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Advanced combustion for low emissions and high efficiency
Part 1: Impact of engine hardware on HCCI combustion

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ABSTRACT

Two single-cylinder diesel engines were optimised for advanced combustion performance by means of practical and cumulative hardware enhancements that are likely to be used to meet Euro 6 emissions limits and beyond. These enhancements included high fuel injection pressures, high exhaust gas recirculation levels and charge cooling, increased in-cylinder swirl, and a fixed combustion phasing. These enhancements achieved low engine-out emissions of NOx and particulate matter emissions with engine efficiencies equivalent to today's diesel engines. These combustion conditions approach those of Homogeneous Charge Compression Ignition, especially at the lower part-load operating points.

Four fuels exhibiting a range of ignition quality, volatility, and aromatics contents were used to evaluate the performance of these hardware enhancements on engine-out emissions, performance, and noise levels.

KEYWORDS

Internal combustion engine technology, diesel, fuel, combustion, Homogeneous Charge Compression Ignition (HCCI), Controlled Auto-Ignition (CAI), EGR, exhaust emissions, cetane number, aromatics, volatility, kerosene, particulate matter (PM).

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SUMMARY

This study investigated the influence of different engine configurations and fuel properties on engine performance, efficiency, and emissions including noise. Two single-cylinder engines benchmarked for Euro 5 and Euro 6 emissions levels were optimized for advanced combustion performance. Various hardware configurations were tested that included a lower compression ratio, higher maximum cylinder peak pressure and rail pressure, optimised in-cylinder swirl, adjustment of fuel injection timing, and intensified exhaust gas recirculation (EGR). These hardware enhancements are practical options and will increasingly be used on modern diesel engines to achieve Euro 6+ emissions levels.

Four fuels exhibiting a range of ignition quality, volatility, and aromatics contents in the diesel and kerosene range were used to evaluate the performance of these hardware enhancements on engine-out emissions, performance, and noise levels. The engines could be operated on all four fuels at full load and at all part-load points.

The Phase B engine benchmarked for Euro 6 emissions was optimised individually on all four fuels using a Design of Experiments (DOE) approach. The main findings from the Phase B engine work were:

- Very low engine-out nitrogen oxides (NOx) levels could be achieved using high levels of cooled EGR to give cool combustion, while maintaining acceptable Particulate Matter (PM) emissions. A diesel particulate filter (DPF) would be needed to reduce particulate emissions as well as a diesel oxidation catalyst (DOC) to control hydrocarbon (HC)) and carbon monoxide (CO) emissions.
- If the combustion timing is maintained at the optimum point, fuel efficiency levels similar to current diesel vehicles can be achieved. In practical terms, maintaining the optimum combustion timing could be achieved by using a closed loop combustion control (CLCC) approach and an in-cylinder pressure sensor.
- Optimising the swirl at each load point resulted in a marked reduction of the NOx/PM trade-off. At lower loads, higher swirl levels were found to be especially beneficial.
- Switching off the pilot injection gave lower PM emissions and higher noise at the higher part-load points and higher HC and CO emissions at the lower part-load points. No clear advantage was found at all part-load points by switching off the pilot injection.
- Once the combustion timing had been optimized, all four fuels could be operated with the same fuel injection pressure, boost pressure and pilot injection quantity and offset. No further improvement in performance was found by optimizing these parameters individually for each fuel.

Overall, the engine hardware enhancements included in this study enabled a significant improvement of the emissions behaviour and fuel efficiency. In comparison with these improvements, the influence of the four test fuels on overall engine performance, emissions, and efficiency was relatively small.

This study investigated engine performance and emissions for a fully warmed-up engine at steady-state conditions only. Additional work would be needed to investigate the influence of fuel properties on engine performance under transient cycles and cold start conditions.

1. INTRODUCTION

Emissions of air pollutants from motor vehicles have fallen dramatically as a result of improvements in engine technology levels. Euro 5 emission regulations are now in force and further reductions in emissions will be required by 2014 for light duty vehicles and passenger cars through Euro 5b and 6 emissions standards. More recently, attention has focussed on fuel efficiency in order to address concerns over future energy supplies and Greenhouse Gas (GHG) emissions associated with energy use. Vehicle and engine technology has evolved rapidly in response to these two challenges and fuels have also changed to enable and assist the introduction of new technology.

Exhaust catalyst systems are an important factor in controlling air pollutant emissions and fuel sulphur levels have been dramatically reduced over the past two decades to enable these systems to function effectively. Gasoline and diesel fuels that are effectively sulphur-free (<10ppm) now cover 100% of liquid road transport fuels in Europe from 2009. Although other changes to fuel properties have been introduced, their effects on emissions are small compared to that of engines and advanced aftertreatment systems coupled with sulphur-free fuels. The existing sulphur-free fuels are expected to meet the needs of the vehicles introduced over the next several years.

In the search for improved fuel efficiency, attention is concentrating more closely on advanced engine combustion systems. Diesel engines are already efficient and the challenge now is to maintain or even improve this efficiency as air pollutant emissions are reduced further. Gasoline engines are currently less efficient and various approaches are being explored to bring their fuel consumption closer to diesel engine levels. Significant improvements seem possible through design improvements to existing engines. In particular, the use of a smaller and higher specific power gasoline engine is one approach that is likely to reduce fuel consumption.

For the longer term, engines using new and advanced combustion systems are being developed. If successful, these engines could combine good fuel efficiency with lower engine-out emissions. This could also reduce the demand on exhaust catalyst systems, which can themselves have a negative impact on fuel consumption depending upon the frequency with which the catalyst requires regeneration. Because these advanced combustion concepts combine features of both gasoline and diesel combustion, the optimum fuel characteristics may be different from those provided by conventional gasoline and diesel fuels today.

These advanced engine designs are the subject of this report. The terms ‘HCCI’ (Homogeneous Charge Compression Ignition) and ‘CAI’ (Controlled Auto Ignition) are often used to describe these developments. Broadly speaking, the advanced combustion concept entails substantially premixing the fuel and air and then combusting the mixture without spark initiation at relatively low temperatures. This concept has the potential to reduce both soot and NOx formation.

'True' HCCI combustion involves injecting fuel very early in the combustion cycle in order to provide sufficient time for thorough fuel-air mixing. Early injection makes it difficult to control the ignition process, however, and most researchers now favour later injection with high levels of exhaust gas recirculation (EGR) to control the combustion temperature. Such an approach retains most of the benefit of ‘true’ HCCI while allowing better control of the combustion process. HCCI combustion can be achieved most easily at lower engine speeds and loads and becomes increasingly difficult as power increases. The first production engines may therefore be 'part-time' HCCI engines, reverting to conventional diesel or gasoline operation at higher loads.

The search for practical combustion systems has led to a wide range of new acronyms describing particular engine solutions. In this study, we have used the term HCCI in its most generic sense to cover all those advanced combustion concepts that seek to provide:

- low engine-out emissions (particularly NOx and PM),
- low fuel consumption (comparable to or better than conventional diesel engines), and
- stable engine operation over a wide load range.

Many research and development groups around the world are working on practical ways to achieve this performance [1,2]. A recent literature survey [3] has shown that technology exists to produce combustion with very low soot and NOx emissions over a significant part of the engine load range, with one study reporting HCCI combustion even up to full load [4]. Diesel engines seem likely to be the basis for most HCCI applications because they utilize high fuel injection pressures and have the potential for high EGR rates and high boost pressures that allow the preferred combustion conditions to be achieved.

These technologies are comparatively new and it is not possible to predict precisely how they will develop. Because of their potential importance for future fuels, however, CONCAWE and FEV Motorentechnik GmbH have collaborated to investigate what can be achieved by practical future engine technology and how fuel properties could influence the effectiveness of these technologies.

Part 1 of this collaborative programme, reported here and summarized in [10], investigated how practical and cumulative engine hardware developments can reduce engine-out emissions while retaining acceptable noise levels and maintaining or improving fuel efficiency. A limited set of four fuels have been tested in Part 1, with the engine calibrations optimised for each fuel. These four fuels were designed to evaluate the effects of ignition delay, volatility, and molecular composition.

In a related study, referred to later as Part 2 [5,12], an engine configuration that was optimised in this Part 1 study was then tested under steady-state conditions on a wide range of both practical and experimental fuels. These included some fuels in the gasoline boiling range that are not traditionally associated with diesel engines. In each case, the combustion timing was adjusted to the same optimum position, simulating the behaviour of a future engine using CLCC [6,7].

The objectives of this Part 1 study were to:

- define the potential for advanced combustion to reduce engine-out emissions while maintaining efficiency, and
- investigate the influence of fuel properties on engine performance.

The study is restricted to a thorough evaluation of steady state effects including full load. More work is needed on transient cycles and cold engine operation to present a complete picture.

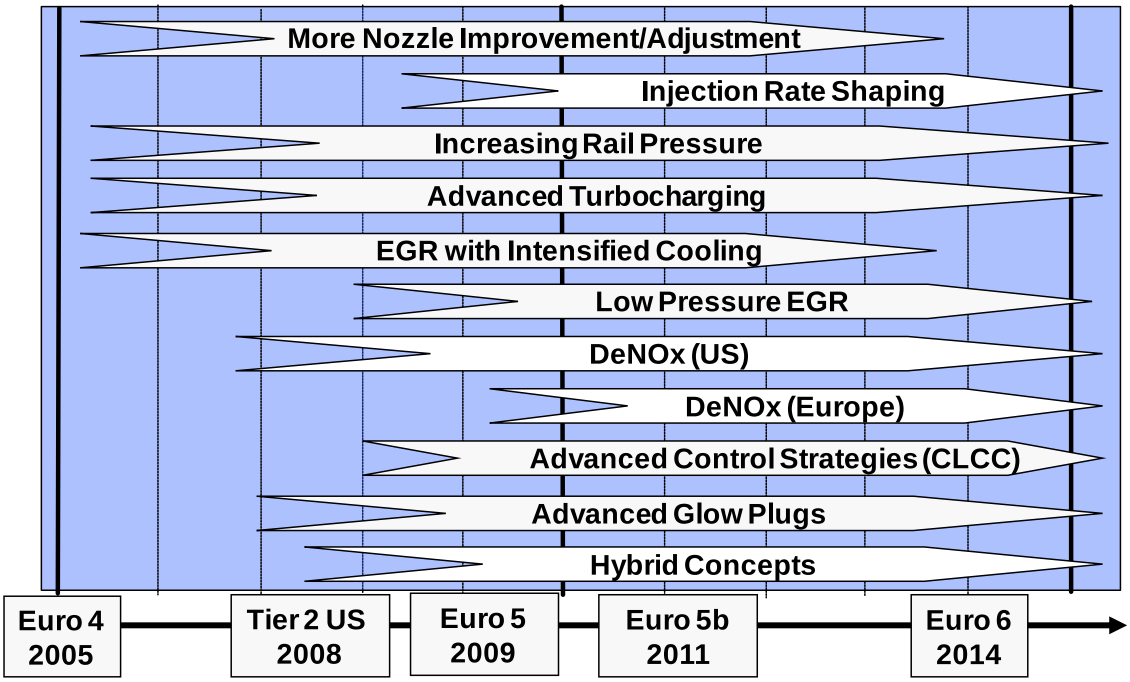
2. TEST PROGRAMME

The engine hardware enhancements in this programme have been selected to achieve optimised combustion behaviour for lowest engine-out emissions and highest fuel efficiency. Utilizing this concept, HCCI combustion is possible especially at lower engine loads but becomes more like conventional diesel combustion as the load increases. The concept applied here was to allow as much premixing of fuel and air as possible before auto-ignition occurs. Our success criterion, however, was the performance of the engine (in terms of emissions, efficiency, and noise) rather than the exact nature of the combustion. We have used engine hardware enhancements that are practical for future engines, including high fuel injection pressures for improved mixture formation, high EGR levels, and intensified charge air and EGR cooling to lower the temperature in the combustion chamber for a cooler combustion condition.

2.1. CONFIGURATIONS OF THE TEST ENGINES

This study was based on our assessment of engine and other hardware configurations that are likely to be required in order for future light-duty diesel vehicles to comply with forthcoming stages of European and US emissions legislation. An estimated hardware timeline is shown in **Figure 1**.

Figure 1 Possible hardware improvements needed to meet European and US emissions limits for light-duty diesel vehicles



In this study, two different engine configurations were evaluated. Initially, a single-cylinder research engine benchmarked for Euro 5 emissions limits was used, called the Phase A engine. This was followed by testing on a second single-cylinder research engine benchmarked for Euro 6 emissions limits, called the Phase B engine[1]. The Phase B engine included additional hardware enhancements that are expected to be needed to meet Euro 6 emissions limits and beyond. A downsizing concept was used because both engines had a comparatively small cylinder swept volume of 390 cm^3. This swept volume would allow the construction of a 1.6L 4-cylinder engine while maintaining the power of today's 2.0L engines.

While maintaining the swept volume of the Phase A engine, the Phase B engine had a reduced compression ratio (CR), a higher maximum cylinder peak pressure, and an optimised cylinder head concept. At the same time, the fuel injection equipment used on this engine was capable of a maximum rail pressure of 2000 bar. Due to the higher maximum rail pressure, a nozzle with a lower hydraulic flow rate (i.e. smaller nozzle hole diameters) was used to improve mixture preparation. For Euro 6 engines, it is assumed that CLCC will be applied, and, for this reason, the Phase B engine was also capable of simulating closed loop combustion control. The centre of combustion (CA50[2]) was kept constant when operating the engine with different fuels. Simulation of intensified EGR cooling was also possible with the Phase B engine configuration by using an improved charge air cooler efficiency.

The specifications of the two single-cylinder research engines evaluated in this study are shown in **Table 1**.

Table 1 Bench e***ngine specifications***

	Units	Phase A Engine	Phase B Engine
Emissions Benchmark	[-]	Euro 5	Euro 6
Single cylinder swept volume	[cm^3]	390	390
Stroke	[mm]	88.3	88.3
Bore diameter	[mm]	75	75
Compression ratio	[-]	16	15
Valves per cylinder	[-]	4	4
Maximum cylinder peak pressure	[bar]	160	220
Fuel injection equipment specifications:		Bosch Piezo Common Rail System	Bosch Piezo Common Rail System
Maximum injection pressure	[bar]	1600	2000
Nozzle	[-]	8 x 153°	8 x 153°
Nozzle hole diameter	[µm]	120	109
Hydraulic Flow Rate (HFR)	[cm^3/30s]	370	310
Charge air cooling level	[-]	Euro 5	Euro 6
Variable swirl	[-]	No	Yes with Variable Valve Lift (VVL)

1 More details on the Phase B engine are given in [8] where the combustion system is referred to as "High Efficiency Combustion System" (HECS).

2 CA50 is the point in the combustion cycle where 50% of the injected fuel has been converted to energy.

For the purpose of engine optimisation and calibration, limits of engine operation achievable by today's production engines were used in this study. The maximum boost pressure was calculated based on the capabilities of a two-stage turbocharger on a multi-cylinder engine using GT-Drive[3]. Additionally, the maximum rail pressure was limited according to the capabilities of a multi-cylinder engine.

Modern Euro 5 engines already have the ability to intensify the in-cylinder swirl by deactivating one inlet port. The Phase B engine used a different approach to investigate the potential of swirl for improving fuel efficiency and reducing emissions. This consisted of a Variable Valve Lift (VVL) concept combined with a seat swirl chamfer at each of the two intake valves. This system allowed a high flow coefficient and a very good filling of the combustion chamber to be achieved while retaining the possibility to vary the swirl over a wide range.

For the engine studies reported here, VVL was achieved by using different camshafts. As shown in **Figure 2,** higher swirl levels in the combustion chamber were possible by using camshafts that shorten the valve lift during the valve opening event.

Figure 2 In-cylinder swirl characteristics as a function of valve lift

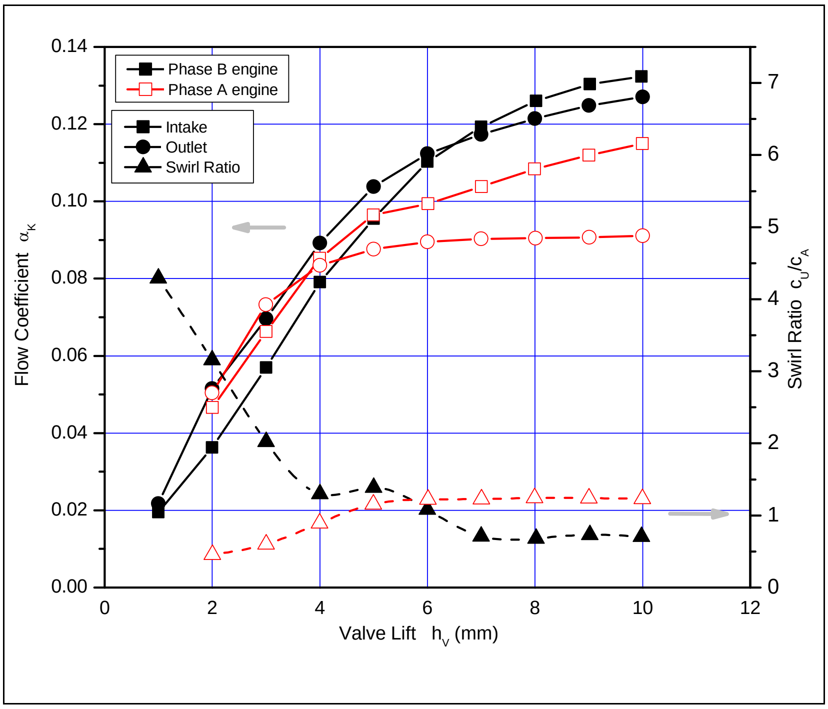
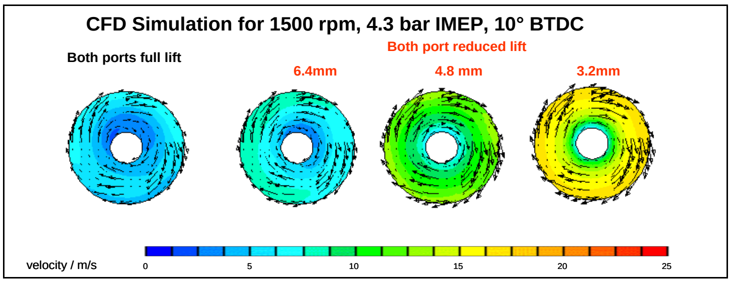


Figure 3 shows CFD[4] simulations of the effect of different valve lifts on the in-cylinder swirl. In addition to the standard valve lift of 8 mm, three shorter valve lifts of 6.4, 4.8, and 3.2 mm were also tested that led to higher swirl levels in the combustion chamber. This figure shows that very good homogeneity of the swirl level inside the chamber can be achieved with this system.

[3] Gamma Technologies software for simulating vehicle test cycles for load point and emissions estimation.
[4] Computational Fluid Dynamics

Figure 3 Cross-sectional simulation of the in-cylinder circumferential charge air velocities with four different valve lifts



2.2. TEST FUELS

For this study, four fuels were selected to investigate the influence of ignition delay, volatility, and molecular composition on combustion performance. **Table 2** shows the properties of these four fuels. In this study, the Derived Cetane Number (DCN), as measured in an Ignition Quality Tester (IQT) [11], has been used to characterize the cetane number (CN) of each fuel. All fuel properties are summarized in **Appendix 1** of [12].

The reference fuel for the Part 1 study was an ultralow-sulphur European diesel fuel that complied with EN590 specifications, called the Baseline Diesel. Compared to this fuel, the Low Cetane Diesel fuel had a lower DCN (reduced from 53 to 44) while the aromatic content and volatility were almost the same. The Low Aromatics Diesel had about the same DCN as the Baseline Diesel but with very low aromatics (2.2% m/m aromatics compared to 24.9% m/m in the Baseline Diesel). The Kerosene was more volatile than the three diesel fuels but had a similar DCN and aromatics level to the Low Cetane Diesel.

Table 2 Properties of the four fuels investigated in t***his*** study

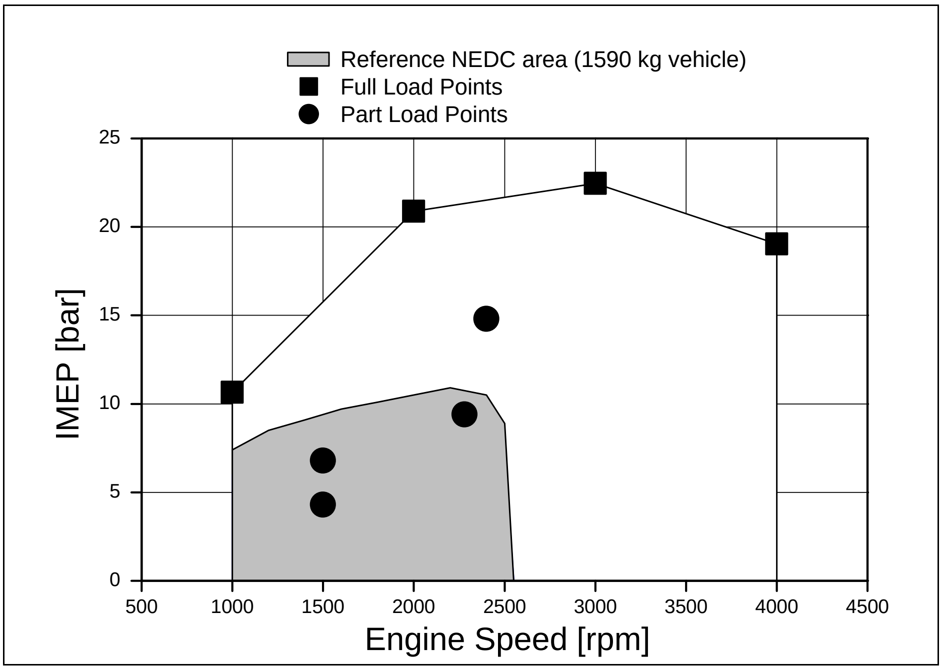
Fuel Property	Units	Baseline Diesel (DCN53)	Low Cetane Diesel (DCN44)	Low Aromatics Diesel (DCN53)	Kerosene (DCN47)
Density @ 15°C	[kg/m³]	832.9	841.8	812.3	801.1
Derived Cetane Number	[DCN]	52.6	44.2	52.9	46.5
Total Aromatics Content	[% m/m]	24.9	27.1	2.2	19.3
Sulphur Content	[mg/kg]	7	39	41	226
Lower Heating Value (LHV)	[MJ/kg]	43.0	42.89	43.56	43.01
Distillation 10%	[°C]	195.4	184.0	205.5	165.6
Distillation 50%	[°C]	274.9	241.0	227.1	199.0
Distillation 90%	[°C]	328.9	338.6	329.3	253.1

3. TEST PROCEDURE AND METHODOLOGY

3.1. ENGINE TEST POINTS

The engine investigations included analysis of the engine performance and the fuel influence at part-load and full load conditions. All load points tested within this programme are displayed in **Figure 4**. In this figure, the engine load is shown as Indicated Mean Effective Pressure (IMEP). The typical speed and load range spanned by the European regulatory emissions cycle (New European Driving Cycle (NEDC)) for a 1590 kg vehicle is also shown.

Figure 4 Load points selected for this study



At the start of this study, it was decided to investigate three part-load points. A fourth lower part-load point at 1500 rpm, 6.8 bar was added later at the start of testing on the Phase B engine.

Typically, HCCI combustion can only be sustained at lower loads. It is important, however, to evaluate the engine's emissions and performance over the entire engine operating envelope, including the higher part-load operating points. Three lower part-load points are positioned within the range of the NEDC (based on a typical vehicle configuration, for example, 1.6L engine, 1590 kg vehicle mass). A fourth and higher load point was selected outside the NEDC range to gain information about the engine performance and the fuel influences at higher loads that are important for real-world driving, other driving cycles (e.g. US06) and possible additional engine downsizing. The four part-load operating points are shown in **Table 3**.

Table 3 Part-load operating points

Engine Speed	IMEP	Note
[1/min]	[bar]	
1500	4.3	
1500	6.8	Load point added for Phase B engine testing
2280	9.4	
2400	14.8	

The full load behaviour was evaluated at four engine speeds to assess overall engine operation. Full load operation was constrained by either the Filter Smoke Number (FSN) or the maximum exhaust gas temperature since the exhaust gas temperature is limited for turbochargers on production engines. The four full load operation points are shown in **Table 4**.

Table 4 Full load operating points

Engine Speed	Smoke Limit	Exhaust Gas Temperature Limit
[1/min]	[FSN]	[°C]
1000	2.6	820
2000	1.7	820
3000	2.2	820
4000	2.4	820

3.2. ENGINE CONFIGURATIONS

Throughout this study, six different engine configurations were used that differed in the hardware and the injection strategy used. The following terminology has been defined:

Configuration 1: Phase A engine (Euro 5 hardware)

Configuration 2: Phase B engine with lower Compression Ratio (CR) and standard swirl (Euro 6 hardware)

Configuration 3: Phase B engine with lower CR and optimised swirl by Variable Valve Lift (VVL)

Configuration 4: Phase B engine with lower CR, optimised swirl, and enhanced charge air cooling

Configuration 5: Phase B engine with lower CR, optimised swirl, and no pilot injection. The CA50 was kept constant when the pilot injection was switched off.

Configuration 6: Phase B engine with lower CR, optimised swirl, and multiple fuel injections. Only a short study was completed with multiple injections, such as, split main injection or multiple pilot injections. These results are presented in **Section 4.4**.

Configuration 1 (the Phase A engine) was benchmarked for Euro 5 emissions limits while Configuration 2 (the Phase B engine) was benchmarked for Euro 6 emissions limits. Configurations 3-6 were enhancements based on Configuration 2 but featured even better emissions potential than Euro 6 emissions limits.

3.3. ENGINE CALIBRATION AND OPTIMISATION

At part-load on the Phase A engine, the calibration was optimised on the Baseline Diesel fuel only, using a Design of Experiments (DOE) approach. This approach was used to define the optimum fuel injection pressure, boost pressure, and injection timing strategy. More details on the DOE approach used in this study are included in **Appendix 1**.

The first tests on the Phase B engine were conducted using the optimisation of the Phase A engine to see only the influence of hardware changes. After this, the calibration of the Phase B engine was optimised individually using the DOE approach for all fuels at part-load conditions (Configuration 2).

When optimising the engine, the following main engine parameters were varied:

- Beginning of Pilot Injection (BOI-Pilot)
- Duration of Pilot Injection (DOI-Pilot)
- Beginning of Main Injection (BOI-Main)
- Fuel Rail Pressure
- Boost Pressure
- Exhaust Gas Recirculation (EGR) Rate

On the test bench, all of these parameters could be varied essentially independently of one another.

The DOE approach was used when optimising the engine for a given load point and fuel. Within a defined experiment range, the selected input parameters (BOI-Main, rail pressure, boost pressure, and EGR rate) were applied in different combinations to the engine. With the resulting engine data from the test bench, models of the emission and fuel efficiency behaviour could be calculated as a function of the input parameters.

With experience, it was found that a full DOE optimisation procedure was not needed in every case and that the individually optimised calibrations for the different fuels could be harmonized. Generally, the primary optimisation requirement was the adjustment of the fuel injection timing to give a constant centre of combustion. The combustion timing giving best efficiency was selected, with the CA50 being about 5-11 degrees Crank Angle (°CA) After Top Dead Centre (ATDC). This approach simulates the behaviour of a future engine operating with CLCC.

The centre of combustion was brought to the optimum position by adjusting the fuel injection timing. Most of the engine tests were performed with a single pilot injection but some tests were also conducted without pilot injection. When a single pilot injection was used, the pilot injection quantity and advance before the main injection were kept constant. The volume of fuel in the main injection varied slightly between fuels in order to maintain the required Indicated Mean Effective Pressure (IMEP), reflecting variations in the energy content of the fuels.

Optimisation of the timing was performed at a NOx level close to Euro 5. Final EGR swings were conducted to characterise the performance of the final calibration without any further adjustment of the injection timing. In this study, the EGR swings are shown as a function of the NOx level, rather than as a function of the EGR rate. This is because the use of EGR is mainly targeted at reducing the engine-out NOx emissions.

In the swirl investigations, the swirl level was optimised for each load point. At each load point, EGR swings with four different swirl levels were conducted on the Phase B engine using the calibration (injection quantity and timing and boost pressure) optimised for the standard swirl. A swirl level was selected based on the emissions and fuel efficiency behaviour. The engine was then DOE-optimised only with the Baseline Diesel. Again, final EGR swings were conducted with all four fuels at all load points using the optimised calibration of the Baseline Diesel (Configuration 3).

As a next step, the EGR cooling was intensified. To evaluate the influence on the engine performance, EGR swings at all part-load points were performed for all four fuels. These tests were performed with the optimised swirl level for each part-load point (Configuration 4).

With the optimised swirl level (but not with enhanced EGR cooling), the effect of switching off the pilot injection was also investigated. When the pilot injection was switched off, the centre of combustion was again kept constant by adjusting the timing of the main injection (Configuration 5).

Finally, a short study was carried out with different multiple injection strategies, such as a split main injection or multiple pilot injections. These investigations were performed with the optimised swirl level for each part-load point (Configuration 6).

3.4. WHAT WAS MEASURED

On single cylinder engines, the load is usually specified by the Indicated Mean Effective Pressure (IMEP) which is calculated on-line from the recorded cylinder pressure traces. The Brake Mean Effective Pressure (BMEP) for a full engine can be calculated by estimating a Friction Mean Effective Pressure (FMEP) for a full engine.

At the test bench, emissions of CO, HC, and NOx emissions were measured directly at the engine-out exhaust. Smoke emissions were measured using an AVL 415S Smoke Meter, which also provided a read-out of PM emissions using an internal conversion formula. These emissions are shown as “indicated specific” (g/kWh) in the figures included in this report.

The fuel consumption was measured volumetrically by determining the engine runtime needed to consume a defined volume of fuel. Then, by using the density of the fuel, the gravimetric fuel consumption could be calculated. Because fuels with different energy contents were used in this study, the engine efficiency is more important than the fuel consumption and was calculated taking into account the energy content of each fuel.

In order to evaluate the noise emissions, combustion noise was calculated from the recorded cylinder pressure traces using an average of about 50 cycles. The methodology is described in [9]. The overall noise outside the engine was then calculated taking into account attenuation through the engine structure and including mechanical noise. This overall engine-out noise is reported here as the Combustion Sound Level (CSL).

3.5. PERFORMANCE CRITERIA

While optimising the bench engines, it was assumed that future production engines will use an HC/CO diesel oxidation catalyst (DOC) and a diesel particulate filter (DPF). To control NOx emissions in passenger cars, active DeNOx systems, such as Selective Catalytic Reduction (SCR) or storage catalysts, will preferably be avoided due to their complexity and cost. To realize this simplification, however, sufficient levels of EGR must be applied to achieve very low engine-out NOx emissions. The bench engine was optimised based on these considerations.

The success criteria of the optimisation included the following factors:

- low engine-out NOx;
- PM, HC, and CO emissions as low as possible and suitable for further reduction by aftertreatment;
- an acceptable engine noise and
- a fuel efficiency that was at least as good as the base configuration of the engines.

Based on the New European Driving Cycle (NEDC), maximum Indicated Specific NOx emissions (ISNOx) targets were estimated for each load point and for different Euro emission levels. The estimated maximum ISNOx levels are shown in **Table 5.** These targets take into account the expected impact of transient operation and cold start and include an appropriate engineering margin.

Table 5 Estimated maximum ISNOx levels for different Euro emission levels

ISNOx [g/kWh]	1500 rpm 4.3 bar IMEP	1500 rpm 6.8 bar IMEP	2280 rpm 9.4 bar IMEP	2400 rpm 14.8 bar IMEP
Euro 4 limits	0.6	0.6	1.8	2.25
Euro 5 limits	0.4	0.4	1.2	1.50
Euro 6 limits	0.2	0.2	0.6	0.75

Although the highest part-load point was outside the NEDC range, ISNOx values were also estimated for this point. This point was selected as a reference for possible future driving cycles or for further engine downsizing.

3.6. REPEATABILITY OF ENGINE TESTS AND EVALUATION OF FUEL EFFECTS

The process of engine optimisation and evaluation at each load point is time-consuming so it was not practical to carry out statistically meaningful repeat tests for the entire programme. However, some repeat tests (including repeat tests within a DOE) were carried out on the Baseline Diesel and these tests gave reassurance that engine drift was not a concern. Based on these results, a comparison of the different engine configurations and fuels could be made with confidence.

To increase the reliability of the fuel comparisons, short-term 'back-to-back' tests were performed at the end of the study using the optimised calibrations (where the fuels are compared at a constant centre of combustion) and hardware tested during the programme.

4. RESULTS

4.1. BASELINE OPTIMISATION AND FUEL EFFECTS ON THE PHASE A AND PHASE B ENGINES

4.1.1. Baseline Optimisation in the Phase A Engine (Configuration 1)

The Phase A engine was optimised on the Baseline Diesel fuel using the Design of Experiments (DOE) approach. **Table 6** shows the “global optimisation” found for the three initial part-load points. The background for this calibration is described in **Appendix 1**.

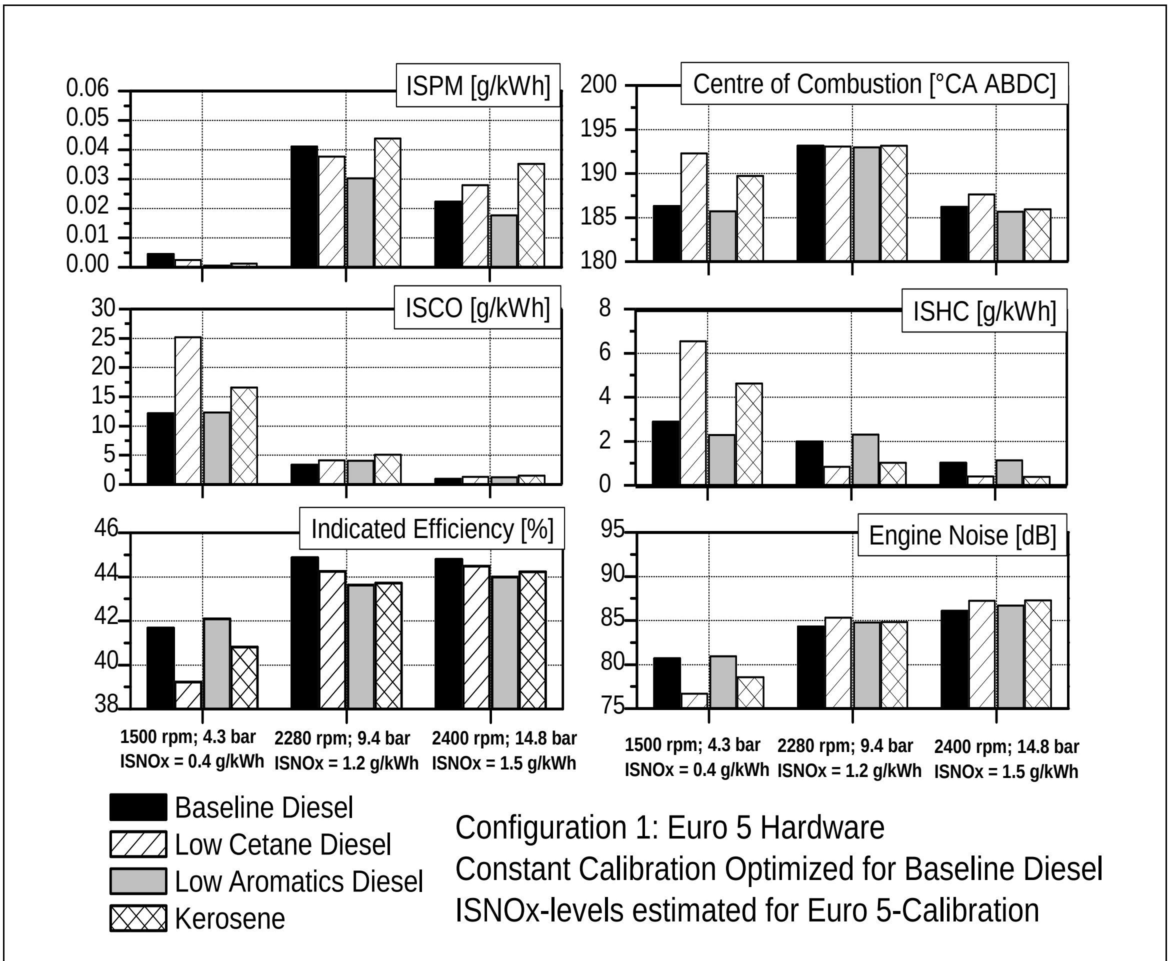
Table 6 Global optimisation of the Phase A engine

Engine Speed	[rpm]	1500	2280	2400
IMEP	[bar]	4.3	9.4	14.8
BOI-Main	[°CA BTDC]	3.1	2.5	3.0
BOI-Pilot	[°CA BTDC]	12.1	32.0	35.0
DOI-Pilot	[µs]	150	145	125
Rail Pressure	[bar]	1046	1120	1600
Boost Pressure	[bar abs]	1.41	2.10	2.55
Exhaust Gas Back Pressure	[bar abs]	1.47	2.20	2.75
Temperature downstream of the intercooler	[°C]	25	35	45

4.1.2. Fuel effects in the Phase A Engine (Configuration 1)

The emissions behaviour of the Phase A engine at the Euro 5 NOx emissions level is shown in **Figure 5** for the four test fuels**.** The engine calibrations are as optimised on the Baseline Diesel and shown in **Table 6**. Three different part-load points are shown.

Figure 5 Comparison of fuel effects on the Phase A engine with the engine calibrated on the Baseline Diesel fuel only



The NOx levels at each test point were those required to meet Euro 5 emissions limits, as indicated in **Table 5**. Because the injection timing was fixed in these tests, the centre of combustion varied between fuels.

At the lowest part-load point, the Low Aromatics Diesel and the Kerosene showed almost no PM emissions even at very low NOx emissions. The Baseline Diesel and the Low Cetane Diesel showed slightly higher PM emissions. (Examples of NOx/PM trade-off curves are shown in **Figures 6** and **7**.) At the same time, the CO emissions increased for the Low Cetane Diesel and Kerosene fuels. At higher engine loads, the Low Aromatics Diesel in particular showed some potential to reduce engine-out PM emissions. The CO emissions at higher loads were very low.

The indicated efficiency was quite similar for all test fuels except at the lowest load point where the lower cetane number fuels gave lower efficiency. Because these tests were conducted at fixed injection timing, this could be related to later combustion on these lower cetane fuels. In general, a slight advantage for the Baseline Diesel could be seen. The HC emissions were generally very low. At higher engine loads, HC emissions were lower for the Low Cetane Diesel and Kerosene fuels.

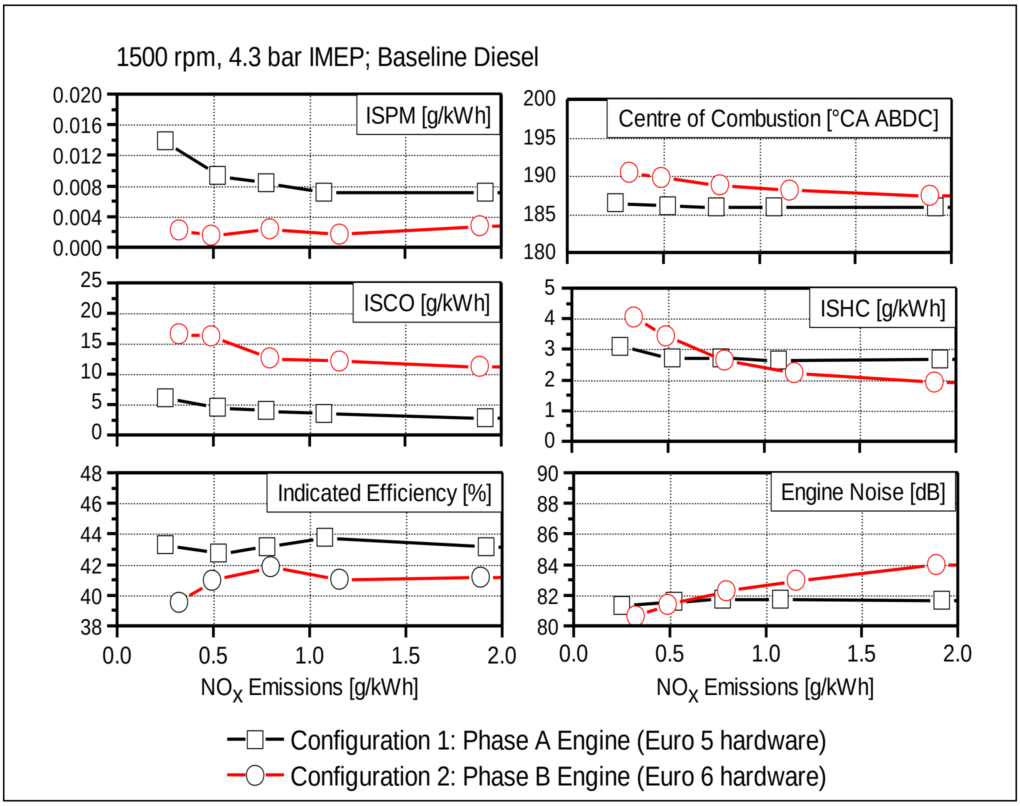
Noise levels increased at the higher load points, but were maintained at acceptable levels for all test points.

In the Phase A engine operated with the same optimised calibration as for the Baseline Diesel, it was observed that the pilot injection did not burn for the Low Cetane Diesel and Kerosene fuels at the lowest part-load point. This was presumably due to the lower cetane numbers for these two fuels.

4.1.3. Switch to the Phase B Engine (Configuration 2)

After switching to the Phase B engine, initial tests were performed on the Baseline Diesel using the same optimised calibration (injection pressure, rail pressure, boost pressure) found for the Phase A engine (Configuration 1). The following figures compare the results obtained on the Phase A and B engines using this same calibration. The following examples show the lowest (**Figure 6**) and the medium part-load points (**Figure 7**).

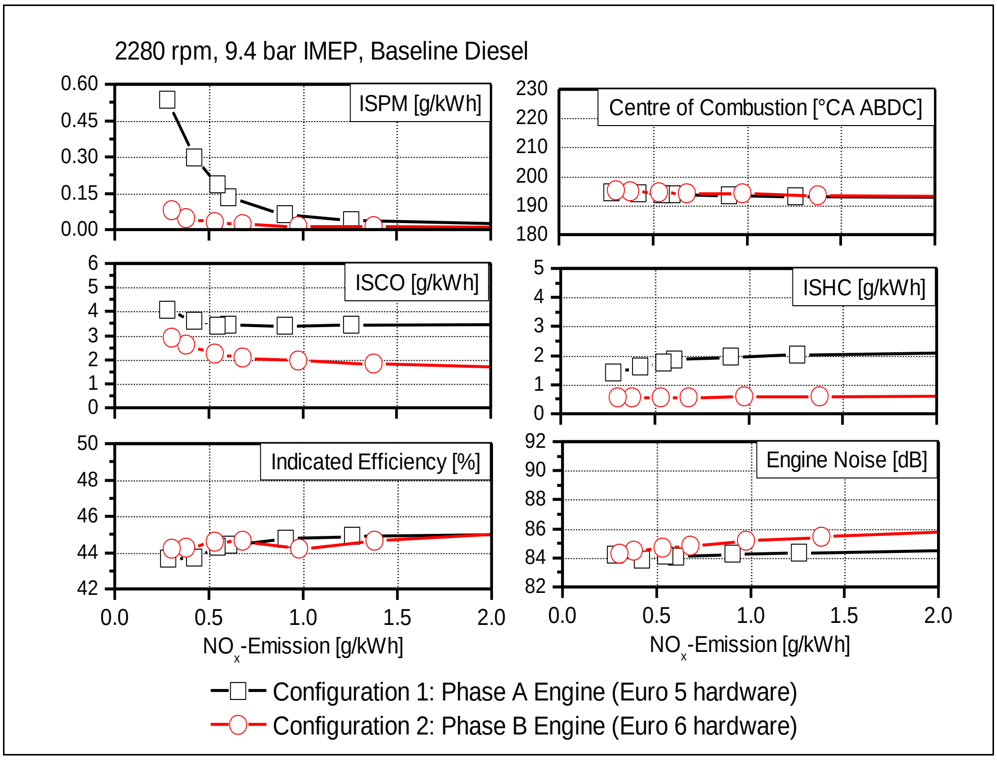
Figure 6 Comparison of the Phase A and Phase B engines at 1500 rpm, 4.3 bar IMEP using the Baseline Diesel fuel and the calibration for the Phase A engine



At the 1500 rpm, 4.3 bar IMEP part-load point, lowering the Compression Ratio from 16 in the Phase A engine to 15 in the Phase B engine resulted in a later (retarded) centre of combustion. As expected, this later centre of combustion led to a lower indicated efficiency and increased the CO emissions by a factor of 3. This increase could be problematic for the oxidation catalyst because of the low exhaust gas temperatures at this part-load point. Therefore, a recalibration was necessary, including a correction of the combustion phasing.

The PM emissions were significantly lower on the Phase B engine while the HC and noise emissions depended strongly on the EGR rate.

Figure 7 Comparison of the Phase A and Phase B engines at 2280 rpm, 9.4 bar IMEP using the Baseline Diesel fuel and the calibration for the Phase A engine

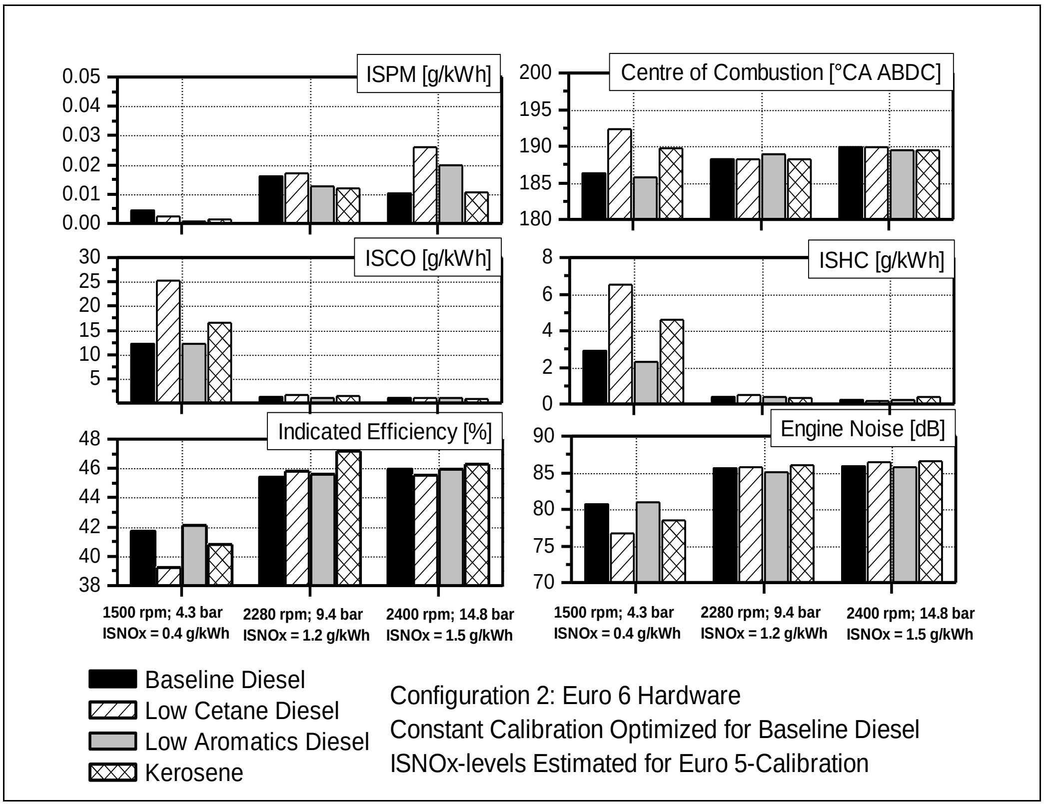


At the 2280 rpm, 9.4 bar IMEP part-load point, there was only a small change in the centre of combustion. For this reason, the indicated efficiency also stayed at about the same level. The Phase B engine had much lower emissions at this part-load point. The PM, CO, and HC emissions were all significantly lower although there was a slight increase in noise emissions.

4.1.4. Fuel Effects in the Phase B Engine (Configuration 2)

Initially, the Phase B engine was DOE-optimised just on the Baseline Diesel fuel. Using this optimised calibration of injection timing, rail pressure and boost pressure, the emissions behaviour of all four test fuels are compared in **Figure 8** at the Euro 5 NOx emission level in the Phase B engine.

Figure 8 Comparison of results on all four test fuels on the Phase B engine using the optimization obtained on the Baseline Diesel fuel



In general, a marked difference in engine performance can be seen for PM emissions at the high part-load point. All other emissions remained on a similar level for the different test fuels. At the lowest part-load point, an influence on CO and HC emissions can also be seen presumably as a consequence of the influence of the different fuels on the centre of combustion. As the centre of combustion moved later in the engine cycle, the indicated efficiency strongly decreased, especially for the Kerosene and the Low Cetane Diesel fuels.

These results stimulated questions regarding what improvement potential would be possible when using additional engine hardware enhancements and individual calibration optimisations for each test fuel.

4.1.5. DOE Optimisations on the Configuration 2 Engine

The Phase B (Configuration 2) engine was then individually DOE-optimised for each fuel to achieve the best Indicated Specific Fuel Consumption (ISFC).

Table 7 shows the ISFC-optimisations for the four fuels where it can be seen that the optimised calibrations are quite similar for all fuels. At the lowest part-load point, the only significant difference is in the beginning of the main injection (BOI-Main). As the cetane number of the fuel was lowered, the BOI-Main shifted to an earlier injection timing in order to keep the centre of combustion at the optimum point for the ISFC-optimisation. At the higher part-load points, the calibration differences were smaller and, at the highest part-load point, the optimised calibrations for all fuels were the same.

It was observed that the calibrations could be harmonized if CLCC was used to keep the centre of combustion constant for different fuels. Final back-to-back tests were performed on all fuels in which the centre of combustion was kept constant and with a fixed pilot offset and injection duration. The results of these final back-to-back tests are given at the end of this report and covered in more detail in the Part 2 report [12].

Table 7 ISFC optimisation of the Phase B engine

	BOI-Main	**Centre of Combustion**	**Pilot Offset**	**DOI-Pilot**	**Rail Pressure**	**Boost Pressure**
	[°CA BTDC]	[°CA BTDC]	[°CA]	[µs]	[bar]	[bar]
1500 rpm, 4.3 bar IMEP					720	1.07
Baseline Diesel	8.3		10	180	720	1.07
Low Cetane Diesel	14.6		10	180	720	1.07
Low Aromatics Diesel	8.0		10	180	720	1.07
Kerosene	11.2		10	180	720	1.07
Harmonised	variable	-6.6 @ 0.5 g/kWh ISNOx	10	180	720	1.07
1500 rpm, 6.8 bar IMEP						
Harmonised	variable	-5.8 @ 0.5 g/kWh ISNOx	11	140	904	1.5
2280 rpm, 9.4 bar IMEP						
Baseline Diesel	6.0		14	135	1399	2.29
Low Cetane Diesel	8.1		20	120	1129	2.30
Low Aromatics Diesel	8.0		20	120	1419	2.12
Kerosene	6.2		14	120	1401	2.12
Harmonised	variable	-9.2 @ 0.5 g/kWh ISNOx	20	120	1399	2.29
2400 rpm, 14.8 bar IMEP						
Baseline Diesel	7.0		28	120	1800	2.60
Low Cetane Diesel	7.0		28	120	1800	2.60
Low Aromatics Diesel	7.0		28	120	1800	2.60
Kerosene	7.0		28	120	1800	2.60
Harmonised	variable	-10.8 @ 1.0 g/kWh ISNOx	28	120	1800	2.60

Figures 9 and **10** show the effect that the fuel had on the centre of combustion at the lowest part-load point. As the cetane number of the fuel was reduced, the same start of injection resulted in a retarded combustion (**Figure 9**). So, in order to keep the centre of combustion constant, the beginning of injection was adjusted. With this adjustment, the pressure traces then overlapped, ensuring that fuel effects were not confounded with engine calibration effects (**Figure 10**).

Figure 9 Pressure traces at 1500 rpm, 4.3 bar IMEP in the Phase B engine for fixed pilot and main injection timings

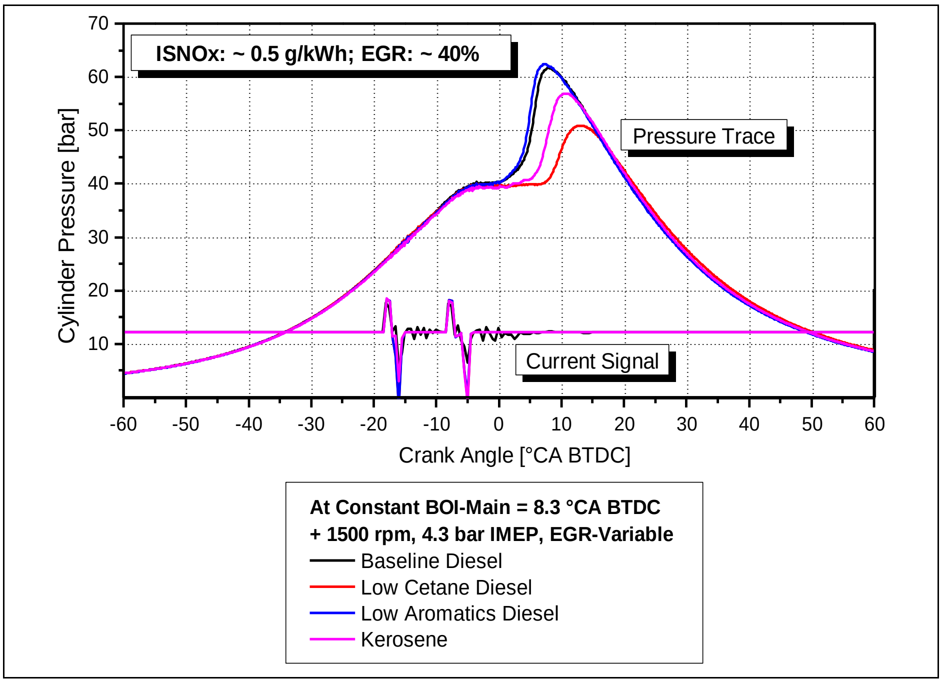


Figure 10 Pressure traces at 1500 rpm, 4.3 bar IMEP in the Phase B engine with injection timings adjusted to achieve a constant centre of combustion

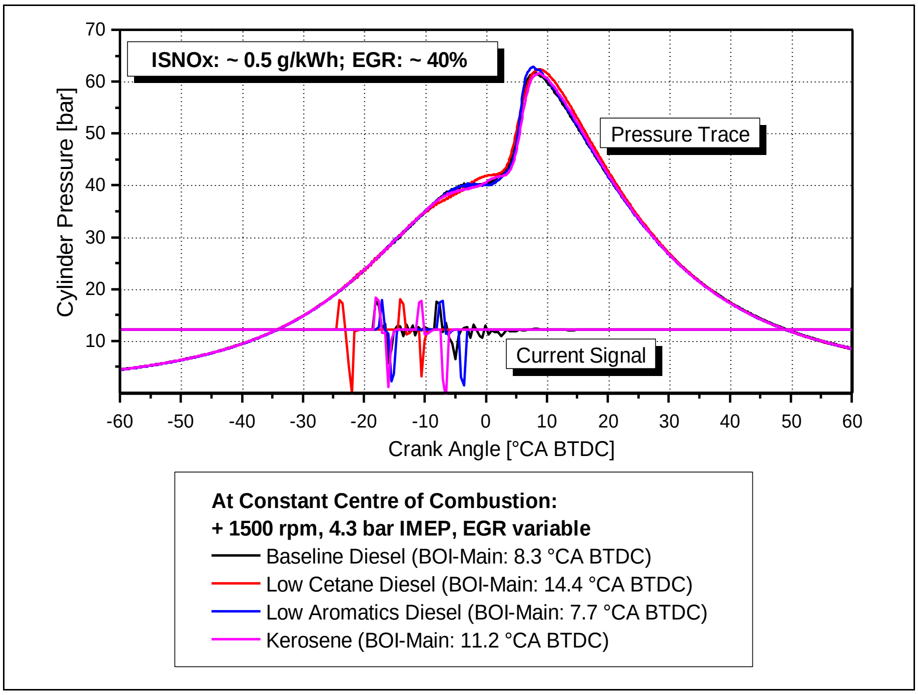
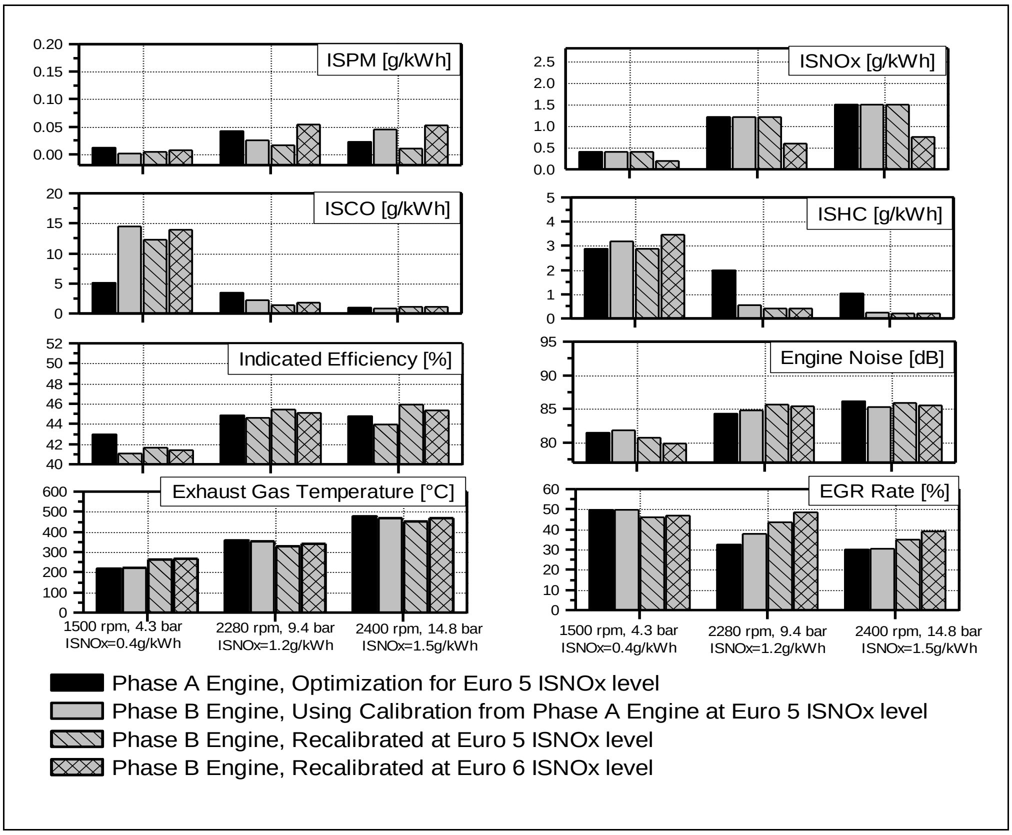


Figure 11 compares the results from Configurations 1 and 2 for the Phase A and B engines, respectively. The comparison of the Phase A and B engines using the same calibration optimised for the Phase A engine was discussed previously. When comparing results at the Euro 5 NOx level, the recalibration of the Phase B engine generally led to better fuel efficiencies and improved HC and CO emissions. At higher engine loads, the PM emissions were markedly lower but the noise emissions increased slightly.

A further reduction of the NOx emissions to Euro 6 levels was achieved by using higher EGR rates. As expected, this increased the PM, HC, and CO emissions and lowered the fuel efficiency and noise emissions. Since the NOx levels were 50% lower for Euro 6 compared to Euro 5, the observed change in emissions and fuel efficiency performance on recalibration is very modest.

Figure 11 Comparison of engine Configurations 1 and 2 with the Baseline Diesel fuel



4.2. OPTIMISED SWIRL

The next engine enhancement included variable swirl (Configuration 3). With this enhancement, the swirl level was individually optimised for all part-load points using the Baseline Diesel. **Figure 12** shows the NOx/PM trade-offs for the different part-load points and swirl levels. Taking the other emissions into account, one swirl level was selected for each part-load point. A higher swirl level was found to be beneficial at the lower part-load points. The relative air/fuel ratios and the EGR rates for the different part-load points and in-cylinder swirl levels are shown in **Appendix 2**.

Figure 12 NOx/PM trade-off curves for the Baseline Diesel fuel and different swirl levels

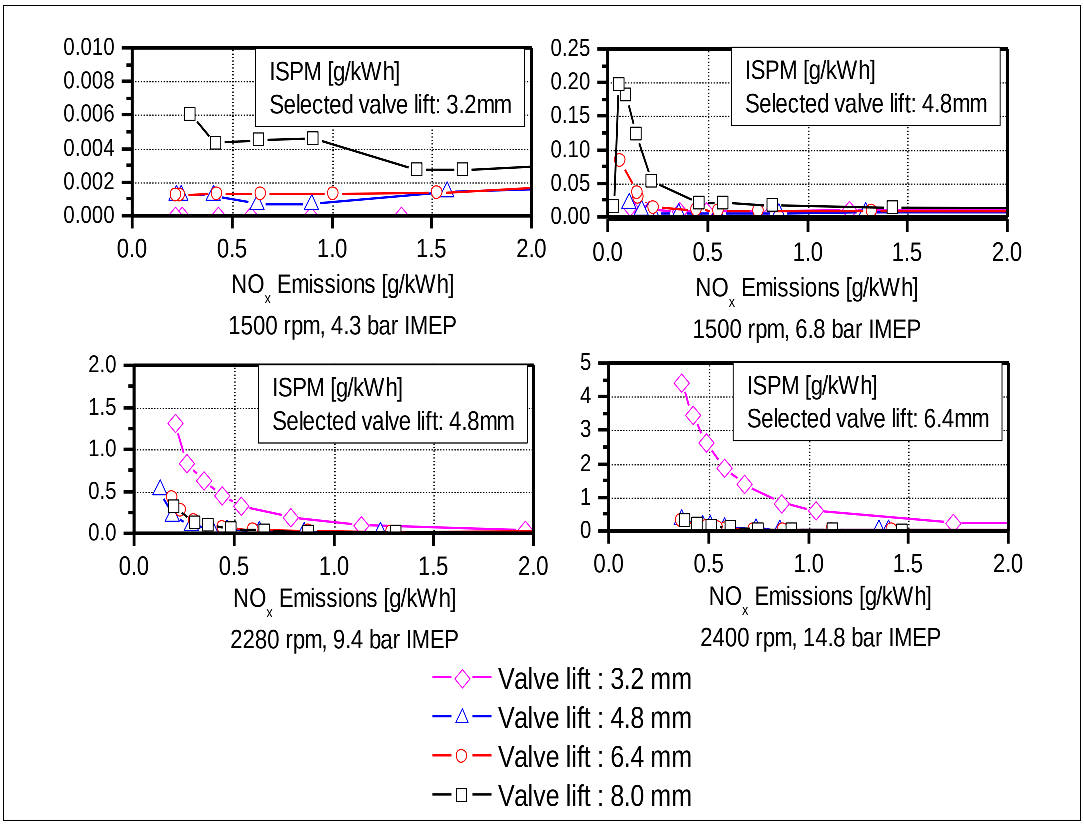


Table 8 compares the valve lifts optimised for each part-load operating point.

Table 8 Optimised valve lifts and swirl levels at each part-load operating point

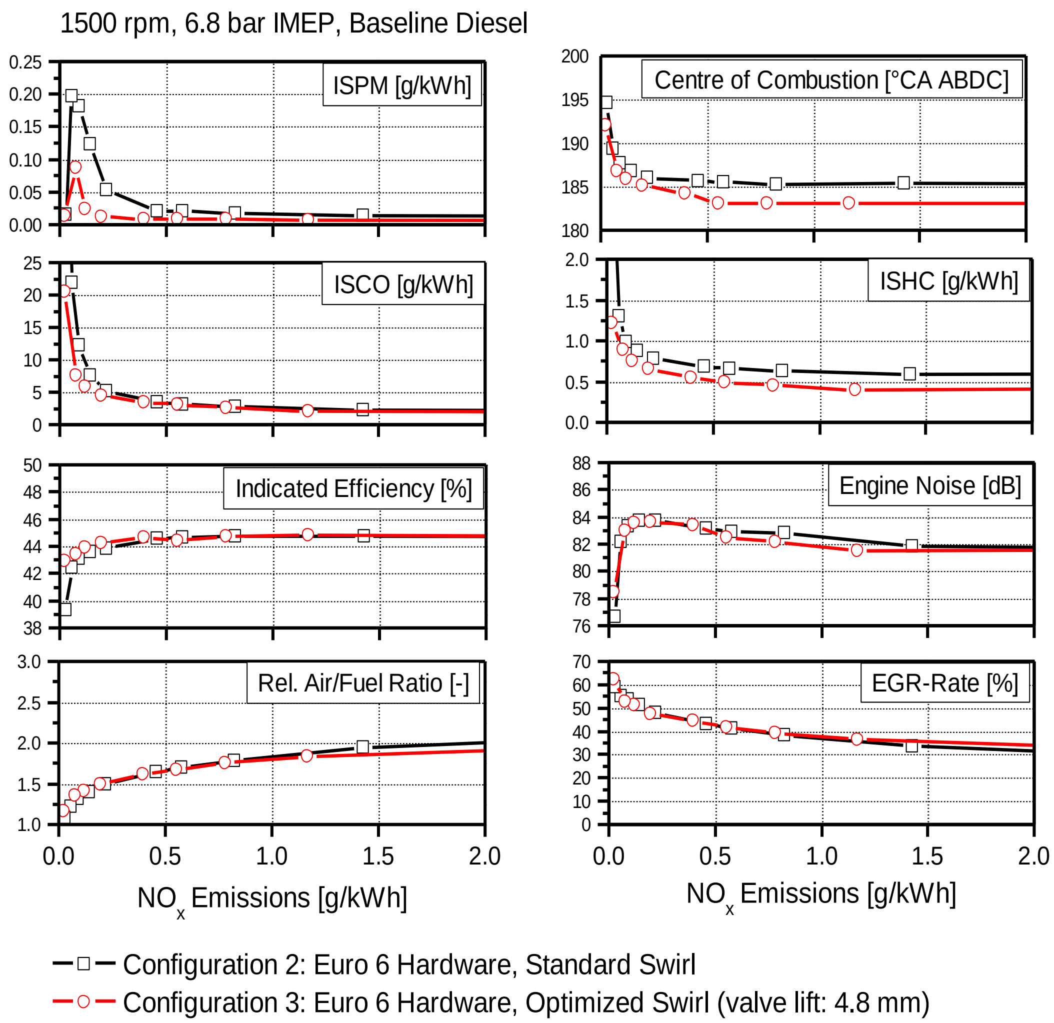
Load Point	Optimised Swirl [mm valve lift]	Swirl Level
Full Load	8.0	
2400 rpm 14.8 bar IMEP	6.4	
2280 rpm 9.4 bar IMEP	4.8	
1500 rpm 6.8 bar IMEP	4.8	
1500 rpm 4.3 bar IMEP	3.2	High Swirl

Each part-load point was DOE-optimised for the Baseline Diesel with the optimised swirl. Finally, EGR swings were performed with the optimised calibration setting.

Figure 13 shows the final EGR swings (Baseline Diesel) for Configurations 2 and 3. At this part-load point (1500 rpm, 6.8 bar IMEP), the optimised swirl leads to a strong improvement of the NOx/PM trade-off and HC and CO emissions without impacting the indicated efficiencies and noise levels.

When moving the NOx/PM trade-off to very low NOx emissions, the PM emissions increased substantially. A further increase of EGR led to a turn-down of the PM emissions at both standard and optimised swirl conditions. At the same time, the HC and CO emissions increased while the indicated efficiency dropped, which means that the fuel consumption increased. This region of engine operation is very unstable. A slight decrease in EGR rate resulted in a strong increase in PM emissions while a slight increase in EGR rate resulted in very high HC and CO emissions. This turn-down in PM emissions with increasing EGR levels is characteristic of HCCI-like combustion and has been observed in previous studies [3]. A similar turndown was seen with the other three fuels at this part-load condition. At the 4.3 bar condition, this turn-down in PM emissions is not seen because emissions are very low throughout the entire NOx range. At higher loads, the NOx/PM trade-off returns to a more diesel-like behaviour (see **Figure 12**).

Figure 13 EGR swings with standard and optimised swirl for the Baseline Diesel fuel



4.3. ENHANCED CHARGE AIR COOLING

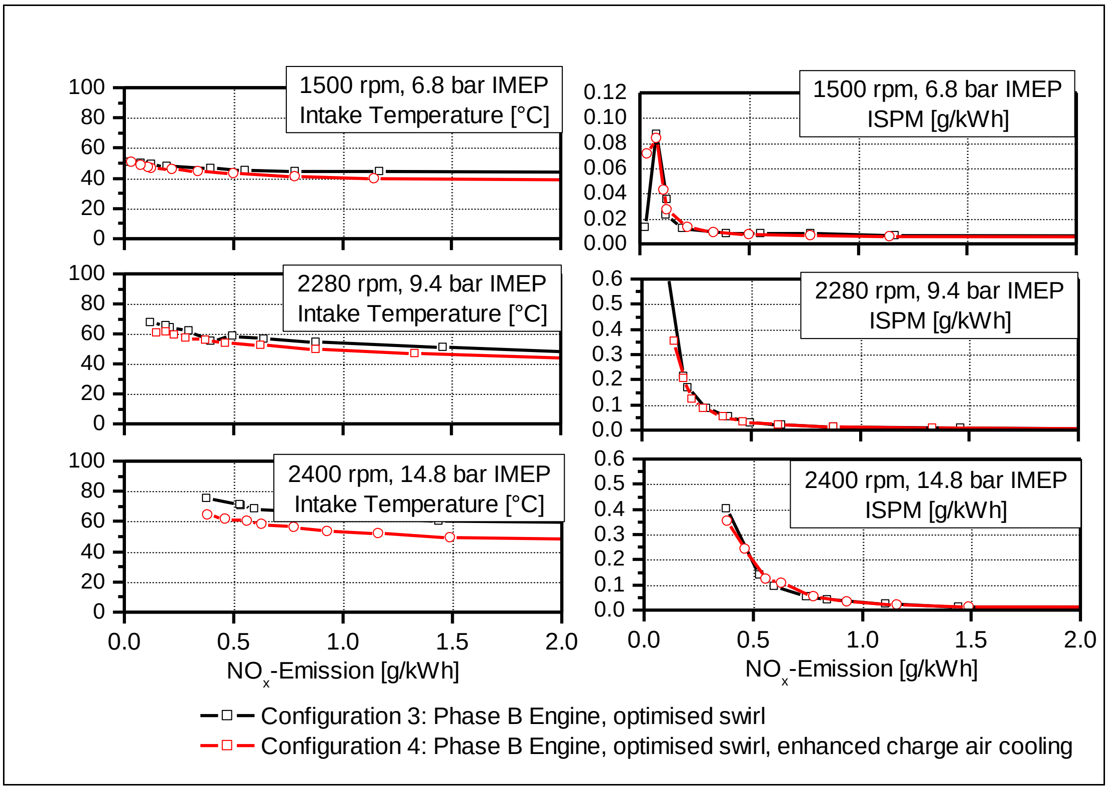
In order to investigate the potential of enhanced charge air cooling on the emission performance, the temperature downstream of the intercooler was further reduced. The initially selected temperature downstream of the intercooler is shown in **Table 9**. These values were selected based on earlier investigations to provide realistic boundary conditions for engine testing. The temperature downstream of the intercooler was reduced to the minimum possible temperature of 25°C for each load point for the investigation of enhanced charge air cooling.

Table 9 ***Global optimisation of the Phase B engine***

Engine Speed	[rpm]	1500	1500	2280	2400
IMEP	[bar]	4.3	6.8	9.4	14.8
Standard Boundary Conditions: Temperature downstream of the intercooler	[°C]	25	30	35	45
Enhanced charge air cooling: Temperature downstream of the intercooler	[°C]	25	25	25	25

Figure 14 compares the air temperature in the intake manifold and ISPM as a function of NOx emissions at the three higher part-load points, both with and without enhanced charge air cooling. Because Configuration 2 already permitted very high levels of cooled EGR, the extra benefits of intensified charge air cooling (Configuration 4) are quite modest. The enhanced charge air cooling does not lead to a visible improvement in the NOx/PM trade-off on this engine.

Figure 14 EGR swings with standard and enhanced charge air cooling for the Baseline Diesel



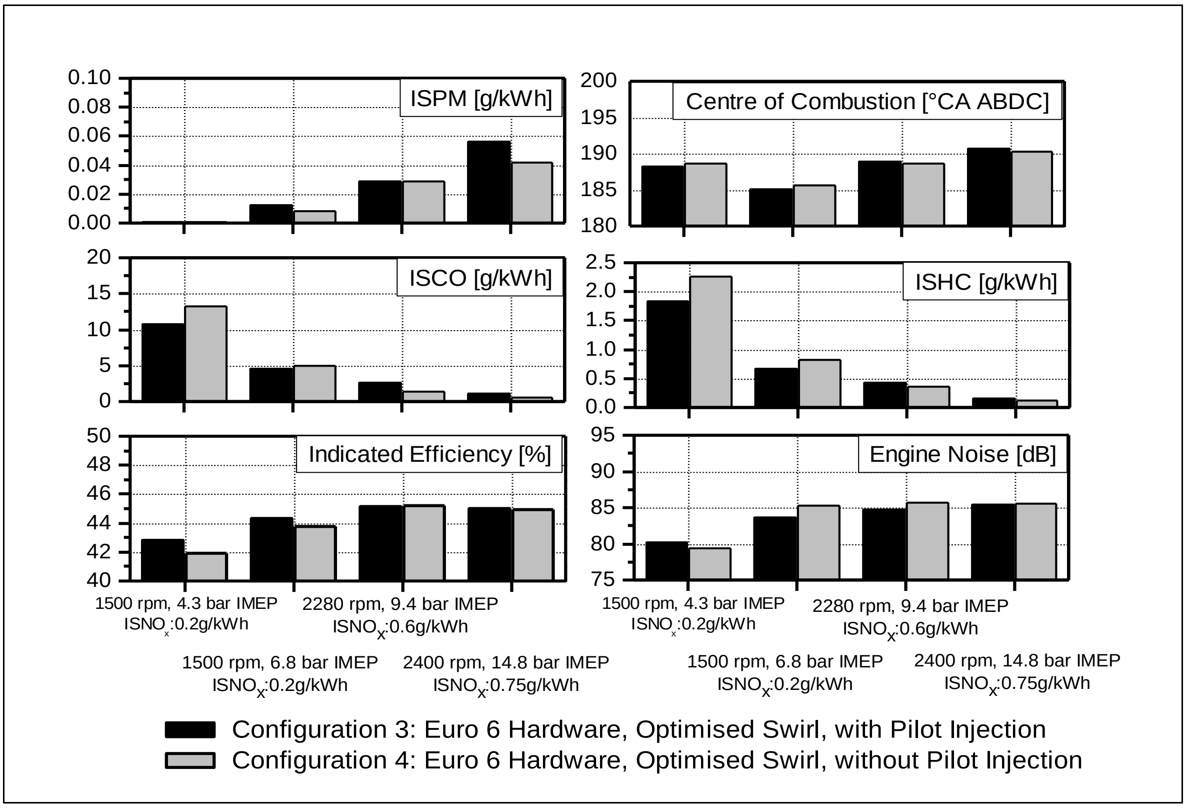
4.4. PILOT SWITCH-OFF AND MULTIPLE INJECTION STRATEGIES

The effect of switching off the pilot injection was also investigated in this study (Configuration 5). The centre of combustion was kept constant for these tests and the results are shown in **Figure 15**.

At the highest part-load point, switching off the pilot injection helped reduce the PM emissions. At the same time, the HC and CO emissions were slightly lower and the noise level was unchanged. At the two medium part-load points, switching off the pilot injection increased the noise emissions without any apparent benefits in the other parameters. At the lowest part-load point, switching off the pilot injection increased the HC and CO emissions. Because of the low temperatures at this part-load point, the ability of the oxidation catalyst to efficiently convert the HC/CO emissions may be reduced so an increase in these emissions should be avoided. For this reason, pilot injection is probably needed to stabilize the combustion at the lowest part load point.

Because switching off the pilot injection did not provide significant benefits, this report concentrates on the results obtained with pilot injection. Exhaust temperatures are considered further in the Part 2 study [12].

Figure 15 Effects of switching off the pilot injection for the Baseline Diesel fuel



To evaluate the potential of advanced injection strategies, a limited study was carried out to investigate multiple injection strategies (Configuration 6). For the different load points, different injection strategies were selected (**Table 10**).

Table 10 Selected injection strategies

Engine Speed	IMEP	Injection Strategy Investigated	Strategy Selected
[1/min]	[bar]	-	-
1500	4.3	2 Pilot Injections	No improvement observed
1500	6.8	Split Main Injection (50/50) / (70/30)	Split Main (70/30) BOI-Main 1: 8° CA BTDC BOI-Main 2: -8 °CA BTDC
2280	9.4	Split Main Injection (50/50) / (70/30)	Split Main (70/30) BOI-Main 1: 6.4 °CA BTDC BOI-Main 2: -5 °CA BTDC
2400	14.8	Split Main Injection (50/50) / (70/30)	No improvement observed

At the three higher part-load points, the separation between the two main injections was optimised first by keeping the start of the first main injection constant and then varying the start of the second main injection. After choosing an optimum separation, the main injection event was varied keeping the gap between the two main injections constant. Finally, an EGR variation was performed to compare the results with those obtained using the conventional injection strategies. In the three higher part-load points, a fuel distribution of 70% for the first injection and 30% for the second was found to be better because a 50/50 distribution led to a significant drop in the indicated efficiency.

At the highest and the lowest part-load points, no significant improvements were observed. For this reason, these results are not shown in this report.

Figure 16 shows the results at the 2280 rpm, 9.4 bar IMEP part-load point. The comparison of the standard injection strategy with the split main injection strategy, including the optimised in-cylinder swirl, shows that the centre of combustion is now slightly earlier. There is in addition an improvement of the PM and CO emissions and a slight increase in the indicated efficiency. At the same time, there is a strong increase in the noise emissions (CSL), which is about 3dB higher than with the standard injection strategy.

Figure 16 Effects of using a split main injection (2280 rpm, 9.4 bar IMEP)

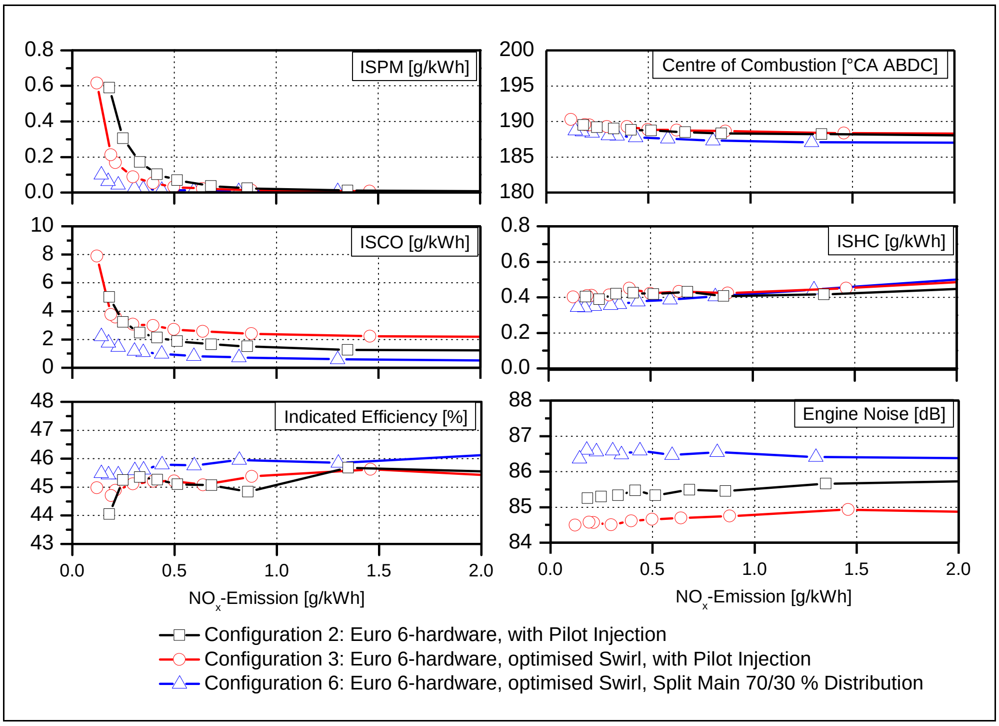
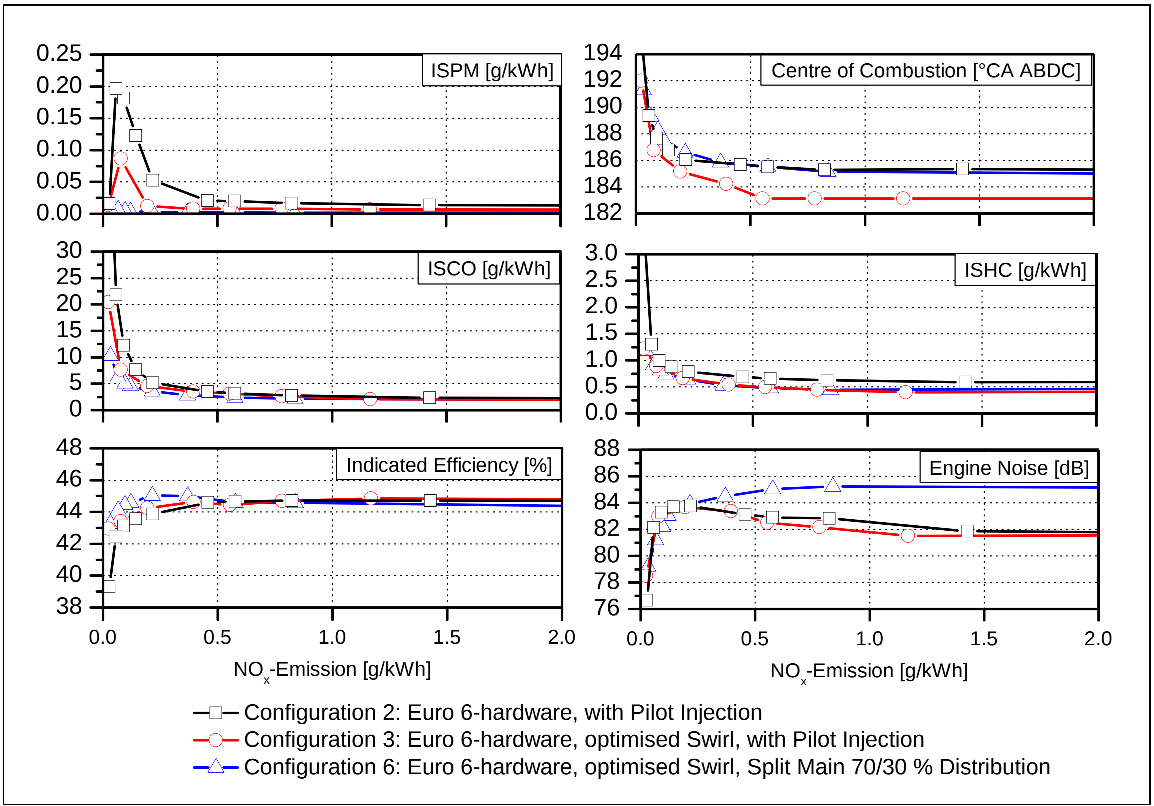


Figure 17 shows the results of the split main injection at the 1500 rpm, 6.8 bar IMEP part-load point. At this part-load point, a potential improvement was observed using the advanced injection strategy. There are almost no PM-emissions and the CO and HC emissions are slightly lower than with the standard injection strategy. At low EGR-rates, the CSL was about 3dB higher than with the standard injection strategy but the noise emissions were at about the same level in the target NOx area.

This short study showed some potential to further improve the engine-out emissions and combustion efficiency with advanced injection strategies even on a highly sophisticated combustion concept. However, a more complete investigation would be needed to evaluate the full potential of such injection strategies.

Figure 17 Effects of using a split main injection at 1500 rpm/6.8 bar IMEP



4.5. FULL LOAD MEASUREMENTS

Full load measurements were performed on both the Phase A and B engines to investigate if the full load targets could be achieved with all four fuels. Optimisation at the full load conditions was not part of this programme. The full load boundary conditions were taken from another project with a similar engine concept and are listed in **Table 11**.

The mechanical capabilities of the Phase B engine exceeded those of the Phase A engine. Therefore, an extended full load was also tested on the Phase B engine in three steps. First, the boost pressure was increased for all load points. Second, the maximum allowed cylinder peak pressure was increased to 190 bar. Finally, the rail pressure at the two highest engine speeds was increased from 1600 to 1800 bar.

Table 11 Full load boundary conditions

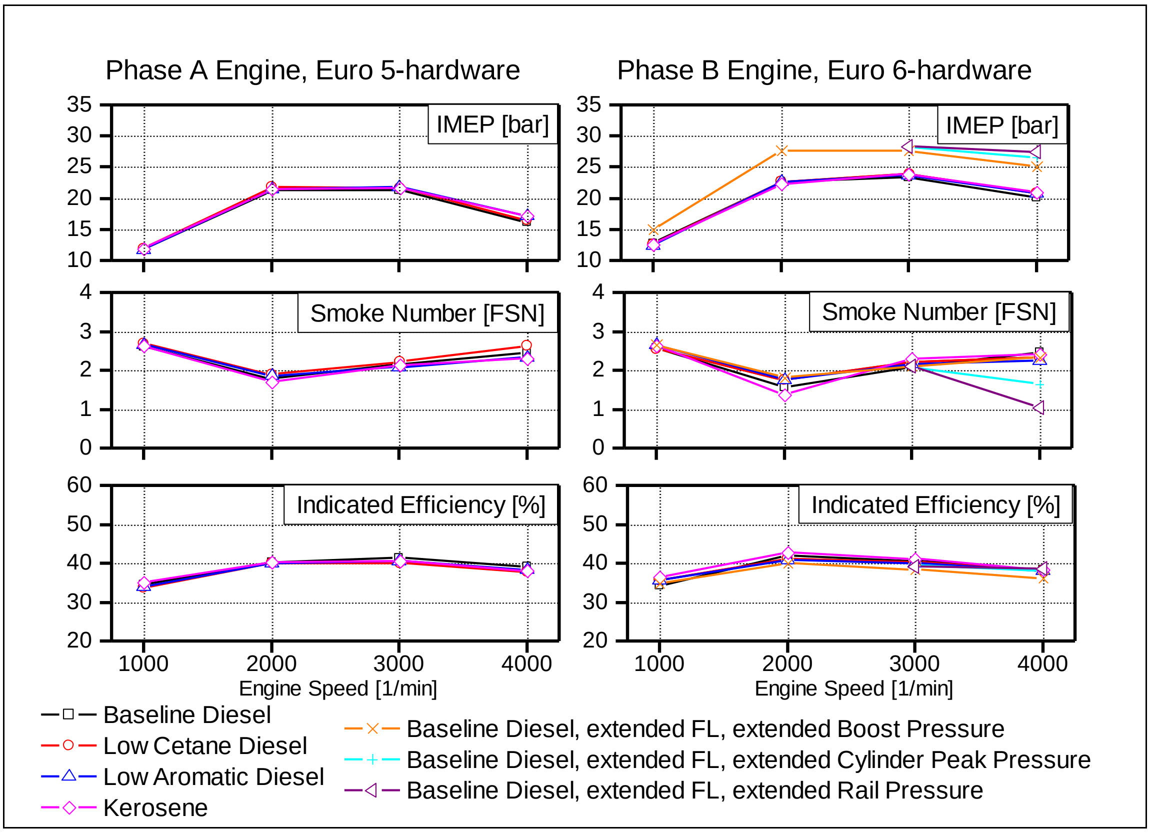
Engine Speed	Boost Pressure	Exhaust Gas Back Pressure	Rail Pressure	Cylinder Peak Pressure	Smoke Limit
[1/min]	[bar, abs.]	[bar/abs.]	[bar]	[bar]	[FSN]
1000	1.22	1.47	800	100	2.6
2000	2.06	2.07	1600	150	1.7
3000	2.30	2.48	1600	160	2.2
4000	2.29	2.99	1600	160	2.4
Extended Full Load on the Phase B Engine only					
1000	1.50	1.90	1000	130	2.6
2000	2.45	2.46	1600	160	1.7
3000	2.70	3.20	1600	160	2.2
3000	2.70	3.20	1600	180	2.2
3000	2.70	3.20	1800	180	2.2
4000	3.00	3.90	1600	160	2.4
4000	3.00	3.90	1600	190	2.4
4000	3.00	3.90	1800	190	2.4

Note: The Exhaust Gas Temperature limit was 820°C

The full load calibrations were defined taking into account the maximum cylinder pressure, the maximum exhaust gas temperature, and the smoke number limit. With increasing load, some adjustments were needed to stay within these limits. For example, the maximum cylinder pressure could be maintained by adjusting the injection timing. Increasing the quantity of fuel injected increased the load and increased the smoke number. The full load point was controlled by the point at which the smoke number limit was reached. If the exhaust gas temperature limit was reached before the maximum smoke number was reached, then the exhaust gas temperature became the limiting factor.

The results of the full load investigations are shown in **Figure 18**. The full load targets could be achieved with all fuels by changing only the injection timing and no significant differences were observed among the four test fuels. The extended full loads allowed a further increase in the maximum power output of the Phase B engine, so that at 4000 rpm the IMEP could be increased by 37.5% from 20.0 to 27.5 bar. At the highest engine speed and extended full load, the limiting factor was the 820°C exhaust gas temperature limit and not the smoke number. The extended full load was tested with all four fuels and no significant differences were observed among the four test fuels.

Figure 18 Full load investigations

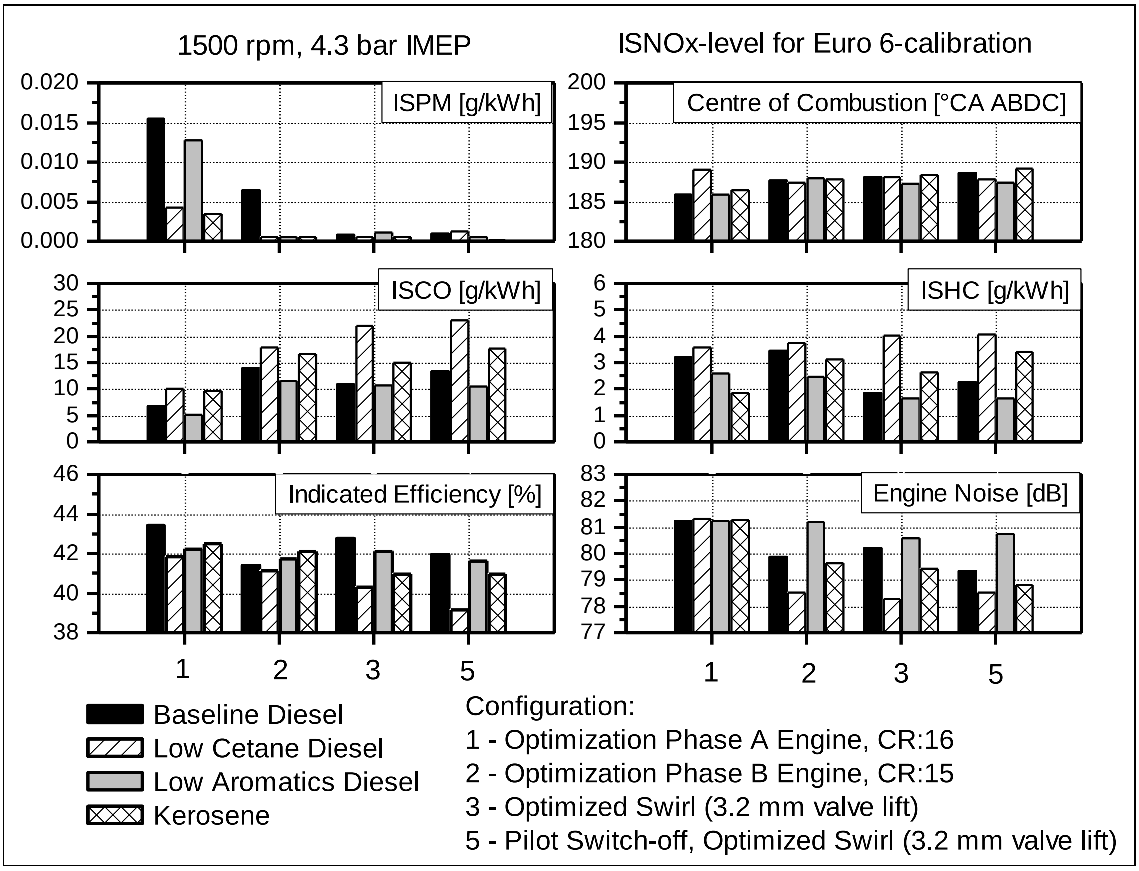


4.6. SYNTHESIS OF ENGINE AND FUEL EFFECTS

To better understand the impact of all the engine improvements, the results of the different engine configurations and test fuels are compared in **Figures 19 to 22** at the NOx level estimated for Euro 6 limits. The calibration used for these comparisons was optimised for the Baseline Diesel and kept constant for the other three fuels. This meant that the combustion phasing could be different for different fuels.

At the lowest part-load point (1500 rpm, 4.3 bar IMEP, **Figure 19**), the PM emissions were very low even for the Phase A engine. Switching to the Phase B engine reduced the PM emissions even further and almost no PM emissions were detected for any of the fuels after optimising the swirl. At this lowest part-load point, switching off the pilot injection showed no advantages.

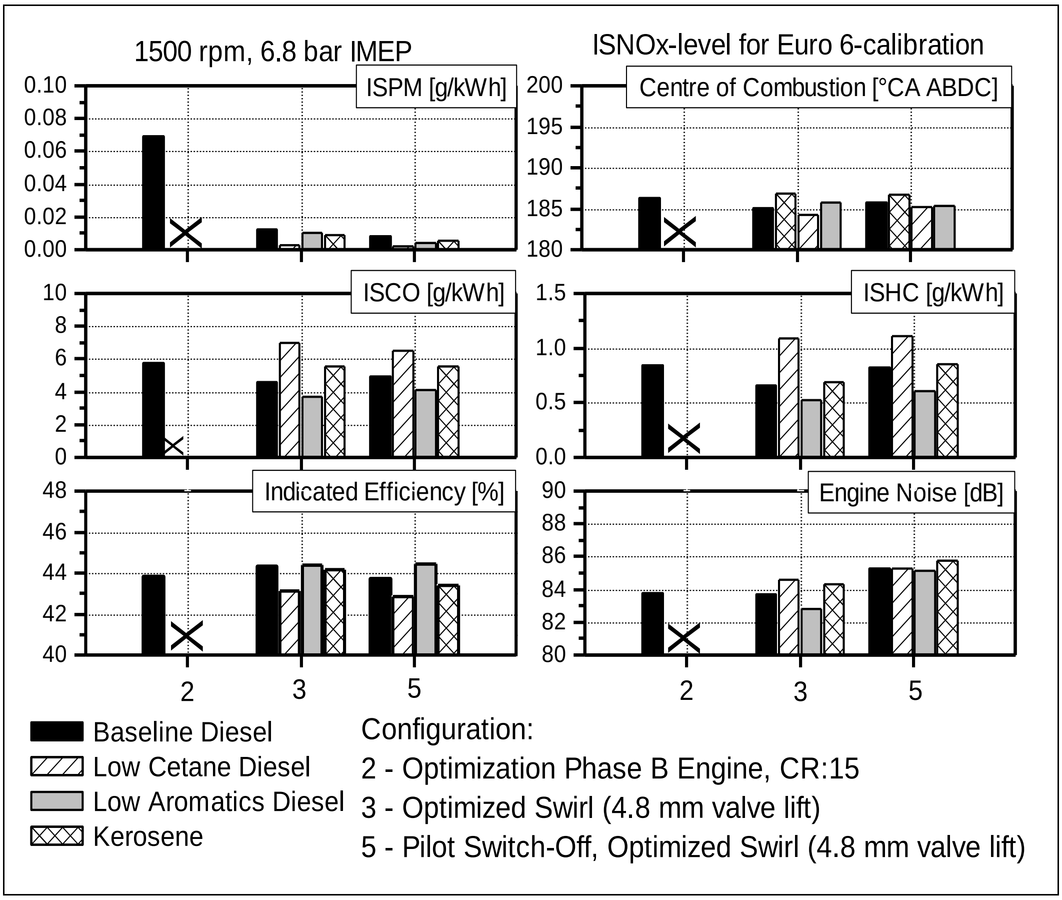
Figure 19 Comparison of four fuels using the Baseline Diesel calibration for four different engine configurations at 1500 rpm, 4.3 bar IMEP (note: combustion phasing was not matched in this testing)



For Configuration 3 shown in **Figure 19**, a fuel effect was observed for HC, CO, noise, and efficiency. However, the centre of combustion was not kept constant in this part of the study. Since the cetane number can give rise to different combustion phasing, these emissions and efficiency differences may not be due solely to fuel differences. Tests at the same centre of combustion provided a much better comparison and the results of back-to-back tests using a constant centre of combustion are shown in **Section 4.7** and in the Part 2 study [12].

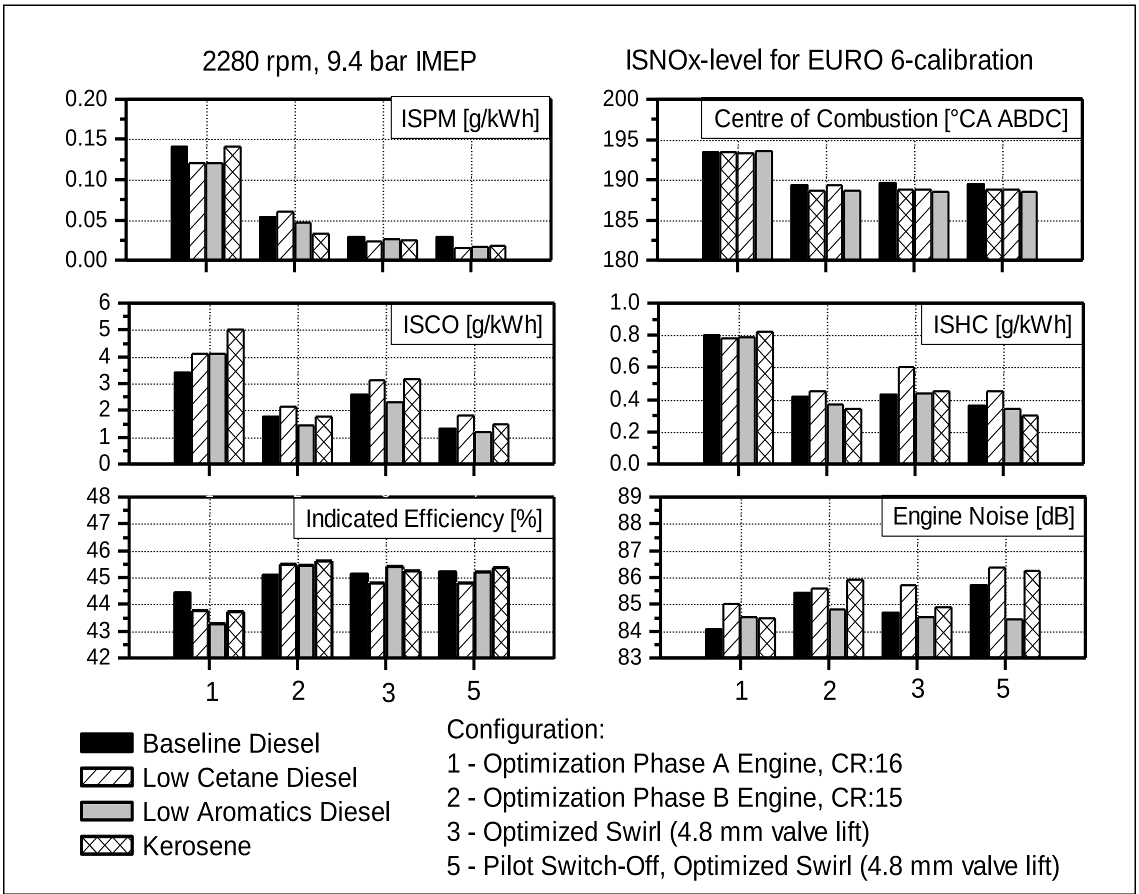
At the 1500 rpm, 6.8 bar IMEP part-load point (**Figure 20**), Configuration 2 was tested only with the Baseline Diesel. A strong reduction of the PM emissions was observed with the optimised swirl. While some fuel effects were evident, the Phase B engine produced low PM emissions with only moderate increases in HC and CO on all four fuels.

Figure 20 Comparison of four fuels using the Baseline Diesel calibration for three different engine configurations at 1500 rpm, 6.8 bar IMEP (note: combustion phasing was not matched in these tests)



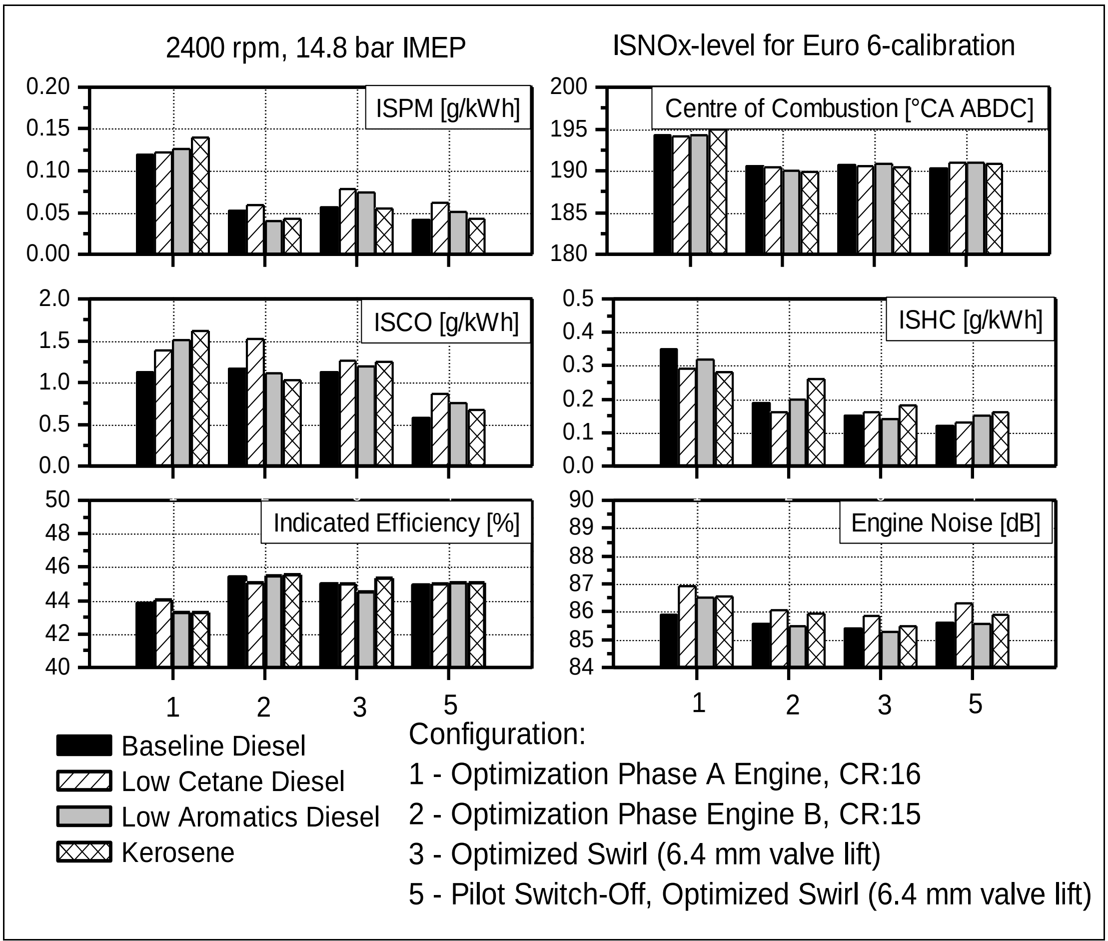
At 2280 rpm, 9.4 bar IMEP (**Figure 21**), all emissions and efficiency improved when switching from the Phase A to the Phase B engine. The Kerosene fuel appeared to have an influence on PM emissions. A further PM improvement was observed by optimising the swirl intensity. All other emissions remained essentially at the same level (except the slightly higher CO emissions). As discussed earlier, switching off the pilot injection increased the noise emissions.

Figure 21 Comparison of four fuels using the Baseline Diesel calibration for four different engine configurations at 2280 rpm, 9.4 bar IMEP (note: combustion phasing was not matched in these tests)



At the highest part-load point (2400 rpm, 14.8 bar IMEP), only a small fuel influence was observed (**Figure 22**). The change in hardware from the Phase A to the Phase B engine led to a marked improvement in fuel consumption and emissions. In comparison to this improvement, the differences caused by the different fuels were quite small. Optimising the swirl was also not beneficial at this load point. While there were some improvements in the HC and noise emissions, the PM emissions increased and the indicated efficiency decreased.

Figure 22 Comparison of four fuels using the Baseline Diesel calibration for four different engine configurations at 2400 rpm, 14.8 bar IMEP (note: combustion phasing was not matched in these tests)



The engine improvements from the Phase A engine to the Phase B engine dramatically reduced the PM emissions at all part-load points. The fuel effects and the variations of the centre of combustion were different at different loads and none of the fuels was clearly the best at all test conditions.

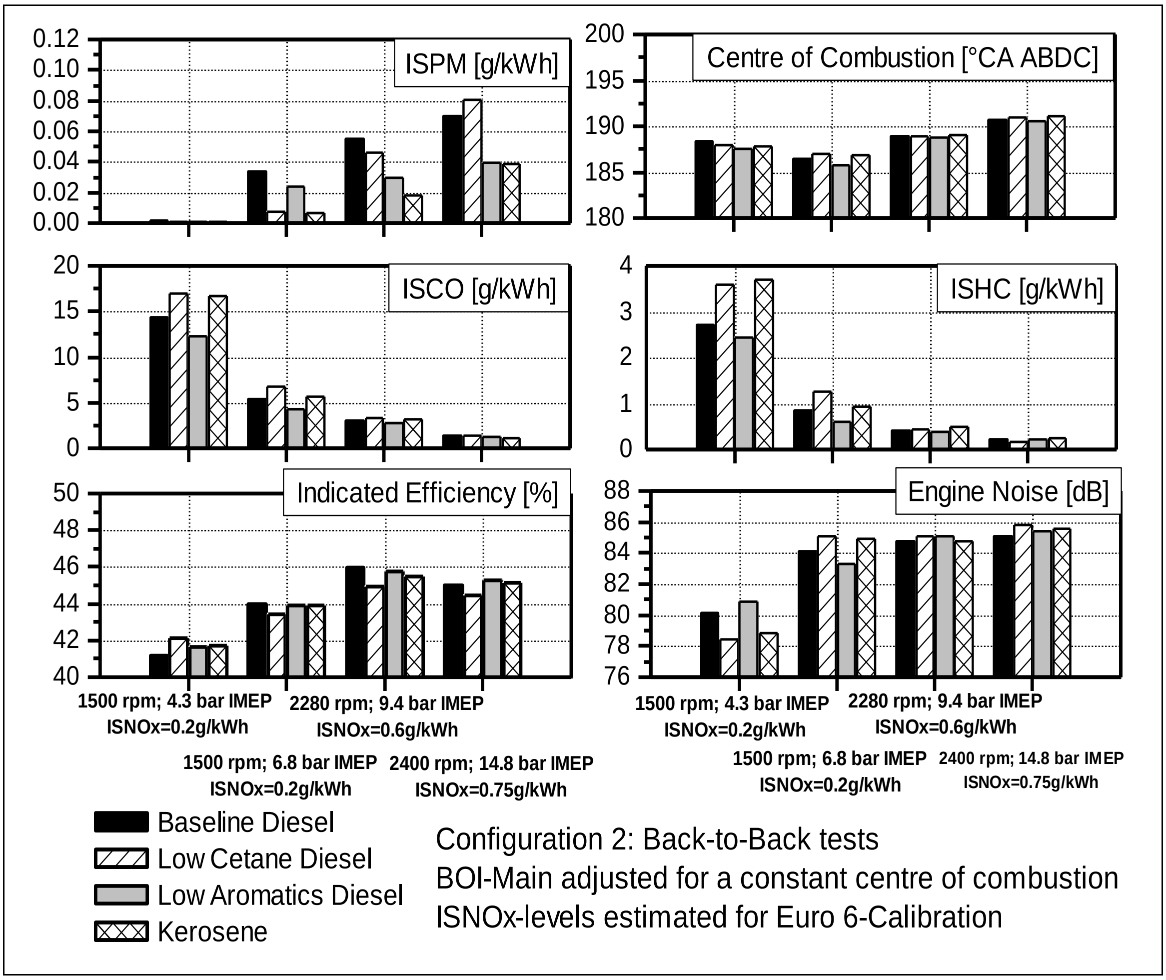
Summarizing the results from the different engine configurations, the transition from the Phase A to the Phase B engine resulted in a significant improvement in fuel efficiency and emissions performance. Optimised swirl resulted in a further reduction in emissions. In comparison with the hardware improvements, the influence of the four fuels on emissions behaviour was relatively small.

4.7. BACK-TO-BACK TESTS AT CONSTANT CENTRE OF COMBUSTION

Although a more detailed evaluation of fuels is presented in the Part 2 study [12], **Figure 23** compares the results of back-to-back tests on the four fuels evaluated in this Part 1 study. These tests were performed on the Phase B engine with Configuration 2 (standard swirl). The harmonized calibration was used and the beginning of fuel injection was adjusted to achieve a constant centre of combustion.

These results therefore simulate the behaviour of the fuels in a future engine with CLCC timing.

Figure 23 Back-to-back tests on all four fuels in the Phase B engine (note: combustion phasing was matched in these tests)



The NOx levels for each test point were those required to meet Euro 6 limits, as indicated in **Table 5**. The centre of combustion was adjusted to be the same for each fuel.

At the lowest part-load point (1500 rpm, 4.3 bar IMEP), the fuels with lower cetane numbers (Lower Cetane Diesel and Kerosene) showed higher HC and CO emissions. On the other hand, the noise emissions were lower for these fuels than for the fuels with higher cetane numbers. The PM emissions were very low on all the fuels at this part-load point.

At the 1500 rpm, 6.8 bar IMEP part-load point, the two fuels with the lower cetane numbers still showed higher HC and CO emissions. The noise emissions were reversed, however, so that these two fuels now produced higher noise emissions. These fuels also showed significantly lower PM emissions, presumably because they achieved a longer premixing time and more HCCI-like combustion.

At the two highest part-load points, there were no significant fuel influences on the HC, CO, and noise emissions. For PM emissions, the Kerosene and the Low Aromatics Diesel showed advantages at these two part-load points.

Due to the constant centre of combustion for the different load points, no significant influence of fuel properties on fuel efficiency was observed.

5. CONCLUSIONS

This study investigated the influence of different engine configurations and fuel properties on engine performance, efficiency, emissions, and noise. Two single-cylinder engines benchmarked for Euro 5 and Euro 6 emissions levels were evaluated that had been optimised for advanced combustion performance. Various hardware configurations were tested that included a lower compression ratio, higher maximum cylinder peak pressure and rail pressure, optimised in-cylinder swirl, adjustment of fuel injection timing, and intensified EGR. Four fuels with a range of ignition quality and aromatics content were used to evaluate the performance of these hardware enhancements on engine-out emissions, performance, and noise levels.

For both engines, it was observed that the engines could be operated on all four fuels at full load and at all part-load points. The full load targets could be achieved with all fuels and only the fuel injection timing needed to be adjusted. In the first phase of this study, some fuel effects were seen on the Phase A engine benchmarked for Euro 5 emissions but the effects were small.

In the second phase of this study, the Phase B engine benchmarked for Euro 6 emissions was optimised individually on all four fuels using a DOE approach. The main findings from the Phase B engine work were:

- The DOE-optimisations for all fuels led to similar calibrations.
- Harmonization of the engine calibrations was possible for different fuels by adjusting the fuel injection timing to give the same centre of combustion. This procedure simulated the operation of an advanced Closed-Loop Combustion Control system.
- The centre of combustion was found to be the most important calibration parameter and satisfactory performance was achieved on all four fuels when this was controlled.
- Optimising the swirl at each load point enabled a marked reduction of the NOx/PM trade-off. At lower loads, higher swirl levels were found to be especially beneficial.
- Very low NOx levels could be achieved while maintaining acceptable PM levels.
- Switching off the pilot injection gave lower PM emissions and higher noise at the higher part-load points while HC and CO emissions increased at the lower part-load points.
- Combustion typical of HCCI operation was observed at the lower part-load points. At higher part-load points, combustion became more like conventional compression ignition.
- At the compression ratio of 15 in the Phase B engine, the combustion system responded more strongly to changing fuel properties, especially at the lower engine loads.

Overall, the engine hardware enhancements included in this study enabled a significant improvement of the emissions behaviour and fuel efficiency. In comparison with these improvements, the influence of the four test fuels on emissions and efficiency was relatively small.

This study investigated engine performance and emissions for a fully warmed-up engine at steady-state conditions only. Additional work would be needed to investigate the influence of fuel properties on engine performance under transient cycle and cold start conditions.

6. GLOSSARY

ASTM	American Society for Testing and Materials
ATDC	After Top Dead Centre
BMEP	Brake Mean Effective Pressure
BOI-Main	Beginning of Injection – Main injection
BOI-Pilot	Beginning of Injection – Pilot injection
BTDC	Before Top Dead Centre
^{o}CA	Degrees Crank Angle
CA10	Point in the combustion cycle where 10% of the injected fuel has been converted to energy
CA50	Point in the combustion cycle where 50% of the injected fuel has been converted to energy. In this report, CA50 is also called the 'centre of combustion'.
CAI	Controlled Auto Ignition
CFD	Computational Fluid Dynamics
CFR	Cooperative Fuel Research
CLCC	Closed Loop Combustion Control
CN	Cetane Number
CO	Carbon Monoxide
CO_2	Carbon Dioxide
CR	Compression Ratio
CSL	Combustion Sound Level
dB	Decibels
DCN	Derived Cetane Number
DOI-Pilot	Duration of Injection – Pilot injection
DOE	Design of Experiments
DeNOx	NOx Reduction Aftertreatment
DPF	Diesel Particulate Filter

EGR	Exhaust Gas Recirculation
EN590	CEN Specification (European Norm) for Diesel Fuel
FMEP	Friction Mean Effective Pressure
FSN	Filter Smoke Number
GT	Gamma Technologies
g/kWh	Grams per kilowatt-hour
HC	Hydrocarbon
HCCI	Homogeneous Charge Compression Ignition
HFR	Hydraulic Flow Rate
IMEP	Indicated Mean Effective Pressure
IQT	Ignition Quality Test or Tester
ISCO	Indicated Specific Carbon Monoxide Emissions
ISFC	Indicated Specific Fuel Consumption
ISHC	Indicated Specific Hydrocarbon Emissions
ISPM	Indicated Specific Particulate Mass Emissions
ISNOx	Indicated Specific Nitrogen Oxides Emissions
LHV	Lower Heating value
MJ	Megajoules
NEDC	New European Driving Cycle
NOx	Nitrogen Oxides
PM	Particulate Matter or Mass
rpm	Revolutions per minute
SCR	Selective Catalytic Reduction
TDC	Top Dead Centre
US06	Supplemental Federal Test Procedure (USA)
VVL	Variable Valve Lift

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APPENDIX 1 DESIGN OF EXPERIMENTS OPTIMISATION

The Design of Experiments (DOE) engine optimisation included the following steps:

1. Pre-optimisation of the calibration-parameters to define the experimental range for the DOE
    a. Definition of the input parameters (BOI-Main, rail pressure, boost pressure and EGR rate)
    b. The engine must properly function with all combinations of the input parameters
    c. Calibrations which combine unfavourable parameters are of special interest, especially “worst-case” calibrations (e.g. combinations of low rail pressure, late main injection and low boost pressure)
    d. Setup of an adequate DOE Plan
        i. Definition of the mathematical model to use, choice of cubic models
        ii. Setup of DOE plan with 4 input parameters
        iii. 36 measuring points, including:
            1. 25 Model points (for the calculation of the model)
            2. 1 Base Point (Base calibration)
            3. 5 Repeatability Points (to identify possible drifts within the DOE)
            4. 5 Validation Points (used to validate the models, not used to calculate the models)

- Conduct DOE test on the engine test bench
- Evaluate the engine data
    a. Optimisation of the engine
    b. The boundary conditions of the DOE optimisation are chosen based on reasonable limits for production engines

While optimising the engine, two different strategies were used to balance the different performance measures. The first strategy, the “global optimisation”, takes into account all relevant limited emissions as well as the engine noise emissions:

- Reduction of engine-out NOx emissions is the primary target
- Smoke emissions to be constrained to fixed levels acceptable for DPF treatment
- Fuel consumption and noise are not allowed to increase above conventional diesel levels
- Some increase in HC and CO emissions is permitted.

The second optimisation strategy, the “ISFC Optimisation” minimized the fuel consumption at a targeted NOx level. When using this strategy, other emissions were not regarded. This strategy was used to determine the optimum centre of combustion and understand the general behaviour of the engine and fuels without constraining emissions.

As an example, the DOE results on the Baseline Diesel at 2280 rpm, 9.4 bar IMEP are shown in **Figure A1-1**. The DOE model displays those points that can be calculated using the entire experiment range of the DOE. The constraint points include those points that meet the constraints for the global optimization. The global optimization takes the constraints into account and optimises the engine with regards to the limiting emissions (HC, CO, PM, noise) and the indicated efficiency.

Figure A1-1 DOE optimisation at 2280 rpm, 9.4 bar IMEP on the Phase B engine using the Baseline Diesel fuel

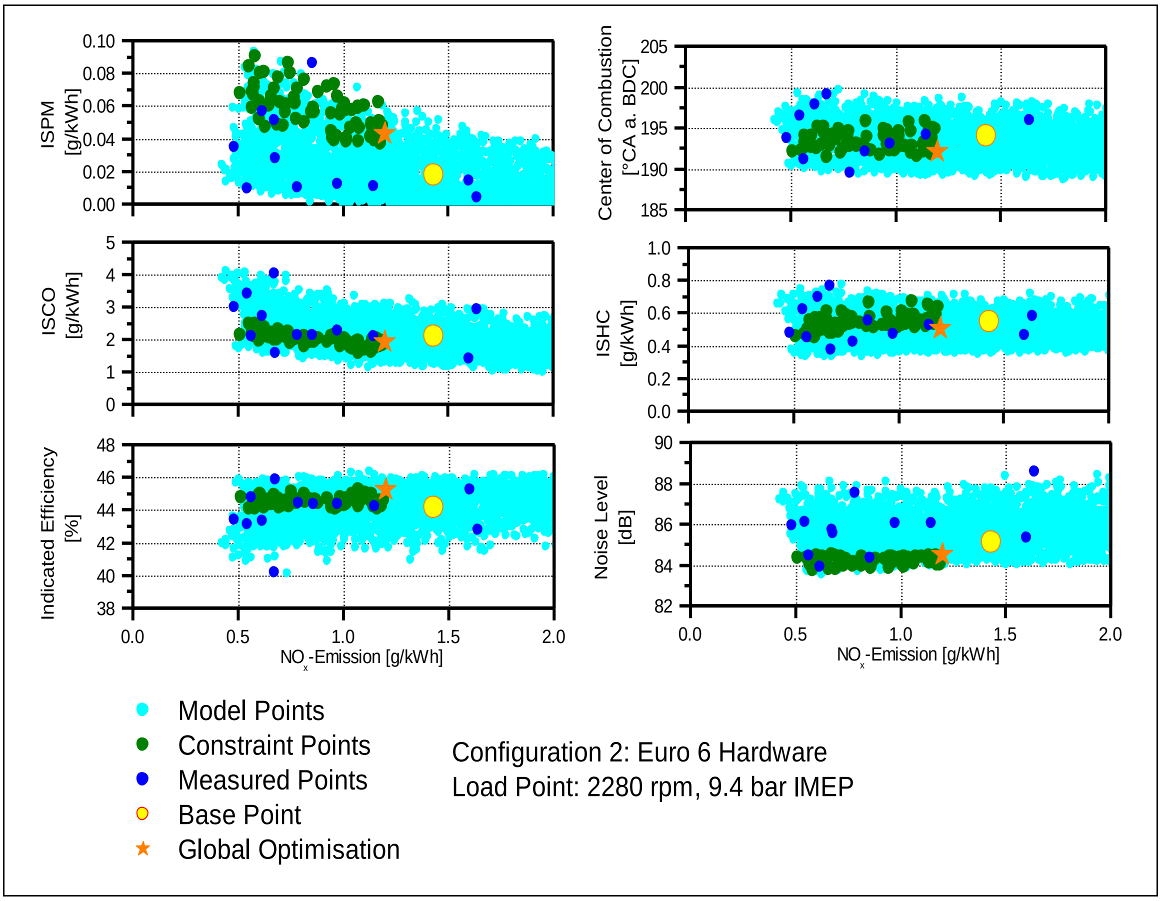


Figure A1-2 DOE optimisation strategies

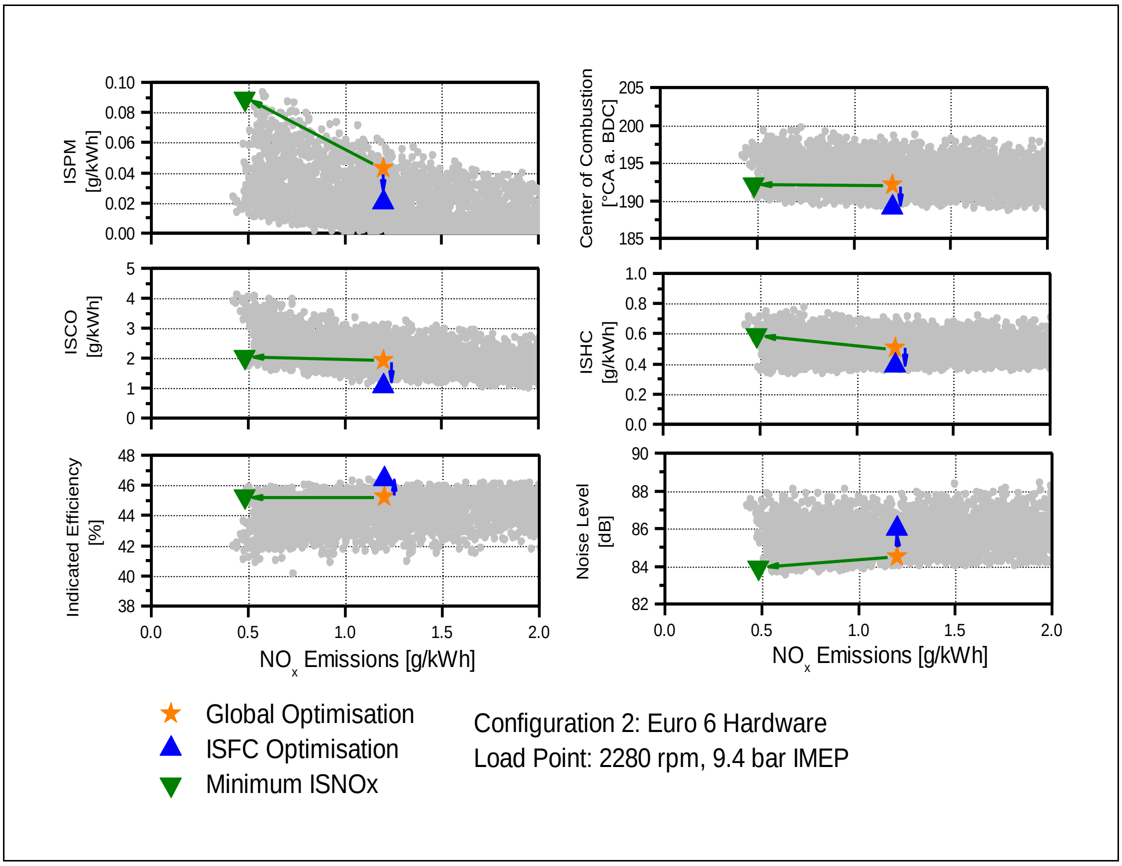


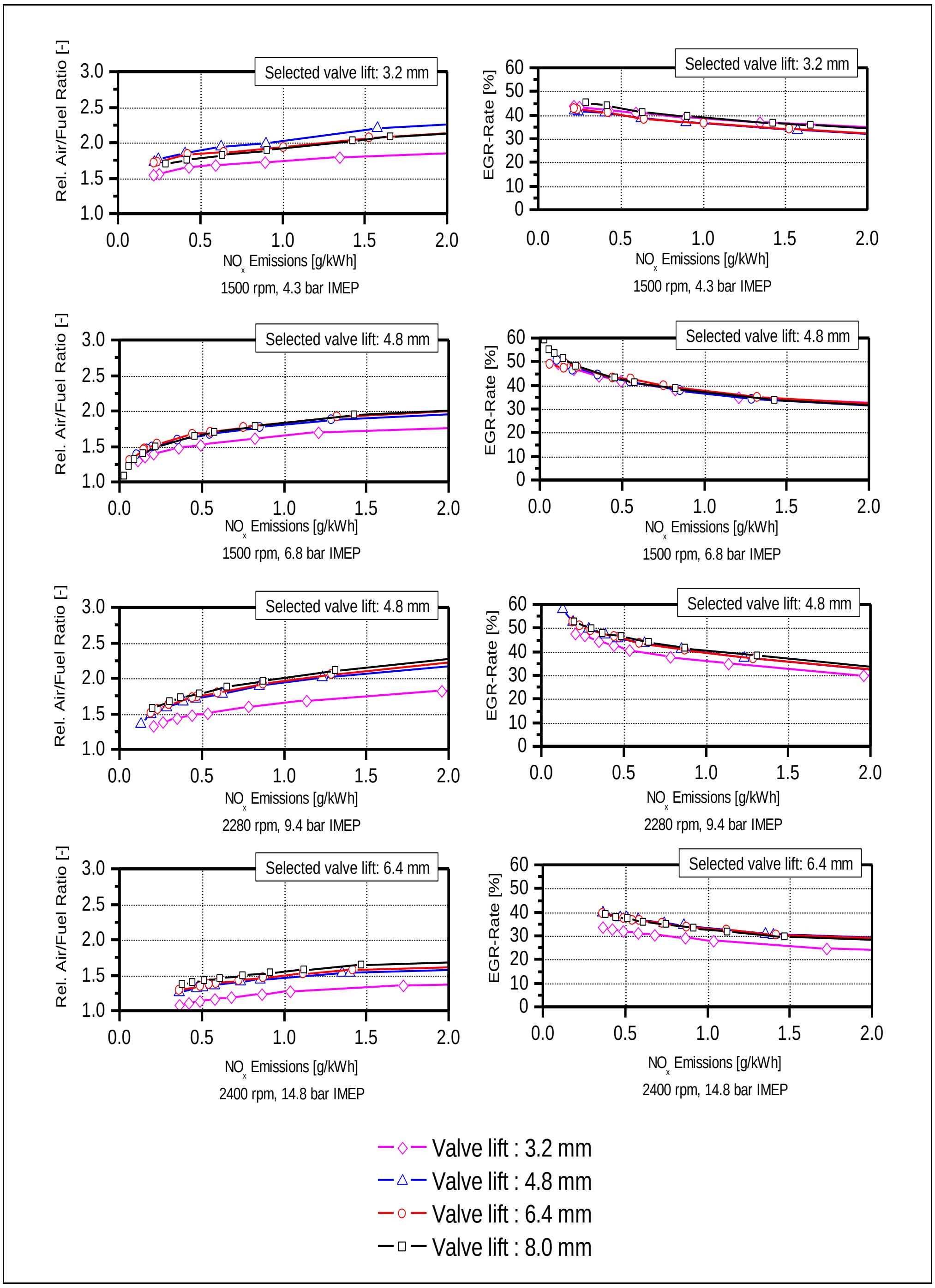
Figure A1-2 shows optimisation strategies that are possible by using the DOE models and how they affect the different emissions. For example, beginning with the global optimisation, the indicated efficiency might be optimised without regard for the other emissions (ISFC optimisation). It can be seen that the improved indicated efficiency also leads to a further reduction of the PM, HC, and CO emissions. However, at the same time, the noise emissions increase because they had been constrained for the global optimisation. If, for example, NOX emissions are minimized (minimum ISNOx), a strong increase in the PM emissions is observed (NOx/PM trade-off).

After optimising the four mentioned engine parameters, the pilot injection was conventionally optimised using separate swings for the beginning and the duration of the pilot injection. A pilot injection is used especially to reduce the engine noise emissions. At lower loads, the pilot injection is also necessary to stabilize the combustion process. Taking both of these criteria into account, the optimum beginning and duration of the pilot injection were selected from each corresponding swing.

APPENDIX 2 RELATIVE AIR/FUEL RATIOS AND EGR RATES

The relative air/fuel ratios and EGR rates for the different part-load points and swirl levels are shown in **Figure A2-1**.

Figure A2-1 Relative air/fuel ratios and EGR rates for the Baseline Diesel fuel at different swirl levels



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