

Characterization of Reactivity Controlled Compression Ignition (RCCI) Using Premixed Gasoline and Direct-Injected Gasoline with a Cetane Improver on a Multi-Cylinder Engine

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ABSTRACT

The focus of the present study was to characterize Reactivity Controlled Compression Ignition (RCCI) using a single-fuel approach of gasoline and gasoline mixed with a commercially available cetane improver on a multi-cylinder engine. RCCI was achieved by port-injecting a certification grade 96 research octane gasoline and direct-injecting the same gasoline mixed with various levels of a cetane improver, 2-ethylhexyl nitrate (EHN). The EHN volume percentages investigated in the direct-injected fuel were 10, 5, and 2.5%. The combustion phasing controllability and emissions of the different fueling combinations were characterized at 2300 rpm and 4.2 bar brake mean effective pressure over a variety of parametric investigations including direct injection timing, premixed gasoline percentage, and intake temperature. Comparisons were made to gasoline/diesel RCCI operation on the same engine platform at nominally the same operating condition. The experiments were conducted on a modern four cylinder light-duty diesel engine that was modified with a port-fuel injection system while maintaining the stock direct injection fuel system. The pistons were modified for highly premixed operation and feature an open shallow bowl design. The results indicate that the authority to control the combustion phasing through the fuel delivery strategy (e.g., direct injection timing or premixed gasoline percentage) is not a strong function of the EHN concentration in the direct-injected fuel. It was also observed that NO_x emissions are a strong function of the global EHN concentration in-cylinder and the combustion phasing. In general, NO_x emissions are significantly elevated for gasoline/gasoline+EHN operation compared with gasoline/diesel RCCI operation at a given operating condition.

CITATION: Dempsey, A., Curran, S., and Reitz, R., "Characterization of Reactivity Controlled Compression Ignition (RCCI) Using Premixed Gasoline and Direct-Injected Gasoline with a Cetane Improver on a Multi-Cylinder Engine," *SAE Int. J. Engines* 8(2):2015, doi:10.4271/2015-01-0855.

INTRODUCTION

In an effort to reduce NO_x and soot emissions in-cylinder from compression ignition engines, while improving thermal efficiency to meet challenging fuel economy regulations, many new compression ignition combustion strategies have been proposed. These combustion strategies are known as low temperature combustion (LTC) and produce simultaneous reductions in NO_x and soot emissions compared with conventional diesel combustion (CDC). Simply speaking, the goal of the LTC process is to achieve sufficient premixing between the fuel and air to avoid soot formation and to maintain low combustion temperatures to avoid NO_x formation, which is achieved through premixing and dilution with either air or exhaust gas recirculation (EGR).

Dual-fuel Reactivity Controlled Compression Ignition (RCCI) is a form of LTC that has been shown to address a number of issues that have plagued other LTC concepts. Two fuels of different reactivity (i.e., different octane or cetane number) are used by RCCI as described by

Kokjohn et al. [1]. In this strategy, a low reactivity fuel, such as gasoline, is premixed via port-fuel injection (PFI), and a high reactivity fuel, such as diesel fuel, is direct-injected (DI) midway through the compression stroke. Fuel reactivity gradients in the combustion chamber are developed by the process of in-cylinder fuel blending [2].

The stratification of fuel reactivity results in an elongated combustion event and reduced pressure rise rates compared with other premixed compression ignition strategies [3]. In addition, in-cylinder fuel blending of two fuels with different reactivities, allows the global fuel reactivity to be varied on a cycle-to-cycle basis. This allows a wide range of loads to be achieved by simply varying the ratio of the two fuels, thereby addressing one of the major challenges with single-fuel premixed combustion. A wide range of engine loads on a heavy-duty engine was demonstrated by Kokjohn et al. by varying the gasoline-to-diesel ratio while keeping the diesel injection strategy fixed. [4]. Additionally, the diesel fuel is injected early enough during the compression stroke and in a small enough quantity that NO_x and soot emissions remain low. Although RCCI operation has been shown

to produce significant reductions in NO_x and soot emissions, unburned hydrocarbons (UHC) and carbon monoxide (CO) emissions tend to increase compared with CDC.

Gasoline and diesel fuels have been used by the majority of RCCI research conducted to date. There are practical challenges of a combustion strategy that requires the use of two separate base fuels (i.e., two fuel tanks). Vehicle systems analysis has shown that the fueling split over federal regulatory drive cycles is approximately 60% gasoline and 40% diesel [5]. However, RCCI combustion requires only that a fuel reactivity gradient be spatially present in the combustion chamber. Therefore, it has been proposed to use port-injected low reactivity fuel and to direct inject the same low reactivity fuel mixed with a small percentage of a cetane improver (i.e., ignition improver) to serve as the high reactivity fuel. With this approach, the vehicle would require a single primary fuel tank (i.e., with low reactivity fuel, such as gasoline) and an additive tank, which would be significantly smaller in size. This is illustrated in [Figure 1a](#), in which pure fuel (i.e., gasoline) and pure cetane improver (i.e., EHN) would be stored in separate tanks and a blending mechanism would mix them to a fixed additive concentration for the DI fuel stream. To date, the practicality of such a technique in a production viable system has not been explored. This is simply the proposed system configuration. For simplicity, the current study used two separate tanks in which the gasoline and EHN are mixed in the DI tank, as shown in [Figure 1b](#).

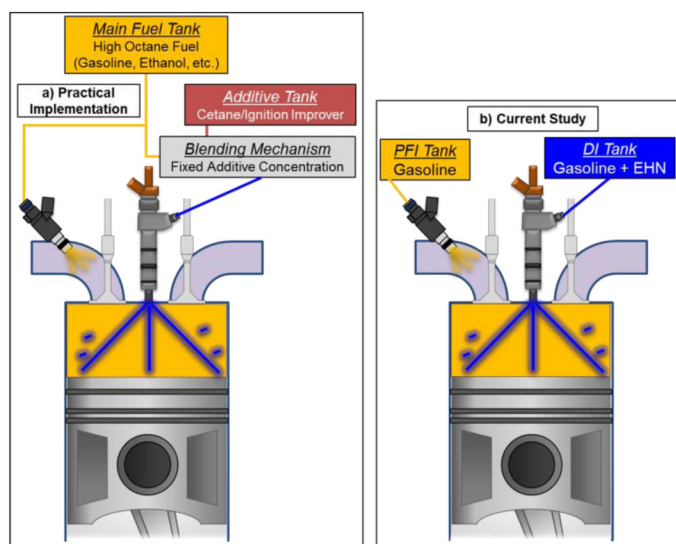


Figure 1. Fuel injector layout for single-fuel RCCI operation with port-injected low reactivity fuel (i.e., gasoline) and direct-injected low reactivity fuel (i.e., gasoline) mixed with a cetane improver (i.e., EHN). a) System configuration for a practical application such as a vehicle. b) System configuration for the current study with gasoline and EHN to allow various levels of cetane improver in the DI fuel to be easily investigated.

As an example, if the cetane improver were added to the direct-injected base fuel at a fixed rate of 5%, and the vehicle uses, on average, 60% port-injected base fuel and 40% direct-injected fuel, the rate of cetane improver consumption would potentially be only 2% of the total fuel. Thus, the additive tank would need to be replenished far less frequently than the base fuel tank and perhaps lessen the practical burden of RCCI combustion in passenger car applications.

Splitter et al. were among the first to investigate the aforementioned concept on a heavy-duty single-cylinder engine, in which they port-injected gasoline and direct-injected gasoline mixed with small amounts of di-tert-butyl peroxide (DTBP), a commonly used additive to improve the cetane number of diesel fuel [6]. RCCI combustion was found to be achievable by mixing as little as 2% by volume DTBP into the direct-injected gasoline.

Similar research on a light-duty single-cylinder engine was conducted by Kaddatz et al. [7]. They used E10 (90% gasoline/10% ethanol) mixed with 3% by volume 2-ethylhexyl nitrate (EHN) as the high reactivity fuel for RCCI combustion. Comparisons with gasoline/diesel RCCI were made, and it was found that compared with diesel fuel more E10+EHN was required to achieve similar combustion phasing. This indicates that diesel fuel is more reactive than E10 mixed with 3% by volume EHN. It was also shown by Kaddatz et al. [7] that for RCCI operation with gasoline/diesel, if the direct injection timings were sufficiently advanced, the NO_x emissions went to essentially zero (i.e., zero thermal NO production). However, as the direct injection timings were advanced using E10+EHN as the high reactivity fuel, the NO_x emissions converged to approximately one-third of the fuel bound NO_x from the EHN. Similar NO_x increases from EHN under LTC operating conditions have been observed for conditions in which the thermal NO production was essentially zero with non-nitrogen containing fuels [8], [9], [10]. This potential disadvantage of using EHN as a reactivity enhancer is discussed in the present study.

DESCRIPTION OF CURRENT STUDY

Cetane Improver Selection-EHN

In an effort to characterize the effects of commercial cetane improvers on high octane fuels, fully premixed and dilute homogeneous charge compression ignition (HCCI) experiments with gasoline, ethanol, and methanol were used by Dempsey et al. [8]. The effective primary reference fuel (PRF¹) number (i.e., octane number) of gasoline as a function of the mass percentage of cetane improver additive is shown in [Figure 2](#). The effective PRF number in the study by Dempsey et al. was determined by comparison-to-combustion phasing maps constructed using the PRF fuels, iso-octane and n-heptane [8]. As shown in the figure, as the additive amount is increased, the effective PRF number is decreased, indicating an increase in the base fuel reactivity. Under dilute conditions, it was shown by Dempsey et al. that EHN is a more effective ignition improver compared with DTBP for a conventional gasoline fuel [8]. Thus, to achieve a given fuel reactivity, less cetane improving additive will be required. Based on these findings, EHN was chosen as the ignition improving additive for the current study.

The EHN used in this study had a purity of 97% and was purchased from the Sigma-Aldrich Corporation. The properties of EHN are shown in [Table 2](#). The chemical formula of EHN is C₈H₁₇NO₃ and the molecular structure is shown in [Figure 3](#). The basic structure of EHN is that of an ethylhexane molecule with one of the hydrogen atoms replaced with an NO₃ nitrate functional group.

1. The PRF number is the percentage of iso-octane in a binary fuel mixture of iso-octane and n-heptane.

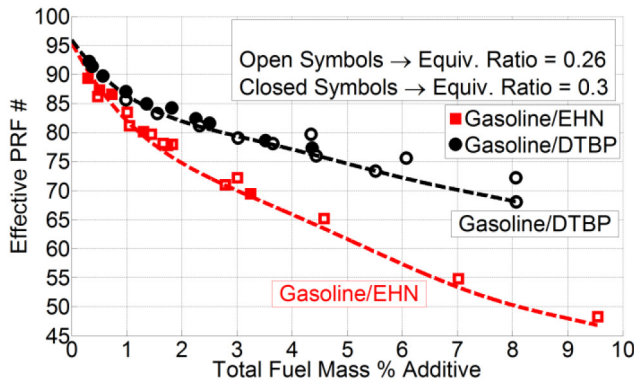


Figure 2. Impact of EHN and DTBP on gasoline fuel reactivity during heavily diluted HCCI operation [8].

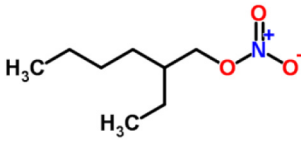
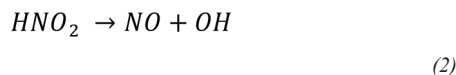
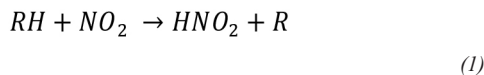


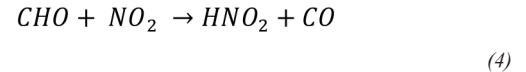
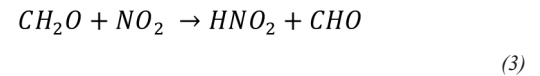
Figure 3. Molecular structure of 2-Ethylhexyl Nitrate (EHN).

EHN is the most commonly used cetane improver on the market today. In commercial diesel fuels, EHN is known to be added in concentrations as high as 0.4%, which has been shown to improve the cetane number by up to ~10 points, depending on the base fuel [11]. Additionally, the thermal decomposition threshold temperature is increased to more than 150°C when mixed in a fuel solution at high pressure such that premature decomposition is not expected to take place in the fuel system [12].

When EHN is injected into the cylinder, it undergoes rapid thermal decomposition into NO_2 , formaldehyde (CH_2O), and other less reactive radicals, including the heptyl radical ($\text{C}_7\text{H}_{15}\cdot$) [12]. The low temperature reactions and hence the reactivity of the base fuel is enhanced by EHN mainly through the production of hydroxyl radicals (OH) from CH_2O and NO_2 . The following reaction pathways that show this effect have been identified by Clothier et al. [13], [14] and Das Adhikary [15]. (Note that in these reactions RH represents the base fuel.) Hydrogen from the base fuel is abstracted by NO_2 produced from the EHN thermal decomposition to create HNO_2 . The HNO_2 subsequently forms NO and OH .



Ignition of the base fuel is accelerated by OH reacting with the base fuel and other fuel fragments to produce heat and buildup aldehydes and H_2O_2 . Ignition delay is thus reduced by shortening the time it takes to reach the critical temperature for H_2O_2 decomposition and thus autoignition [16], [17]. The NO is recycled to NO_2 to restart the sequence of producing more OH radicals [13]. Another efficient route to the production of HNO_2 and thus OH is through the formaldehyde and NO_2 created by the thermal decomposition of EHN.



Naturally, because of the nitrate content of EHN, increased NO_x emissions can be a concern. For CDC however, it has been shown by Kaddatz et al. [7] and Schwab et al. [18] that NO_x emissions do not increase because of the addition of EHN, even at large doping levels of 7.5% EHN by volume in diesel fuel. This is thought to be because the NO_2 produced by the decomposition of EHN is in a hydrocarbon-rich environment during CDC. Thus, the NO_2 can participate in hydrocarbon oxidation and be reduced to N_2 and O_2 , much like it does across a three-way catalyst.

On the contrary, it has been shown that for lean, low temperature combustion conditions EHN yields an increase in engine-out NO_x emissions [7], [10]. This is thought to be because during low temperature combustion the injection events are early and sufficiently separated from the autoignition event such that there are no rich mixtures at the start of combustion. Thus, the dominant oxidant for the hydrocarbons will be oxygen, and engine-out NO_x emissions will ultimately be created by some of the NO and NO_2 formed from EHN decomposition.

Objectives of the Current Study

Dual-fuel RCCI operation has been shown to provide excellent authority over the timing of combustion, which is something that has plagued other low temperature combustion concepts [3]. The focus of the current study was to assess the combustion phasing controllability as a function of the EHN level in the direct-injected fuel for the proposed single-fuel RCCI concept. A light-duty, multi-cylinder engine was operated on port-injected gasoline and direct-injected gasoline mixed with various levels of EHN. For comparison, results with port-injected gasoline and direct-injected diesel are also presented. For operation with gasoline/EHN mixtures, only the NO emissions are presented, as discussed in the next section and Appendix 1. However, for gasoline/diesel operation, the total NO_x emissions are presented. These differences are clearly labeled in each figure in the subsequent sections.

NO_2 Emissions Measurements

In this study exhaust NO_x emissions ($\text{NO} + \text{NO}_2$) were measured with a heated chemiluminescence detector (HCLD), which is the typical measurement principle for combustion engine research. In the reaction chamber of the detector, nitric oxide (NO) reacts with ozone (O_3), and chemiluminescence of an intermediate product occurs. The number of photons produced is proportional to the concentration of NO in the sample.

To measure NO_2 as well, and thus the total NO_x concentration, the sample must first be passed through a converter, which typically contains a carbon-based molybdenum catalyst. Here the NO_2 is converted to NO , and thus total NO_x is measured in the reaction chamber. It was discovered in the experimental campaign that the

catalyst material had been poisoned, rendering the NO₂ converter completely ineffective. This was confirmed by sending a known quantity of NO₂ through the analyzer and measuring the total NOx emissions, which were found to be zero. It is thought that exposing the catalyst material to high levels of hydrocarbon emissions from low temperature combustion led to its poisoning. Thus, the present measurements for gasoline/EHN RCCI operation are focused primarily on just the NO emissions.

However, a Fourier Transform Infrared (FTIR) analyzer from MKS was used in conjunction with standard emissions analyzers to take measurements of gasoline/EHN mixtures. These results are shown in [Appendix 1](#). The HCLD measurements in total NOx mode agree very well with the FTIR measurements of NO in the exhaust over a range of conditions. Again, this also confirms that the NO₂ converter was ineffective and only NO was being measured by the HCLD.

As shown in [Appendix 1](#), there is a significant amount of NO₂ present in the exhaust from RCCI operation with gasoline/EHN mixtures. However, as the combustion phasing changes, the NO emissions change to a much greater extent than the NO₂ emissions. Thus, measuring only NO is a reasonable indicator of the total NOx emissions trends as conditions change. However, it is important to keep in mind throughout this work that a non-negligible fraction of the total NOx emissions was not measured with the HCLD for the gasoline/EHN mixtures.

Calculation of Fuel Bound NOx Emissions

It is of interest to calculate the concentration of NOx emissions in the exhaust that could have potentially stemmed directly from the EHN molecule. Note that these calculations neglect any fuel bound nitrogen in the base gasoline fuel, which, as shown in [Table 2](#), is an acceptable assumption for the fuel used in this study. First, the molar flow rate of exhaust will be required ($\dot{n}_{exh}$), which is given as

$$\dot{n}_{exh} = \frac{\dot{m}_{exh}}{MW_{exh}}, \quad (5)$$

where $\dot{m}_{exh}$ is the mass flow rate of exhaust, which is assumed to be the sum of the fresh air flow rate and the total fuel flow rate, and MW_{exh} is the molecular weight of exhaust. The molar flowrate of EHN in the fuel is

$$\dot{n}_{EHN_{fuel}} = \frac{\dot{m}_{EHN_{fuel}}}{MW_{EHN}}, \quad (6)$$

where $\dot{m}_{EHN_{fuel}}$ is the mass flow rate of EHN through the DI injector and MW_{EHN} is the molecular weight of EHN, which is 175.23 g/mole. Because of the molecular structure of EHN, which is shown in [Figure 3](#), the molar flow rate of nitrogen from EHN is equal to the molar flow rate of EHN. Thus, the maximum mole fraction of NOx in the exhaust stream stemming directly from EHN is

$$X_{fuelNOx} = \frac{\dot{n}_{EHN}}{\dot{n}_{exh}}. \quad (7)$$

This quantity will be referred to in this work as fuel bound NOx emissions. In molar form it is not possible to distinguish between NO and NO₂ stemming from the EHN molecule. To convert this molar quantity to a mass based, fuel bound NOx an assumption would need to be made about its composition because a molecular weight is required. Throughout this investigation, the fuel bound NOx is presented as a molar concentration for simplicity and to avoid the need for an assumed molecular weight.

EXPERIMENTAL ENGINE AND LABORATORY SETUP

Multi-Cylinder Light-Duty Compression Ignition Engine

The engine used for this study was a modified model year 2007 General Motors (GM) direct injection 1.9 L ZDTH diesel engine. The geometric specifications of the engine are shown in [Table 1](#).

Table 1. Geometric specifications of a model year 2007 GM 1.9 L compression ignition engine.

Number of Cylinders	4
Number of Valves per Cylinder	4
Bore [mm]	82.0
Stroke [mm]	90.4
Connecting Rod Length [mm]	145.4
Modified Compression Ratio [-]	15.1
Total Displacement [L]	1.9

The original equipment manufacturer (OEM) pistons were replaced with pistons that are more suited for premixed combustion concepts. The modified piston features a wide/shallow bowl geometry, which creates a more quiescent combustion chamber and reduces the piston crown surface area compared with the OEM reentrant bowl geometry [19]. Using single-cylinder engine experiments, the modified piston bowl geometry yielded an increase in combustion efficiency and indicated efficiency for RCCI operation compared with the OEM piston using a variety of fuels [20]. In addition, a reduced compression ratio (15.1 down from 17.1) is used by the modified piston to allow for higher load operation during premixed combustion (e.g., reduced peak cylinder pressures and pressure rise rates). A schematic drawing and photograph of the stock OEM piston and the modified piston bowl geometries are shown in [Figure 4](#).

The direct injection fuel system was unmodified from the production form. High-pressure fuel was delivered to a common rail by an engine-driven Bosch CPH pump. The direct injectors were Bosch CR12.2 solenoid-driven common rail injectors. The nozzle tips had seven holes with a nominal diameter of 0.14 mm and an included angle of 148 degrees. The engine is equipped with the stock turbocharger (VGT), but a aftermarket charge air cooler was

installed to control the intake temperature. The intake manifold was modified to incorporate extended-tip, narrow-spray-angle PFIs, which deliver the low reactivity gasoline for RCCI operation. The overall engine and fuel system layout for RCCI operation is shown in Figure 5.



Figure 4. Stock OEM and modified piston bowl geometries for the multi-cylinder GM 1.9 L engine. The modified piston bowl was used for the present research.

The stock engine control unit was replaced with a full-pass control system from National Instruments-Powertrain Controls (formerly Driven, Inc.). This allowed simultaneous control of the PFI and DI fuel systems, including cylinder-to-cylinder balancing, as well as all other relevant engine parameters, such as rail pressure, VGT position, EGR valve position, and swirl vane position.

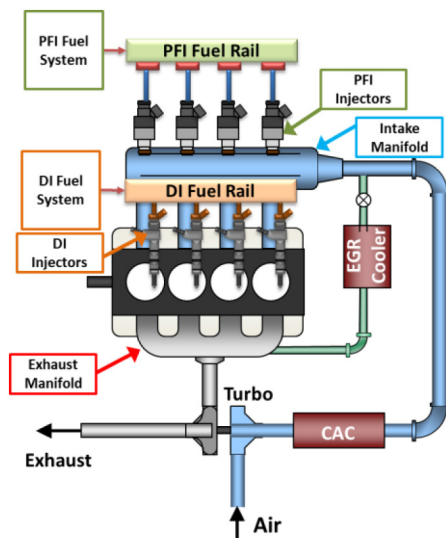


Figure 5. Intake, exhaust, and fuel system schematic for the GM 1.9 L engine.

Data Acquisition and Analysis

For RCCI operation, the DI fuel flow rate was measured with a Max Machinery 710-213 positive-displacement volumetric flow measurement system, while the PFI fuel flow rate was measured using a Micro Motion Coriolis fuel flowmeter. The intake fresh air flow rate was measured using a laminar flow element from Meriam Process Technologies (model #50MC2-4F). Conditioned air was supplied to the engine at a constant temperature of 25°C and a relative humidity of 58%.

High-speed, in-cylinder pressure data were acquired using Kistler model 6058A pressure transducers installed in glow plug adapters of all four cylinders. Individual Kistler Type 5010 dual-mode amplifiers were used to amplify the signals, and software from National Instruments-Powertrain Controls group was used to acquire and process the cylinder pressure data. Cylinder pressure was pegged to the intake manifold pressure at bottom dead center of the intake stroke and was sampled at a resolution of 0.2 crank angle degrees (CA). For each operating point, 300 consecutive cycles of cylinder pressure data were acquired. The individual cycle's cylinder pressure data were not digitally filtered in any way. The collected cycles were ensemble averaged to yield a representative cylinder pressure trace.

Combustion analysis was conducted using National Instruments-Powertrain Controls software known as the Driven Combustion Analysis Toolkit (DCAT). Each individual cycle's apparent heat release rate (AHRR) was calculated and subsequently ensemble averaged. The AHRR is essentially the chemical heat release rate minus the heat loss rate to the combustion chamber walls and is given by

$$AHRR = \frac{dQ}{d\Theta} \Big|_{chem} - \frac{dQ}{d\Theta} \Big|_{wall} = \frac{1}{\gamma - 1} V \frac{dP}{d\Theta} + \frac{\gamma}{\gamma - 1} P \frac{dV}{d\Theta}, \quad (8)$$

where P is the average in-cylinder pressure, V is the cylinder volume, and γ is the ratio of specific heats of the working fluid. In DCAT, γ is approximated as the average polytropic coefficient of the compression and expansion processes for each individual cycle.

From the AHRR, the accumulated heat release can be determined. The start of combustion was calculated to be the crank angle at which the accumulated heat release had reached 10% of its maximum, denoted as CA10. The combustion phasing is the crank angle of 50% accumulated heat release and is denoted as CA50.

In a multi-cylinder engine there are variations in conditions from cylinder to cylinder, such as wall temperatures, EGR level, trapped residuals, total fuel delivered, etc. In addition, each fuel injector is unique and will yield a slightly different fuel flow rate for a given command duration. Kinetically controlled combustion strategies like RCCI are especially sensitive to changes in these types of parameters. This can lead to significant cylinder-to-cylinder imbalances in the combustion phasing (CA50), indicated mean effective pressure (IMEP²), peak pressure rise rate (PPRR), and emissions. To adjust for these variations, the total fueling and ratio of gasoline to diesel fuel was varied independently for each cylinder. Typically, the PFI and DI injection durations needed to be varied by $\pm 10\%$ from cylinder to cylinder. Throughout this study the IMEP and CA50 from each cylinder was within 5% and 1.0 crank angle of the nominal value for the engine, respectively.

Exhaust emissions were measured using standard gaseous emissions analyzers. A heated flame ionization detector (FID) was used to measure total unburned hydrocarbons (UHC). A heated

2. Indicated means the cycle work was calculated from the full cycle of in-cylinder engine friction. Mean effective pressure is equal to the cycle work divided by the engine displacement volume.

chemiluminescence analyzer (HCLD) was used to measure total NO_x emissions (NO + NO₂). However, as mentioned previously, the NO₂ converter was not functioning during this study. Thus, only NO emissions were being measured. Appendix 1 shows that the HCLD measurement in total NO_x mode agrees very well with the NO emissions from an MKS FTIR.

Both CO and CO₂ were measured using non-dispersive infrared (NDIR) instruments. Intake and exhaust O₂ were measured using a paramagnetic detector (PMD). Both the intake and exhaust sample

streams were conveyed from heated filters to the instruments through heated lines maintained at 190°C. An AVL 415S smoke meter was used to measure the filter smoke number (FSN), which is an indicator of black carbon containing soot in the exhaust. Throughout this study the FSN was 0.1 or less and will not be reported further. These FSN results are similar to what has been observed in previous multi-cylinder RCCI experiments with conventional fuels [21]. The low-speed data (airflow, fuel flow, boundary conditions, and emissions) were recorded for 180 seconds at ~2 Hz. The average results over the 180 seconds are presented.

Table 2. Fuel properties for the various compounds and mixtures used in the current study. All gasoline/EHN mixtures were additized with 350 ppm by volume of Infineum R655, a commercial lubricity additive, for use in the common rail direct injection system.

	<i>Diesel</i>	<i>Gasoline</i>	<i>2-Ethylhexyl Nitrate</i>	<i>Gasoline + 2.5% EHN</i>	<i>Gasoline + 5% EHN</i>	<i>Gasoline + 10% EHN</i>
Designation in Body Text	Diesel	Gasoline	EHN	2.5% EHN	5% EHN	10% EHN
Product Code	Haltermann ULSD HF-00582	Haltermann EEE HF-00437	Sigma-Aldrich 293784	-	-	-
Chemical Formula	-	-	C ₈ H ₁₇ NO ₃	-	-	-
Composition						
~Volume % EHN-in-Gasoline (Splash Blended)*	-	-	-	2.5%	5%	10%
~Mass % EHN-in-Gasoline (Splash Blended)*	-	-	-	3.21%	6.38%	12.57%
Aromatics [vol. %]**	28.0%	30.3%	-	-	-	-
Olefins [vol. %]**	1.0%	0.5%	-	-	-	-
Saturates [vol. %]**	71.0%	69.2%	-	-	-	-
Carbon [mass %]**	85.78%	87.05%	54.86%	85.79%	84.89%	82.89%
Hydrogen [mass %]**	13.42%	13.57%	9.71%	13.32%	13.23%	12.99%
Nitrogen [mass ppm] **	-	7.3	80,000	2747	5463	10,576
Volume % EHN-in-Gasoline (Fuel Analysis)***	-	-	-	2.67%	5.36%	10.53%
Mass % EHN-in-Gasoline (Fuel Analysis)***	-	-	-	3.43%	6.83%	13.22%
Reactivity and Energy						
Lower Heating Value [MJ/kg]**	42.8	43.0	27.4	42.5	42.0	41.0
Research Octane Number [-]**	< 40	96.6	-	64.3	< 40	< 40
Motor Octane Number [-]**	< 40	88.6	-	64.7	46.6	< 40
Cetane Number [-]**	44.2	< 19	-	25.4	33.0	62.6
Distillation						
Heat of Vaporization [KJ/kg]	~270	~300				
IBP [C]**	179	30	-	31	29	31
T10 [C]**	214	53	-	53	52	53
T50 [C]**	259	105	-	106	107	109
T90 [C]**	317	158	-	167	173	179

*Targeted values for volume percentage of EHN-in-gasoline that was splash blended on-site at Oak Ridge National Laboratory.

**Experimental results of fuel analysis conducted at Southwest Research Institute's Fuel Laboratory according to ASTM specifications.

***Experimental amount of EHN-in-gasoline that was calculated from experimentally determined nitrogen content in the fuel mixture.