

An Experimental Investigation of Dual-Fuel Combustion in a Light Duty Diesel Engine by In-Cylinder Blending of Ethanol and Diesel

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ABSTRACT

This study investigated dual-fuel operation with a light duty Diesel engine over a wide engine load range. Ethanol was hereby injected into the intake duct, while Diesel was injected directly into the cylinder. At low loads, high ethanol shares are critical in terms of combustion stability and emissions of unburnt hydrocarbons. As the load increases, the rates of heat release become problematic with regard to noise and mechanical stress. At higher loads, an advanced injection of Diesel was found to be beneficial in terms of combustion noise and emissions. For all tests, engine-out NO_x emissions were kept within the EU-6.1 limit.

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INTRODUCTION

Today, a vast effort is undertaken in order to reduce pollutant and Greenhouse gas (GHG) emissions of the transportation sector. Compared to spark ignition (SI) engines, compression ignition (CI) engines feature a higher thermo-dynamical efficiency, but require intense aftertreatment of the exhaust gas. With the current emission limits, typically a Diesel Oxidization Catalyst (DOC) for oxidation of hydrocarbon (HC) and carbon monoxide (CO) is used. Emissions of nitrogen oxides (NO_x) are prevented by utilization of e.g. a NO_x Storage Catalyst (NSC) or a Selective Catalytic Reduction (SCR) catalyst. Emitted particulate matter (PM) is reduced by a Diesel Particulate Filter (DPF). Next to these aftertreatment systems, engine-out emissions can be reduced at by in-cylinder measures as well. NO_x is formed at high temperatures and air-excess. Therefore, exhaust gas is recirculated (EGR) into the combustion chamber in order to reduce the combustion temperatures and lower NO_x emissions. However, this measure typically reduces the amount of oxygen available in the cylinder and thus worsens the soot oxidation. This phenomenon is also known as the "Diesel-dilemma." It has been proven that soot emissions can effectively be reduced by lowering the local equivalence ratio Φ at the start of combustion (SOC). For this reason, several concepts like Homogenous Charge Compression Ignition (HCCI) or Partially Premixed Compression Ignition (PPCI) have been suggested to achieve overall lower engine-out emissions [1,2,3]. It appears that the optimal fuel for such combustion concepts features an auto-ignition quality that is in between Diesel and gasoline fuel [4]. As a result, it is still challenging to utilize these combustion concepts within the whole engine load range. On the one hand, HCCI typically causes very high rates of heat release (ROHR), and thus steep cylinder

pressure gradients, also resulting in insufficient transient response. On the other hand, combustion concepts with highly homogeneous mixtures suffer from high levels of unburned hydrocarbon (HC) and carbon monoxide (CO) emissions particularly in the low load regime. Therefore, another combustion concept was sought after that combines both a low PM/ NO_x -trade-off and low HC- and CO-emissions. Consequently, a combustion strategy was proposed that utilizes a port fuel injection (PFI) of a low-reactivity fuel and direct injection (DI) of a high Cetane fuel. This dual-fuel operation allows the control of the overall mixture reactivity and the load-dependent combustion phasing. Hence, this in-cylinder fuel blending with early injection of the reactive fuel was called "Reactivity Controlled Compression Ignition" (RCCI) by Kokjohn et al. [5]. The RCCI concept is proven to provide a better controllability of combustion compared to e.g. (dual-fuel) HCCI and PCCI. It also contributes to very low PM- and NO_x -emissions for both light-duty (LD) and heavy-duty (HD) engines at various loads [6,7,8]. Moreover, gross thermal efficiencies in HD engines above 60% were reported, which is mostly related to reduced wall heat losses resulting from lower combustion temperatures. [9]

In this study dual-fuel operation with Diesel and ethanol was investigated in detail. Ethanol is regarded as promising fuel alternative for gasoline engines, which is why it was chosen as low reactivity fuel for port fuel injection. Ethanol has a higher octane rating than conventional gasoline RON95, so engine knocking is less problematic. This is of particular interest for Diesel engines that have a relatively high compression ratio (CR) when compared to gasoline engines. Furthermore, ethanol can easily be produced from renewable feedstock e.g. cellulose and hemicellulose via fermentation.

The engine load range investigated covers low part load as well as full load condition. The focus of the study was dual-fuel operation with late Diesel pilot injection that triggers combustion. During these experiments, the ethanol mass fraction varied from 0% to above 90% m/m. The engine-out NO_x -emissions were controlled via high EGR-rates and met EU-6.1 legislation for all part load points. In order to preclude effects of e.g. retarded combustion on efficiency and emissions, the center of combustion (CA50) was kept constant and independent of the ethanol mass share. In addition to the investigations with Diesel pilot injection, investigations with a very early Diesel injection were performed at several operating points. The investigations with early Diesel injection will be referred to as “RCCI” in the following. At full load operation the highest possible ethanol fuel mass share was targeted.

METHODOLOGY

In the following subsection, the characteristics of the utilized fuels, the specifications of the engine as well as the measurement devices will be introduced.

Fuel Characteristics

For PFI, a highly volatile and knock resistant fuel was chosen, namely ethanol. For DI, standard EN590 Diesel pump fuel was selected. The physical characteristics of the fuels are summarized by [Table 1](#).

Table 1. Fuel characteristics

	EN590 Diesel	Ethanol
Carbon mass fraction / %	84.47	52.14
Hydrogen mass fraction / %	13.27	13.13
Oxygen mass fraction / %	2.26	34.73
Density (25 °C) / kg/m ³	820	787
Boiling temperature / °C	170-350	78
Vapour pressure (20 °C) / kPa	<0.1	5.8
Specific enthalpy of vaporization / kJ/kg	358	912
Stoichiometric air requirement / 1	14.8	8.98
Lower heating value / MJ/kg	42.8	26.84
Research octane number (RON)	-	109
Cetane Number (CN)	53	-
Purity / %		>99.0

Single Cylinder Diesel Research Engine

The engine that was used for this study is a single cylinder engine (SCE) with a displacement of 0.39 l. The fuel injection equipment (FIE) for direct injection (DI) is near to series production, state-of-the-art and features an injection pressure up to 220 MPa. For port fuel injection (PFI) a Bosch valve was used. The engine control unit is based on a Rapid Control Prototyping (RCP) computer enabling a free

determination of the injection parameters for both DI and PFI. The engine features a maximum specific power output of 80 kW/l with peak firing pressures of 19.0 MPa. Due to intense charge-air and exhaust gas recirculation (EGR) cooling the engine meets EU-6 NO_x -level while simultaneously achieving low particulate matter (PM) emissions even in pure Diesel operation. For dual-fuel operation, the piston bowl geometry was not adjusted. Thus, the piston has a ω -shaped reentrant bowl with a volume of 21.6 cm³. The squish height is 0.78 mm and the compression ratio results in $\epsilon = 15:1$.

More detailed information of this engine can be found in previous publications [10,11]. [Table 2](#) summarizes the main specifications of the engine.

Table 2. Engine specifications

	Unit	
Displacement	cm ³	390
Stroke	mm	88.3
Bore	mm	75
Compression ratio	-	15:1
Valves	-	4
Max. cylinder pressure	MPa	19.0
Fuel injection equipment DI	-	Bosch Piezo Common Rail System
Max. injection pressure DI	MPa	220
Nozzle DI	-	8 hole, 109 μm hole diameter, 153° cone angle 620 cm ³ /min @10.0 MPa
PFI Valve	-	2-spray, 347 cm ³ /min @ 0.3 MPa
Boosting	-	Max. 0.38 MPa abs.

Investigated Load Regime

This study focuses on the application of a dual-fuel approach for a passenger car Diesel engine. Therefore, almost the whole engine load regime was investigated, covering low part load operation to full load operation. For all engine loads, an optimization of the engine calibration was carried out with Diesel only in previous studies. For the initial characterization of dual-fuel operation, most of the calibration parameters such as boost pressure, DI rail pressure, and center of combustion (CA50) were kept constant at part load operation. [Table 3](#) summarizes the main engine settings at part load operation. The transition of the EU-6.1 NO_x -level of 80 mg/km is into g/kWh for the four stationary load points is explained in [11].

The maximum engine power and torque is limited by the Filter Smoke Number (FSN), exhaust gas temperature, and peak firing pressure (PFP). These limits as well as the further calibration parameters (rail pressure, boost pressure etc.) were adapted from current state-of-the-art applications. The most important calibration parameters as well as the restraints at full load are presented in [Table 4](#).

Table 3. Engine calibration at part load

Speed / Load	CA50	Rail press. DI	Boost press.	Charge air temp.	Exh. gas press.	EU-6.1 NO _x level
rpm / MPa	° CA bTDC	MPa	MPa	°C	MPa	g/kWh
1500 / 0.43	-6.6	72	0.107	25	0.113	0.2
1500 / 0.68	-5.8	90	0.150	30	0.160	0.2
2280 / 0.94	-9.2	140	0.229	35	0.239	0.4
2400 / 1.48	-10.8	180	0.26	45	0.28	0.6

Table 4. Engine calibration and restraints at full load

Speed	IMEP DICI	Boost press.	Exh. gas press.	Rail press.	Charge temp.	Max. FSN	Max. exh. temp.	Max. cyl. press.
rpm	MPa	MPa	MPa	MPa	°C	-	°C	MPa
1000	1.47	0.15	0.19	100	35	2.6	850	13
2000	2.44	0.245	0.246	180	46	1.7	850	16
3000	2.77	0.27	0.32	200	53	2.5	850	19
4000	2.62	0.3	0.39	200	60	2.8	850	19

By Figure 1 the investigated part and full load points are shown within the range of the whole engine map.

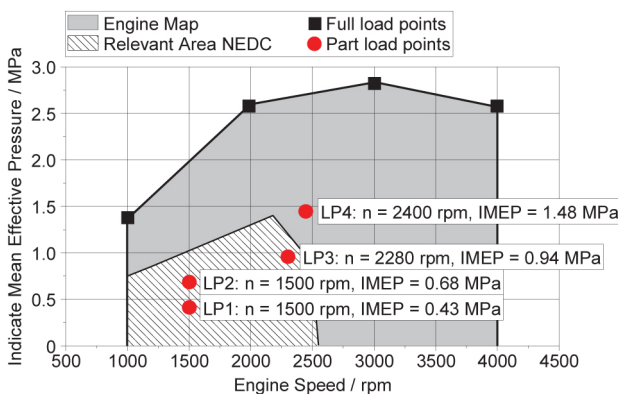


Figure 1. Engine map and investigated load points

Measurement Equipment

Regulated and unregulated emissions, i.e. CO, CO₂, HC, NO_x and PM are measured at engine exhaust by means of following sampling systems:

- HC: Flame ionization detector (Rosemount NGA 2000)
- O₂: Paramagnetic oxygen analyzer (Rosemount NGA 2000)
- CO: Infrared gas analyzer (Rosemount NGA 2000)
- CO₂: Infrared gas analyzer (Rosemount NGA 2000)
- NO_x: Chemiluminescence analyzer (Eco Physics 700 EL ht)
- PM: Filter paper method (AVL 415S)

For the HC-emissions, it was assumed that in common Direct Injection Compression Ignition (DICI)-operation all HCs are present as propane (C₃H₈). In contrast, in dual-fuel and RCCI-operation, HC-emissions were assumed to be ethanol (C₂H₅OH). During low load operation, the extraction site for exhaust gas is located in the exhaust gas manifold. During medium to high load operation, it is placed downstream of the back pressure valve, due to high exhaust gas back pressure. The sampling lines are heated up to 180 °C for the HC, CO, and NO_x measurements to avoid condensation. The sample line of the Smoke Meter is heated to 75 °C. The EGR-rate is calculated based on CO₂ concentration in the intake manifold and the exhaust gas. The fuel consumption of both ethanol and Diesel fuel is measured by a Coriolis-type fuel flow meter. An ultrasonic gas meter “Flowsic” by Sick is utilized to measure air volume flow. With the air temperature and its water content the air mass flow is calculated. The boost pressure can be adjusted independently by an external three-stage charging system which also provides low intake air temperatures via three charge air coolers. An electric, position controlled EGR-valve is used for adjusting the EGR-rate. The exhaust gas backpressure is controlled by two valves and a bypass in the exhaust gas track. For measuring the in-cylinder pressure, a water-cooled piezoelectric pressure transducer by Kistler (6041A) is utilized. FEV’s “Combustion Analyzing System” (CAS) records the cylinder pressure trace. It is used for determination of combustion related parameters such as CA50. All pressures are recorded in increments of 0.5 crank angle degrees (° CA) except the cylinder pressure, which is recorded at 0.1 ° CA. Indicated values, such as indicated mean effective pressure (IMEP) and displayed cylinder pressure traces, are recorded as an average of 50 cycles unless stated otherwise. The position of the crank angle and the differentiation between high pressure phase and gas exchange phase is measured by a crank angle sensor and an additional sensor at the intake camshaft. For calculation of burning rates, ignition delay, etc., the FEV software “Engine Process Optimization System” (EPOS) V4 was utilized. The ignition delay of the fuel is deduced from the time difference of the in-cylinder pressure rise due to ignition (5% of fuel mass converted) and the start of fuel injection (SOI). The time lag between the start of injector energizing (SOE) and SOI is 160 μs, almost independently of the injection pressure.

RESULTS AND DISCUSSION

Within this section, the experimental test results at part load and full load operation are presented.

Part Load Operation

In total, four part loads were investigated, ranging from 1500 rpm and 0.43 MPa IMEP up to 2400 rpm and 1.48 MPa IMEP. Before performing detailed investigations, the influence of end of PFI on combustion and emissions has been assessed. For these tests, a fixed ratio of Diesel to ethanol mass share of 50/50 been chosen, and no EGR was applied. It was determined that neither end of injection nor injection pressure has a measurable influence on the combustion and emission performance. For this reason, all investigations presented in this paper have been performed with a PFI pressure of 0.4 MPa and start of injection (SOI) at intake valve opens (IVO) during the same color code: conventional DICI operation (EN590 Diesel

100% m/m) is plotted in dashed grey lines, dual-fuel operation with Diesel pilot injection is shown by red rhombs, and RCCI measurements are displayed by blue stars.

n = 1500 rpm, IMEP = 0.43 MPa

At the lowest engine load tested, the ethanol mass share was limited to 22.5% m/m, due to engine misfire. The most important findings at this engine load for 0%, 10% and 22.5% m/m ethanol are given by Figure 2. As seen in the upper graph, the FSN is very low for pure DICl as well as in dual-fuel operation. The reason might be that the low in-cylinder temperatures and pressures lead to a long ignition delay and therefore, sufficient time for fuel and air mixing. The indicated specific NO_x -emissions are illustrated in the graph below. For all measurements these emissions were controlled via EGR to be compliant with EU-6 legislation, which is why there is no difference to be seen between the combustion concepts. A significant difference can be seen in HC-emissions. Only 10% m/m ethanol cause the HC-emissions to rise by almost 100% compared to Diesel-only operation. With 22.5% m/m ethanol, the HC-emissions are even the threefold of those in Diesel-only operation. Previous studies have reported an increase in HC-emissions with increasing ethanol content [12,13]. At low ethanol fraction of 10% m/m the COV of IMEP is even below that of pure Diesel-operation, and thus misfiring is not the main cause of the high HC-emissions. Instead it is believed that ethanol is trapped in the crevices and does not take part in combustion. [14] The temperature drops rather rapidly, so the HCs cannot be oxidized during the exhaust stroke. In addition, flame quenching is contributing to the elevated HC-emissions. A similar trend, although not as badly affected as the HC-emissions, can be observed for the CO-emissions. They increase by about 4 g/kWh, which contributes to a rise of 40% with only 10% m/m ethanol. While the ethanol fraction continues to increase, an incline in CO-emission can be observed: With 22.5% m/m, the CO-emissions are 17.3 g/kWh compared to 11 g/kWh with 100% m/m Diesel. These observations are in contrast to what was presented by Padala et. al. [12] who have shown that CO-emissions decrease when adding small amounts of ethanol as a result of its higher combustion temperature.

The reason for this may be that, on one hand the low compression ratio of the investigated configuration and the fact, that for all ethanol fractions investigated in this study, high EGR rates up to 50% were applied, and thus the combustion temperatures were significantly decreased. Moreover, the engine is boosted, so an overall equivalence ratio of $\Phi \sim 0.62$ -0.64 was maintained despite the high EGR-rates. This resulted in a significantly diluted cylinder charge. Furthermore, this load point is characterized by quite low temperatures, making the low temperature heat release (LTHR) an important parameter for combustion. It was shown previously that ethanol reduces the LTHR and cool flame compared to Diesel-surrogates. The reason is that the fuel mixture is mostly oxidized by reacting with OH radicals. However, when substituting Diesel with ethanol, the production of OH radicals is less since the concentration of Diesel, which can easily peroxydize and yield chain branching, is reduced. [15,16,17] Despite slightly increased losses in incomplete combustion, the overall efficiency is not affected by increasing the ethanol fraction up to 22.5% m/m. This is mainly due to the rather similar combustion and heat release trace with same CA50. However, the slightly reduced heat release rate with 22.5% m/m ethanol (see Figure 3) might reduce wall heat losses and compensate the higher losses caused by incomplete combustion.

Furthermore, the reduction in ROHR causes a drop in the maximum pressure rise rate (MPRR) and a reduction in the combustion noise. Still, the heat release rate is high and thus combustion temperature is high, which is why EGR-rates of 50% are required in order to meet EU-6.1 conform NO_x -emission of 0.2 g/kWh.

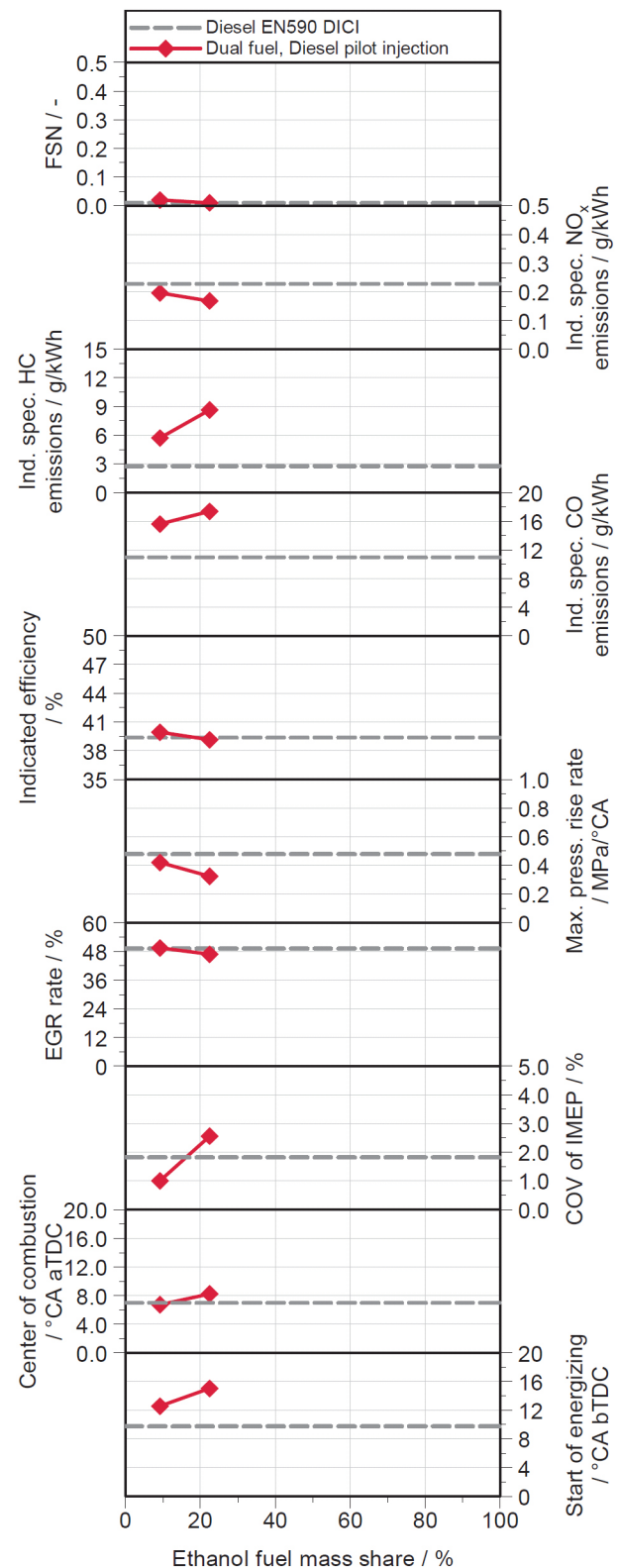


Figure 2. Experimental results with pure EN590 Diesel and dual-fuel operation at 1500 rpm and 0.43 MPa IMEP.

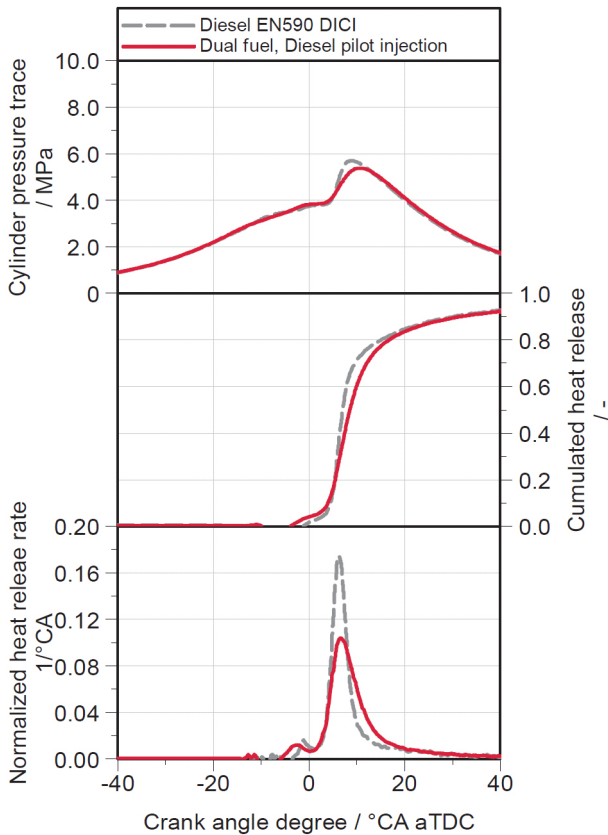


Figure 3. Averaged cylinder pressure trace and heat release with pure EN590 Diesel and Diesel/ethanol dual-fuel operation (22.5% m/m ethanol) at 1500 rpm and 0.43 MPa IMEP

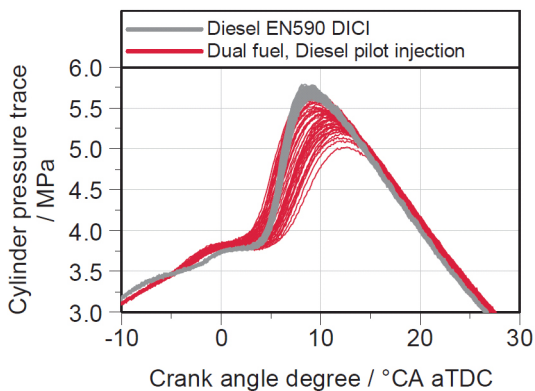


Figure 4. 50 consecutive cylinder pressure trace and heat release with pure EN590 Diesel and Diesel/ethanol dual-fuel operation (22.5% m/m ethanol) at 1500 rpm and 0.43 MPa IMEP

Even though the COV of IMEP (see Figure 2) is not above the misfiring limit of 5% [18], the cyclic fluctuations of the cylinder pressure trace are much more distinct than with pure Diesel operation.

Figure 4 shows 50 consecutive cylinder pressure traces for 0% and 22.5% m/m ethanol. Both measurements in average have a similar CA50, but obviously the cycle-to-cycle variation is high with 22.5% m/m ethanol. With increasing ethanol fraction, this problem became even worse. Starting at the engine settings for 22.5% m/m ethanol, the combustion could not be controlled anymore with only slightly higher ethanol shares of about 25% m/m. Furthermore, utilization of a very early Diesel injection, as it is proposed by [5], was not favorable, since combustion did not take place when advancing the injection. Because combustion was not stable for 30s that are required for proper data acquisition, no data could be extracted.

n = 1500 rpm, IMEP = 0.68 MPa

Another load point with same engine speed of 1500 rpm, but higher load of 0.68 MPa IMEP was investigated as well. The findings are displayed in Figure 5.

As it can be concluded, with higher load the ethanol mass fraction could be further extended. Compared to the maximum of 22.5% m/m at an IMEP of 0.43 MPa, a share of exact 50% m/m ethanol could be introduced without sacrificing combustion stability.

As it appears in the upper graph, within this load point a huge benefit in FSN can be gained already with only 28% m/m ethanol while still meeting EU-6.1 NO_x-legislation (0.2 g/kWh) without the need of DeNO_x-aftertreatment (see second graph).

Similar to the lower load point, an almost linear correlation of HC-emissions and ethanol mass fraction can be observed. But, due to higher temperatures and power output, specific HC-emissions are lower than at the previously discussed load point with similar ethanol content. However, at 50% m/m ethanol specific HC-emissions are 10 g/kWh. Even being not critical in terms of emissions due to appropriate exhaust gas temperature and catalyst conversion efficiency, it finally identifies a lack in combustion efficiency. This is a relative increase by the tenfold compared to pure Diesel operation. At this operating condition, the COV of IMEP is 1%, and the corresponding consecutive cylinder pressure traces show no hint of misfiring (see Figure 7). Thus, the rise in HC-emissions is believed to be mainly caused by ethanol trapped in parasitic volumes and crevices. When looking at the specific CO-emissions, again a similar trend can be observed. As in the lower load point, the incline in emission is not as dramatic as with the HC-emissions. This increase should be due to flame quenching of the premixed highly diluted ethanol/air mixture. Previous studies already pointed out that low temperature combustion concepts with high mixture homogenization temperatures at the cylinder wall and piston surface are too low to enable a sufficient oxidization of CO to CO₂ [19,20].