

An Investigation into the Effects of Fuel Properties and Engine Load on UHC and CO Emissions from a Light-Duty Optical Diesel Engine Operating in a Partially Premixed Combustion Regime

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ABSTRACT

The behavior of the engine-out UHC and CO emissions from a light-duty diesel optical engine operating at two PPCI conditions was investigated for fifteen different fuels, including diesel fuels, biofuel blends, n-heptane-iso-octane mixtures, and n-cetane-HMN mixtures. The two highly dilute (9-10% O₂) early direct injection PPCI conditions included a low speed (1500 RPM) and load (3.0 bar IMEP) case-where the UHC and CO have been found to stem from overly-lean fuel-air mixtures-and a condition with a relatively higher speed (2000 RPM) and load (6.0 bar IMEP)-where globally richer mixtures may lead to different sources of UHC and CO. The main objectives of this work were to explore the general behavior of the UHC and CO emissions from early-injection PPCI combustion and to gain an understanding of how fuel properties and engine load affect the engine-out emissions.

The UHC and CO emissions from an emissions certification diesel fuel and several soy- and palm-based biofuel blends were found to partially collapse when plotted as a function of the premixed burn duration for the low speed and load condition. Significant reductions in both UHC and CO were observed with the biofuel blends. The engine-out UHC and CO emissions and premixed burn duration, which directly correlates with the magnitude of the peak heat-release, were shown to depend on three general parameters: (1) the injection timing, which controls the targeting and the initial thermodynamic state of the ambient gases; (2) the mixing time between the fuel and ambient gases during the ignition delay period; and (3) the premixed combustion phasing, which controls the temperature, pressure, and mixture state

during premixed combustion. In order to investigate the effects of ignition quality and volatility in greater detail, a controlled parametric study involving nine different fuels was conducted. The fuels were selected to provide an orthogonal fuel matrix in which ignition quality and volatility were varied independently. The advancement of the combustion phasing for fuels with higher ignition quality was found to significantly decrease the UHC and CO emissions, while fuel volatility was determined to have a much smaller effect on the engine-out emissions. The emissions from the higher speed and load condition were found to be mainly dependent on the injection timing and were less sensitive to fuel-type. This is consistent with previous studies that have attributed the behavior of the UHC and CO emissions at higher loads to bulk mixing processes controlled by the targeting of the injected fuel spray.

INTRODUCTION

In recent years, growing concerns over the emissions of greenhouse gases have prompted automobile consumers and manufacturers to consider alternative fuel sources and more efficient power plant options. Diesel engines, in particular, have become more attractive in automotive applications because of their superior efficiency compared to spark ignition engines and their compatibility with the current energy infrastructure. However, the wide-spread use of diesel engines in automobiles is currently constrained by regulations on the emissions of nitrogen oxides (NO_x) and particulate matter (PM). This has led engine manufacturers and researchers to investigate methods to reduce NO_x and PM, while maintaining the high energy conversion efficiencies inherent in traditional diesel combustion. One approach to this problem is the utilization of advanced combustion

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strategies that limit the production of NO_x and PM by avoiding high temperatures and fuel-rich combustion. This work focuses on the advanced combustion strategy called partially premixed compression ignition (PPCI). PPCI combustion uses high levels of exhaust gas recirculation (EGR) and relatively low compression ratios to decrease in-cylinder temperatures and increase ignition delay times, allowing for partial premixing of the injected fuel and limiting the production of both NO_x and PM. The low in-cylinder temperatures and dilute mixtures, however, also limit the ability of the injected fuel to fully oxidize, potentially leading to increased emissions of unburned hydrocarbons (UHC) and carbon monoxide (CO) and decreased combustion efficiency. An attractive method for mitigating these fuel oxidation issues is the use of biofuels, which have been found to reduce UHC and CO emissions in traditional diesel engines [1,2]. This study examines the general behavior of the UHC and CO emissions from biofuel blends at two PPCI operating conditions and investigates the effects of ignition quality, volatility, and engine load on the engine-out emissions.

Several previous studies have concentrated on determining the main sources of UHC and CO emissions from the optically accessible diesel engine used in the current study. The engine was operated in a highly dilute (10% O_2 mole fraction), slightly boosted ($P_{\text{intake}} = 1.5$ bar), low load (3.0 bar IMEP), PPCI regime [3,4,5,6]. These investigations used a UHC planar laser-induced fluorescence (PLIF) technique to measure in-cylinder distributions of a wide range of UHC species and a spectrally resolved CO LIF technique to determine the late-cycle (30° and 50° aTDC) distributions of CO in the clearance volume above the piston-crown. It was determined that the UHC and CO emissions mainly originated from over-mixed, overly lean fuel-air mixtures located within the squish region. Other sources of UHC and CO included: (1) liquid fuel dribble from the injector well after the main injection event, (2) piston fuel films, (3) crevice flows, and (4) overly-lean mixtures near the injector tip [3].

Additional measurements [7], using the same diagnostic techniques in the same engine, investigated how several biofuel blends influenced the above sources of UHC and CO. Six different fuels or fuel blends were tested, including a 2007 emissions certification diesel, 20% and 50% by volume blends of either soy methyl ester (SME) or palm methyl ester (PME) biofuel, and neat SME fuel. It was demonstrated that the addition of the SME and PME fuels resulted in a substantial decrease in UHC and CO emissions, while the engine-out soot and NO_x levels remained low. This study [7] also highlighted that at a highly-dilute (10% O_2 mole fraction in intake), low load (3.0 bar IMEP) operating condition, the bulk flows and mixing that occur within the combustion chamber were similar for all of the tested fuels-leading to the

same gross sources of UHC and CO as listed above for the diesel fuel. However, the relative amount of emissions generated by each source was found to be altered by the addition of the biofuels. The magnitude of the UHC and CO emissions in the squish region was found to decrease for the biofuel blends. There was also evidence that the biofuels resulted in smaller late-cycle piston-top fuel films, fewer UHCs in the crevice flows, and less overly-lean UHC in the near-injector region.

At higher loads, the globally richer fuel-air mixtures potentially lead to different sources of UHC and CO. Several studies [9,10] have investigated a highly dilute (9% O_2), slightly boosted ($P_{\text{intake}} = 1.5$ bar), medium load (6.0 bar IMEP) PPCI operating condition over a range of start of injection (SOI) timings using a standard diesel fuel. Optimal injection timings were observed where the emissions of UHC and CO were minimized. Opat et al. [9] suggested that the optimal injection timings coincided with the spray targeting that most efficiently utilized the available O_2 within the combustion chamber. Koci et al. [10] added species-resolved measurements of the UHC emissions from the same engine. These authors also concluded that the behavior of the UHC and CO emissions was a result of bulk mixing processes controlled by the injection timing, and resulted from a combination of different behaviors of the light ($<C_6$) and heavy ($>C_6$) hydrocarbon species.

The current work adds to these previous studies by examining the general behavior of the UHC and CO emissions with respect to the combustion process for several biofuel blends and investigates the effects of fuel ignition quality, volatility, and engine load on the engine-out emissions. The operating conditions were chosen to match those used in the previous studies discussed above. The two highly dilute (9-10% O_2), boosted ($P_{\text{intake}} = 1.5$ bar), PPCI conditions consisted of a low speed (1500 RPM) and load (3.0 bar IMEP) case-where the UHC and CO emissions have been found to stem from overly-lean fuel-air mixtures-and a higher speed (2000 RPM) and load (6.0 bar IMEP) case-where the increased fueling rates may lead to different emissions sources. The behavior of the UHC and CO emissions from a 2007 emissions certification diesel fuel and the five biofuel blends was initially investigated at the low speed and load condition to determine the relationship between the emissions and several combustion metrics (ignition delay, combustion phasing, burn durations, etc.). Further experiments were then conducted to isolate the effects of ignition quality and volatility on the engine-out UHC and CO emissions. An orthogonal fuel matrix was constructed with several n-heptane-iso-octane, n-cetane-HMN, and diesel fuel mixtures in order to vary the ignition quality and volatility independently. Finally, the emissions from the slightly higher speed and load condition were examined to contrast the emissions behavior between different load and speed conditions.

EXPERIMENTS

The following sections describe the engine facility, data acquisition systems, operating conditions, and fuels that were used throughout this study. Additional details regarding the engine and its performance relative to an all-metal test engine can be found in [11,12].

ENGINE

The single-cylinder optical research engine used in this work was derived from a General Motors 4-cylinder 1.9 L light-duty diesel engine. A cross-section of the optically accessible engine is shown in [Fig. 1](#), with the engine and fuel injection system specifications provided in [Table 1](#). The engine features a fused-silica piston top that includes the reentrant bowl and valve cut-out geometries that are typical of a production piston. The only significant deviation from a production piston geometry is the elongation of the top ring-land crevice to accommodate imaging within the piston bowl. The lower ring-pack position results in a crevice volume 3-4 times larger than that of a typical engine. The engine has a geometric compression ratio (CR) of 16.7, but an effective CR of 14.0 more closely matches the pressure behavior predicted by a one-dimensional engine model [11]. The discrepancy between the geometric and effective compression ratios is due largely to heat transfer within the increased crevice volume. Two gapless compression rings were used to minimize blowby. A previous study conducted on the same engine [12] has shown that the UHC and CO emissions trends from a similar single-cylinder all-metal engine with a more typical ring-land crevice volume were well captured by the optical engine.

(See [Figure 1](#) after last section of paper)

Table 1. Engine, Valve, and Fuel Injection Specifications.

Engine Geometry	
Bore	82.0 mm
Stroke	90.4 mm
Displacement Volume	0.477 L
Geometric CR	16.7
Effective CR	14
Squish Height	0.78 mm
Valves	
Intake Valves	2
IVO	-359 °CA
IVC	-152 °CA
Exhaust Valves	2
EVO	132 °CA
EVC	360 °CA
Injection Parameters	
Injector	Bosch CRI2.2
Sac Type	Mini Sac
Sac Volume	0.23 mm ³
Holes	7
Nozzle Diameter	0.14 mm
Nozzle Area	0.0154 mm ²
Included Angle	149°
Rail Pressure	860 bar

DATA ACQUISITION

In-cylinder pressure measurements were acquired with 0.25° CA resolution using a Kistler 6125 quartz pressure transducer mounted in the glow-plug position within the head of the engine. The pressure measurements were used to determine the apparent heat-release rate (AHRR) and the indicated mean effective pressure (IMEP) for each experimental run by averaging over 50 fired cycles. The exhaust gases were sampled through a heated sampling line and filter connected to the exhaust plenum. A California Analytical Instruments (CAI) flame ionization detector (CAI 600 MHFID) was used to measure the UHC content and the CO was measured with a CAI Non-Dispersive Infrared (NDIR) gas analyzer after the UHC and water were removed by passing the exhaust sample through a condenser.

OPERATING CONDITIONS

The UHC and CO emissions were examined over a wide range of injection timings at two PPCI operating conditions. A light load (3 bar IMEP), highly dilute (10% O₂), boosted intake ($P_{intake} = 1.5$ bar), 1500 RPM, baseline operating condition was chosen to match the conditions studied

extensively in the past using the same engine [5,7]. Previous engine-out soot, UHC, CO, and NO_x emissions measurements acquired over a wide range of O₂ concentrations, injection timings, and intake pressures have shown that this condition provided an adequate combustion efficiency (~98%), although the UHC and CO emissions were high compared to conditions with higher oxygen concentrations [11]. In other words, this condition represents a limit to the amount of charge dilution that can be achieved before the combustion deteriorates to unacceptable efficiency levels. A further understanding of the mechanisms leading to UHC and CO emissions at this condition could lead to improvements in light load PPCI operation-where UHC and CO emissions are most problematic. The second operating condition had a higher speed (2000 RPM) and load (6.0 bar IMEP), and represents one of the higher speed/load conditions expected in a typical urban drive cycle. This condition has been studied in Refs. [9,10], which used an all-metal engine with a similar geometry as the optical engine used here. The UHC and CO emissions from the all-metal engine have previously been found to follow trends similar to those observed with the current optical engine [12]. The operating conditions used in this study are summarized in Table 2.

Table 2. PPCI Operating Conditions.

	Condition 1	Condition 2
Engine Speed [RPM]	1500	2000
Intake Pressure [bar]	1.5	1.85
Intake Temperature [°C]	99	72
Oxygen Mole Fraction	10%	9%
Load (IMEP) [bar]	3.0	6.0
Swirl Ratio	2.2	2.2
Air Mass Fraction	0.460	0.412
N ₂ Mass Fraction	0.402	0.425
CO ₂ Mass Fraction	0.138	0.162

To limit the combustion chamber surface temperatures and to minimize optical surface fouling, the engine was fired once every 5 cycles. The skip-fired operation required simulated EGR, consisting of N₂ and CO₂, to be delivered to the engine in order to obtain the desired intake O₂ concentration. The diluents N₂ and CO₂ were added to the intake stream in proportions such that the specific heat ratio of the simulated EGR matched the specific heat ratio of complete combustion products at 600 K (estimated in-cylinder temperature at the time of injection). Mass fractions of the intake flows used in this study are included in Table 2. These mass fractions correspond to an EGR rate of 73% and 67% for conditions 1 and 2, respectively. A previous study using the same engine facility [12] investigated the effects of including H₂O, CO and C₃H₈ in the simulated EGR stream. It was determined

that the inclusion of these species affected the ignition delay and emissions levels modestly. However, the emissions trends were closely matched to those obtained with the less reactive simulated EGR (N₂ and CO₂).

To avoid excessive wear of the oil-less piston rings and enable regular cleaning of the optical components, the engine run times were limited. Consequently, a steady-state condition was not achieved and a strict 8 minute schedule was followed to ensure that the thermal state of the engine was consistent for each run. The schedule began with 105 seconds of motored operation to stabilize the intake plenum pressures and was followed by 60 seconds of skip-firing to preheat the combustion chamber surfaces. Emissions data were then collected within the next 30 seconds, and the engine was stopped and allowed to cool for the remainder of the 8 minute cycle.

FUEL PROPERTIES

Certification Diesel and Biofuel Blends

Six different fuels, including a 2007 emission certification diesel fuel and five biofuel blends, were initially tested to determine the general behavior of the UHC and CO emissions. The biofuel blends consisted of 20% and 50% by volume mixtures of either SME or PME biofuel with the certification diesel. Neat diesel fuel and SME were also tested. The main fuel properties for each of the constituent fuels are provided in Table 3. The SME and PME biofuels were processed using straight vegetable oil (SVO) and were made from 100% soy and palm oil, respectively. The compositional data for the SME and PME fuels were determined by an independent lab and the composition of the diesel fuel was supplied by the manufacturer. The compositions of the SME and PME fuels used in this study (provided in Table 3) are typical of SME and PME biofuel compositions cited in the literature [2].

(See Table 3 after last section of paper)

Table 4 provides the properties and nomenclature of the biofuel blends used in this study and the distillation curves for each blend are shown in Fig. 2. The distillation curves were estimated by fitting a curve to the data reported in [2] for each constituent of the fuel blends. The composite distillation curves were created by weighting the volume % recovered values for each fuel by its volume fraction, as shown in Eq. (1):

$$Vol(T) = \sum_{i=1}^N v_{f,i} \cdot Vol_i(T) \quad (1)$$

where $Vol(T)$ is the estimated volume percent recovered of the fuel blend at a given temperature T . N is the total number

of fuels that comprise the blend, v_{fi} is the volume fraction of fuel i , and $Vol_i(T)$ is the volume percent recovered of fuel i . The cetane number values reported in Table 4 were determined in a similar manner. The composite distillation curves indicate that the biofuels (SME and PME) and their blends are less volatile than the diesel fuel at low distillation temperatures. However, the distillation behavior of all the fuels becomes similar above the T_{95} point.

Table 4. Biofuel Blend Properties and Corresponding Nomenclature.

	Diesel [% vol]	PME [% vol]	SME [% vol]	LHV [MJ/kg]	Cetane Number
Cert. Diesel	100	0	0	42.9	47.1
PME-20	80	20	0	41.5	50.3
PME-50	50	50	0	39.5	55.1
SME-20	80	0	20	41.6	47.0
SME-50	50	0	50	39.9	46.9
SME-100	0	0	100	36.9	46.7

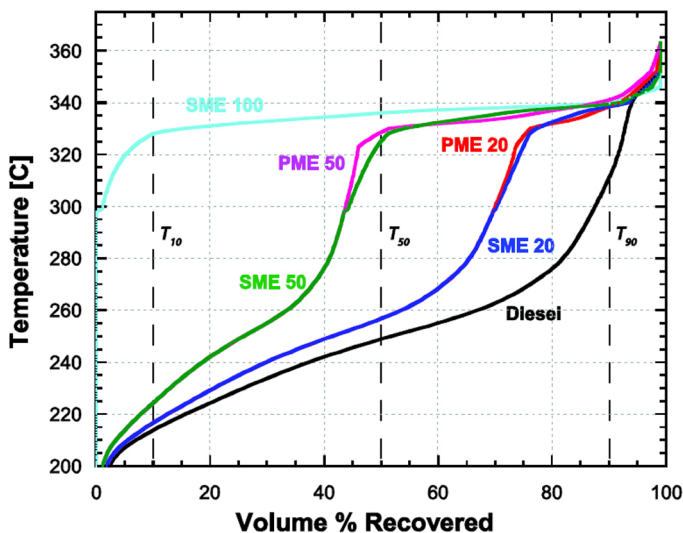


Figure 2. Distillation curves for each of the investigated fuels. The biofuel blends are less volatile than the diesel fuel at low volume percent recovered values. Above the T_{95} point, distillation behavior of all the fuels is similar.

Ignition Quality-Volatility Matrix

To investigate the effects of fuel properties in a more controlled manner, a fuels matrix was constructed to isolate the effects of ignition quality and volatility. The fuels matrix is displayed in Fig. 3 and the properties for all of the fuel mixtures are provided in Table 5. Three levels of ignition quality and volatility were chosen to provide an orthogonal matrix in which the ignition quality and combustion phasing were varied independently. The three levels of volatility consisted of: (1) mixtures of the high volatility primary research fuels (PRFs), n-heptane and iso-octane, which have

boiling points ($T_{boil} \sim 90^\circ\text{C}$) well below the initial boiling point temperature of the certification diesel fuel, (2) mixtures of n-cetane and 2,2,4,4,6,8,8-heptamethylnonane (HMN), which have mid-level volatility with boiling points ($T_{boil} \sim 260^\circ\text{C}$) that are located near the middle of the certification diesel's distillation curve, and (3) mixtures of two diesel fuels from the Fuels for Advanced Combustion Engines (FACE) fuel matrix [15] (FACE 2 and FACE 6) that have T_{90} temperatures similar to the biofuel blends. Note that volatility is correlated strongly with fuel density, and a concurrent variation in the latter property was also incurred. The three levels of ignition quality were developed by mixing each group of fuels in proportions such that the combustion phasing of the mixture matched fuels with approximate cetane numbers of 38, 47, and 56. The combustion phasing was matched at a single injection timing of 23.1° bTDC. However, the phasing between matched fuels remained similar over the entire range of injection timings that were tested.

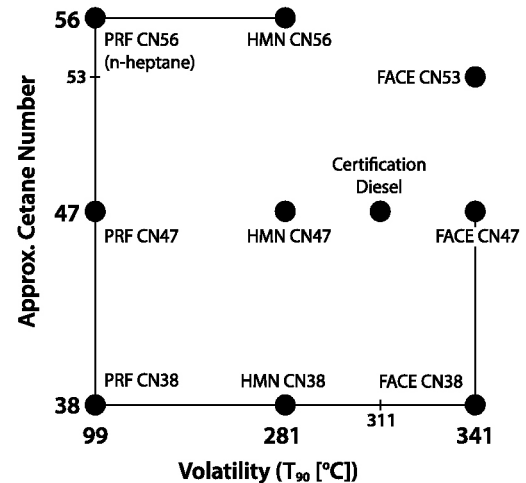


Figure 3. Fuels matrix for ignition quality-volatility study. Fuels were mixed such that the combustion phasing at an SOI of 23.1° bTDC was appropriately matched to other fuel mixtures. Cetane numbers are approximate (see Table 5).

(See Table 5 after last section of paper)

RESULTS

The results are divided into two sections, which present (1) the engine-out UHC and CO emissions from the 1500 RPM, 3.0 bar IMEP condition and (2) the corresponding emissions from the 2000 RPM, 6.0 bar IMEP condition. The first section includes an analysis of the relationship between the UHC and CO emissions and several combustion parameters for the diesel and biofuel blends. This analysis provides a foundation to understand the behavior of the engine-out emissions and how they are affected by different fuel-types.

This is followed by an in-depth investigation into the effects of ignition quality and volatility using the fuels matrix described above. The second section contains an examination of the UHC and CO emissions from the 2000 RPM, 6.0 bar IMEP condition to highlight the differences in emissions behavior at higher speeds and loads.

1500 RPM, 3.0 BAR IMEP CONDITION

Certification Diesel and Biofuel Blends

Sample apparent heat-release rate (AHRR) profiles are presented in Fig. 4 for the certification and biodiesel blends at an injection timing of 23.1° bTDC. Although the data presented in Fig. 4 were acquired at a single injection timing, the trends observed in the heat-release parameters (peak heat-release rate, combustion phasing, low-temperature heat-release, etc.) remain similar over the range of tested injection timings.

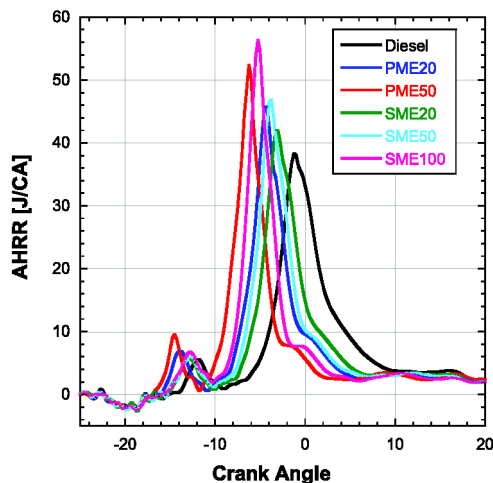


Figure 4. Apparent heat-release rate (AHRR) profiles at the low speed and load condition (SOI = 23.1° bTDC, 1500 RPM, 3.0 bar IMEP, 10% O₂) for the diesel and biofuel blends [7].

The AHRR curves in Fig. 4 show that the premixed combustion phasing advances with the addition of the PME and SME biofuels, with the PME advancing the combustion farther for a given blending fraction. The peak heat-release rate also increases as the main combustion event is advanced. Note that the peak heat-release increases monotonically as the amount of a given biofuel is increased. However, the rate of increase is different for the SME and PME fuels. The behavior of the low-temperature heat-release (LTHR) is also different between the two biofuels. The LTHR of the SME blends have a similar phasing and peak magnitude, while the LTHR advances and increases in magnitude as PME is added to the diesel fuel.

The engine-out UHC and CO emissions for the certification diesel and biofuel blends are shown in Fig. 5 as a function of injection timing. These data for the biofuel blends have previously been presented in [7] with complementary NO_x, soot, and optical measurement data.

(See Figure 5 after last section of paper)

The behavior of the engine-out UHC and CO emissions with respect to the injection timing is similar for all of the tested fuels. Fuel effects are apparent at all injection timings for both the UHC and CO emissions and are most noticeable at retarded injection timings, where the UHC emissions vary greatly over the range of fuels. Minima in UHC are observed near an SOI of 21° bTDC while minima in CO are slightly retarded at 19° bTDC. The presence of these minima have previously been attributed to a competition between excessive fuel being injected into the squish volume at advanced injection timings and the formation of overly-lean mixtures in the squish volume at retarded injection timings [6, 9]. The UHC emissions are also roughly symmetric at SOI timings advanced and retarded from the location of the minima. However, the CO emissions increase more slowly at the retarded injection timings. It is interesting to note that the emissions behavior is similar for all of the fuels even though the fuels cover a wide range of ignition qualities and, therefore, have differing combustion phasing at a given SOI (see Fig. 4). This suggests that the targeting of the injected fuel spray plays a central role in the behavior of the UHC and CO emissions at this condition.

UHC EMISSIONS

The dependency of the engine-out UHC and CO emissions on several combustion parameters, including: ignition delay, premixed combustion phasing (CA50), premixed combustion duration (B50-10, the crank angle duration between the points at which 10% and 50% of the fuel energy has been released), peak heat-release rate, and mixing-controlled combustion duration (B90-70), were investigated to demonstrate how the combustion process affects the emissions of UHC and CO. It was determined that the UHC emissions were best correlated by two closely related parameters, the premixed burn duration (B50-10) and the peak heat-release rate. Since the engine load and, therefore, the approximate amount of energy released from each fuel were held constant in all of the experiments, the premixed burn duration is a good indicator of the peak AHRR-short premixed burn durations lead to high peak release rates and high in-cylinder temperatures. Figure 6 shows that the measured UHC emissions partially collapse when plotted against B50-10. The premixed burn duration (B50-10), instead of the peak AHRR, was investigated in this study because it can be broken down into more fundamental components as will be shown later in this section.