

Emission and Ignition Effects of Alternative Fuels at Conventional and Premixed Diesel Combustion

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ABSTRACT

The growing availability of different biofuels and synthetic fuels is leading to increased diversity of automotive fuels. Understanding how fuel properties affect combustion and how engine calibration strategies can compensate for variations in fuel composition is crucial for ensuring proper engine operation in this world of increased fuel diversity. This study looks at the ability to compensate for wide changes in cetane quality.

Four different fuels with variations in cetane number, volatility and composition have been tested in a single cylinder engine and compared to diesel fuel. The selected operating conditions represent the entire engine map of a passenger car diesel engine. In part load the effects were investigated for conventional and premixed Diesel combustion.

The results show that part load operation is especially relevant for the detection and compensation of varying fuel properties and that, depending on engine load, different control strategies have to be applied.

INTRODUCTION

Most recent energy and transport scenarios [1] foresee a growing diversification in the fuel market driven by

environmental, supply and financial factors. Fuels derived from biomass offer the opportunity to significantly decrease the emission of CO₂ and are increasingly being mandated by governments across the globe. In addition to the widespread use of “first generation” ethanol as fatty acid methyl esters we are also seeing the emergence of advanced biofuels developed from no-food biomass sources such as straw and wood waste. Synthetic fuels such as Gas-to liquids gasoil derived from natural gas are also being used increasingly in the automotive fuel pool.

With a growing demand for automotive fuels and the limited supply of easily recoverable conventional oil we can expect to see continued growing demand for biofuels and synthetic fuels in the near future.

Even with today's limited fuel diversity engine operation is very sensitive to fuel properties. With the expected further diversification of the fuel market, a detection and compensation of the fuel property variations will be necessary to minimize calibration cost and guarantee safe and satisfactory engine operation.

The need to reduce diesel engine emissions and the requirement to increase the specific engine power has led to a reduction of diesel engine compression ratios over the past years. Compression ratios have declined from an average of 22 to around 17. The most recent increases in power density

and the trade-off between nitrogen oxide and particulate matter has been achieved via a further reduction of compression ratios down to approximately 16 [2]. For very low nitrogen oxide (NO_x) emissions all recent passenger car diesel engines also use high levels of exhaust gas recirculation (EGR)

Besides incorporating engine modifications to cope with future fuel property diversity, further diesel engine development will be driven by the need for increased combustion efficiency whilst complying with emission regulations and customer requirements. The strong demand to reduce CO_2 emissions from diesel engines may lead to a re-increase of compression ratios in the near future [2]. Independent of these changes a partially premixed combustion is being introduced at low part load. This minimises smoke emissions, reducing particle trap load and therefore prolonging regeneration intervals and improving fuel consumption.

Since all these measures to improve efficiency and reduce emissions require finely tuned combustion to achieve clean, reliable and efficient diesel combustion on a wide range of fuels it will be necessary to detect and compensate for the variations in fuel quality.

STATE OF RESEARCH

Fuel research in the automotive sector aims to minimize emissions at the highest efficiency possible. Most research focuses on part load engine operation since clean combustion is most sensitive under these conditions and because most passenger car emission regulations apply in this engine map area.

For diesel-like fuels most publications agree that cetane number, aromatics content and heating value are the key fuel properties that influence emissions and efficiency of diesel combustion. Higher cetane numbers lower combustion noise and enhance cold start ability and low aromatics content lowers particulate matter (PM) emissions [3, 4, 5, 6].

Fuels with oxygen content are reported to give less PM emissions with NO_x emission levels similar or slightly higher than reference diesel fuel [7, 8, 9, 10].

Viscosity and density are important for fuel distribution and evaporation in the combustion chamber [11].

Some papers mention the necessity to adjust combustion phasing and injection parameters [11, 12] to compensate for fuel differences. In fact, one engine that achieves this with closed loop combustion control via cylinder pressure sensors was launched in the US market in 2008. Testing it with different fuels showed that it is able to compensate variations in cetane quality [13].

Interestingly the volatility of the fuels studied is rarely mentioned as a major factor that affects emission levels.

Today's diesel engines use a variety of operating strategies in the engine map. Especially at part-load there are three alternative strategies. These are the conventional diesel combustion, partly premixed combustion (PCCI) and the homogeneous combustion (HCCI) (figure 1).

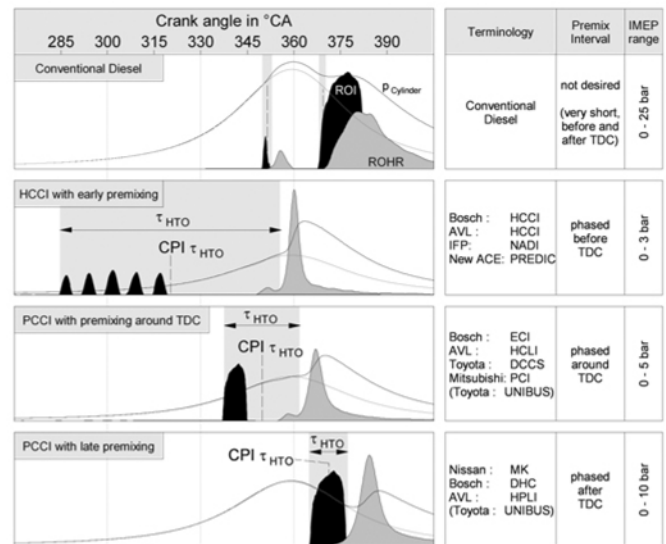


Figure 1. Diesel combustion strategies

Premixed Diesel combustion strategies achieve virtually soot and NO_x -free combustion and can benefit from fuels giving a long ignition delay and fast vaporisation. Fully homogenised processes (HCCI) use early multiple injections supported by high EGR rates to extend the time for mixture preparation. PCCI combustion is designed to realize just enough premixing for a suppression of the soot emission at very low NO_x . PCCI typically uses one early single injection and high EGR rates.

In contrast to HCCI and PCCI, conventional diesel combustion needs a short ignition delay to control the combustion noise. Therefore one or more pilot injections (PI) are typically applied especially in low part-load. The combustion effects resulting from the pilot injection play a very important role for conventional diesel combustion. Unfortunately fuel effects on pilot injection (PI) and combustion are seldom studied in detail [5]. With pilot injection being a major tool to adjust emissions and noise in part load this is clearly an important topic for future scientific investigations.

METHODOLOGY

TEST ENGINE

The data for this paper was collected on a single cylinder test bench equipped with a Diesel engine of the specifications presented in [table 1](#):

Table 1. Specifications of the test engine

Engine Type	Diesel
Compression Ratio	15.88
Combustion chamber	Omega
Injection system	Common rail
Injector	Servo magnetic 7 hole, 153°
Nozzle flow	820 cm ³ / 60 s
Bore / Stroke	88 mm / 88.34 mm
Displacement	537 cm ³
Max cylinder pressure	220 bar

The test bench located at Technische Universitaet Berlin (TU Berlin) comprises of a temperature controlled test cell, a direct current (DC) dynamometer, auxiliary units and data acquisition systems.

It is equipped with an external compressor, a cooling system and a heater to provide conditioned charge air to the engine. The air flow is measured by a rotary piston volumetric flow meter. Acquisition of pressure and temperature at the flow meter allow calculation of air mass flow.

Coolant water and engine lube oil are provided independently from the engine to minimize feedback on engine operation. Separate heating and cooling systems allow individual conditioning of both circuits.

The fuel system is run independently by an electric motor. Additional pressure accumulators are installed after the high pressure fuel pump to ensure minimal pressure waves at the injector. The injector and the common rail pressure are controlled by a fully accessible engine control unit (ECU) allowing up to five individual injection events of independent timing at pressures from 200 to 1600 bar. The pressure in the injector line is measured by a Kistler sensor of the type 4067A2000. The fuel consumption is determined by a fuel scale of the type AVL 7131.

The cylinder pressure is measured by two individual piezoelectric sensors, a Kistler 6043 ASP and an AVL GU23D. The cylinder pressure traces are recorded at a resolution of 0.2° crank angle whereas the injection line

pressure traces are recorded with a resolution of 0.1° crank angle. The high resolution data acquisition is done by an AVL Indimodul 621.

The exhaust gas backpressure is adjusted via a choke valve. To simulate a turbocharger and allow high pressure EGR the exhaust backpressure was controlled to a level 200 mbar above the intake pressure.

The EGR rate is calculated from fresh air flow capacity and adjusted by a choke valve. The recirculated exhaust gas is cooled to allow low intake temperatures even at high load and high EGR rates.

An exhaust gas sample is taken close to the exhaust valve and routed to NDIR, CLD, FID and PMD analysers by a conditioned pipe to ensure gas temperatures above 190 °C at all times. In addition with cooling and drying the sample gas for NDIR and PMD analysers the procedure complies with ECE regulations. The smoke number is measured by a smokemeter (AVL 415) located after the exhaust backpressure choke valve. Smoke is calculated from filter smoke number (FSN) by the formula given in the smokemeter product description.

Indicated mean effective pressure (IMEP), combustion noise (noise), combustion phasing and combustion stability are calculated from the cylinder pressure signal by the AVL Indimodul 621. Because the charge air is provided externally, IMEP for the high pressure phase (IMEP_{HP}) is considered relevant for this study neglecting the gas exchange phase.

The test bench and all auxiliary units are controlled by proprietary test bench software developed in Labview® which in addition records all necessary data apart from the high resolution pressure traces.

Data post processing provides emissions according to ECE regulations and heat release rates.

TEST CONDITIONS AND FUELS

The operating points for the test schedule were selected in order to represent engine operation over the entire engine map.

To determine the full load capability of the test fuels the maximum torque at $n = 4000 \text{ min}^{-1}$ was determined for all fuels ([table 2](#), OP3). The low load operating point ([table 2](#), OP1) was selected because of its high frequency of occurrence with given vehicle/engine combinations during the NEDC. This operating point was also chosen to investigate fuel effects in regular und premixed charge compression ignition (PCCI) engine operation modes. In this part of the study special attention was paid to the extension of the PCCI operating range due to cetane number variations. To

Table 2. Operating points

Operating Point	Speed	IMEP _{HP}	p _{rail}	p _{intake}	T _{intake}	m _{PI}
-	min ⁻¹	bar	bar	Bar	°C	mg/stroke
OP1	1650	5	750	1.0	60	conv.: 3.0 PCCI: w/o
OP2	2350	15.0	1350	3.0	70	1.5
OP3	4000	22.0	1600	2.5	50	w/o

address fuel effects relevant to recent downsizing trends, an operating point at upper part load (table 2, OP2) was selected. All operating points and relevant driving cycles for a given mid-class vehicle / engine combination are shown in figure 2. To represent the fully heated engine the intake temperature was controlled to 60°C for lower part load (OP1), 70°C for upper part load (OP2) and 50°C for full load (OP3). Engine coolant and engine oil were conditioned to 80° C for OP1 and to 90°C for OP2 and OP3.

<table 2 here>

For a better understanding of the fuel effects on combustion and emissions a variation of the engine operation conditions at constant load was carried out for all three operating points. At part load a variation of the centre of heat release (COHR), charge air pressure (p_{intake}), EGR rate, rail pressure (p_{rail}) and pilot injection mass (m_{PI}) was conducted whilst only COHR was varied for the full load operation point (OP3).

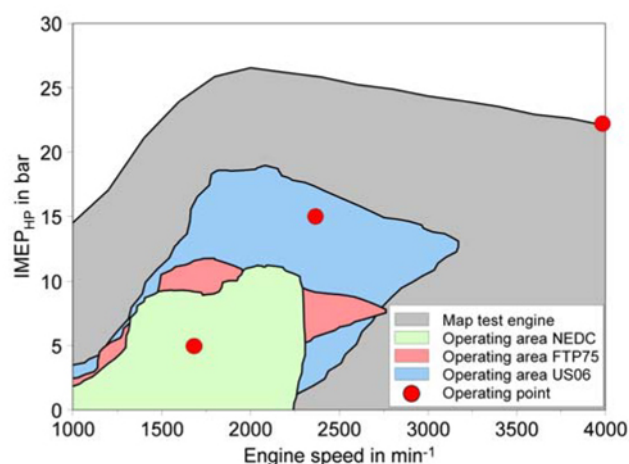


Figure 2. Test engine map

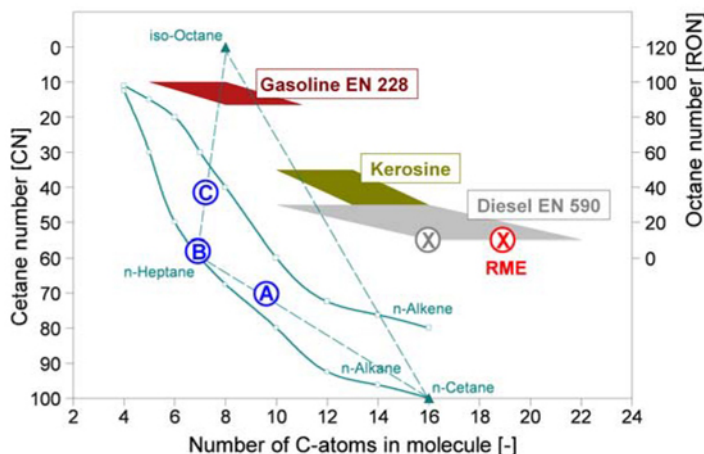


Figure 3. Cetane number vs. C-atoms in molecules

Differences in ignition delay based on fuel properties have a drastic impact on fuel air mixture generation and combustion. Therefore the primary objective in the selection of fuels for this study was to identify fuels of varying cetane number. Hence mixtures of primary reference fuels (PRF) with cetane numbers of CN = 42, 54, 69 were selected. The mixtures consist of n-heptane, iso-octane and n-cetane in varying fractions (figure 3). A detailed analysis of the fuels is presented in table 3. All PRF based fuels contain virtually no aromatics and have a very low boiling range, with T50% = 98 °C, compared to regular Diesel fuel, which has a boiling temperature of T50% ≈ 284 °C.

Since oxygen content is known to alter PM emissions significantly a fuel containing oxygen was added to the fuel matrix. Because of its relevance for the European market a biodiesel derived from rapeseed oil with 19 C-atoms was selected (RME). RME has a boiling temperature of T50% = 350 °C.

<table 3 here>

The baseline fuel for this study was Diesel EN 590. The baseline Diesel, Fuel B and RME have similar cetane numbers whilst varying in chemical composition. The differences in chemical composition lead to very different

Table 3. Test fuel properties

Fuel	Distillation T50% T90%	CN BASF	Density	Viscosity	Heating value
	°C	-	kg/m³	mm²/s	MJ/kg
Diesel EN 590	284 - 355	54	0.84	3.42	42.9
RME EN 14214	351 - 357	55	0.88	4.17	37.4
Fuel A	98 - 279	69	0.72	0.84	42.4
Fuel B	98 - 106	54	0.69	0.61	44.6
Fuel C	98 - 104	42	0.69	0.65	44.9

volatilities. This allows conclusions, how the volatility affects combustion and emissions independent of cetane number effects.

RESULTS

FULL LOAD

Figure 4 shows the peak load at $n = 4000 \text{ min}^{-1}$ for a maximum cylinder pressure of $p_{\text{cyl,max}} = 160 \text{ bar}$ and a maximum exhaust temperature of $T_{\text{exhaust,max}} = 735 \text{ °C}$. To validate the effects a variation of the COHR was carried out at a slightly lower load.

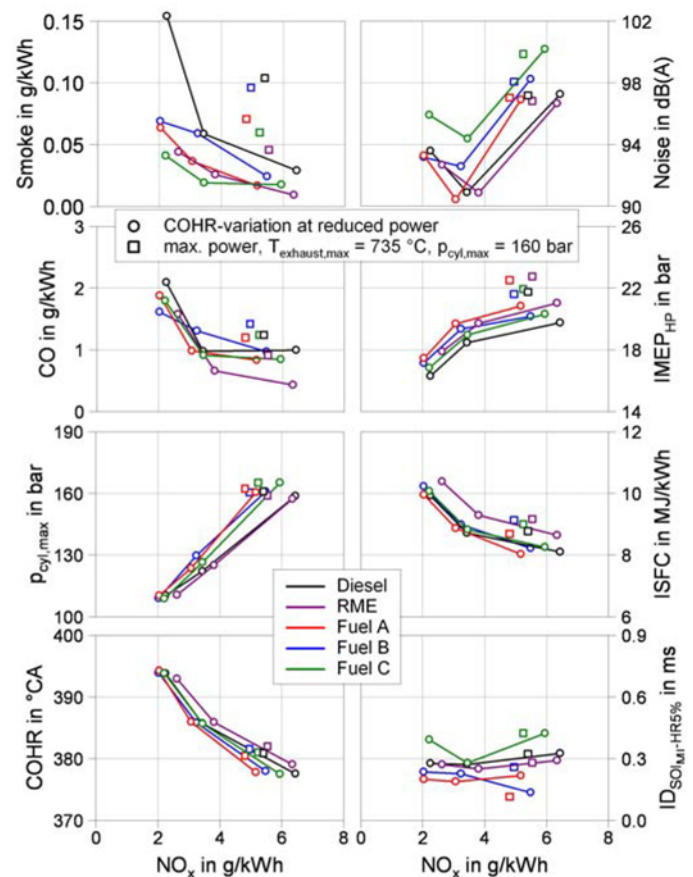


Figure 4. Full load, emission and COHR variation, OP3

The results indicate the following:

- Rated power at the given boundary conditions can be achieved with all selected fuels.
- Engine operation at full load is only slightly affected by fuel properties. No significant influence of boiling range, aromatics content and number of C-atoms per molecule can be detected. This is caused by the high in-cylinder pressures

at full load leading to a short ignition delay of $ID_{IBMI-ROHR5\%} \approx 0.3$ ms showing a marginal reduction with increasing cetane number.

- Improved fuel air mixture generation due to low cetane numbers leads to a slightly more pronounced premixed combustion and therefore to a higher heat release rate (figure 5) and combustion noise.
- Smoke emissions from RME are significantly lower compared to regular Diesel fuel although the cetane number of both fuels is similar and RME has a higher T50%. Hence the oxygen content of RME is believed to lower the engine out smoke emissions in this case.
- To achieve constant load, beginning of injection (BOI) and injection duration had to be adjusted according to heating value, density and viscosity of the fuel.

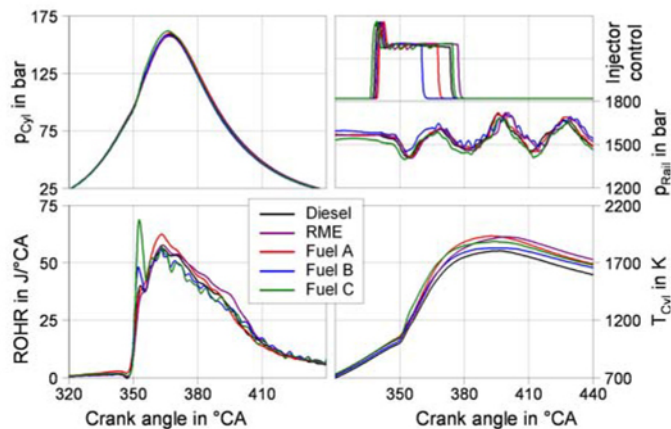


Figure 5. Pressure traces and heat release at full load, $T_{exhaust} = 735$ °C, OP3

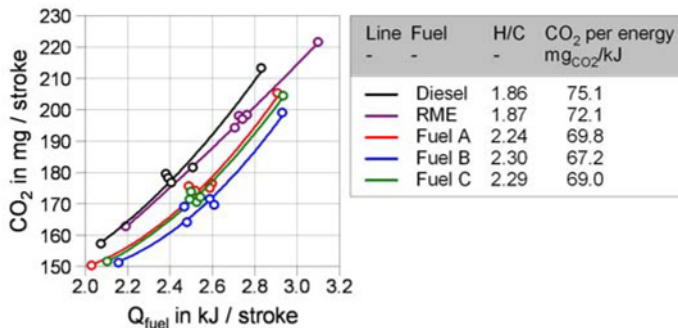


Figure 6. CO₂ emission vs. injected energy per stroke at full load, OP3

Figure 6 presents CO₂ emissions in mg per kJ fuel energy content in respect to the injected energy content per stroke. At full load there are distinct differences in CO₂ emissions due to varying H/C ratios and heating values. The short chain

length and therefore highest H / C ratio of fuel B leads to the lowest CO₂ emission.

UPPER PART LOAD

To assess fuel effects at upper part load (OP2) a reference point at EGR = 45 %, COHR = 377 °CA and $m_{PI} = 1.5$ mg was defined. This reference point was chosen, because it exhibits low exhaust gas pollutant emissions with regular Diesel fuel. In a first step the individual fuels were tested with the injection timings (start and duration of pilot and main injection) held constant at diesel fuel settings. In a second step the injection timings were adjusted for each fuel to meet the defined reference point (figure 7).

First step - results with constant injection timing

Without adjustments to the injection timing fuels with higher volatility show no combustion of the pilot injection. Fast evaporation of small amounts of fuel leads to lean, not ignitable, fuel air mixtures (Fuel A, B, C).

This causes an increasingly premixed combustion during the main combustion event and therefore higher pressure gradients (figure 7, left). With Fuel C a higher combustion noise is observed.

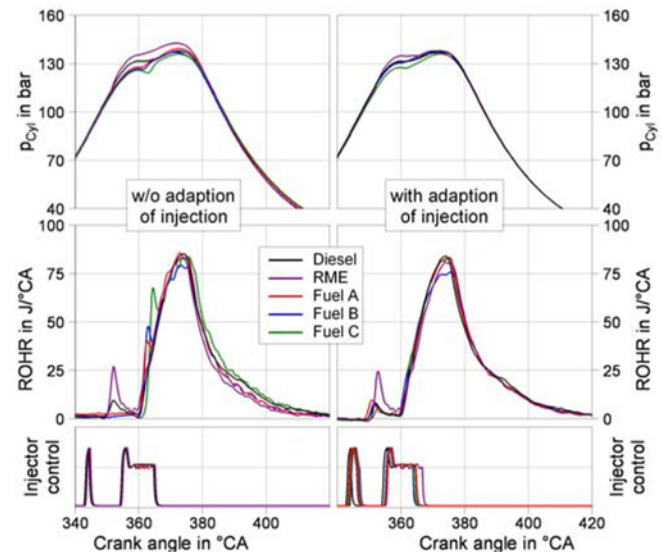


Figure 7. Pressure traces and heat release rate at reference point in upper part load, OP2

Second step - results with adjusted injection timing

To reach the reference point in terms of IMEP_{HP} and COHR beginning and duration of the main injection event were adjusted. In addition, the pilot injection mass was adjusted to $m_{PI} = 1.5$ mg with constant timing gap between start of pilot and main injection. This strategy required increased injection duration for pilot injection and main injection for RME while