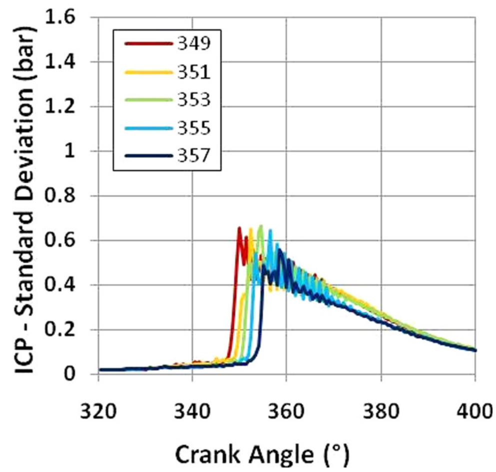

Fig. 8 ICP-SD for 7 deg split injection at 600 bar rail pressure with G50 across CA50 phasing

partially mitigate over-penetration with further segmentation of injection events into multiple injections. A typical graph of tHC and CO emissions through a test-string can be seen in Fig. 4.

Work performed by Musculus and Shinichi [14] suggests the increase in tHC in particular may be due to a portion of the fuel burning incompletely in the region near the injector, particularly just before or shortly after the end of injection events.

LTC Variability

In-cylinder pressure variability was demonstrated in previous work to exhibit some typical characteristics: it always rises quite sharply as soon as combustion commences, stays very high for a few crank-angles and then drops just as sharply to a plateau, at which it stays for most of the combustion before petering-off towards the end [1].

In order to understand this behavior, it is necessary to consider what exactly in-cylinder pressure standard deviation (ICP-SD) represents: high ICP-SD essentially means a reduced level of repeatability in in-cylinder pressure at that specific crank angle, on a cycle-by-cycle basis. Therefore, it is natural that the onset of combustion displays a sharp rise in instability: occasionally, a cycle will display an unusually early or late start of combustion compared to the mean. This means that at that specific crank angle, on some cycles the pressure will be identical to the motored pressure trace (as combustion may not have started), on some it will be moderately high (as combustion has just started), and on some it will be very high (as combustion will have already started a few degrees ago). This, in part, accounts for the very high spike in ICP-SD around the start-of-combustion point.

In the single injection scenario we observe trends similar to those expected in conventional combustion. There is nearly no effect on ICP-SD caused by the cool-flame combustion, an effect that is similar to the burning of a pilot injection in conventional combustion. When compared to similar tests performed in conventional combustion, single-injection LTC shows a significant decrease in ICP-SD. While single-injection conventional combustion of comparable engine speed and load demonstrates peaks of up to 1.6 bar, single-injection LTC has a highest peak test-point of 1.2 bar, with most other points peaking at 0.8 or below, making single injection LTC demonstrate ICP stability comparable to that of piloted injection regimes in conventional combustion. The test string with the highest peak ICP-SD is shown in Fig. 5.

Split injection tests were not too dissimilar in profile or peak, though their plateau lasts longer thanks to a longer combustion duration. As CA50 progressed closer to TDC in some cases the combustion of the first part of the split injection shows a shorter peak than that of the second part. This is logical: as injection occurs at a higher cylinder pressure there is less room for discrepancy in cycle-to-cycle variance of when combustion initiates. This can be observed in Fig. 6 in the 356 and 358 CA50 cases. Throughout, ICP-SD magnitude was globally reduced with

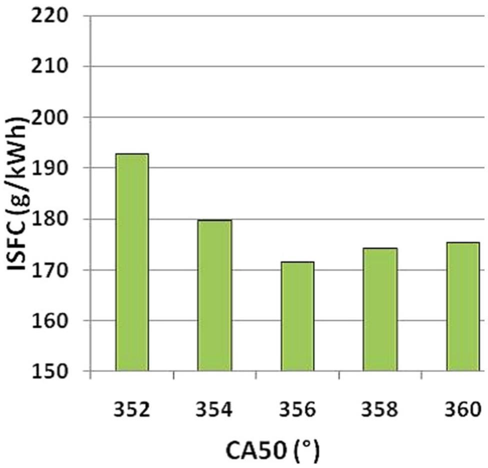

Fig. 9 ISFC for single injection at 700 bar rail pressure with G30 across CA50 phasing

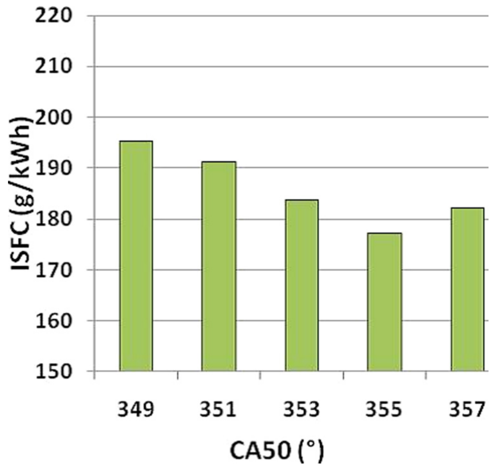

Fig. 10 ISFC for 7 deg split injection at 600 bar rail pressure with G30 across CA50 phasing

increased GTL blend ratio (Figs. 7 and 8). This suggests that GTL behaves in a more consistent way under LTC, with less cyclic variation.

As with the NO_x reduction, there seems to be little difference between the ICP-SD of G30 and G50 suggesting once again that a small amount of GTL may offer the majority of the fuel's benefits.

IMEP variability is highly consistent throughout LTC combustion regimes, showing no particular trend with respect to either injection regime or GTL blend ratio. This is to be expected to an extent, as the very stable ICP shows little variance, and by association IMEP shows high consistency. Typical values of IMEP coefficient of variation (IMEP-CoV) are between 1.8 and 2%, which is well within the limits of what is expected of modern automotive diesel engines [15].

There are no visible trends correlating ISFC with fuel composition. This may seem unusual as GTL has a slightly lower density than diesel, but its calorific value is slightly higher and the net effect is comparable efficiency [14,16–18].

There is an expected but minor trend of improved ISFC with CA50 phased closer to TDC, as mentioned before, as more of the combustion occurs at a higher cylinder pressure and is therefore more thermodynamically efficient.

ISFC is slightly worse when implementing 7 deg split injection but not significantly so (between 4% and 8% for equivalent CA50). It is presumed this is related to the broadening of the combustion duration: in order to achieve similar CA50, a much advanced first injection must be specified and so combustion starts earlier and ends later than an equivalent single injection scenario. This results in a percentage of the fuel being burnt at lower cylinder pressures and consequently reduces efficiency. ISFC for the 10 deg split injections was universally inferior to the 7 deg split by between 5% and 10%.

A typical example of this single and split injection ISFC is shown in Figs. 9 and 10.

Discussion

Using short-dwell split injection in lieu of single injection for LTC can achieve comparable or better emissions while requiring a usefully lower amount of EGR. Improved cycle-by-cycle variation in split injection renders it one of the smoothest and most consistent operating conditions possible. The potential for zero or near-zero smoke and NO_x emissions makes LTC a very likely candidate for implementation in commercial vehicles. High CO and tHC emissions may be problematic to address as the exhaust gas temperature during LTC may be as low as 150 °C, which may not be sufficient to light-off conventional diesel-oxidation catalysts.

Substituting GTL in diesel has been shown to have good results; significant reductions in NO_x coupled to similar or slightly reduced other emissions make GTL blends a good prospect for the future of LTC, provided the engine map is tailored to the fuel's characteristics. For the most part, GTL's benefits appear to be of most value at the 30% by volume substitution; the 50% blend did not demonstrate any further significant gains in comparison. GTL blends also showed increased cycle-by-cycle repeatability in LTC which compares favorably to conventional diesel combustion.

For all the emission advantages of short-dwell split-injection LTC, efficiency is slightly decreased. The effect of slowing the rate of heat release down and spreading it over a larger duration did not improve its specific fuel consumption. The trend for even worse fuel consumption exhibited by the 10 deg split tests does not suggest that further segmentation of the injection regime (into three or four parts) will do anything but worsen this trend. This does not mean to say that there may not be a more efficient complex injection regime; the effect of nonsymmetrical split injection was not investigated and may harbor a breakthrough in combustion efficiency. Perhaps increasing the ratio of fuel delivered in the first part of the split will partially mitigate the inefficiency by ensuring less fuel is being oxidized late in the cycle where nearly no remaining oxygen is available and the piston has commenced to depressurize the chamber.

From the perspective of minimization of fossil-fuel dependency, the research shows that natural gas can be a suitable part-substitute for diesel by refining it into GTL fuel and blending it with fossil diesel. The complexity of developing engines capable of running diesel-GTL blends is far lower than other methods involving natural gas in a gaseous state.

Conclusions

The investigation started out with the intention of determining the effect of implementing a split-injection regime in LTC in terms of emissions and cyclic variability, as well as the effect of significant percentages of gas-to-liquid fuel substitution in the fuel. The results lead to specific key conclusions:

(1) Implementing short-dwell split injection LTC improves NO_x significantly over single injection with a small loss in fuel efficiency.
(2) Using GTL blends in LTC improves NO_x emissions with comparable PM emissions. This improvement is more apparent in single injection LTC. The improvement is not proportional to the amount of GTL in the fuel: the 30% GTL blend performed nearly as well as the 50% blend.

(3) GTL blend's PM emissions are more sensitive to combustion phasing as the higher Cetane index reduces mixing time at a faster rate than diesel.
(4) CO and tHC emissions in LTC, while high, are improved by swapping from single to short-dwell split injection. Their consequent increase due to spray penetration could be mitigated or eliminated by the use of injectors and swirl valves specifically designed to avoid over-penetration. Increasing the dwell time between the injections deteriorates both emissions significantly as some of the fuel fails to oxidize.
(5) In-cylinder pressure standard deviation in single injection LTC scenarios compared favorably to conventional combustion.
(6) Split injection LTC is not significantly different in in-cylinder pressure standard deviation to single injection. While comparable in magnitude, they are different in profile, with split-injection occasionally demonstrating improved repeatability during the combustion of the first half of the injection charge.
(7) Increased GTL blend ratio improves in-cylinder pressure repeatability, though as with the NO_x reduction, little difference was visible between the 30% and 50% GTL blends.
(8) IMEP is highly stable in LTC and its stability is unrelated to GTL blend ratio or injection regime.
(9) ISFC in LTC is between 4% and 8% worse when implementing short-dwell split-injection throughout the fuels. It is not clear whether further segmentation of the injection regime into a more complex arrangement will overcome this, though it is unlikely as longer-dwell split injection is markedly worse.

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Nomenclature

LTC = low-temperature combustion
GTL = gas-to-liquid fuel
ECU = electronic control unit
CA = crank angle
CA50 = crank angle when 50% heat released
ICP = in-cylinder pressure
ICP-SD = standard deviation of in-cylinder pressure
TDC = top-dead center
ISFC = indicated specific fuel consumption
NO_x = nitrous oxides
PM = particulate matter (smoke)
tHC = total hydrocarbons
CO = carbon monoxide
IMEP = indicated mean effective pressure
IMEP-CoV = coefficient of variation of indicated mean effective pressure

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