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Development of a Fuel Injection Strategy for Diesel LTC

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ABSTRACT

A production V-8 engine was redesigned to run on low temperature combustion (LTC) with conventional Diesel fuel. Two fuel injection strategies were used to attain reduction in soot and NOx; a) early premixed injection strategy: fuel injected early during the compression stroke and b) late premixed injection strategy: fuel injected close to TDC with heavy EGR. The early premixed injection strategy yielded low NOx and soot but struggled to vaporize the fuel as noted in unburned hydrocarbons readings. The late premixed injection strategy introduced the fuel at higher in-cylinder temperatures and densities, improving the fuel's vaporization and limited the unburned hydrocarbon and carbon monoxide. The use of high EGR and high injection pressure for late premixed injection strategy provided sufficiently long ignition delay that resulted in partially premixed cylinder charge before combustion, and thereby prevented high soot, even in presence of high EGR. The engine operation was enhanced by a specially designed controller to balance all engine cylinders with the aide of in-cylinder pressure transducers. Under early premixed injection timings, where combustion phasing control was more challenging, the controller was responsible for reducing the soot in some cases by over 50%. The engine mapping was performed for a range of BMEP up to 7 bar.

The CHEMKIN simulation of the combustion process reproduced the experimental results, specially the start of cool flame reactions for the early premixed injection strategy. The simulations provided a useful development tool to frame the boundary conditions towards optimum ignition timing.

INTRODUCTION

The conditioning of lean, well premixed mixtures has come to be known as Homogeneous Charge Compression Ignition or HCCI and shown to yield low smoke and low NOx by creating conditions where locally rich fuel/air mixtures are not formed and combustion temperatures are low [1]. The lack of homogeneity has expanded the combustion treatment under a variety of names [2,2], but one or another has come to be known under the more general term of Low Temperature Combustion (LTC). The degree of desired homogeneity varies with the fuel type. In the case of two stage fuels that display cool flame chemistry such as Diesel, homogeneity is desirable to bring local equivalence ratios down. This is because fuel-rich regions will ignite early if in-cylinder temperatures run up, before top-dead-center (TDC), causing poor power output and large rates of pressure rise that can cause high combustion noise and damage the engine.

Two main variables are commonly used to provide for the LTC like conditions in a Diesel application namely EGR and high injection pressure. High EGR rates are capable to lower combustion temperatures to limit the NOx formation reaction which are triggered at around 2000K and can restrict the chemical reaction from polycyclic aromatic hydrocarbons to dry soot [4]. Improved mixture formations also contribute to lowering the local equivalence ratio and combustion temperatures. Mixing is enhanced by means of swirl, piston bowl geometry or injection characteristics – usually increased injection pressure. Both EGR and mixture formation become difficult to attain at higher loads and speeds. Furthermore, the LTC approach has been demonstrated in real engine light duty applications and has been successfully integrated into strategies to regenerate exhaust gas aftertreatment systems such as LNT and DPFs [4].

A significant challenge to LTC is the increased unburned hydrocarbons (UHC) emissions which are also associated with poor fuel economy. The sources for UHC are incomplete combustion arising from long mixing times (long ignition delays) due to local equivalence ratios (ϕ) below the lean combustion limit [5], from locally rich regions due to over fueling or injector dribble at the end of injection, and from decreasing in-cylinder temperatures as the piston moves away from top-dead-center. The local lean and rich ϕ regions have been identified with a process of over and under mixing respectively [6]. Quenching of the fuel-charge air mixture when coming in contact with the relatively cold surface of the piston or piston cylinder wall contributes to the UHC. Crevice volumes can trap fuel which is later released as UHC in the exhaust stroke [7]. The latter two sources are especially relevant in the case of Diesel fuel, which is characteristically difficult to vaporize at low temperatures. Yet when attempting to create a homogenous charge, the injection timing is typically advanced, thus the risk of impinging fuel on the cylinder wall and diluting the lubricating oil.

A second challenge for the application of LTC is the controllability of the ignition timing [8]. This is particularly challenging in a Diesel application as well because a Diesel mixture tends to ignite at relatively low temperatures (or too early in the compression stroke) and too abruptly. Extending the operating region to high loads require the introduction of techniques to lower compressed gas temperatures by means of EGR, variable valve actuation (VVA) to adjust the intake valve close time to adjust effective compression ratio or preferably yet a variable compression ratio (VCR) device to adjust the geometric ratio.

This paper addresses the UHC and ignition control challenges in a production engine platform by studying two distinct fuel injection strategies. The early premixed strategy applied by means of a multi-hole / multi-row injector to yield optimum NOx and soot emissions but generally in excess of UHC. The second approach was the late premixed injection strategy consisting of an injection closer to TDC, governed by criteria to limit UHC and competitive fuel economy. The injection timing resulted in identifying the temperatures wherein the fuel was vaporized and sufficient ignition delay was allowed for the fuel to mix with the charge air to limit both NOx and soot. The work scope herein will extend in time to introduce the aforementioned VVA and VCR techniques to extend the operating regime.

ENGINE SETUP

ENGINE CONFIGURATION - The base engine was the International 6.4L V8 engine used in medium duty trucks and school buses. The engine was modified as described in Ref [9] and is summarized in Table 1. Here the base engine data is compared with present build.

Table 1 Test Engine Specification

	Base engine	Test Engine
Displac. Bore Stroke	6.4L 98.5mm 105mm	6.4L 98.5mm 105mm
FIE	DI Common Rail	DI Common Rail
CR	16	12-16.5
Turbo Charger	Single Stage VNT	Dual Stage VNT
EGR system	HP loop Single Cooler	HP loop Dual Cooler
IVC EVO	-133 BTDC 132 ATDC	-133 BTDC 132 ATDC

The engine was tested with a range of compression ratios (CR) ranging from 12 to 16.5. The injectors tested encompass the base injector and multi row - multi hole injectors, with hole diameter sizes ranging from 100 to 180 µm and number of holes up to 20. The criteria for evaluating the injector characteristics consisted in maximizing the homogeneity of fuel and charge air mixture. The methodology is illustrated in Figure 1. Here the local equivalence ratio is illustrated by color contours over three crank angles, 310, 330 and 350º (360º being TDC). The injection timing was set at 270º. As noted earlier multiple hole diameters were simulated. Shown here are 111 and 157µm. The contours are quantified with probability distribution functions. The narrower the distribution the homogeneity is better. The tests here used the 111µm injector. Further details on how the present hardware configuration was selected are contained in the above reference.

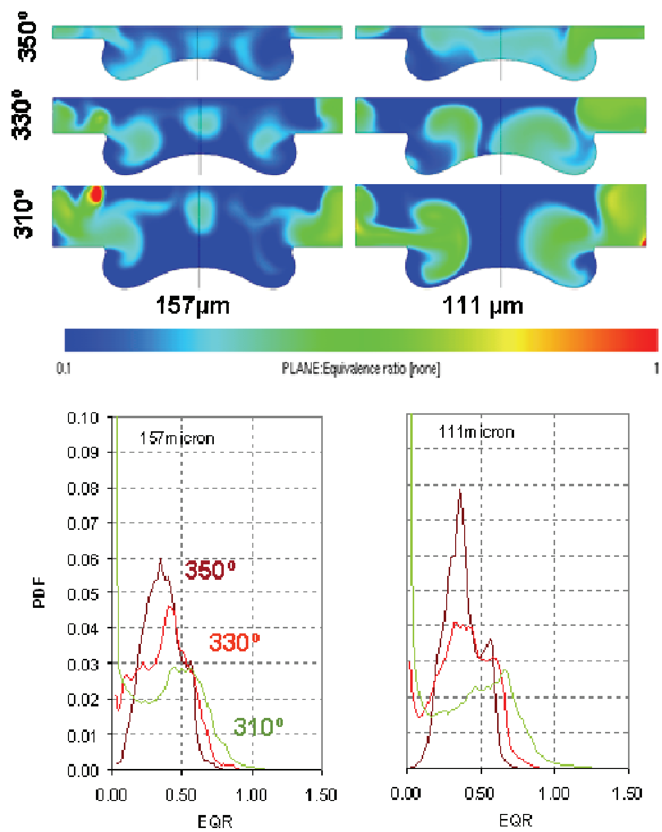

Figure 1 Bowl details of CR14 used in the experiments

Engine Design - The engine layout was accomplished by a procedure illustrated in Figure 2 encompassing (1) defining the boundary conditions to sustain low temperature combustion, (2) defining the engine torque curve, (3) necessary engine hardware, (4) optimized fuel injection equipment, and (5) set up of control supervisor. The boundary conditions primarily target levels of equivalence ratio (Φ) between 0.3 and 0.4 and EGR levels between 40-50%. The timing of the injected fuel was early so as to attain a homogenous charge, yet was constrained to maximize vaporization which depends on the temperature of the charge air in the cylinder.

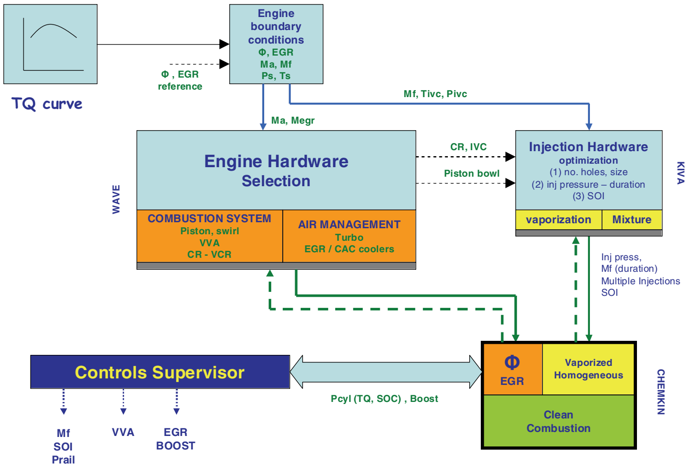

Figure 2 System Optimization

Experimental Layout - The engine layout and experimental setup is shown in Figure 3. The engine air system incorporated a heater system in parallel to the charge air cooler and was capable to adjust the intake manifold temperature by adjusting two bypass valves. The heater was used primarily to see the sensitivity of ignition timings to temperatures as illustrated later in this paper. The fuel in this study proceeded from three batches of fuel with tight control over the cetane number. The present results correspond to a cetane number of 48.

The total hydrocarbons were measured via a heated flame ionized detector, CO and CO2 via a non-dispersive infrared analyzer, and oxides of nitrogen with a heated chemiluminescense detector. The smoke was measured by means of a classical filter paper method with resolution of 0.01 FSN or soot concentrations of 0.1mg/m3. The cylinder pressure was measured with a piezo type transducer and was used to calculate cycle indicated torque and the heat release trace used later to determine the ignition timing.

EXPERIMENTAL RESULTS

The benefit of early premixed injection strategy was readily apparent in low NOx, soot emissions owing to well premixed characteristics. The challenges of such a strategy however was avoiding fuel impingement and improved vaporization of the Diesel fuel; factors which lead to poor fuel economy and high levels of HC emissions. The lack of vaporization was overcome by delaying the injection timing closer to TDC, where the in-cylinder temperatures were higher. The late premixed cases reduced HC emissions while still yielding low NOx and soot emissions. These cases are discussed next.

EARLY PREMIXED INJECTION CHARACTERISTICS

The early premixed combustion was attained by shifting fuel from a conventional combustion provided by single shot injection at near TDC to an earlier timing (typically around 60°BTDC). This is illustrated in Figure 4 and Figure 5. Three distinct stages are highlighted next.

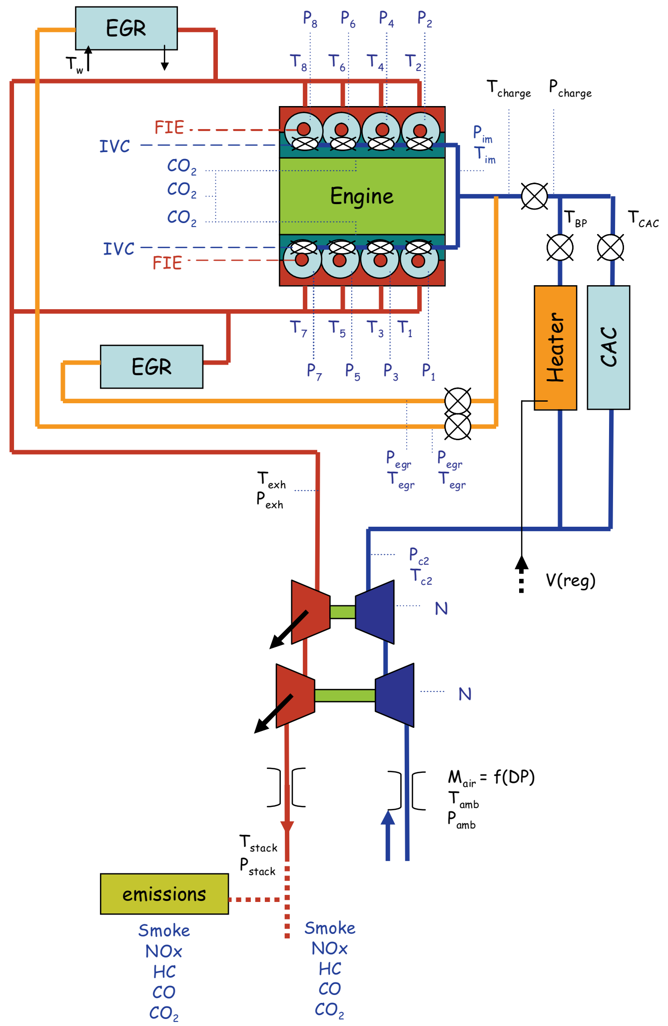

Figure 3 Experimental Layout

A: Conventional combustion: This is the initial condition when the engine was operating in the conventional combustion mode, with the injection timing configured to obtain a CA50 of 5°ATDC.

B: Intermediate stage: During this stage a small fraction of fuel was injected as pilot fuel (usually at 60°BTDC) while a second injection event was still configured close to TDC. The main injection timing was adjusted wherever possible to obtain a CA50 of 5°ATDC. Similarly the main injection quantity was also adjusted by the rapid prototype engine controller to keep the engine torque of the intermediate stage same as the conventional combustion. The points B, B1 show the emission history at the intermediate points as the pilot quantity was continuously increased (the main injection quantity was also continuously decreased to maintain the same torque as conventional combustion).

C: Early premixed injection: During this stage all the fuel was injected as the pilot fuel very early during the compression stroke at 60°BTDC.

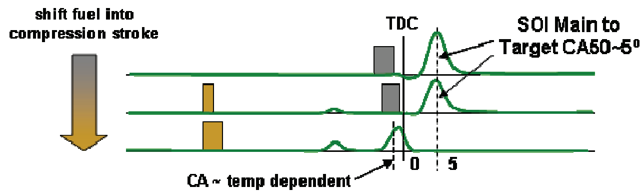

Figure 4 Fuel injection shift procedure from conventional to early pre-mixed

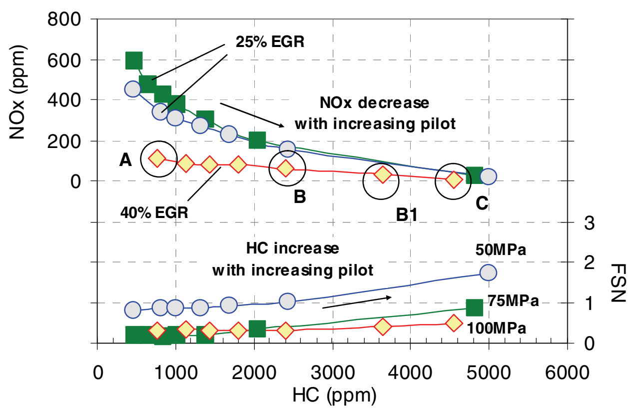

Figure 5 Fuel distribution impact on performance, N= 1000rpm, IMEP=4.5-4.7bar

The results of NOx and soot are shown for varying EGR levels and injection pressures with respect to UHCs in Figure 5 at a representative speed and load. EGR was effective in lowering NOx, whereas injection pressure lowered soot, with exception of very early injections (as the cylinder pressures are smaller, high injection pressure will cause the spray to over penetrate into the chamber). The corresponding heat releases are illustrated in Figure 6. For cases A, B the main injection event is capable to retain the combustion phasing, whereas in case C, where all fuel was introduced early on the compression cycle, the temperature determined the combustion phasing, which in this case shifts shortly before TDC.

Cylinder Pressure Feedback - The engine control module was substituted by a Rapid Prototype System (RPS) with complete authority over all engine actuators including the injector drivers. A unique part of the control system was the cylinder pressure feedback. Each cylinder was equipped with a pressure transducer. The pressure signal was acquired at a 0.5º crank angle resolution over 620º, nearly the entire engine cycle. A dedicated board was used to acquire the data for all 8 cylinders and was made available to the main processor unit. The main processor then calculated cylinder torque, heat release, and estimated the start of combustion and 50% burnt mass fraction (CA50) for each cylinder before the next cycle. The main processor was then programmed to be able to correct both fueling amount and injection timing to each cylinder to yield similar combustion traces.

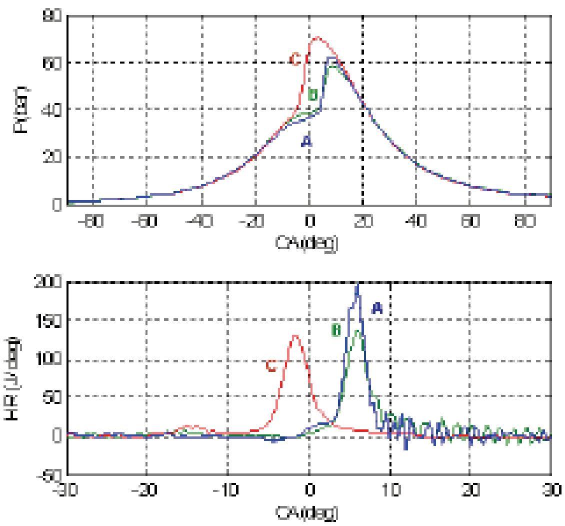

Figure 6 Pressure and heat-release rates showing the transition from conventional to LTC with early premixed injection strategy

The procedure was applied to conventional as well as LTC like combustion strategies. For very early injection timings the adjustments were limited to trimming the fuel amount (fueling amount correction only. No timing corrections were applied for the case of early premixed injection strategy) for equalizing the indicated cylinder torque. The application to early premixed injection strategy is illustrated in Figure 7. The large variation of the open loop system was largely corrected. In this particular case most of the variation is appreciated in the hot reaction as seen in the peak of the second heat-release curve.

Impact on emissions – Figure 8 shows the impact of the pressure feedback system applied to conventional and LTC like conditions. The impact over a conventional injection strategy is shown in the high NOx points, where the NOx is lowered from 3.3 to 3.1 g/kW-hr, with no impact on smoke. Without changing the engine boundary conditions, injection timing was adjusted to early in the compression stroke. The lean and homogenous conditions provided by the early injection yielded a 94% drop in NOx, though soot rises, owing to the variability captured in the heat release traces of Figure 7. The impact of the close loop control was considerably greater for the LTC conditions. Soot was reduced from 0.08 to 0.03 g/kW-hr (around 50% soot reduction). The EGR used for all data points shown here was 20%.

THE LATE PREMIXED INJECTION STRATEGY

The lack of vaporization associated with early premixed injection strategy was overcome by delaying the injection timing, closer to TDC, where the in-cylinder temperatures were higher. The late premix reduced the HC emissions while still yielding low NOx and soot emissions.

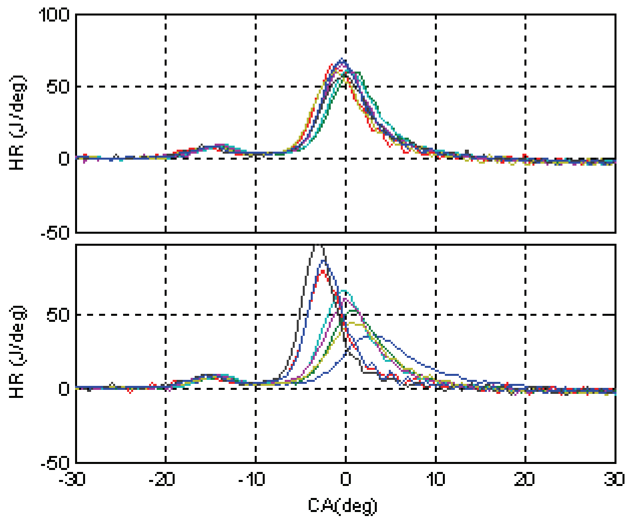

Figure 7 Fuel-trim applied to balance cylinder under HCCI like operation (N=1000rpm, iMEP=3.3bar)

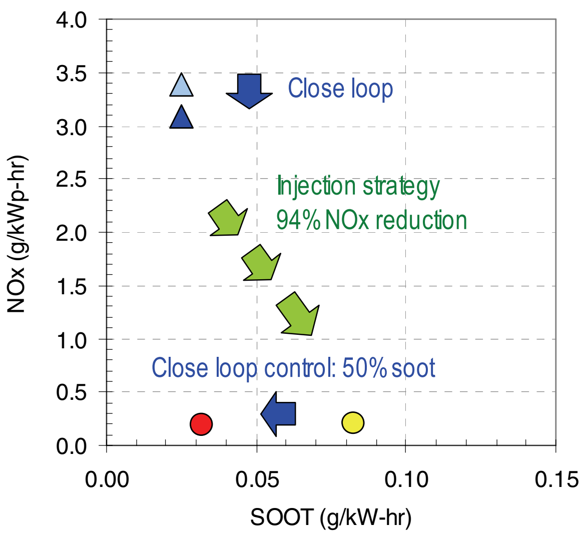

Figure 8 Impact of fuel distribution and cylinder trim on NOx-FSN trade-off (N=1000rpm, iMEP=3.3 bar)

Optimization of SOI and EGR – The effect of SOI at a fixed EGR level was considered, with the expected reduction of hydrocarbon emissions as the timing was retarded towards TDC. The capability to vaporize the Diesel fuel was improved against elevated temperatures. The impact is illustrated in Figure 9. At 55% EGR HCs was reduced from 5000 to 1800ppm. Within this range, NOx was kept at or below 15ppm. The soot was below 0.5 FSN, yet not sufficiently low to indicate a fully homogenous mixture. The effect of EGR was captured in points A and B of the same figure. For these points the CA50 is maintained at 5º ATDC (SOI: 11.0BTDC & 6.BTDC). The HC levels drops below 1000ppm at the loss of NOx which climbed over 150ppm, an order of magnitude above point B. The pressure curves and corresponding heat release traces for these points are included in Figure 10. The heat release traces illustrate the increased ignition delay contributing to the low NOx and effectively lower combustion temperatures seen in the cylinder.

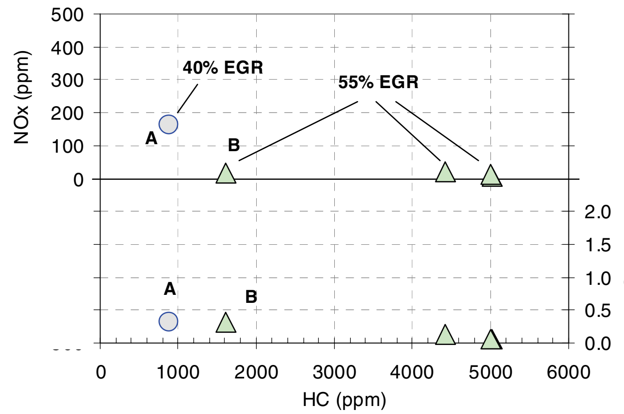

Figure 9 Impact of optimizing SOI on engine performance at N=1000 rpm, iMEP=4.7bar

The ignition delay was estimated from the end of the rate of injection profile to the beginning of the hot flame chemistry obtained from the heat release trace. Other definitions consider the delay from the start of injection command, but the present definition was preferred as it relates better to an actual delay. For the 55% EGR case the delay was approximately 7º or 1.2 milli-seconds. Current testing needs to be expanded to fully understand the relation of this delay time to conditions of low temperature combustion. Generally, a delay time of 1.2 to 1.5 milli-seconds seems to be a necessary threshold to allow for adequate mixing.

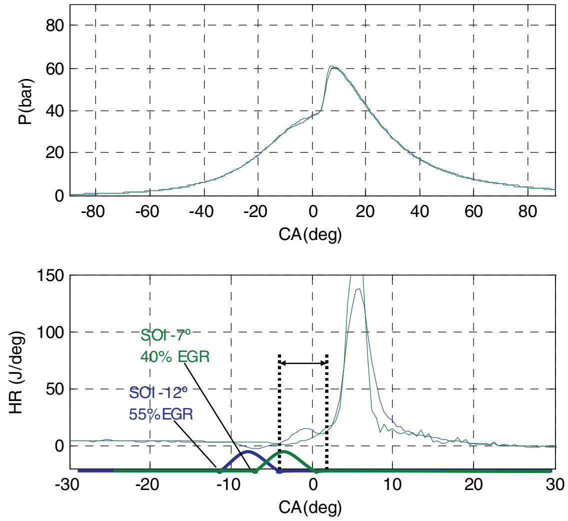

Figure 10 Pressure and heat-release accompanying Figure 9

Other Effects – In addition to injection timing, the mixture preparation leading to low temperature combustion was attained with lean type conditions. The effect of equivalence ratio or air to fuel ratio is illustrated in Figure 11. The effect of injection timing is coupled with increased boost showing a soot reduction of over 1 FSN when increasing the boost by 15kPa. The soot drops further from 2 to 0.25 FSN with an additional boost of 50kPa. The corresponding heat release traces designated A to B are shown in Figure 12.

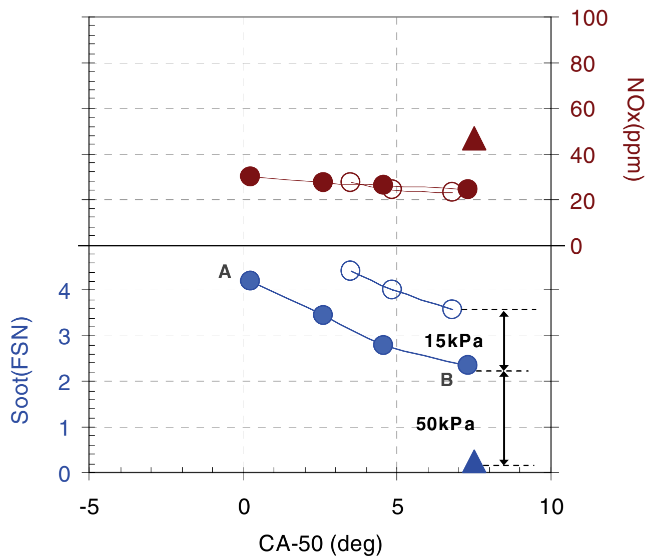

Figure 11 Effect of injection timing and boost, N=1200rpm, iMEP=7.2 bar, EGR=46%

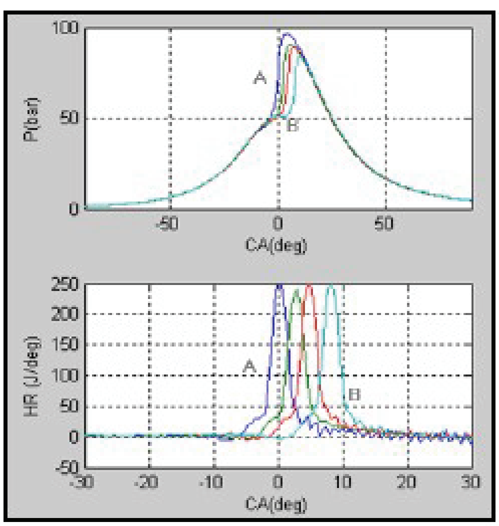

Figure 12 Effect of injection timing on heat-release rate (constant boost)

In addition to boost, a second major effect was provided by the engine compression ratio. Two different bowls illustrate the impact over the unburned hydrocarbons. A decrease in compression ratio by 2 yielded a reduction of 1000 ppm through the range of injection timing tested. The NOx was maintained in the range of 20 to 50 ppm. The injection timing and the resulting ignition delay (ID) however have a strong impact on soot, on the order of 1 FSN per 5º of ignition delay.

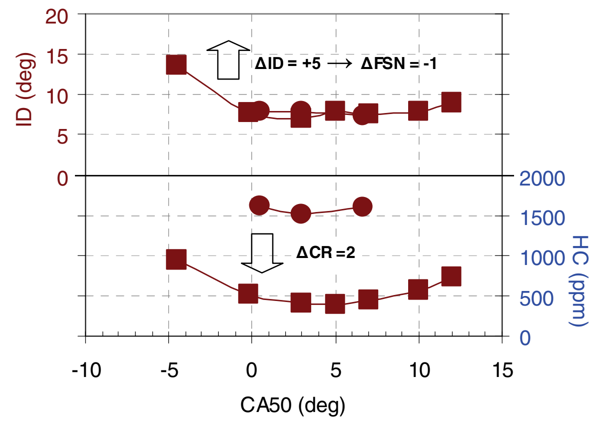

Figure 13 Effect of compression ratio, N=12000rpm, iMEP=7bar, 46%EGR

PREDICTIVE SIMULATION

The computational investigations were conducted using the Senkin application of the CHEMKIN-III kinetics rate code [10] with the full chemistry mechanisms for nheptane from the Engine Research Centre at University of Wisconsin [11]. The engine specifications used for the simulation are shown in Table 1.

The performance and emission predictions for HCCI and LTC combustion use models of varying degrees of complexity. The simplest models use the simplified single-zone model of ref [12,13]; detailed ones use the fluid mechanics-multi-zone chemical kinetics model of ref [14,15]. The single zone model assumes that the cylinder charge in the combustion chamber forms a completely homogenous mixture with a uniform temperature, pressure and composition. Since the single zone model does not take into account the effect of temperature and air-fuel ratio inhomogeneities, all the cylinder charge ignites nearly instantaneously when the ignition temperature (for high temperature reaction) is reached. Therefore, a single zone model under-predicts the burn duration, and over-predicts peak cylinder pressure, the rate-of change of pressure and NOx emissions. This type of a single zone analysis has shown to predict the start of combustion with reasonable accuracy and has been used extensively in parametric studies to understand the effects of boundary conditions such as EGR, compression ratio and equivalence ratio [16]. For the present paper only the SOC values (cool-flame reactions) based on the simulation are relevant rather than the predicted pressure and the bulk temperature values.

The CHEMKIN simulations are compared with the experimental cylinder pressure data for the case of single injection at 60º BTDC. For this case of very early injection, a sufficiently long ignition delay was available to produce a lean homogenous cylinder charge for HCCI type of combustion. The comparison of the experimental and CHEMKIN cylinder pressure show an agreeable match for the start of combustion (as detected by the deviation from the motoring pressure trace) for low-temperature reaction as illustrated in Figure 14. The simulated pressure and bulk temperature profiles show the cool flame reaction at 340º, quite nearly as those shown in the experimental data.

Since the CHEMKIN showed an acceptable trend, CHEMKIN simulations were used for parametric studies to investigate the effect of the two important HCCI boundary conditions namely EGR and intake manifold temperature.

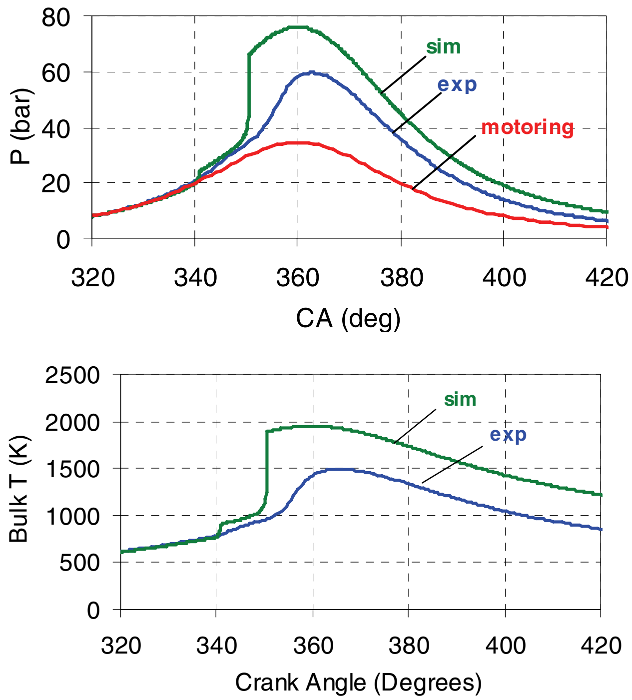

Figure 14 CHEMKIN simulation at no EGR, N=1000rpm, iMEP=3.3 bar

The application of simple kinetics to predict the start of combustion was studied in cases where the fuel is nearly homogenous with the charge air. A parametric effect of intake-temperature and EGR was investigated on the start of combustion. Figure 15 shows a sweep of intake manifold temperatures corresponding to 33, 44 and 55ºC. These conditions were attained by using the CAC/heater bypass shown earlier in Figure 3. The top of the graph is experimental data of pressure and the calculated heat release. The bottom plot shows the estimated bulk temperature using experimental pressure data and the gas composition; it is superimposed with the temperature calculated by the CHEMKIN kinetics prediction. The experiment and prediction show an offset over the start of cool flame and hot flame reactions; the shift in start of combustion introduced by the intake temperature manifold is well captured here however as shown in the figure.

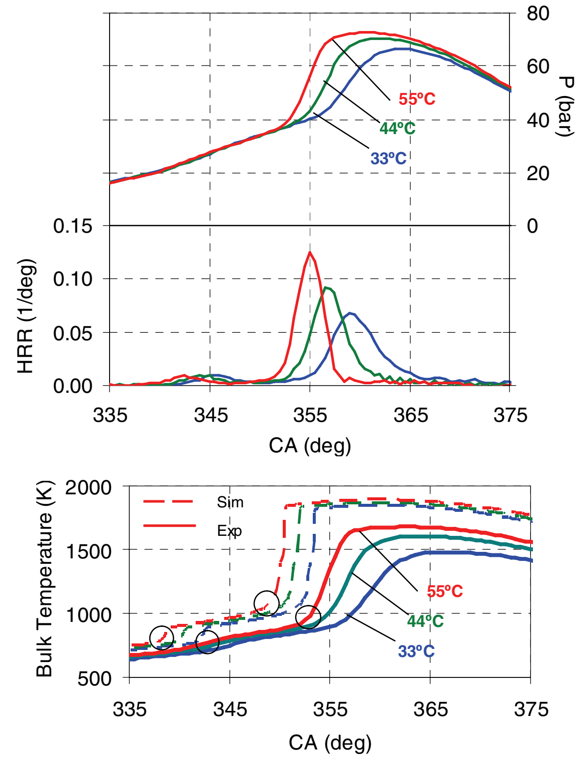

Figure 15 Intake manifold temperature sweep and effect over pressure and heat release with estimated and predicted bulk temperatures. N=1000rpm, iMEP=3.3 bar

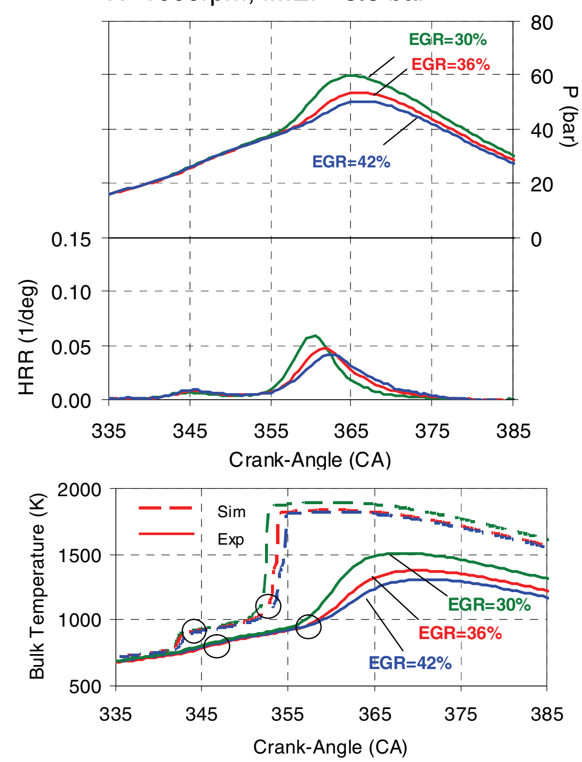

Figure 16 EGR sweep and effect over bulk temperature, N=1000 rpm, iMEP=3.3 bar

Data corresponding to EGR of 30, 36 and 42%, illustrated in Figure 16, shows very similar trends correlating well with the start of combustion of the engine data. The start of combustion was based on bulk temperatures calculated from the experimental pressure traces and from the chemical kinetic model. The simplified model yields values within 3° degrees of the experimental data.

Based on the kinetic simulations contour plots of Figure 17 can be generated for the start of combustion for a given compression ratio and boost. These contour plots provided an estimate of the boundary conditions in terms of EGR, equivalence ratio and intake temperature required for the desired combustion phasing. For instance, according to the simulations a minimum EGR of 55% with an intake temperature of 310~320K is required to for close-to-TDC combustion phasing.

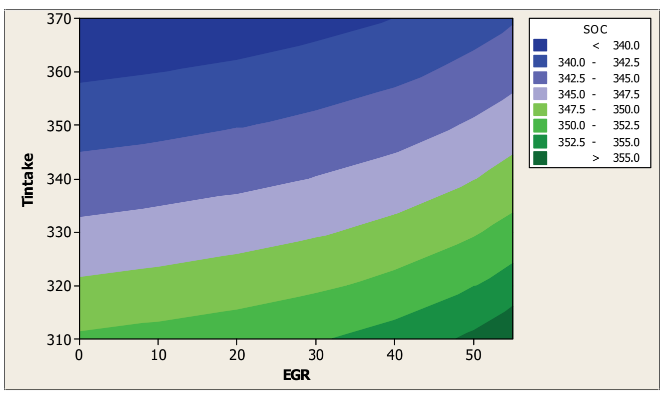

Figure 17 Contour plot for SOC as a function of T_{intake} and EGR (CR=15:1)

Auto-ignition temperatures - The effect of the engine's intake manifold temperatures, or temperatures at intake valve closing (IVC), on the motoring in-cylinder temperatures is shown in Figure 18 and it is compared against the range of the cool flame temperature of auto-ignition for Diesel fuel, approximately 700K, and the band of vaporization temperatures, approximately 310°CA.

The two bands highlight (I) the limit for injection timing to avoid fuel impingement, a limit that may be increased partially with multiple injections, and (II) the limit of intake temperatures to avoid early auto-ignition, a limit that may be extended by lowering the compression ratio or the IVC timing. The result is the identification of an operating region (III) to contain the combustion process

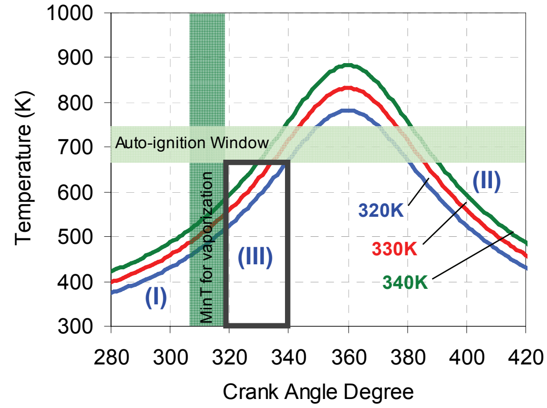

Figure 18 Compression temperatures for varying intake temperatures corresponding to N=1000rpm, iMEP=5bar, and Φ=0.4.

CONCLUSIONS

Low NOx and soot emissions were attained with varying degrees of pre-mixed fuel and charge air conditions. The present work shows the impact of the injection timing on the pre-mixed characteristics as illustrated in the traces of combustion heat release along with impact on unburned hydrocarbons.

As the injection timing was delayed, the fuel mixture favored better vaporization which limited the production of unburned hydrocarbons. The mixture necessary to maintain a range of ignition delay was assured with appropriate levels of EGR and equivalence ratio defined by boost. Such a combination provided a burn angle optimum for power output, approximately corresponding to a CA50 between 3-10º ATDC, and an ignition delay of 1.2~1.5 milli-seconds for sufficient mixing.

Under near-homogeneous conditions, the estimates of start of combustion using a simplified CHEMKIN mechanism proved to be accurate, predicting the cool flame reactions. These predictions coupled to the vaporization characteristics of the fuel provided estimates of injection timing window.

Low temperature combustion like conditions rely on heavy EGR, which coupled with uneven distributions on port geometry and flows or uneven cooling, can see significant variation on the combustion process from cylinder-to-cylinder. The application of a close loop control system based on in cylinder pressure measurements was found to be effective in compensating for the lack of distribution, with significant impacts on emissions. The improvements were more significant with early injection timings, where a soot reduction on the order of 50% was observed.

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