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Exploring a Gasoline Compression Ignition (GCI) engine concept

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Exploring a Gasoline Compression Ignition (GCI) engine concept

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ABSTRACT

This report describes initial engineering and experimental steps to assess the potential to improve efficiency and reduce both regulated and CO_2 emissions, and maintain acceptable noise performance by running gasoline in an advanced diesel engine. An engineering paper study was carried out to analyse critical engine and fuel parameters and judge what speed/load range might be feasible for a Gasoline Compression Ignition engine concept.

Using an advanced diesel bench engine having a higher compression ratio, optimised valve timing, and flexible fuel injection, the engine could be operated on a European market gasoline over full to medium part loads. The combustion was found to be highly sensitive to exhaust gas recirculation (EGR) rates, however, and the simultaneous optimisation of all regulated emissions and combustion noise was a considerable challenge. An advanced glow plug was tested to improve low load performance but did not extend the engine operating range as much as expected. From a commercial perspective, it is well understood that there are significant challenges associated with bringing both a new engine concept and a dedicated fuel into the market at the same time. Although compression ignition (CI) using gasoline was not successful in this study, the potential benefits of fuelling advanced compression ignition engines with market gasoline merited further consideration for the following reasons:-

First, CI engines have a clear efficiency advantage over spark ignition (SI) engines and extending their capability to use a broader range of fuels could be advantageous. Second, the ability of CI engine concepts to use an already available market gasoline would allow these concepts to enter the fleet without fuel constraints. Third, more gasoline consumption in passenger cars would help to rebalance Europe’s gasoline/diesel fuel demand on refineries and reduce GHG emissions from fuel supply. Fourth, a successful new CI vehicle of this type could potentially compete in predominantly gasoline markets in other parts of the world.

Computational fluid dynamics and KIVA simulations were completed on the same single cylinder bench engine configuration operating on market gasoline to identify ways of improving low load performance. This modelling has shown that variable valve timing offers considerable potential for increasing the temperature inside the combustion chamber and reducing the ignition delay. The simulations have also identified the preferred placement of combustion assistance, such as a glow plug or a spark plug, to extend the operating range and performance on gasoline, especially under the lowest load and cold engine starting conditions

KEYWORDS

Gasoline, compression ignition, EGR, computational fluid dynamics, variable valve timing, simulations

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SUMMARY

Future vehicles will increasingly be required to improve their efficiency, reduce both regulated and CO_2 emissions, and maintain acceptable levels of driving, safety, and noise performance. To achieve this high level of performance, they will be configured with more advanced hardware, sensors, and control technologies that will also enable their operation on a broader range of fuel properties. These capabilities offer the potential to design future vehicles to operate on the most widely available and GHG-reducing fuels.

In previous studies, fuel flexibility has been demonstrated on a compression ignition bench engine and vehicle equipped with an advanced engine management system including closed-loop combustion control. An unresolved question is whether engines of this sort can operate routinely on market gasoline while achieving diesel-like efficiency and acceptable emissions and noise levels.

This report describes initial engineering and experimental steps to assess this potential. Using an advanced diesel bench engine having a higher compression ratio, optimised valve timing, and flexible fuel injection, the engine could be operated on a European market gasoline over full to medium part loads. The combustion was found to be highly sensitive to EGR rates, however, and the simultaneous optimisation of all regulated emissions and combustion noise was a considerable challenge. An advanced glow plug was tested to improve low load performance but did not extend the engine operating range as much as expected. From a commercial perspective, it is well understood that there are significant challenges associated with bringing both a new engine concept and a dedicated fuel into the market at the same time. Although CI using gasoline was not successful in this study, the potential benefits of fuelling advanced CI engines with market gasoline merited further consideration for the following reasons:-

First, CI engines have a clear efficiency advantage over SI engines and extending their capability to use a broader range of fuels could be advantageous. Second, the ability of CI engine concepts to use an already available market gasoline would allow these concepts to enter the fleet without fuel constraints. Third, more gasoline consumption in passenger cars would help to rebalance Europe’s gasoline/diesel fuel demand on refineries and reduce GHG emissions from fuel supply. Fourth, a successful new CI vehicle of this type could potentially compete in predominantly gasoline markets in other parts of the world.

To identify ways to improve the lower load performance of a GCI engine concept, computational fluid dynamics (CFD) and KIVA simulations were completed on the same single cylinder bench engine configuration operating on market gasoline. This modelling has shown that variable valve timing offers considerable potential for increasing the temperature inside the combustion chamber and reducing the ignition delay. The simulations also identified the preferred placement of combustion assistance, such as a glow plug, to extend the operating range and performance on gasoline, especially under the lowest load and cold engine starting conditions. The placement of a spark plug was also simulated requiring a wider nozzle cone angle in combination with an optimized nozzle tip protrusion to produce the optimum spray pathway to allow a more fuel rich mixture in the area above the spark plug. The CFD studies, therefore, suggest that it may be possible to achieve lower loads using the GCI concept and future work is planned to carry out engine testing to investigate further.

1. INTRODUCTION

As pollutant emissions from motor vehicles continue to fall to meet lower regulated emission limits, attention is increasingly focused on vehicle efficiency and on fuel consumption to address future concerns with energy supplies and transport's contribution to greenhouse gas (GHG) emissions. Engine, aftertreatment, and vehicle technologies are evolving rapidly to respond to these challenges.

Considerable research is now concentrating on improving the combustion performance of light-duty engines. Compared to spark ignition (SI) engines, compression ignition (CI) engines are already very efficient so the challenge is to maintain or improve CI engine efficiency while further reducing pollutant emissions. Engines using advanced combustion technologies are being developed that combine improved efficiency with lower engine-out emissions, thus reducing the demand on exhaust aftertreatment systems and potentially on vehicle costs. Because these concepts combine features of both SI and CI combustion, the optimum fuel characteristics could be quite different from those needed by today's conventional gasoline and diesel engines [1,2,3,4].

In general, these advanced combustion concepts substantially homogenize the fuel-air mixture before combusting the fuel under Low Temperature Combustion (LTC) conditions without spark initiation. These approaches help to simultaneously reduce soot and NOx formation [5,6]. A literature review [7] has found that there are now a significant number of advanced combustion variations that provide lower engine-out emissions (especially NOx and particulate matter (PM)), lower fuel consumption (comparable to or better than today's CI engines); and stable engine operation over a wide load range.

Light-duty diesel engines are well suited for such advanced combustion because the higher fuel injection pressures, exhaust gas recirculation (EGR) rates, and boost pressures that aid conventional CI combustion also enable future variations of advanced combustion. In addition, the duty cycle of light-duty diesel engines emphasizes lighter loads where advanced combustion is most easily achieved. Many of the necessary hardware enhancements exist today although they may be expensive to implement in production engines. Nonetheless, advanced combustion engines are rapidly moving from research into engine development and commercialisation.

To achieve a nearly homogeneous fuel-air mixture, fuel is commonly injected very early in the engine cycle to provide sufficient time to achieve thorough fuel-air mixing. Although this achieves good fuel dispersion, it also makes it difficult to control the start of the autoignition process as the engine power increases. For this reason, most studies now favour fuel injection later in the engine cycle in order to retain most of the benefits of good fuel dispersion and achieve better control of the ignition process [8,9]. Using this approach, low engine-out emissions can be achieved especially at lower engine loads. The first production engines are therefore expected to operate in a premixed combustion mode at lower loads, reverting to conventional diesel or gasoline operation at higher load conditions [10]. As long as this is the case, fuels must be compatible with both engine operating modes.

Previous engine and vehicle tests in this series [11,12,13] have shown that Low Temperature Combustion (LTC) can be achieved on a surprisingly wide range of fuels using a CI engine designed for diesel fuel. In the same studies, however, European market gasoline proved to be too resistant to ignition to operate satisfactorily using a compression ratio suitable for diesel fuel. Others have worked with US market gasoline at lower CRs with some success [14].

From a commercial perspective, it is well understood that there are significant challenges associated with bringing both a new engine concept and a dedicated fuel into the market at the same time. So, although CI using gasoline was not successful in our previous study, the potential benefits of fuelling advanced CI engines with market gasoline merited further consideration for the following reasons:-

First, CI engines have a clear efficiency advantage over SI engines and extending their capability to use a broader range of fuels could be advantageous. Second, the ability of CI engine concepts to use an already available market gasoline would allow these concepts to enter the fleet without fuel constraints. Third, more gasoline consumption in passenger cars would help to rebalance Europe's gasoline/diesel fuel demand on refineries and reduce GHG emissions from fuel supply. Fourth, a successful new CI vehicle of this type could potentially compete in predominantly gasoline markets in other parts of the world.

Because of these potential benefits, it was decided to investigate more completely the 'gasoline compression ignition' (GCI) engine concept, specifically to determine over what range of conditions an engine could operate successfully in CI mode on a European market gasoline. In addition to an engineering paper study and a bench engine study on the GCI concept [15], computational fluid dynamics (CFD) in-flow and combustion simulations were also carried out [16].

2. ENGINEERING PAPER STUDY

2.1. ANALYSIS OF CRITICAL PARAMETERS

An engineering paper study was first completed to analyse critical engine and fuel parameters and judge what speed/load range might be feasible for a GCI engine concept. The following sections summarise the key learnings from this engineering study. Following this analysis, bench engine results are then presented describing how these learnings were applied to a practical test of the GCI engine concept on a single cylinder CI engine.

For this engineering study and for the bench engine work that follows, it was assumed that the GCI engine concept would be fuelled with a typical European market gasoline. A single batch of European reference fuel containing 5% v/v ethanol (E5) was purchased for the bench engine study and the target and measured properties are shown in **Appendix 1**. This gasoline (CEC RF-02-08) was used in Europe to complete type approval emissions testing on Euro 4 gasoline passenger vehicles and is still the major gasoline in the market today. The gasoline was treated before use with a commercially available ester-type lubricity additive.

2.2. BASIC ENGINE REQUIREMENTS

Beginning with the fuel injection requirements, current gasoline fuel injection systems operate at pressures around 200 bar and would need considerable modification to produce the pressures required for CI combustion. Even with the increased volatility of gasoline compared to diesel, it is expected that fuel injection pressures up to 1000 bar will be needed. Fuel injection equipment for diesel fuel applications is therefore still the best choice, using optimised injectors and perhaps with larger nozzle holes. The injector design may also need to be adapted especially to provide consistent fuel injection at the anticipated lower injection pressures. The lower injection pressures should beneficially reduce pump power demand. For example, reducing the rail pressure from 1460 to 900 bar was calculated to reduce power demand by about 100W.

The lubricity of market gasoline is not adequate to protect today's fuel injection components, so either the engine components must become more robust or the routine use of fuel lubricity additives will be needed. Because of gasoline's higher volatility, a GCI vehicle would also need to be equipped with an evaporative emissions control system.

Smoke and NOx emissions can be expected to be lower on a GCI engine compared to a comparable engine using diesel fuel. From earlier tests using a diesel engine [11,12], engine-out NOx emissions could be maintained at or below the levels needed to meet Euro 6 emissions limits for passenger cars on all but the lowest CN fuels. Although lower engine-out NOx emissions could allow some simplification of the aftertreatment system, retaining a DeNOx catalyst could enable easier control of engine transients and allow better engine efficiency calibration. A future GCI vehicle would need an efficient oxidation catalyst, because emissions of HC and CO are usually higher under LTC conditions compared to normal diesel combustion. A gasoline particulate filter (GPF) would also be needed although lower engine-out PM emissions would lengthen the GPF regeneration interval, saving as much as 1.5% in fuel consumption compared to a DPF-equipped diesel vehicle.

Engine efficiency should be the same as for a diesel CI engine, as long as the combustion phasing is maintained at the optimum point. Previous engine studies [12,13] have demonstrated that this is feasible on a wide range of fuels using closed-loop combustion control (CLCC). Even with a single gasoline fuel, CLCC would probably be needed to accommodate the high sensitivity of ignition delay to the fuel's autoignition characteristics as well as the engine's production tolerances and ambient conditions.

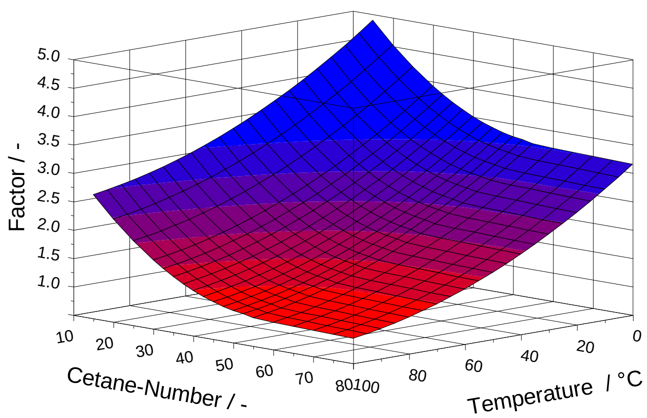
It is well-known that gasoline is naturally resistant to autoignition, so the engine will need to provide suitable fuel/air mixing and temperature conditions if stable combustion is to be achieved. Two stage boosting will be needed to give high charge pressures and keep cylinder temperatures high. High EGR levels will also be required to reduce combustion temperatures and NOx emissions. Variable Valve Timing (VVT) is expected to be needed to provide internal EGR, supplemented by sufficient external EGR.

2.3. IGNITION AND COMBUSTION CHALLENGES

The major challenge for GCI is gasoline's very low Cetane Number (CN) which is usually estimated to be no higher than about CN15. The ignition process was therefore studied in some detail using a simplified model for ignition delay based on temperature in the intake manifold, the cylinder pressure at the beginning of the main fuel injection and the O_2 concentration in the charge gases. A very low CN implies much longer ignition delays. In an engine, this means that a low CN fuel must be injected earlier in the compression stroke when temperatures and pressures are lower than during typical diesel fuel injection. These conditions make achieving ignition even more difficult. Ignition is also retarded at lower temperatures and at the lower O_2 concentrations resulting from the high levels of EGR required to reduce NOx emissions.

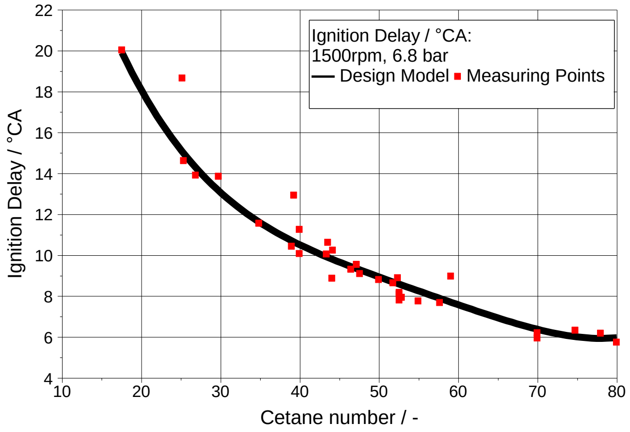
The simplified ignition delay model was initially developed for diesel fuel combustion in a warmed-up engine [12], but was modified by a factor (**Figure 1**) to account for the increased ignition delay seen at lower engine temperatures and with different CN fuels. For example, the ignition delay is expected to lengthen by a factor of three as the engine coolant temperature decreases from 100°C to 0°C, and by at least a factor of two for market gasoline (about CN15) compared to diesel fuel.

Figure 1 Modification of the ignition delay model for different cetane numbers and combustion temperatures



This simple model excludes many complexities of the ignition process, but nevertheless describes well the earlier engine test results on a demonstrator engine (both test bench and vehicle) using a range of fuels (**Figure 2**) [12,13].

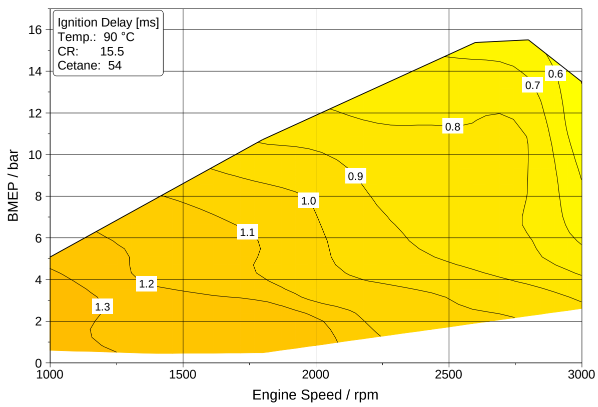
Figure 2 Predicted ignition delay versus CN at 1500 rpm/6.8 bar IMEP



Engine parameters were based on a demonstrator vehicle equipped with FEV's High Efficiency Combustion System (HECS [9]). This engine has a 1.6-litre 4-cylinder engine with a specific power of 80kW/litre, a 2-stage boosting system, hot and cold EGR, an exhaust cam phaser and CLCC. In a previous study [13], using a compression ratio of 15.5:1, this vehicle was able to achieve around 130g/km CO_2 emissions at a 1700 kg curb weight while achieving Euro 6 NOx levels without aftertreatment.

Calculations based on European diesel fuel and a baseline CR 15.5 showed that ignition delays would be below 1.5 ms at an engine coolant temperature of 90°C, even down to the lowest engine speeds and loads (**Figure 3**).

Figure 3 Calculated ignition delays for European diesel fuel at various engine speeds and loads, CR 15.5, and a 90°C engine coolant temperature

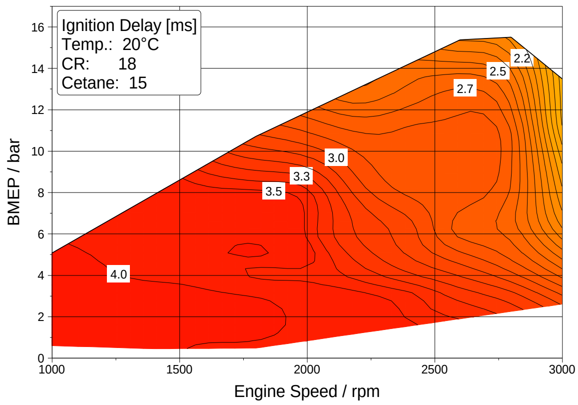


For a coolant temperature of 20°C, representing starting under mild ambient conditions, ignition delays were longer, ranging from around 1.5 ms at higher engine speeds, to just over 3 ms at the lowest speeds and loads. For the evaluation of gasoline ignition, a target of no more than 3 ms ignition delay was set. A crosscheck with engine data showed that calculated cylinder pressures gave similar results to measured pressures, so calculated pressures were used in the remainder of the study.

With the baseline CR 15.5, calculations for European gasoline at a 90°C engine coolant temperature showed that ignition delays would be more than 3 ms for a large part of the operating range, generally below 2000 rpm and 8 bar BMEP. Increasing the CR reduced ignition delays, until at CR 18, ignition delays less than 3 ms were predicted over the entire operating range. Increasing the CR to 19:1 and 20:1 gave further improvements in the calculated ignition delay.

At a coolant temperature of 20°C, however, calculated ignition delays were more than 3 ms over a large part of the speed/load range (**Figure 4**). This was the case at CR 18 and increasing further to CR 20 did not help significantly.

Figure 4 Calculated ignition delays for European gasoline at various engine speeds and loads, CR 18, and a 20^{o}C engine coolant temperature



It was clear, therefore, that some form of combustion assistance (e.g. glow plug or spark plug) would be needed to support combustion under cold engine conditions and perhaps also at light loads even with a warmed-up engine. Other options to more rapidly warm-up the engine might also be beneficial, such as by-passing the intercooler and EGR cooler.

2.4. ADVANTAGES AND DISADVANTAGES OF A GCI CONCEPT

The efficiency of a GCI vehicle is expected to be about the same as for a diesel engine. The additional energy consumption needed to support sustained glow plug operation would be approximately offset by savings from lower fuel injection pressures and longer DPF regeneration intervals. Importantly, replacing conventional gasoline vehicles by GCI vehicles would reduce the overall fleet fuel consumption and CO_2 emissions. Additionally, a successful commercial development of GCI vehicles could open the way for more efficient vehicles in markets where diesel fuel is not widely available for passenger car applications.

The cost of a GCI engine should be similar to a modern diesel engine, with potentially lower cost for the fuel injection and aftertreatment hardware if high vehicle sales are achieved. Compared to a conventional gasoline engine, GCI engines would produce high torque, similar to a diesel engine, but would require more development time to achieve good noise and smoothness.

The fuel consumption benefits for consumers are more complex. Because of its lower density, gasoline has a higher volumetric fuel consumption than diesel at the same engine efficiency. Comparing typical conventional gasoline and diesel vehicles at 1500 kg curb weight, we can estimate the following figures (**Table 1**).

Table 1 Fuel consumption comparisons for different engine types

Engine Type	MJ/100km	gCO_2/km	Litres/100km
Gasoline NA	267	196	8.29
Gasoline DI-TC	225	165	6.99
Diesel CI	210	154	5.85
Gasoline CI	210	154	6.52

NA = naturally aspirated
DI = direct injection
TC = turbocharged
CI = compression ignition

Volumetric fuel consumption for the GCI engine would be higher than for a diesel CI engine, but would be about 7% lower compared to a Gasoline DI-TC engine. CO_2 emissions (on a gCO_2/km basis) would be at about the same low level as a diesel CI engine[1]. Clearly, customer choice can be influenced by fuel prices. Unlike today, when gasoline is taxed higher in many European countries compared to diesel fuel, harmonisation of fuel taxes to an energy content basis would be expected to level the playing field somewhat.

[1] gCO_2 emissions per MJ of fuel differ by only 0.26% for average European gasoline and diesel fuels [22]

3. BENCH ENGINE SPECIFICATIONS

A previous study [11] investigated how engine hardware capabilities could reduce engine-out emissions while maintaining acceptable engine efficiency and noise levels. In this study, four diesel and kerosene fuels were tested with engine calibrations optimized for each fuel. In a companion study [12], a more diverse set of fuels was tested on one of the two engine configurations optimized in the first study.

In these studies, it was assumed that future production engines will use a diesel oxidation catalyst (DOC) to mitigate HC/CO emissions and a diesel particulate filter (DPF) to mitigate PM emissions. To control NOx emissions, an active DeNOx system, such as selective catalytic reduction (SCR) or NOx storage catalysts, will preferably be avoided for passenger cars due to their complexity and additional cost. To achieve such a simplification, however, sufficiently high levels of EGR must be used to achieve very low engine-out NOx emissions. The diesel bench engine was optimized based on these considerations.

The success criteria for the bench engine optimization included the following factors: low engine-out NOx; PM, HC, and CO emissions as low as possible and suitable for further reduction by DOC and DPF aftertreatment systems; acceptable engine noise; and fuel efficiency at least as good as the base engine configuration (CR 15 and HFR 310 injectors) when operating on diesel fuel. The specifications for this bench engine are shown in **Table 2**.

Table 2 Specifications for the compression ignition bench engine

	Units	Engine
Benchmark	[-]	Euro 6
Displacement	[cm^3]	390
Stroke	[mm]	88.3
Bore diameter	[mm]	75
Compression ratio	[-]	17:1 / 19:1
Valves per cylinder	[-]	4
Maximum peak pressure	[bar]	220
Fuel injection system specifications	[-]	Bosch Piezo Common Rail System
Maximum injection pressure	[bar]	2000
Hydraulic Flow Rate (HFR)	[cm^3/30s]	310 / 520
Nozzle hole diameter	[µm]	109 / 141
Number of spray holes	[-]	8
Spray Cone Angle	[°]	153
Charging	[-]	Max. 3.8 bar absolute

The bench engine included hardware enhancements that will enable Euro 6 emissions limits and beyond to be achieved. A comparatively small cylinder swept volume of 390 cc was used, so this concept is consistent with engine downsizing. This swept volume would allow the construction of a 1.6L 4-cylinder engine while maintaining the power of today's 2.0L engines.

The compression ratio in this study was varied from 17:1 to 19:1 by adjusting the volume of the ω-type piston bowl. The maximum cylinder peak pressure of this engine is 220 bar, and the cylinder head concept was optimized to achieve a better intake and exhaust flow performance for reducing gas exchange losses and improved swirl levels and swirl homogeneity for optimized mixture preparation. To optimize the flow characteristics, one intake port was designed as a filling port, the second one as a classic tangential port. Creating the charge movement was supported by seat swirl chamfers on both intake valves. The layout of the combustion chamber geometry showed a conventional recess shape, which was further optimized together with the nozzle geometry (8-hole) in order to achieve the best possible air utilization. The recessed valves made it possible to eliminate valve pockets in the piston and thus further improve the flow quality near the recess. At the same time, the fuel injection equipment was capable of a maximum rail pressure of 2000 bar. Due to this high pressure, a nozzle with smaller diameter nozzle holes was used to improve mixture preparation.

Earlier studies [11,12] had shown that near optimum engine operation could be achieved on a wide range of fuels by keeping the combustion timing constant at a few degrees crank angle (°CA) after top dead centre (ATDC). For this reason, the engine simulated CLCC by keeping the centre of combustion (CA50) constant when operating the engine with different fuels. This was achieved by continuously adjusting the beginning of injection (BOI) using an in-cylinder pressure sensor. This approach successfully maintained the CA50 within a very narrow range, even with major changes in fuel properties and EGR. For improving combustion of the low CN fuel in particular at lower engine loads, the intake air temperature was increased to simulate an EGR-cooler bypass. Additionally, heating of the intake air by heat exchanging with the engine coolant was used for low load operation.

Most part load tests were conducted at an engine speed of 1500 rpm and 6.8 bar IMEP. Intake and exhaust back pressure were adjusted according to typical values for modern passenger car diesel engines, but injection related parameters such as rail pressure and fuel injection phasing were adjusted for gasoline.

Full load capability was investigated at two engine speeds, using the standard diesel calibration. The maximum IMEP was limited by either soot or exhaust gas temperature. The maximum cylinder pressure was also set to 160 to190 bar (**Table 3**) in order to limit the mechanical stress. The full load calibration and limits for both tested engines speeds are given below.

Table 3 Engine calibration at full load operation

Engine Speed	Rail Pressure	Boost Pressure	Exhaust Pressure	Intercooler Temperature	Max. Filter Smoke Number	Max. Cylinder Pressure	Max. Exhaust Gas Temperature
[rpm]	[bar]	[bar]	[bar]	[°C]	[-]	[bar]	[°C]
2000	1800	2.45	2.46	46	1.7	160	850
4000	2000	3.00	4.00	60	2.8	190	850

4. BENCH ENGINE RESULTS

The engineering paper study showed that the principal challenge of achieving GCI is to obtain reliable ignition with an acceptably short ignition delay, given the very low CN of gasoline. A higher compression ratio is needed to keep cylinder pressures and temperatures high, and some assistance for ignition will be needed in the form of glow plugs or another heat source.

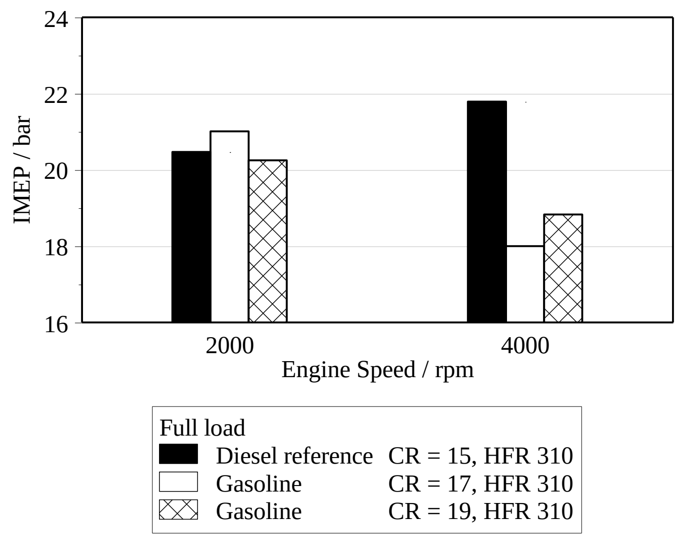
To test the learnings from the paper study, a bench engine study was carried out to provide a proof of the GCI engine concept and determine what hardware measures are most effective for extending the range of acceptable operation. The engine and methodology for the investigation were described earlier and the key results from the tests are presented in the next section.

4.1. COMPRESSION RATIO AND FUEL INJECTORS

Two compression ratios (CR 17 and CR 19) were evaluated with two injectors having different nozzle hole sizes and hydraulic flow rates[2] (HFR 310 with a 0.109mm nozzle diameter and HFR 520 with 0.141mm nozzle diameter).

With CR 17 and HFR 310 nozzles, full load power at 2000 rpm matched that when operating on diesel fuel. At 4000 rpm, full load power was limited by smoke resulting in lower exhaust temperatures in these initial tests that were performed without pilot fuel injection (**Figure 5**).

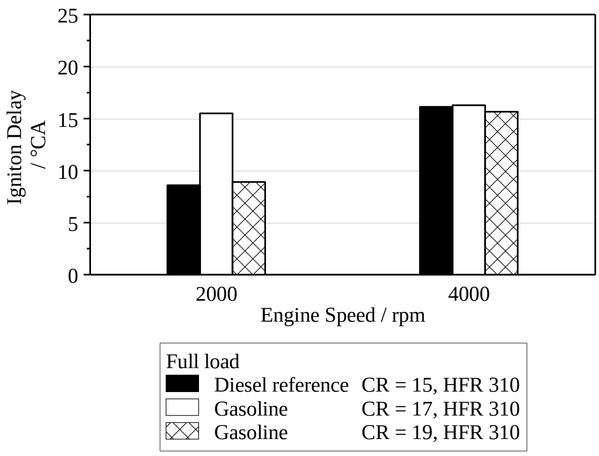
Figure 5 Full load IMEP achievable with acceptable smoke and exhaust temperature



At full load, performance was better with CR 19 than with CR 17, giving shorter ignition delays (**Figure 6**).

2 Hydraulic Flow Rate (HFR) is reported as the amount of fuel (in cm^3) injected in 30 seconds at 100 bar injection pressure

Figure 6 Ignition delays at full load



Contrary to the expectations from the paper study, the lower HFR 310 nozzles (with smaller nozzle holes) were found to perform better than the HFR 520 nozzles. Using the HFR 520 nozzles at CR 17, the maximum cylinder pressure varied dramatically at 2000 rpm so that full load could not be achieved.

At high load, better combustion stability was obtained with the HFR 310 nozzles. At medium to low load and high speed, the HFR 310 nozzles produced lower HC and CO emissions as well as lower combustion sound levels (CSL) compared to the HFR 520 nozzle.

At 1500 rpm and 6.8 bar IMEP, ignition could not be achieved at CR 17 using the standard intake air pressure. To achieve ignition, the intake pressure had to be increased from 1.50 bar to 1.65 bar with the HFR 310 injectors and to 2.00 bar with the HFR 520 injectors in order to achieve stable combustion.

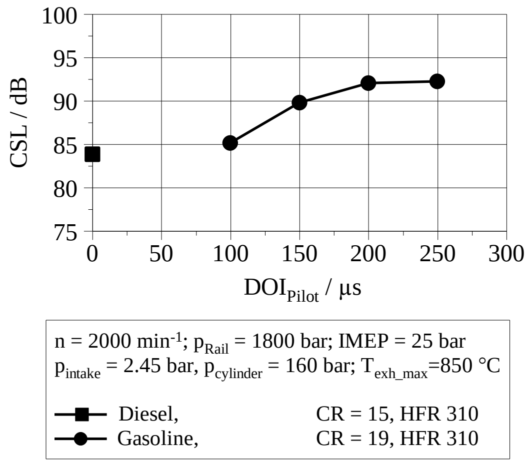
The higher CR 19 and HFR 310 injectors were therefore chosen for further work.

4.2. FULL LOAD OPERATION AT CR 19

Although the CR 19 engine could achieve full load using a single fuel injection, the level of pre-mixed combustion was such that the rate of pressure rise and noise were high. Reducing the fuel rail pressure from 1800 bar to 1000 bar produced a significant noise reduction of around 4 dB. However, because the duration of fuel injection increased, the fuel injection was still occurring after combustion had started, leading to an unacceptable increase in smoke emissions (Filter Smoke Number (FSN) about 4.3).

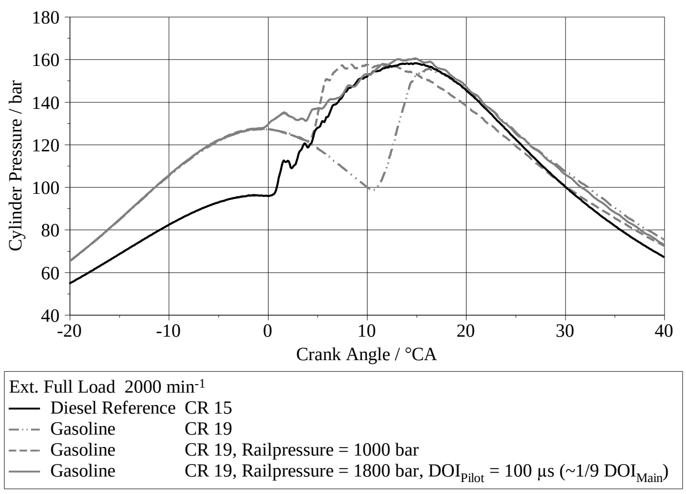
By introducing a small pilot injection, it was found that noise levels were reduced by 9 dB from 94 dB without pilot injection, giving levels close to those when operating on diesel fuel, and with a more modest increase in soot emissions. Further increasing the amount of pilot fuel injection increased noise again (**Figure 7**). The best offset for the pilot injection was found to be 5.5 to 7.0 degrees before the main fuel injection.

Figure 7 With pilot injection, the noise at full load using gasoline was similar to that measured using European diesel fuel



Using the pilot injection, reducing fuel injection pressure from 1800 bar to 1400 bar did not reduce noise and smoke emissions increased. For this reason, it was concluded that pilot injection at higher fuel injection pressure was the most effective strategy. With this approach, engine operation was found to be within the boundaries of 850°C maximum exhaust temperature and 160 bar peak cylinder pressure, while matching the power achieved by the diesel engine (**Figure 8**).

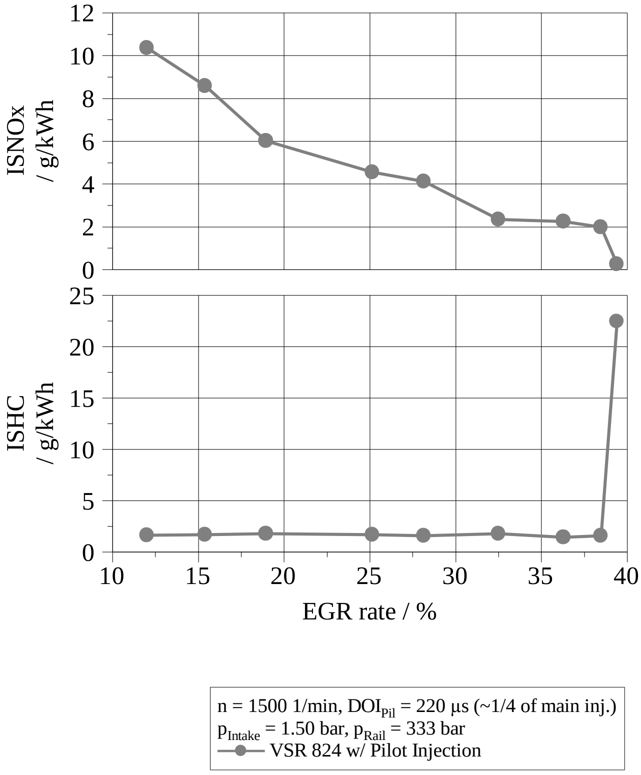
Figure 8 Pilot injection and a high injection pressure gave the best performance at 2000 rpm and full load



4.3. PART LOAD OPERATION AND FUEL INJECTION STRATEGY

At 1500 rpm and without pilot injection, stable operation could not be achieved below 7 bar IMEP even without EGR, but combustion improved with pilot injection. To achieve the best combustion performance, the injection parameters had to be further adapted for gasoline, with the injection pressure reduced from 900 bar to around 400 bar. The proportion of fuel injected in the pilot was slightly higher, but because of the lower injection pressure, the injection duration was 450 µs for gasoline, compared with 140 µs for diesel fuel. The pilot offset was also slightly adjusted but the air intake temperature and pressure remained the same. At 6.8 bar IMEP, a rail pressure of 333 bar and EGR of 40%, NOx emissions of 2g/kWh were achieved.

Figure 9 Effect of EGR on NOx and HC emissions at 1500 rpm and 6.8 bar IMEP showing the sudden transition to unstable combustion



As EGR increased, NOx emissions were reduced (**Figure 9**). The ignition delay increased from 15°CA without EGR to over 30°CA at the highest EGR rates with the combustion duration stable around 25°CA. However, the combustion was very sensitive to slight increases in EGR beyond this point and the transition to poor combustion was dramatic. With a small increase in EGR above 38%, the combustion duration rose to 40°CA and IMEP fell by nearly two bars with very high HC emissions. Similar behaviour was seen at lower loads.

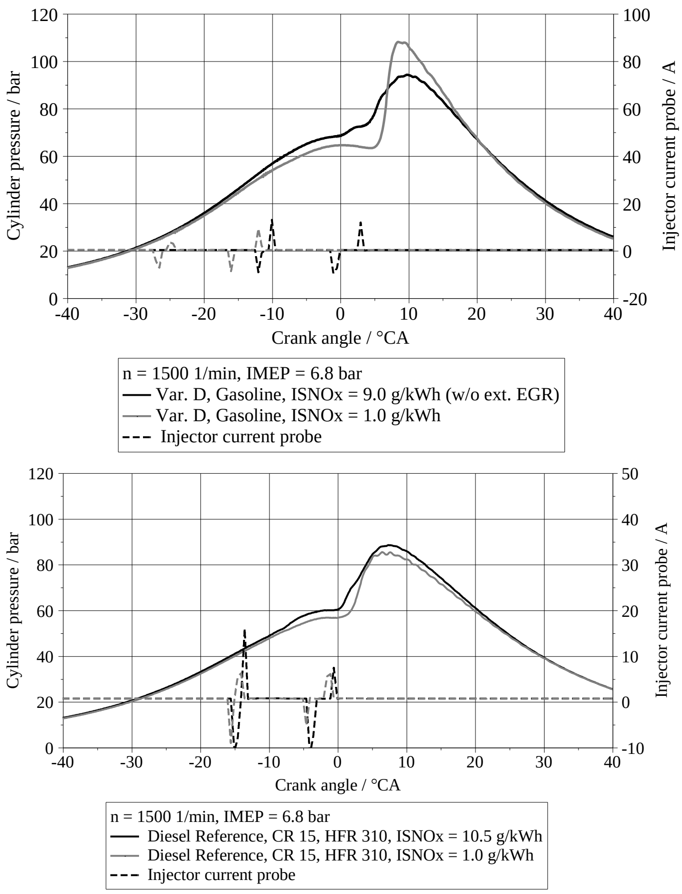
4.4. IMPROVING PART LOAD PERFORMANCE

The results reported above show that the limits of combustion with gasoline are reached suddenly and dramatically. For this reason, three different strategies were evaluated to enhance combustion at low NOx levels: increased air intake temperature, internal EGR using VVT, and improved fuel injection strategies.

Continuing the tests at 1500 rpm and 6.8 bar IMEP, the temperature downstream of the intercooler was increased from 30°C to 75°C. This higher temperature would be realistic in a situation where the EGR cooler is by-passed or the intake air is heated by the engine coolant water. In addition, with the cam phasing adjusted to provide more internal EGR, the configuration chosen (marked 'Var D' in **Figure 10**) moved the exhaust valve opening earlier by 24°CA and intake valve opening later by 24°CA.

With no external EGR, NOx emissions were 9.0g/kWh, slightly lower than for the diesel engine without internal EGR, and the cylinder pressure trace was similar for the two fuels. When external EGR was used with gasoline, NOx emissions could be brought down to 1g/kWh, but the ignition delay increased significantly with a consequent increase in noise. In comparison, the baseline diesel engine was able to operate with external EGR at a similar NOx level with no significant change in ignition delay or increase in noise.

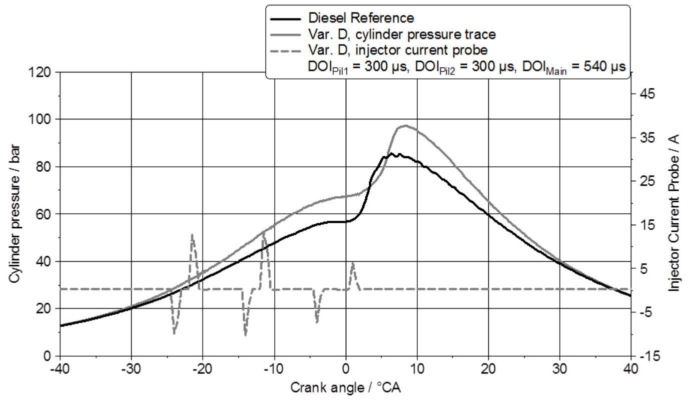
Figure 10 With internal EGR and single pilot injection, gasoline gave a similar pressure trace to diesel, but was less tolerant of additional external EGR. Results are shown at 1500 rpm and 6.8 bar IMEP



To address these limitations, a revised injection strategy was used to phase the introduction of the fuel into the cylinder. Early attempts with a split main injection were not encouraging, but additional tests using a triple injection had more success consisting of two large and early pilot injections before a main injection close to TDC. This injection strategy is similar to that reported in [20,24].

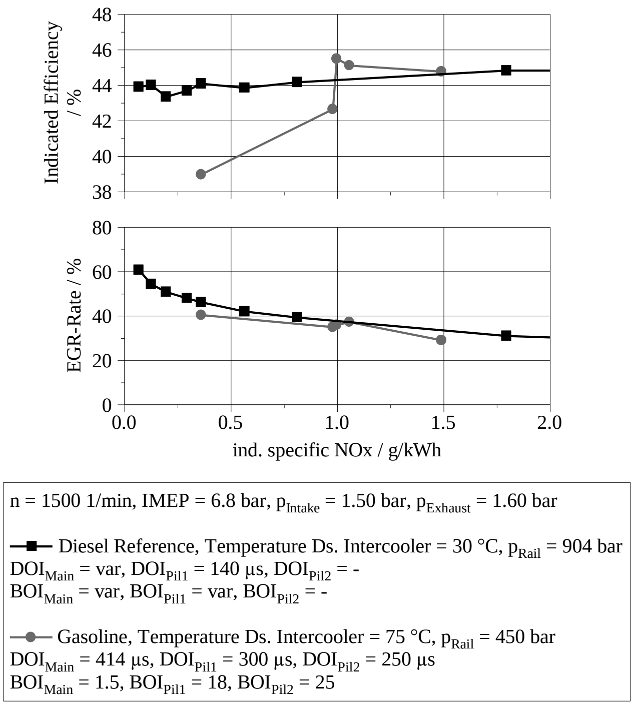
Using this multiple injection strategy, NOx emissions were reduced to 1.0 g/kWh while keeping noise, efficiency, combustion stability and CO emissions close to the diesel reference level and with an acceptable increase in HC and soot emissions. The cylinder pressure trace was similar to that for the diesel engine, with higher peak pressure reflecting the higher CR 19 (**Figure 11**).

Figure 11 With triple fuel injection and internal and external EGR, gasoline gave a smooth combustion at 1.0g/kWh NOx. Results are shown at 1500 rpm and 6.8 bar IMEP



In earlier studies with the baseline diesel engine [12], it was possible to further reduce NOx emissions below 1.0 g/kWh by increasing EGR. However, with gasoline, further increases in EGR (around 40%) led to unstable combustion and a rapid loss in efficiency (**Figure 12**). Higher HC emissions were also observed.

Figure 12 NOx emissions were limited to 1.0 g/kWh. Increasing EGR beyond this level resulted in unstable combustion

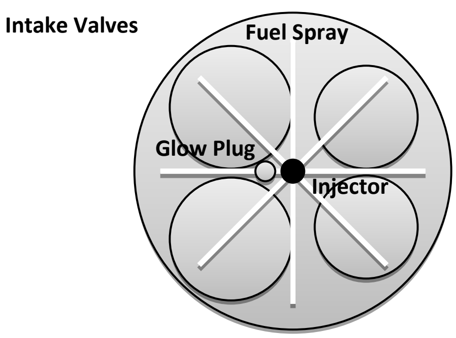


4.5. COMBUSTION ASSISTANCE WITH A GLOW PLUG

As predicted by the engineering paper study, it proved difficult to sustain reliable combustion on the gasoline at lower loads. Light load operation could be achieved, but NOx levels were higher than desired. The combustion was also unstable and would not tolerate additional EGR. For this reason, the engine was fitted with a state-of-the-art glow plug which was capable of a sustained glow temperature of around 1200°C. For these tests, the engine coolant temperature was also reduced to 48°C to simulate the engine warm-up period.

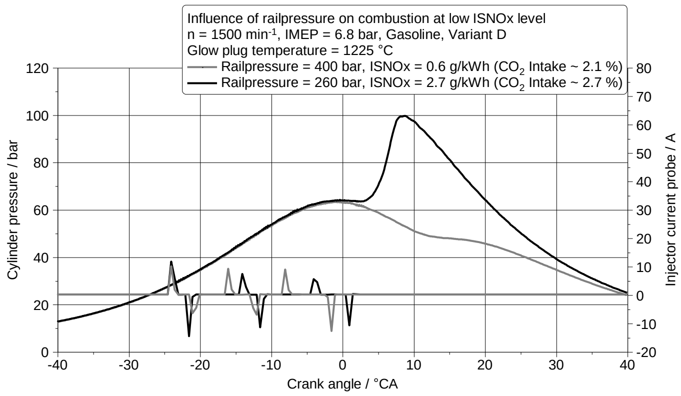
The orientation of the glow plug to the injector spray is known to be critical (**Figure 13**). The position was adjusted by changing the orientation with respect to one individual injector spray by 1 degree increments, while monitoring engine performance. A position close to the spray centre line giving the lowest CO/HC emissions and combustion duration was chosen. Additional simulation work is described in **Section 5** that was completed after the experimental study to investigate the potential to improve the positioning of ignition assistance.

Figure 13 Orientation of the glow plug with respect to the fuel injector spray



With the glow plug installed, low load operation was possible at normal boost pressure levels, even at this cooler engine temperature condition. Under hot engine conditions, however, the glow plug did not help to reduce the NOx emissions. At 400 bar injection pressure, combustion quality was poor with a higher EGR rate. Reducing the injection pressure further to 260 bar improved the combustion, but the increased heat release led to higher NOx emissions even though the EGR level was already quite high (**Figure 14**).

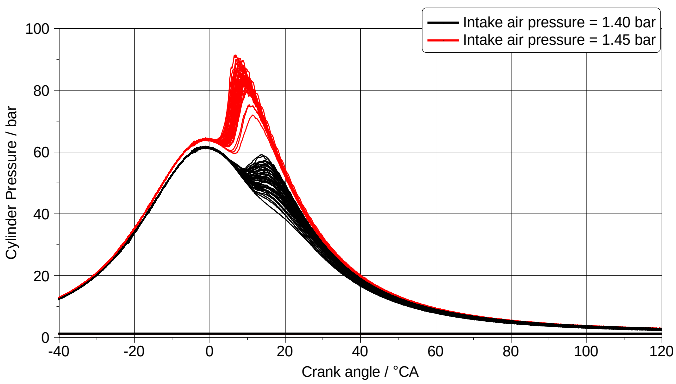
Figure 14 Operation at very low NOx levels was not improved by the use of a glow plug. Results are shown at 1500 rpm and 6.8bar IMEP



4.6. COMBUSTION STABILITY

Combustion with gasoline fuel was very sensitive to EGR, and the transition from stable to unstable combustion was very rapid [23]. In some cases, even where a stable combustion cycle was observed, it could be followed by combustion failure in the following cycles. As noted above, increasing the air intake pressure at low load improved combustion, but variability remained at an unacceptably high level in some cases, as shown in **Figure 15**.

Figure 15 Increasing boost pressure improved combustion at low loads, but combustion variability remained high. Results are shown at 1500 rpm and 3.4 bar IMEP



The stability or repeatability of combustion is therefore an important metric that has been used to evaluate performance throughout this study.

5. MODELING OF THE GASOLINE SPRAY

5.1. MODELING OF THE GASOLINE SPRAY

The complex turbulent reacting flow in the combustion chamber and intake port was modeled using Computational Fluid Dynamics (CFD). In order to reduce the computation time, a Reynolds-averaged Navier-Stokes equations (RANS) approach was used with a time-averaged approximation for the turbulent flow and analogies to reproduce the unsteady flow. For this investigation, the standard k-ε model for high Reynolds numbers was used.

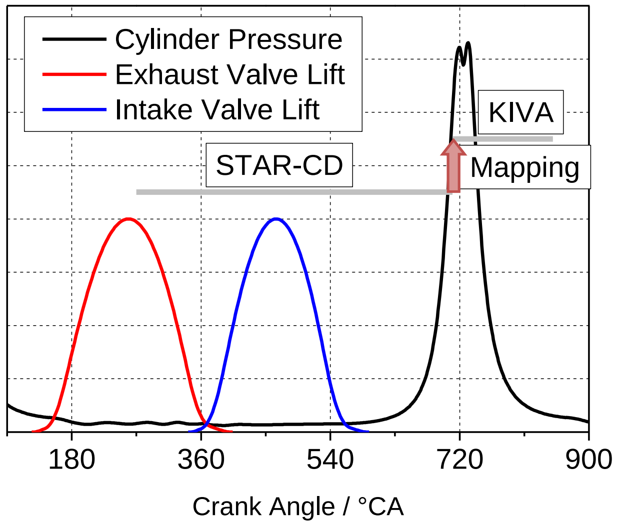
The gas exchange and compression strokes were simulated with STAR-CD software to analyze the effects of using VVT for internal EGR on flow, temperature and residual gas distribution in the combustion chamber. Therefore, a complete mesh considering inlet/outlet ports, piston, cylinder head and walls was generated using a commercial mesh-generating tool and imported into STAR-CD. Required boundary conditions such as temperature and pressure traces, in the intake/exhaust (I/E) ports were delivered by an adjusted one-dimensional simulation of the single cylinder bench engine [25].

To investigate the additional potential by using both a VVT strategy and a glow plug at low load operating conditions, the STAR-CD in-flow results were then coupled with combustion simulations performed with the CFD code KIVA. The multi-dimensional modeling code KIVA 3V Release 2 with Engine Research Center (ERC) model extension was adopted to carry out the combustion simulations and to study the impact of internal EGR on gasoline ignitability and combustion stability at the low load operating conditions. This KIVA CFD code includes a modified RNG k-ε turbulence model, a Kelvin-Helmholtz (KH) and Rayleigh-Taylor (RT) spray model, the Shell auto ignition model, a laminar/turbulent Characteristic Timescale Combustion (CTC) model, a crevice flow model and a spray/wall impingement model [26].

The combustion simulations were performed for the high pressure cycle, with a segment of the combustion chamber representing the computational domain. The detailed flow field computed in the STAR-CD was included as boundary conditions for the KIVA simulations by means of a mapping procedure [27,28] and an example is shown in **Figure 16**. Pressure, temperature, gas composition and flow velocity resulting from the STAR-CD models were transferred to the KIVA models. The flow field during compression stroke and gas exchange phase was mapped for all the different VVT strategies at a few degrees before the injection event. Thus, the detailed flow field of STAR-CD was coupled with the KIVA combustion modeling, increasing the accuracy of the in-cylinder simulation results. For all the VVT strategies investigated, the same injection timing and phasing (pilot and main injections) were applied, to isolate the effect of the internal EGR on the combustion stability and in-cylinder temperature.

The KIVA simulations were performed on the GCI bench engine configuration with a 19:1 CR and an injector with HFR of 310 cm^3/30 sec at 100 bar. A gasoline fuel from the KIVA fuel library was adopted to carry out the combustion simulations.

Figure 16 Mapping methodology scheme from STAR-CD to KIVA



6. DISCUSSION OF RESULTS

Before starting the experimental study on the GCI engine concept, an engineering paper study was completed to identify the key challenges and issues in designing a GCI engine capable of running over the full speed and load range on market gasoline. Having completed the experimental programme, it is useful to compare the expectations from the paper study with the practical experience observed on the GCI bench engine.

6.1. FUEL LUBRICITY, VISCOSITY AND VOLATILITY

It had been anticipated that the lubricity of the gasoline would not be adequate and a commercial treat of a common lubricity improving additive was used throughout. Although this treatment prevented mechanical damage to the injector, there was still some evident mechanical damage to the high pressure fuel pump. More work would be needed to identify the additive treat rate that is needed to provide adequate protection. It remains an unresolved question as to whether fuel injection equipment could be developed (at acceptable cost) to operate on market gasoline.

The low viscosity of gasoline leads to high leakage at high pressure pump. It was often difficult in practice to achieve the injection pressures required for full load. The pressure in the injector's fuel return line had to be higher than is typical for diesel fuel in order to ensure acceptable performance of the injector used in the study. Only a simple modification of the injector return line was required in order to achieve good performance.

6.2. INJECTION PRESSURE AND NOZZLE HOLE DIAMETER

During the completion of the paper study, Won et al. [20] reported that a larger diameter nozzle and lower injection pressures could be beneficial for this type of combustion system. Contrary to this expectation, in our work the smaller diameter nozzle was found to perform better throughout the entire speed and load range.

Having fixed on the smaller hole size, it was found that a double pilot or split main injection strategy provided a more effective means, compared to a reduction in rail pressure, to control the combustion noise at full load without smoke emissions debits.

6.3. COMPRESSION RATIO

The paper study correctly showed that the length of the ignition delay would be a key issue. In the initial phase of this work, CRs of 17 and 19 were tested, and, as expected, the higher CR 19 led to a shorter ignition delay and permitted operation at lower loads. At full load, CR 19 gave better efficiency (with no power loss) compared to CR 17 but performance was limited by smoke, noise and exhaust temperature.

Increasing CR beyond 19 was considered, but was not tested. It is expected that higher CR will be beneficial at low loads, but could cause difficulties at higher load. For this reason, a Variable Compression Ratio (VCR) system could be advantageous for a practical embodiment of this GCI engine concept, even if the bowl geometry could not be varied.

6.4. EGR AND INTAKE AIR

External EGR was used to control the NOx levels, but this led to longer ignition delays, producing poor combustion stability or even failed combustion at the extreme limits. Increasing the temperature of the external EGR and intake air from 30°C to 75°C improved combustion stability. The higher temperature would be realistic in a situation where the EGR cooler is by-passed or the intake air is heated by the engine coolant water. Even higher temperatures than this would require a more complex approach for heating the intake air, for example heat exchange with exhaust gases or electric heating.

The potential of internal (uncooled) EGR using negative valve overlap has been highlighted in other literature [13,17]. Four different VVT strategies were tested as part of this study but it is not clear that this approach will overcome problems with the very long ignitions delays found for GCI. However, the extent of negative valve overlap used in this study was less than that used in [17].

In this study, internal (uncooled) EGR was found to be advantageous for reducing HC emissions and improving fuel consumption in the mid-load range. There are a number of competing effects that occur when more internal EGR is used. For example, the higher temperature by itself shortens the ignition delay but also leads to higher NOx levels which require higher EGR levels to control. With higher EGR levels, the decrease in local oxygen levels and inhomogeneities associated with the internal EGR concentration lead to higher smoke levels and a tendency to lengthen the ignition delay. For this reason, it is not obvious how beneficial internal EGR will be under a given set of conditions.

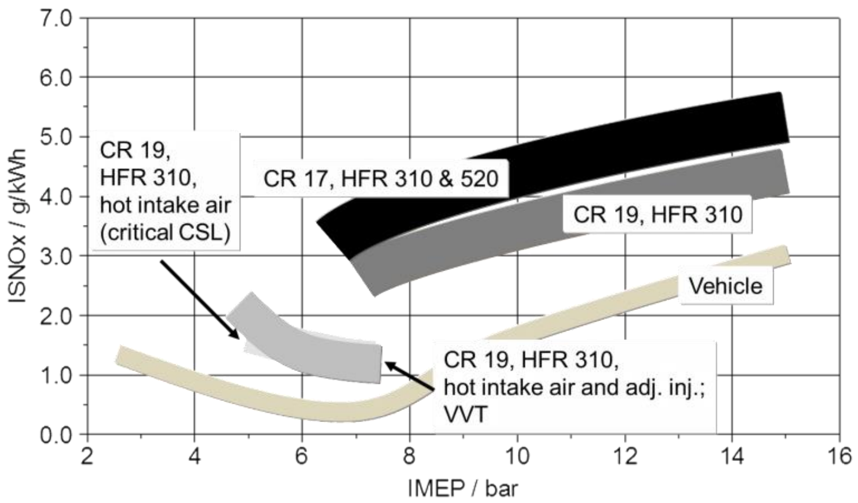
Boost pressure can also allow the engine to operate at lower loads, but high levels of boost are not achievable from the turbocharger at lower loads.

6.5. NOx LEVELS

As part of the paper study, it was expected that the engine-out NOx/PM trade-off would be better compared to diesel engines at low engine loads. **Figure 17** shows the NOx levels achieved in this study for the various hardware options tested. The target NOx levels at 1500 rpm for various loads are shown by the solid band marked "vehicle".

Even with the optimized injection strategy, higher CR, VVT, and hot intake air, the engine was not able to achieve the target NOx levels without exceeding a reasonable level of HC emissions. With combustion assist in the form of a glow plug, it was possible to achieve loads down to 4.3 bar IMEP, but not with the EGR levels required to meet the target NOx levels. At higher engine loads, the target NOx levels could not be achieved with stable combustion behaviour although more effort in this region would be expected to improve performance.

Figure 17 NOx emissions achievable at 1500 rpm as a function of IMEP



6.6. SMOKE, HC AND CO EMISSIONS

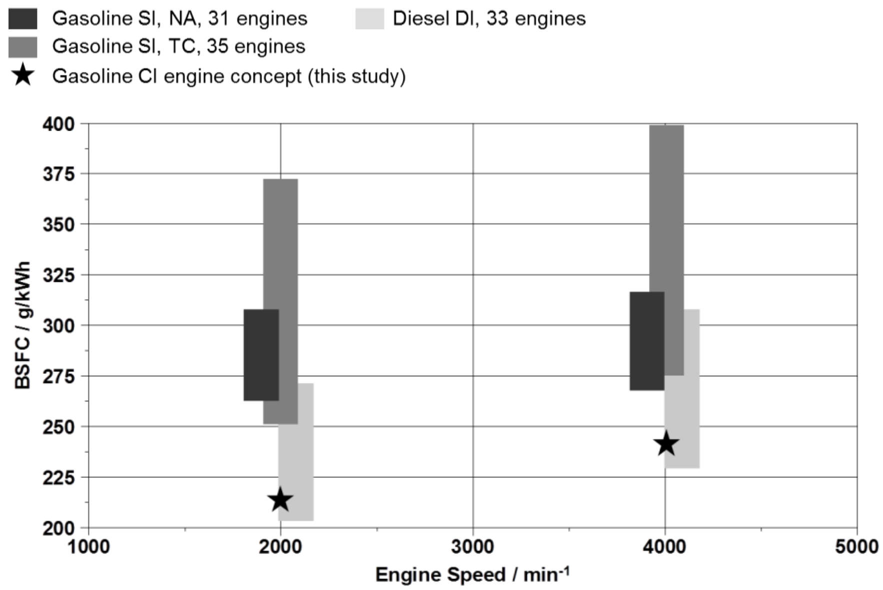
From the engineering paper study, smoke emissions were expected to be reduced significantly in the GCI engine concept compared to diesel. In practice, however, the higher CR and longer injection duration compared to diesel effectively eliminated any smoke benefit. For low loads, and at the EGR levels needed to simultaneously achieve low NOx and low combustion noise, it was observed that smoke levels could exceed diesel levels.

HC and CO emissions were expected to be a particular challenge for the GCI concept and a five-fold increase was anticipated compared to diesel. Due to the higher CR 19, advanced injection strategy, and cam phasing, HC and CO emissions in fact only doubled in this study compared to diesel at a comparable engine load/speed, NOx level and CSL.

6.7. FUEL CONSUMPTION

Based on the paper study, fuel consumption on an energy or gravimetric basis was expected to be comparable to diesel-fuelled CI engines. As shown in **Figure 18**, the fuel consumption (on a gravimetric basis) of the GCI engine concept was at the low end of the diesel DI engine band and better than all comparable SI engines at both the full load and lower load test conditions. Broadly speaking, expectations with respect to fuel consumption were met.

Figure 18 Comparison of the measured fuel consumption of the GCI engine tested in this study (stars) and state-of-the-art gasoline SI and diesel CI engines[3]



6.8. COMBUSTION ASSIST FOR LOW LOAD OPERATION

The paper study recognized the formidable challenge of getting the fuel to ignite at low loads and during cold starts. With this in mind the experimental programme tested the use of a glow plug to assist combustion at lower load conditions.

Before testing glow plug technology, the engine was able to achieve stable combustion at a speed load point of 1500 rpm and 6.8 bar IMEP using the other engine modifications discussed above (CR 19, VVT, and optimized injection strategy). This was a better result than had been expected.

The relative position of the glow plug and fuel spray were adjusted but additional optimization of the glow plug position and spray cone angle might be possible. From the results obtained so far, however, the glow plug option does not appear to be the single solution that would allow this technology to operate at low loads on market gasoline.

In particular, the glow plug appears to be extremely sensitive to EGR rate. Using a reduced injection pressure it was possible to achieve lower load operation, but with only limited amounts of EGR. Use of sufficient EGR to achieve NOx levels below 1g/kWh resulted in unstable combustion. Other workers have recently reported similar findings with the use of glow plugs [21].

[3] Based on FEV engine benchmarking data

6.9. OTHER HARDWARE OPTIONS

This paper describes initial engineering and experimental steps to assess the potential of operating a compression ignition engine on market gasoline. Using an advanced compression ignition bench engine having a higher CR 19, optimised valve timing, advanced engine management system, and flexible fuel injection, the engine could be operated on a typical European gasoline over full to medium part loads. The combustion was found to be highly sensitive to EGR rates, however, and the simultaneous optimisation of all regulated emissions and combustion noise was a considerable challenge.

Nevertheless the efficiency of operation of the engine was comparable with a CI engine operating on diesel and superior to any current SI technology at the conditions tested. In order to meet advanced emission control standards such as Euro 6 and beyond, a vehicle based on this technology would require a similar aftertreatment configuration to a CI engine running on diesel as well as an evaporative emissions control system.

The anticipated problem was to find a way to operate this GCI engine concept in a stable combustion mode at low load and glow plug technology was not as effective as had been expected.

There are several other possible approaches which could be envisaged to allow better operation at low load, some of which could be examined in future studies:

- Use of a spark plug to initiate combustion and provide combustion assist at low load;
- A more complex cam profile allowing a more flexible VVT arrangement;
- Use of a supercharger as well as a turbocharger to increase the boost pressure at low load; and
- Use of Variable Compression Ratio (VCR).

It is important to note that the approaches listed above would add complexity and cost to the vehicle, with better fuel consumption benefits expected in most cases.

6.10. GASOLINE'S IGNITION QUALITY

The low CN of gasoline implies longer ignition delays. To a certain extent this is desirable to achieve LTC, because it provides more time for fuel mixing after injection and before combustion starts. However, as ignition delays become longer, gasoline must be injected earlier in the compression stroke where temperatures and pressures are lower than for diesel fuel combustion. This makes achieving ignition even more difficult for an already ignition-resistant fuel.

High levels of EGR are needed to cool the combustion and reduce NOx emissions, but the higher CO_2 and lower O_2 concentrations associated with EGR further slow the ignition reactions. Although internal EGR provides higher temperatures, it is believed that it leads to inhomogeneities and high soot emissions and for that reason was not as effective as expected. Because of the higher temperatures, more EGR is needed to get the same level of NOx relative to diesel, but the gasoline fuel lacked the resilience to ignite under these more difficult conditions.

Although compression ignition with gasoline remains challenging, the potential benefits merit further study. CI engines have a clear efficiency advantage over spark ignition engines, and the ability to use a readily available market gasoline would provide an easy access into the market and a convenient way to accommodate increasing demand. From a technical perspective, fuels with lower octane numbers, such as diesel/gasoline blends, remain interesting but the challenges associated with bringing both a new fuel and a new vehicle concept into the market at the same time must be addressed.

6.11. CFD MODELING RESULTS

As mentioned above, a VVT strategy was found to support the autoignition of gasoline fuel in the GCI bench engine at low engine operating conditions. Thus, several VVT strategies were modelled at 1500 rpm and 4.3 bar IMEP. As shown in **Figure 19**, different configurations of valve lift and cam phasing were simulated to investigate the role of internal EGR on gasoline ignitability. The details of the VVT strategies are shown in the **Table 4**.

Figure 19 VVT strategies investigated at 1500 rpm and 4.3 bar IMEP

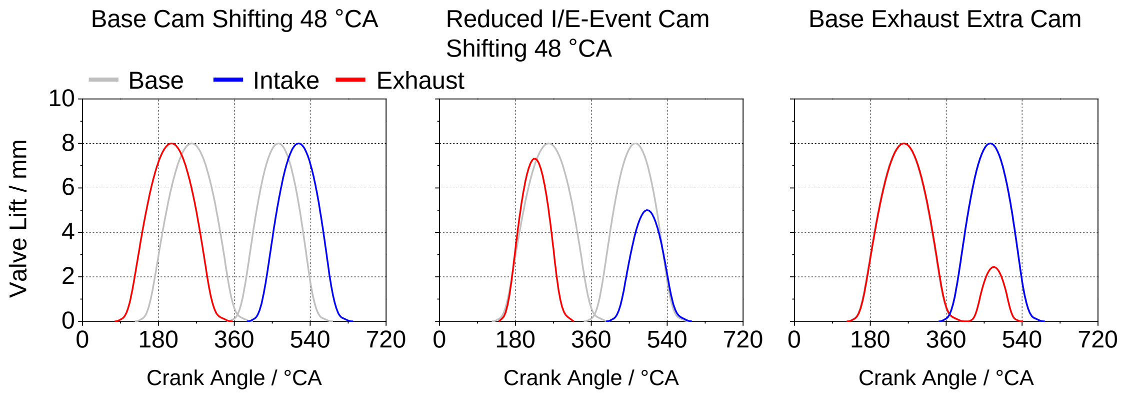


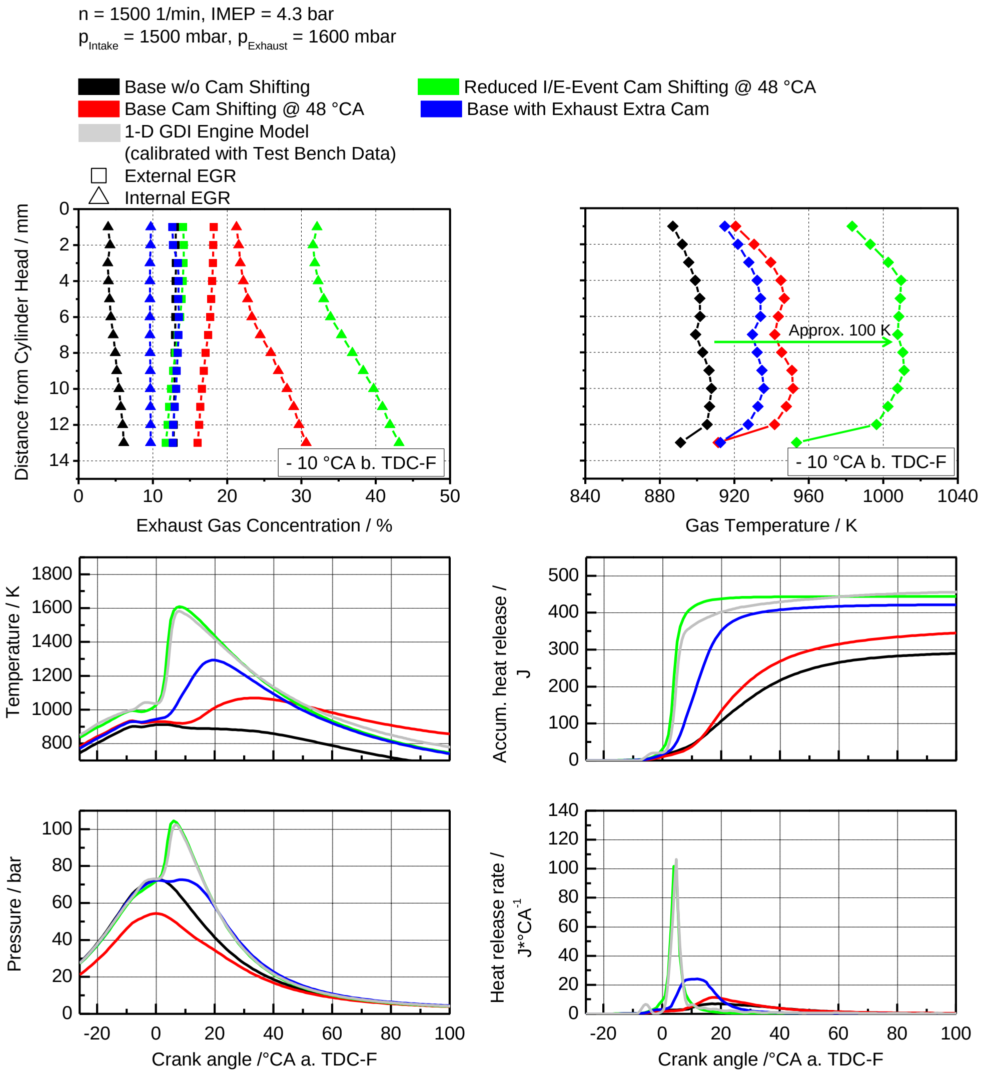
Table 4 Specifications of the four VVT strategies investigated in the CFD modelling study

VVT	Cam Duration[1]		Cam Lift		Extra Cam	Valve Overlap[1]	Internal EGR
Units	°CA TDC-F		mm		-	°CA	%
	Intake	Exhaust	Intake	Exhaust			
Base 0°CA	342	124	8	8	no	+62	4
Base 48°CA	390	76	8	8	no	-34	18.78
Reduced I/E Event Cam Shifting 48°CA	395	140	5	7.3	no	-76	20.53
Base Exhaust Extra Cam	342	124	8	8	Exhaust[2]	-62	12.88

[1]Value refers to a 0 mm valve lift.
[2] Extra cam specifications: maximum valve lift equals 2.4 mm and a valve duration equal to 135°CA.

The results of the CFD modeling for all VVT strategies are shown in **Figure 20**. In order to gain information regarding the distribution of the internal/external EGR, independently from the global residual exhaust gases, distinct scalar tracers were implemented in the STAR-CD model to detect and monitor internal/external EGR distribution inside each cut plane. To distinguish between them, different symbols are used in this figure. The same post-processing approach was adopted to analyze the in-cylinder gas temperature, plotted on the x axis of the top left graph in **Figure 20**. Here the trend lines represents an averaged value of the temperature within the circumferential cut plane with the adoption of the same y-axis as for the right side graph. More details on this averaged value post-processing approach can be found in [25].

Figure 20 CFD results overview of the VVT strategies investigated at 1500 rpm and 4.3 bar IMEP



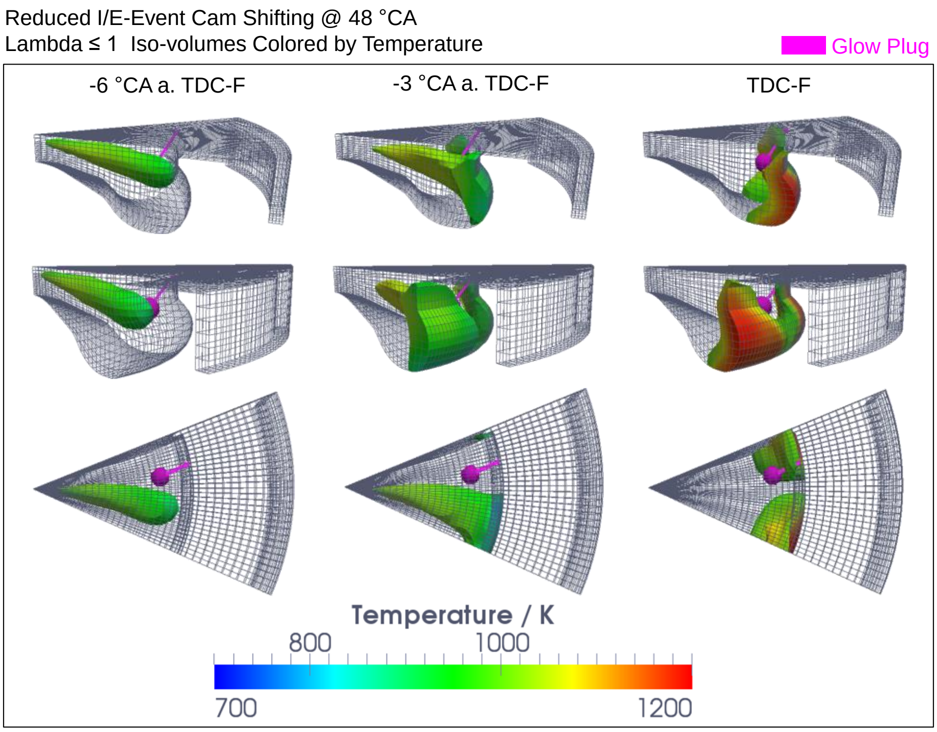
The results of the STAR-CD in-flow simulations are shown in the first two graphs. In the top right graph, the EGR trapped inside the combustion chamber of the GCI engine is represented. In order to easily show the total EGR distribution within the cylinder, circumferential cut planes of the cylinder volume at the SOI were performed. Within the cut-planes, the total EGR concentration was averaged and plotted versus the cut plane distance from the cylinder head.

In the bottom four graphs, the results of the combustion analysis performed with KIVA are exposed. Test bench data were used to calibrate a 1D model of the GCI single-cylinder bench engine. The 1D model results, shown in light grey colour, were used to ensure a realistic modelling of the spray formation, ignition and combustion with KIVA. In counterclockwise order, the in-cylinder temperature, pressure, heat release rate and cumulative heat release of the KIVA results are shown plotted versus ºCA.

Regarding the STAR-CD results, the different VVT strategies are shown with different colours. It is clear that only the reduced I/E cam event with a shifting of 48°CA (**green** curves) matches the required ignition features and ensures a stable combustion for this low operating point. The negative overlap of this variant allows a higher internal EGR content (see **Table 4**) which also increases the in-cylinder temperature, assuring a complete ignition and a more stable combustion. As shown in the bottom left graph, the KIVA model for the reduced I/E cam event with a shifting of 48°CA [27,28] exhibits cool flame behaviour, associated with the hotter temperature reached at the end of the compression phase. Although the pilot ignition does not ignite properly, this variant seems to be the most promising in terms of ignition and combustion behaviour. Weak ignition behaviour was also observed for the variant equipped with an extra exhaust cam, although the cumulative heat release is unsatisfactory. Thus, only the results of the variant with a reduced I/E cam event and a shifting of 48°CA will be discussed in more detail.

Because the engineering study had shown that a glow plug would be required to assist combustion at this low load operating condition, the spatial distribution of the fuel spray and the local lambda (i.e. air/fuel ratio) were also analyzed. **Figure 21** shows the results of the mixture formation analysis performed when the piston is approaching Top Dead Center Firing (TDC-F). Here the spray is visualized inside a 45° mesh sector of the piston bowl, by means of rich lambda iso-volumes coloured by increasing temperature. Thus, the rich zones which will probably ignite with the glow plug are identified.

Figure 21 3D mixture formation analysis for the reduced I/E-event cam shifting @ 48°CA at 1500 rpm and 4.3 bar IMEP



With accurate positioning of the glow plug, the modelling results have shown that a favourable interaction between the fuel spray and glow plug is possible with the chosen nozzle cone angle of 153°. For the VVT variants examined here, wider fuel rich zones were also observed in the range of interest due to the negative valve overlapping.

It must be mentioned that the results discussed above for the variant with reduced I/E cam event with a shifting of 48 °CA refer to the bench engine with advanced boosting conditions, in order to sustain the fuel mixture ignitability [13]. In a diesel passenger car, these advanced boosting conditions would reach the surge limit of a series compressor and the further adoption of a mechanical turbocharger would greatly increase the engine cost. Thus, from here onwards the best VVT configuration will be analyzed with standard diesel boosting conditions listed in **Table 5**.

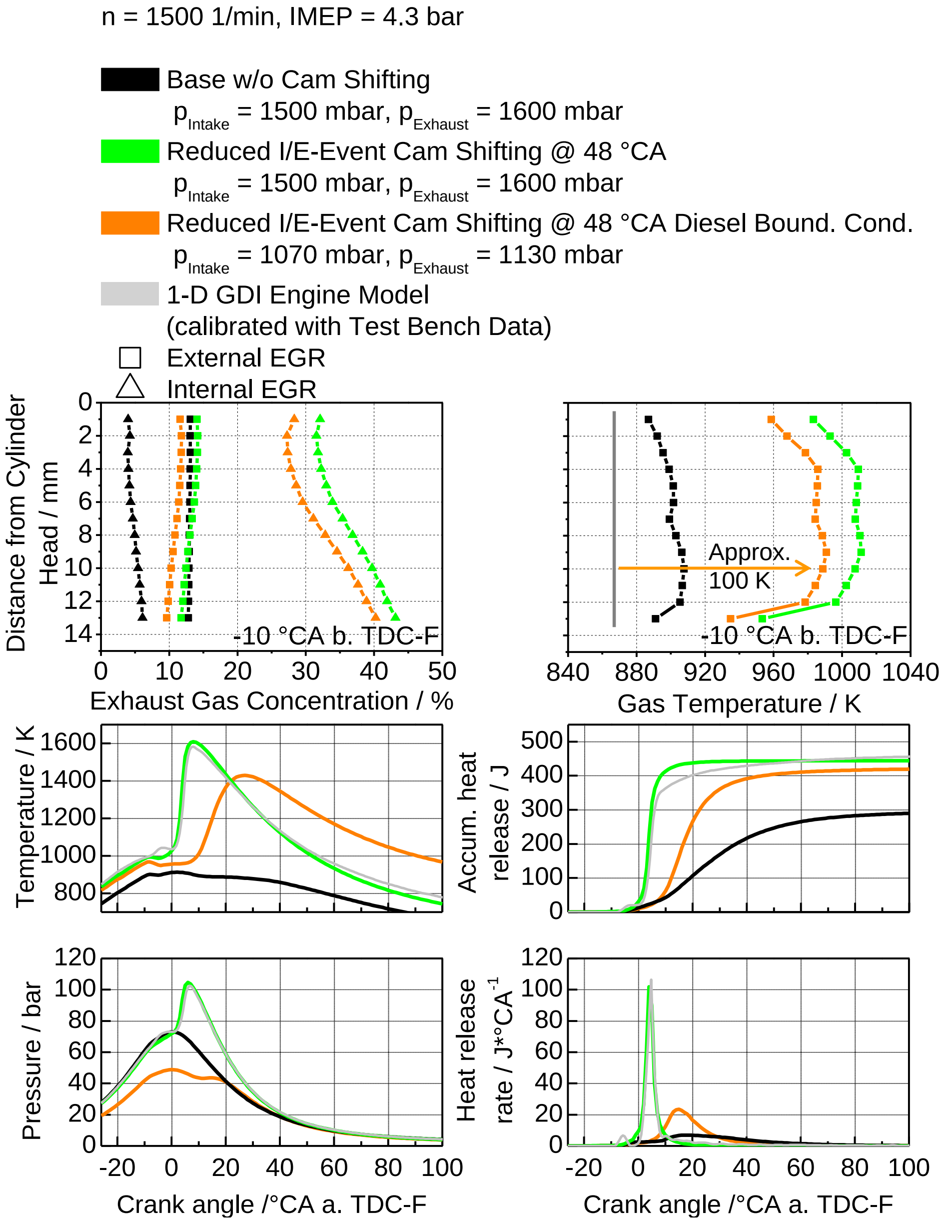
Table 5 Standard diesel boosting conditions for the best VVT configuration

Parameter	Unit	Value
Engine Speed	1/min	1500
Engine Load	bar	4.3
Boost Pressure	mbar	1070
Engine Back Pressure	mbar	1130

As for the CFD results overview given in **Figure 20**, the same post-processing strategy was applied to analyze the variant with standard diesel boosting conditions. In **Figure 22**, the reduced I/E cam event with a shifting of 48°CA with diesel boundaries is shown (**orange**) and compared with the same variant with advanced boosting condition (**green**) and with the base VVT strategy (**black**). Again the results of the CFD flow simulation are presented in the top two graphs. It can be observed that the amount of internal EGR is nearly constant by comparing advanced and standard diesel boosting conditions due to a comparable pressure difference of intake and exhaust side (approx. 60-100 mbar).

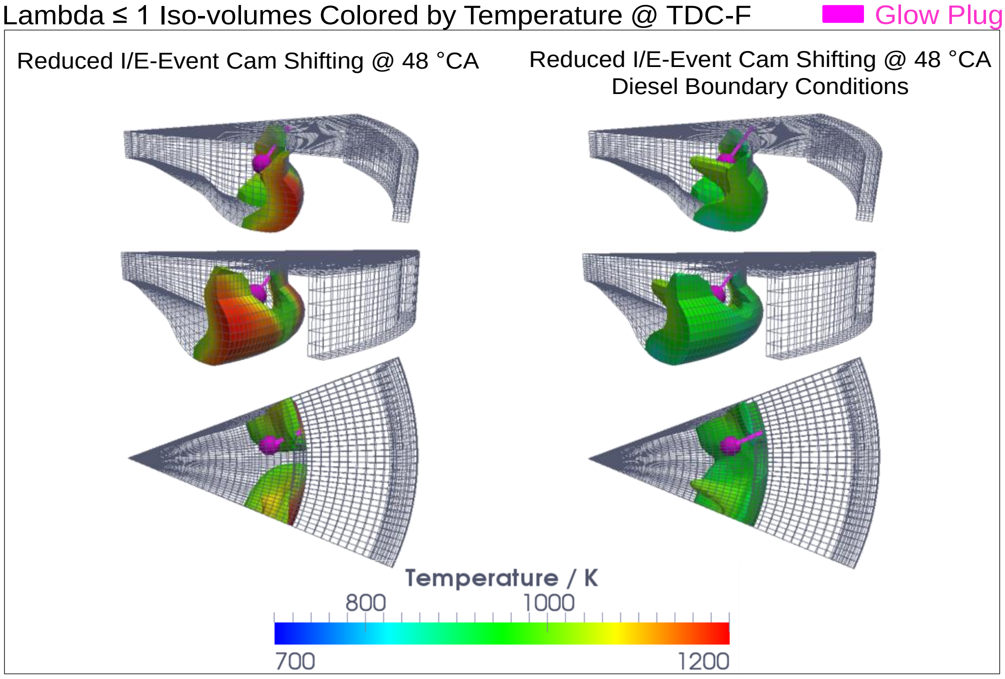
In the top right side graph, the influence of hot internal EGR is analyzed by means of the average temperature [25]. To evaluate the results of STAR-CD, the temperature estimation at SOI of the 1D simulation with standard diesel boosting conditions is illustrated by the gray line. Within this, the great potential (more than 100°K) is always represented by the reduced I/E cam event with a shifting of 48°CA operating at standard diesel boosting conditions. In the bottom four graphs, the results of the combustion analysis performed with KIVA are shown. It is clear that the I/E cam event with a shifting of 48°CA and diesel boundary conditions does not ignite as well as the same variant with advanced boosting conditions. The reduced boosting conditions lead to lower end of compression temperatures (see left top plot), thus resulting in a weak ignition as clearly visible in the heat release rate curves (bottom right plot).

Figure 22 CFD results overview of the different boosting conditions investigated for the I/E cam event with a shifting of 48°CA at 1500 rpm and 4.3 bar IMEP



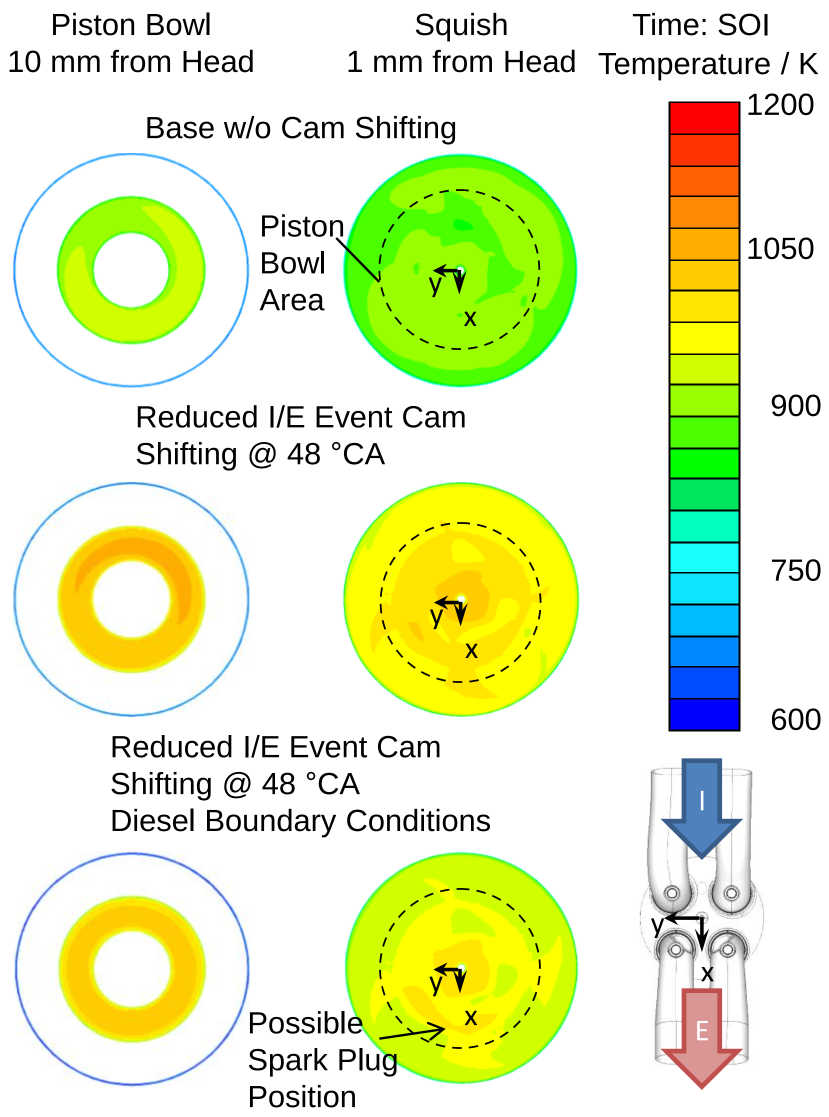
Due to the weak ignitability observed at TDC-F, further investigations on the spray spatial distribution and the local lambda were performed. **Figure 23** shows a detail of the mixture formation analysis performed for the reduced I/E cam event with a shifting of 48°CA operating at standard diesel boosting conditions in KIVA. Here a comparison between advanced and standard boosting conditions has been carried out for iso-volumes of rich lambda ranges. The snapshots on the right hand side of **Figure 23** confirm what was stated above: the temperature reached at TDC-F with standard diesel boosting conditions of about 1000ºK is not enough to properly ignite the air/fuel mixture. Thus, the feasibility of an external energy source application (i.e. spark plug) was also investigated to support the mixture ignitability at this low load engine operating condition.

Figure 23 3D mixture formation analysis of the reduced I/E-event cam shifting @ 48°CA at 1500 rpm and 4.3 bar IMEP for advanced and standard diesel boundary conditions



To investigate the applicability of the spark plug, a preliminary study on the in-cylinder temperature distribution was performed in STAR-CD. Cross-sectional cut planes of different positions in the combustion chamber give an overview of the temperature profile and indicate regions of temperature where a gasoline mixture can ignite. Thus, **Figure 24** shows cross-sectional cut planes of the investigated standard diesel boundary conditions in comparison to results of advanced boosting for two different distances from cylinder head (squish position 1 mm and bowl 10 mm).

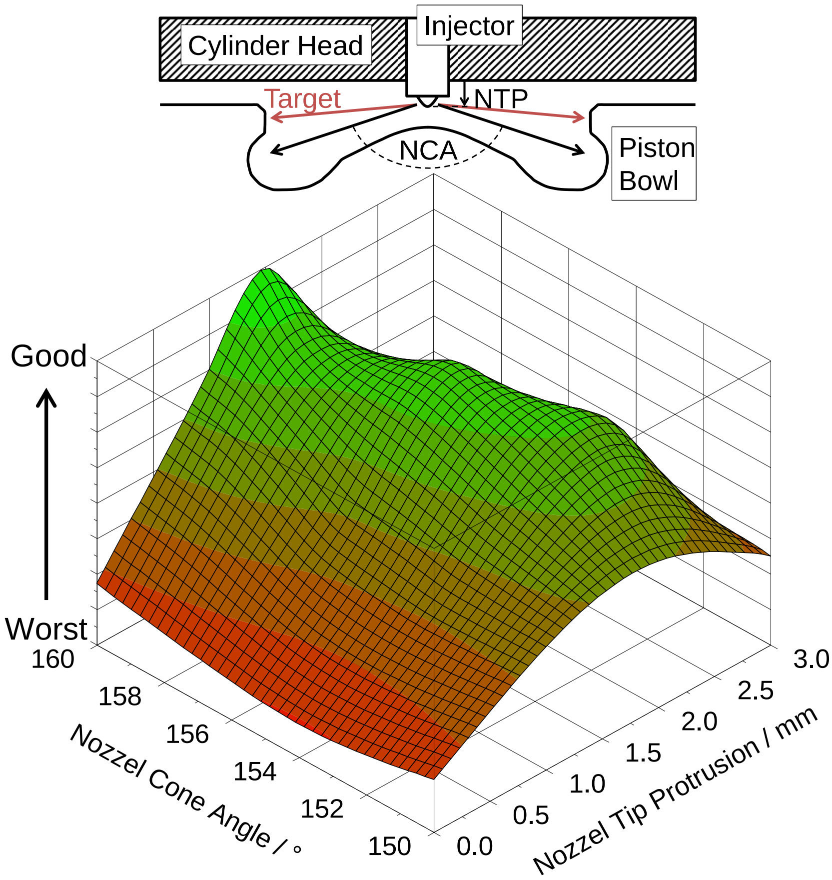
Figure 24 In-cylinder temperature distribution analysis performed in STAR-CD at SOI



As mentioned above, the variant reduced I/E cam event with a shifting of 48°CA operating in advanced boosting leads to highest temperatures in the combustion chamber and identifies a hot spot in the piston bowl on intake side. Using this VVT strategy with standard diesel boosting conditions, the overall temperature drops but shows a more homogeneous distribution within the piston bowl and a temperature hot spot inside the squish cross section of the exhaust side is visible. The local temperature in this region reaches values of up to 1000ºK, which will be too low for gasoline autoignition but enough to be externally ignited with a spark plug application.

To find the best configuration between the geometric spray distribution and the optimal spark plug position suggested by the in-flow STAR-CD analysis, two different nozzle configurations were analysed in KIVA. A simple schematic is shown in **Figure 25** to explain the nozzle parameters investigated in this work.

Figure 25 Nozzle configuration optimization map and explanatory sketch of nozzle parameters varied for the I/E cam event with a shifting of 48°CA operating in standard diesel boundary conditions



The Nozzle Tip Protrusion (NTP) indicates how far the nozzle tip penetrates from the cylinder head. The Nozzle Cone Angle (NCA) is a geometric parameter of the injector and represents the angle formed by the nozzle holes axis. The correct configuration of these parameters allows targeting the optimum turbulence ring inside an omega-shaped piston bowl (indicated in **Figure 25** by the red arrows).Thus, the mixing process is enhanced and a more homogenous mixture is guaranteed when the spark plug energizes. In this paper wider NCAs of 156° and 160° were analysed to optimize the spray targeting for a spark plug application. In order to gather information also on the in-cylinder behaviour for higher engine operating loads, the two different NCA were analysed at varying NTP values (see **Table 6**). It must be mentioned that the NTP values would need to be checked at higher load conditions because higher in-cylinder load and turbulence will tilt up the spray pathway.

Table 6 Nozzle configuration for the spray targeting study

NCA / °	NTP = 1.0 mm	NTP = 1.5 mm	NTP = 2.0 mm	NTP = 2.5 mm
156ºCA	X	X	X	X
160ºCA	X	X	X	X

As the optimization map in **Figure 25** shows, the best compromise between the nozzle parameter varied in KIVA is found by adopting a wider NCA of 160° and an NTP of 1.5 mm.

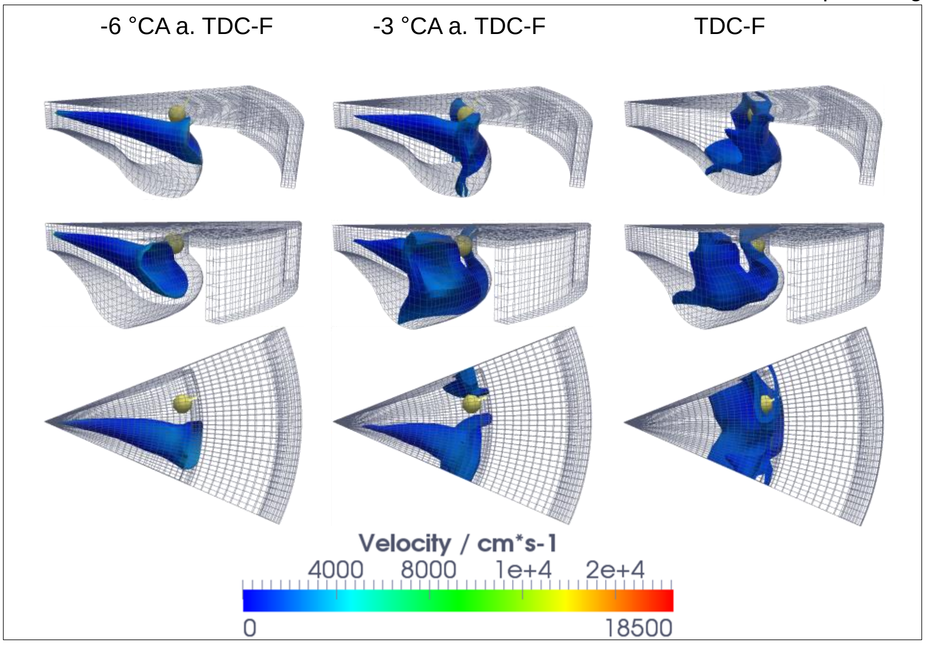
In order to find the optimum configuration, a study was also carried out in KIVA to analyze the velocity field of the air/fuel mixture when the piston is approaching TDC-F. A preliminary benchmarking study was carried out to investigate lambda distributions and spray velocities ranges of interest for passenger car SI engines [30]. In this study, it was found that a local in-cylinder lambda interval of 0.85 up to 1.15 coupled with spray axial mean velocities no greater than 4000 cm/s would guarantee mixture ignition with a spark plug at about TDC.

Figure 26 shows the velocity analysis for the lambda range of interest. It is clear that, when the piston is approaching TDC-F, the mixture velocities are all below 4000 cm/s. With this velocity field, this result confirms that the spark plug would have sufficient time to ignite the air/fuel mixture.

Figure 26 Results of the spray targeting analysis for a spark plug application, evaluated for the I/E cam event with a shifting of 48°CA at diesel boundary conditions

Reduced I/E-Event Cam Shifting @ 48 °CA Diesel Boundary Conditions
Lambda = 0.85 – 1.15 Iso-volumes Colored by Velocity

Spark Plug



7. CONCLUSIONS

This study explored the basic engineering steps needed to achieve a GCI engine concept, that is, stable and reliable combustion of European market gasoline in a CI engine. An engineering paper study was first completed to analyse critical engine parameters followed by a practical evaluation of these parameters on a single cylinder CI bench engine.

This involved choosing an adequate CR (19:1), bowl geometry, and injector nozzle design (HFR 310) as well as using advanced injection strategies (double and triple injections depending on the working point), thermal management (EGR cooler bypass) and VVT to promote internal EGR. These measures allowed satisfactory operation of the engine from full load to relatively low loads in terms of fuel consumption, NOx emissions and noise. However, gasoline's resistance to autoignition prevented the engine from using very high amounts of EGR (which also limited engine-out NOx reduction) or achieving very low loads (which limited the operating range). A first attempt to enhance performance using a glow plug was not successful which warranted additional simulation studies which are presented in this report.

The key conclusions from this study, including the results of the engineering paper study, GCI bench engine tests, and the CFD modelling studies are:

1. Achieving ignition and maintaining combustion stability in a GCI engine concept is challenging with market gasoline, but can be achieved over a large part of the speed/load range by applying well known optimization techniques. Over these operating regions, fuel consumption on an energy or gravimetric basis was better than today's state-of-the-art gasoline engines and comparable to state-of-the-art diesel engines.

2. NOx emissions and combustion noise were comparable to state-of-the-art diesel engines but with higher HC/CO emissions that would need to be controlled using a diesel oxidation catalyst.

3. The autoignition resistance of gasoline prevented the successful operation of this engine configuration at lower loads or when operating with higher EGR amounts.

4. Initial engine tests using a glow plug did not succeed in enabling this operation and CFD modelling studies were carried out to study whether an active ignition system might overcome the difficulties encountered at lower loads.

5. The flow simulations showed that VVT strategies can increase the in-cylinder gas temperature, enhancing gasoline's ignitability at low loads. The simulations also demonstrated that with about 20% internal EGR a temperature benefit of about 100K could be achieved at the SOI.

6. At low load operating conditions, modelling results indicate that stable combustion of gasoline in the GCI engine could be ensured only by a minimum of 20% internal EGR with the reduced I/E-Event cam shifting @ 48°CA. The spray spatial distribution and the local lambda field within the combustion chamber showed that the nozzle configuration selected for the bench engine study is suitable for a glow plug application.

7. Wider Nozzle Cone Angles (NCA) are required for a spark plug application at this lower load operating condition. Adopting an angle of 160° allows a spray pathway which would impact on the upper side of the piston bowl, guaranteeing more fuel-rich mixture in the area above the spark plug position.

8. An increase of the Nozzle Tip Protrusion (NTP) must be coupled with the use of wider NCAs to guarantee the proper share of mixture between the piston bowl and the squish volume for high load operating conditions. For the NCA of 160°, a value of NTP = 1.5mm guarantees the spark plug applicability also at higher loads.

8. GLOSSARY

ATDC	After Top Dead Centre
BMEP	Brake Mean Effective Pressure
CA50	Point in the combustion process where 50% of the injected fuel mass has been converted, also called the Centre of Combustion
^{o}CA	Degrees Crank Angle
CFD	Computational Fluid Dynamics
CI	Compression Ignition
CLCC	Closed Loop Combustion Control
CR	Compression Ratio
CSL	Combustion Sound Level
CTC	Characteristic Timescale for Combustion
DI	Direct Injection
DOC	Diesel Oxidation Catalyst
DPF	Diesel Particulate Filter
EGR	Exhaust Gas Recirculation
ERC	Engine Research Centre
FSN	Filter Smoke Number
GCI	Gasoline Compression Ignition
GHG	Greenhouse Gas
GPF	Gasoline Particle Filter
HFR	Hydraulic Flow Rate
I/E	Intake/Exhaust
IMEP	Indicated Mean Effective Pressure
KH	Kelvin-Helmholz
KIVA	Open access software for modelling chemically reacting sprays

LTC	Low Temperature Combustion
NA	Naturally Aspirated
NCA	Nozzle Cone Angle
NTP	Nozzle Tip Protrusion
RANS	Reynolds-averaged Navier-Stokes (equation)
RNG	Re-normalisation Group
RT	Rayleigh-Taylor
SI	Spark Ignition
SOI	Start of Injection
STAR CD	CFD-based modelling software
TC	Turbocharged
TDC-F	Top Dead Centre-Firing
VVT	Variable Valve Timing

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APPENDIX 1 GASOLINE PROPERTIES

A European CEC RF-02-08 Type Approval reference fuel containing 5% v/v ethanol (E5) was used for this study. A standard treat of a commercial European lubricity additive was also added but the lubricity of the resulting gasoline blend was not measured.

Test	Method	Units	Limit		Result
			Min	Max	
RON	ASTM D2699		95.0		95.8
MON	ASTM D2700		85.0		85.3
Density @ 15°C	IP 365	kg/L	0.743	0.756	0.7532
DVPE @ 37.8°C	EN 13016-1	kPa	56.0	60.0	59.8
Aromatics	ASTM D1319	% v/v	29.0	35.0	32.5
Olefins	ASTM D1319	% v/v	3.0	13.0	3.7
Saturates	ASTM D1319	% v/v			58.8
Benzene	ASTM D6730	% v/v		1.0	0.4
Oxidation Stability	ISO 7536	min	480		480
Existent Gum, Washed	ISO 6246	mg/100mL		4.0	<1.0
Copper Corrosion at 50°C-3hours	ISO 2160		Class 1		1a
Sulfur	EN ISO 20846	mg/kg		10.0	0.2
Ethanol	ASTM D6730	% v/v	4.7	5.3	5.0
Carbon	ASTM D5291	% m/m	Report		85.15
Hydrogen	ASTM D5291	% m/m	Report		13.03
Oxygen	ASTM D6730	% m/m	Report		1.82
Gross Calorific Value	IP 12	MJ/kg			45.01
Net Calorific Value	IP 12	MJ/kg			42.25

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