

Effects of Biofuel Blends on RCCI Combustion in a Light-Duty, Multi-Cylinder Diesel Engine

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ABSTRACT

Reactivity Controlled Compression Ignition (RCCI) is an engine combustion strategy that utilizes in-cylinder fuel blending to produce low NO_x and PM emissions while maintaining high thermal efficiency. Previous RCCI research has been investigated in single-cylinder heavy-duty engines [1, 2, 3, 4, 5, 6]. The current study investigates RCCI operation in a light-duty multi-cylinder engine over a wide number of operating points representing vehicle operation over the US EPA FTP test. Similarly, previous RCCI engine experiments have used petroleum based fuels such as ultra-low sulfur diesel fuel (ULSD) and gasoline, with some work done using high percentages of biofuels, namely E85 [7]. The current study was conducted to examine RCCI performance with moderate biofuel blends, such as E20 and B20, as compared to conventional gasoline and ULSD. The engine experiments consisted of in-cylinder fuel blending using port fuel-injection (PFI) of gasoline or E20 and early-cycle, direct-injection (DI) of ultra-low sulfur diesel (ULSD) or B20 fuel. At the selected load points, the results from RCCI combustion using biofuels and petroleum fuels are compared. Preliminary results show that with E20, the peak load was able to be raised from 8 to 10 bar BMEP, due to ethanol's lower propensity for autoignition. Ethanol's low reactivity enabled an increased reactivity gradient and reduced the pressure rise rate (PRR) compared to gasoline. This increase in peak load also allowed for a 5% relative increase in brake thermal efficiency (BTE). Replacing ULSD with B20 was shown to increase combustion efficiency at low loads. The unique fuel properties of these biofuel blends increases the benefits of the use of RCCI combustion compared to conventional diesel combustion (CDC).

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INTRODUCTION

Rising fuel costs and a focus on the reduction of greenhouse gases has driven the need for increased efficiency from the internal combustion engine. This need for increased efficiency has placed increased attention on the diesel engine due to its higher efficiency compared to other engines. Furthermore, current and future legislation place stringent standards on the exhaust emissions and vehicle fuel efficiency. These demands require the development of advanced engine technologies, which can simultaneously yield high efficiency and low emissions. Many combustion strategies have been investigated to meet the future requirements of clean, efficient diesel engines. Most of the current strategies fall into the category of low temperature combustion (LTC). LTC is desired because low combustion temperatures result in NO_x reductions, due to the high

activation energy of NO formation reactions [8]. In addition, low temperature combustion strategies typically utilize injection timings that allow for increased air/fuel mixing prior to the start of combustion; thus, rich regions are reduced and soot formation is inhibited.

Many researchers have shown that LTC strategies such as Homogeneous Charge Compression Ignition (HCCI) and Premixed Charge Compression Ignition (PCCI) are promising techniques for simultaneous NO_x and soot reduction [9, 10, 11, 12, 13, 14]. Due to the existing fuel infrastructure, most HCCI and PCCI research has been conducted using either gasoline or diesel fuel. However, in their neat forms, each fuel has specific advantages and shortcomings for LTC. For instance, gasoline has a high volatility; thus, evaporation is rapid and a premixed charge can be obtained using port fuel-injection. However, because gasoline resists auto-ignition, it becomes difficult to achieve combustion at low-load

conditions [15,16]. Conversely, diesel fuel has superior auto-ignition qualities; however, this can result in difficulty controlling the combustion phasing as engine load is increased.

To find the optimum fuel for HCCI over a wide engine speed/load range, experiments were performed by Bessonette et al. [17]. They suggested that the best fuel for HCCI operation may have auto-ignition qualities between those of diesel fuel and gasoline, depending on the engine speed and load. Based on these results and experiments by Inagaki et al. [18], the RCCI strategy was developed to allow the fuel reactivity to be optimized at all speeds and loads, providing an increased engine operating range for premixed LTC. Additional information on RCCI can be found in the References [19, 20, 21, 22, 23].

EXPERIMENTAL SETUP

The engine experiments were performed at Oak Ridge National Laboratory (ORNL) using a production 2007 model year 1.9L 4 cylinder light-duty diesel engine. The engine utilized all stock hardware, except for the high pressure EGR heat exchanger, gasoline PFI system and optimized pistons for RCCI operation. Detailed descriptions of the modified pistons can be found in [24]. Also of note is that the OEM ECU was replaced by a Drivven engine controller to allow full access for control of all engine parameters and sub systems, including the additional PFI system. A schematic of the lab setup is shown in [Figure 1](#) and the engine geometry is summarized in [Table 1](#).

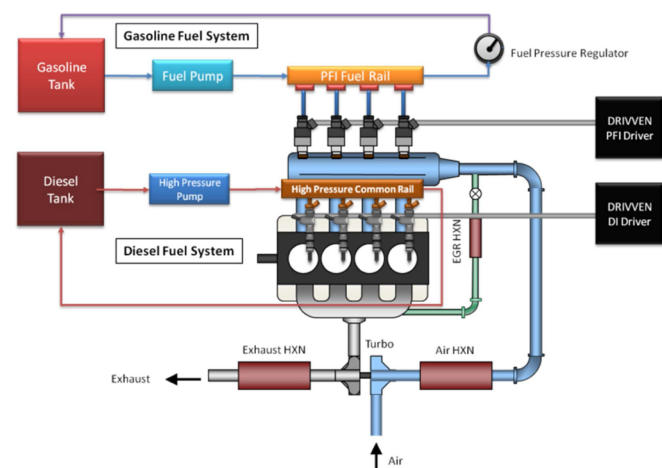


Figure 1. Diagram of the engine lab setup

Table 1. GM 1.9L engine geometry

Displacement (L)	1.9
Number of Cylinders	4
OEM Compression Ratio (-)	17.5:1
Modified Compression Ratio (-)	15.1:1
Bore x Stroke (mm)	82.0 x 90.4
Valve Timings (°ATDC firing, measured at 0.2 mm valve lift)	EVO = 128 EVC = 356 IVO = 361 IVC = 568

EMISSIONS AND PRESSURE MEASUREMENTS

The exhaust gas emissions measurements were taken with a conventional 5 gas bench, measuring intake O₂ and CO₂, exhaust O₂ and CO₂, HC, CO, and NO_x. Emissions measurements were taken after the engine had reached steady state for 30 seconds and were then averaged for 180 seconds. HC emissions were measured on a C1 basis and were uncorrected for response to partial combustion products in the Flame Ionization Detector. Smoke readings were taken with an AVL 415s Smoke Meter with an average of 5 measurements. High speed combustion data were acquired using Kistler model 6058A pressure sensors installed in the glowplug ports of all 4 cylinders. Individual Kistler type 5010 Dual-Mode Amplifiers were used to process the pressure signals. The built-in combustion package from Drivven and a custom MATLAB code were used to post process the data. Cylinder pressure measurements were taken for 300 cycles with the cycle average of each individual cylinder then averaged over all 4 cylinders for all the pressure based calculations.

Because there is potential for thermal and compositional gradients between cylinders in a multi-cylinder engine (MCE), there was a need to adjust both the DI and PFI fueling in each cylinder to maintain constant IMEP and combustion phasing, such that the BMEP and emissions are optimal. To accomplish this, the fuel injection ratio of gasoline and diesel was adjusted in each cylinder to equalize CA50 for each data point. The total fueling amount of gasoline and diesel fuel in each cylinder was then adjusted such that the IMEP across all 4 cylinders were the same. The COV of IMEP was found to vary less than 1% absolute across all 4 cylinders and the COV of MPRR was similarly found to vary less than 1% across all four cylinders after this balancing process was performed. Additional details on this process can be found in [24].

FUEL SYSTEMS

The RCCI engine experiments were performed using port fuel-injection of gasoline, E20 or E85 and direct-injection of ULSD or B20. The OEM common-rail injectors were used

for the DI fuel system and conventional spark-ignition engine gasoline injectors were used for the additional PFI system. The specifications for both injectors are given in [Table 2](#) and [Table 3](#), respectively.

Table 2. Port fuel injector specifications

Fuel Pressure (bar)	3.8
Included Spray Angle	15°
Number of Holes	2
Steady Flowrate (g/s)	2.74

Table 3. Common rail injector specifications

Maximum Injection Pressure (bar)	1,500
Included Spray Angle	143°
Number of Holes	7
Steady Flowrate (cc/30 sec)	440

The gasoline used in the study was a certification grade fuel (UTG 96) with properties shown in [Table 4](#). Similarly, the diesel fuel chosen was also a certification grade ULSD fuel, shown in [Table 5](#).

Table 4. UTG 96 fuel specifications

Initial Boiling Point (°C)	33.9
Temperature 10% evaporated (°C)	50
Temperature 50% evaporated (°C)	107.2
Temperature 90% evaporated (°C)	150
Final Boiling Point (°C)	162.8
Lower Heating Value (MJ/kg)	43.12
MON	88.2
RON	96.1
Anti-Knock Index ((RON+MON)/2)	92.15
Ethanol (%)	0
Specific Gravity (@ 15.6 °C)	0.739
Sulfur (ASTM D5453) (PPM)	29.6

Table 5. ULSD fuel specifications

Initial Boiling Point (°C)	38.9
Temperature 10% evaporated (°C)	69.4
Temperature 50% evaporated (°C)	105
Temperature 90% evaporated (°C)	160.6
Final Boiling Point (°C)	215.6
Lower Heating Value (MJ/kg)	42.91
Specific Gravity (@ 15.6 °C)	0.845
Cetane Number (-)	46
Sulfur (ASTM D5453) (PPM)	9.9

To make the B20 blend, 100% soy methyl ester (SME) was splash blended on site with the same ULSD at 20% by volume. Specifications for the B20 fuel are mentioned in [Table 6](#).

Table 6. B20 fuel specifications

Lower Heating Value (MJ/kg)	41.710
Specific Gravity (@ 15.6 °C)	0.8518
Cetane Number (-)	46

The E20 used in the study was supplied premixed by an outside fuel supplier. Specifications are shown in [Table 7](#).

Table 7. E20 fuel specifications

Lower Heating Value (MJ/kg)	39.642
Ethanol (%)	19.8
Specific Gravity (@ 15.6 °C)	0.755
RON [-] ¹	101.2
MON [-]	90.5
AKI (RON+MON)/2 [-]	95.9

E20 EXPERIMENTAL METHODOLOGY

Engines typically operate over a wide speed and load range in vehicles, so it was of interest to investigate the widest operational range for RCCI combustion as possible. [Figure 2](#) shows E20/diesel operation with speed and load points covered by a light-duty diesel engine over the FTP75 test. The test vehicle used for comparison was a 2007 model year Opel Astra with the same 1.9L engine and a manual transmission. These results were taken at Argonne National Laboratory and shared with the authors.

To make meaningful comparisons of the impact of using E20 in place of gasoline, the present operating map results (as seen in [Figure 2](#) and [Figure 3](#)) were compared to a similar engine mapping exercise, using gasoline and diesel fuels, performed at ORNL on the same engine as the E20 results were taken. Since including both sets of results would result in an extraordinarily large paper, they were split to two papers. Results from the gasoline/diesel study can be seen in [\[25\]](#). To show the effects of using E20, the present results are compared to those from [\[25\]](#) and depicted as contour plots of the difference between the two data sets. Contour plots are ideal for this exercise as they can show the general trends while accounting for some of the minor differences between operating conditions of both data sets. Since high load performance with E20 was the primary focus for the work,

¹The RON and MON were not directly measured for this exact E20 blend, but these RON and MON data are representative of other E20 blends used in the same Dept. of Energy Intermediate Ethanol Blend fuel study.

loads below 3 bar BMEP were not extensively studied but were investigated in more detail in [25].

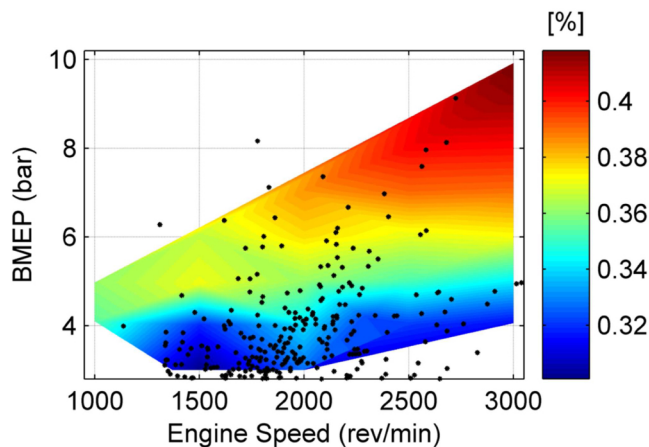


Figure 2. Overall E20/Diesel operating map with CDC light-duty FTP points overlaid

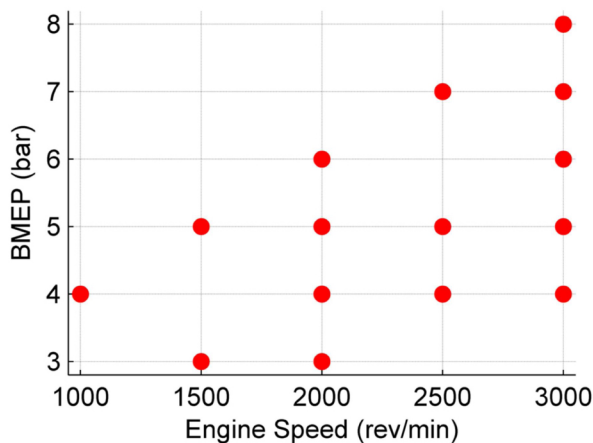


Figure 3. Specific engine operating points used in the E20/D to G/D comparison tests

For the current study, the engine was operated in RCCI mode using E20 as the low cetane number (CN) fuel and ULSD as the high CN fuel. E20 was chosen as the low CN fuel as it supplies a higher resistance to auto-ignition than gasoline and is a possible near-term ethanol blend. Constant engine operating conditions are shown in Table 8. All other parameters (i.e., intake pressure, SOI timing, premixed fuel ratio, etc.) were changed for each data point to reach the optimum emissions and thermal efficiency while being mandated under self-imposed operating limits.

Table 8. E20 Engine operating parameter constants

Coolant Temperature (°C)	85
Intake Temperature (°C)	42 +/- 2
Common Rail Pressure (bar)	500
EGR (-)	0

The self-imposed operating limits were set to keep the maximum pressure rise rate (MPRR) below 10 bar/°CA., CO emissions below 5,000 ppm, HC emissions below 4,000 ppm and maintain acceptable combustion stability. Combustion stability was defined as keeping the COV of MPRR below 20% and the COV of IMEP below 5%.

Generally, for the overall calibration, the intake pressure was set to the highest level that could be achieved with the lowest penalty in pumping work. As shown in Figure 4, the intake pressure varies from 1 bar at low speeds and loads to 1.8 bar at high speeds and loads. The main limitation to the intake pressure was the stock turbocharger, as it is less efficient with the lower exhaust temperatures seen in RCCI.

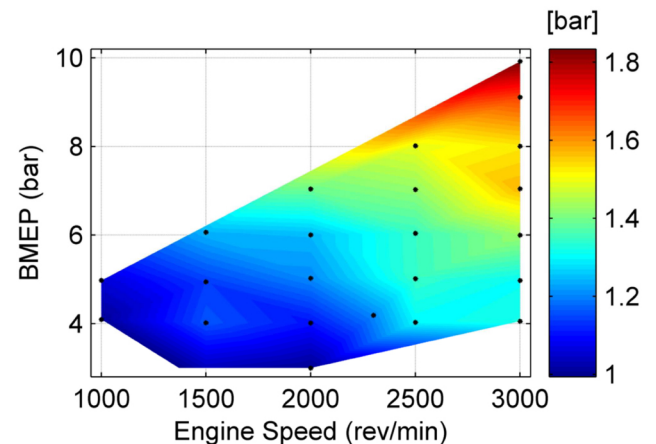


Figure 4. E20/D Intake pressure over the full E20/D experimental space

The premixed gasoline ratio was typically set to the highest value that could be achieved with the highest combustion stability and lowest HC emissions. This resulted in the use of 40% PFI fraction at low loads increasing to nearly 90% at high speeds and loads, as shown in Figure 5.

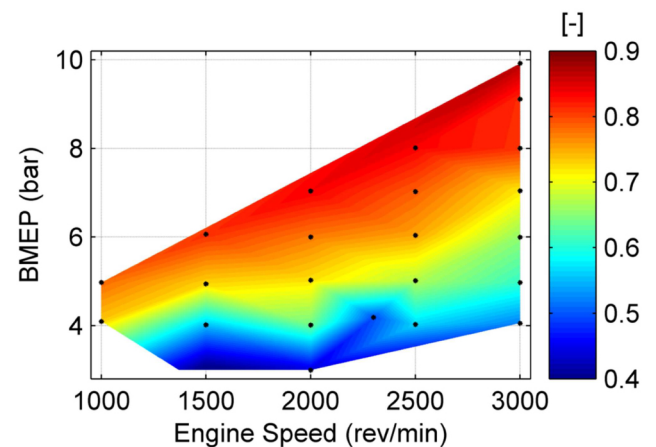


Figure 5. PFI fraction over the full E20/D experimental space

The DI injection strategy varied between a single injection at low engine speeds and loads to a double injection

strategy, characterized by approximately 60% of DI fuel in the first DI injection, which was utilized over the rest of the operating map. The first DI injection typically occurred between 1.75 and 1.9 ms before the second injection, which varied between -40 at low speeds and loads to -60 deg. ATDC at high speeds and loads, as shown in Figure 6.

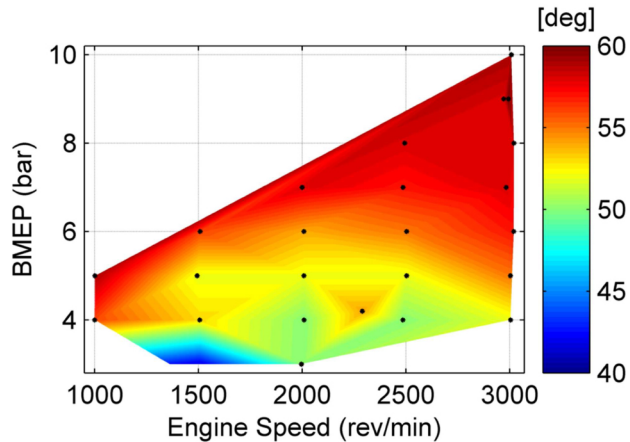


Figure 6. Main SOI timing over the full E20/D experimental space

The DI injection strategy was set to achieve the lowest HC emissions for a given NO_x value, which was set to be 1 g/kg-f. Additional details of this calibration methodology can be seen in [25]

It is important to note that the percent difference between the different fuels is compared in the following graphs, as the relative difference (for percentage variables) between the gas/diesel and E20/diesel results are used rather than the absolute difference. Non-percentage values are the absolute difference. For instance, the difference between 44% and 40% BTE is 10% not 4%.

The fuel injection calibrations were based on previous single-cylinder engine results and computer aided genetic algorithm optimization performed at the UW-Madison Engine Research Center. In spite of this, small differences in cylinder balancing, fuel injection calibration, etc., may exist. Thus, the results are not fully optimized, and more development time using a full design of experiments will likely produce better results.

E20 RESULTS

For the current study, E20 was found to increase brake thermal efficiency (BTE) on average compared to gasoline, as shown in Figure 7. The increase in BTE was caused by a net gain between lower heat transfer and exhaust losses while having higher combustion inefficiency and increased volumetric efficiency (VE). From these gains in VE and reduced thermal losses, the global average² increase in BTE was 1.33%.

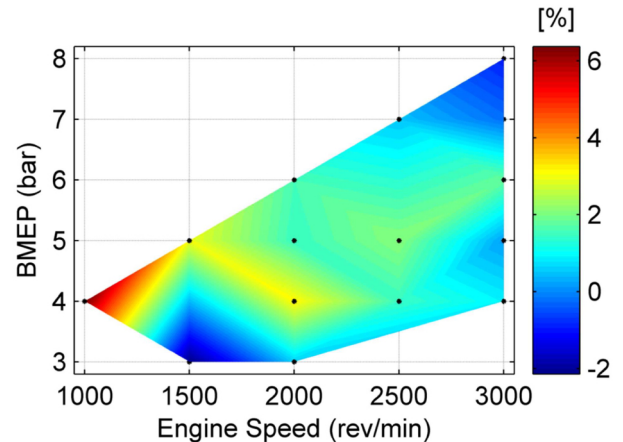


Figure 7. Brake thermal efficiency difference between E20 and gasoline operation

As seen in Figure 8, the difference in VE between E20 and gasoline is shown. The increased VE, 2.2% on average, was caused by the additional air flow from lower intake runner temperatures due to the higher latent heat of vaporization of ethanol as compared to gasoline. The higher VE also reduced pumping losses, by 18.3% on average, as the higher air flow rate increased the turbocharger efficiency, as shown in Figure 9.

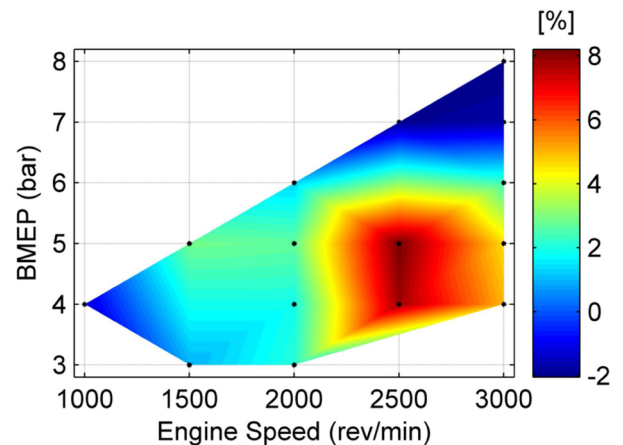


Figure 8. Volumetric efficiency difference between E20 and gasoline operation

²The global average is defined as the % change from gasoline to E20 at each data point divided by all the total number of data points