

Load Limit Extension in Pre-Mixed Compression Ignition Using a 2-Zone Combustion System

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ABSTRACT

A novel 2-zone combustion system was examined at medium load operation consistent with loads in the light duty vehicle drive cycle (7.6 bar BMEP and 2600 rev/min). Pressure rise rate and noise can limit the part of the engine map where pre-mixed combustion strategies such as HCCI or RCCI can be used. The present 2-zone pistons have an axial projection that divides the near TDC volume into two regions (inner and outer) joined by a narrow communication channel defined by the squish height. Dividing the near TDC volume provides a means to prepare two fuel-air mixtures with different ignition characteristics. Depending on the fuel injection timing, the reactivity of the inner or outer volume can be raised to provide an ignition source for the fuel-air mixture in the other, less reactive volume.

Multi-dimensional CFD modeling was used to design the 2-zone piston geometry examined in this study. For experimental evaluation of the design, the geometry was applied to a GM 1.9L ZDTH in-line 4-cylinder engine equipped for dual fuel RCCI operation. The intake system was modified for port injection of gasoline, and diesel was direct injected. Engine tests showed the load could be increased from 9.3 bar IMEP to 12.2 bar IMEP at equivalent noise level when compared to an open geometry using early injections.

CFD modeling using KIVA-3V was used to explore the operation of the novel concept. Using a validated numerical model, the effects of changes in boundary conditions and load extension were examined. The numerical model was used to provide insight into cyclic variability seen in the experiments.

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INTRODUCTION

By tailoring the fuel properties and thermodynamic conditions, premixed, low temperature combustion strategies such as homogeneous charge compression ignition (HCCI) allow low emissions and high engine efficiency. It is possible, although challenging, to control combustion phasing and magnitude without relying on the fuel injection as in a diesel engine, or flame propagation as in a spark ignited engine. Low temperature combustion strategies are hampered by a readily controllable trigger for combustion since they rely on chemical kinetics to drive ignition. The fuel present during compression reaches a state where auto-ignition will occur. A problem arises when the fuel ignites volumetrically, resulting in a high rate of pressure rise that leads to excessive noise and potential engine damage.

Reactivity controlled compression ignition (RCCI) uses a second, high reactivity fuel such as diesel, to provide more control over combustion phasing and magnitude of the peak heat release rate,

which is the driving force beyond the combustion noise. By varying the ratio of pre-mixed to direct-injected (DI) fuel, the phasing and shape of the heat release event can be altered to suit a particular engine load and speed [2, 3, 4, 5, 6]. The high reactivity fuel injection provides better control of the combustion phasing and heat release shape than simply relying on temperature and chemical kinetics. The degree of interaction between the pre-mixed fuel and the direct injected fuel is a function of the injection event(s), the engine geometry, and the time from injection to the start of combustion. Once ignition occurs, combustion continues as heat from the ignition point(s) diffuses rapidly to neighboring regions.

A representative 2-zone piston is shown in the top of Figure 1. It features an axial projection to limit the interaction of the two fuels and thus maintain large gradients in fuel reactivity across the combustion chamber. The green cones indicate the fuel from the direct injection of diesel fuel. The barrier provides several benefits for the combustion system. By isolating the volumes, the reactivity of the

inner or outer volume may be raised relative to the other volume to provide staged ignition, increasing the combustion duration and reducing the peak heat release rate. In the instance shown in [Figure 1](#), the direct injected fuel enters the inner volume and due to the increased reactivity, the inner volume mixture ignites first. The barrier creates a narrow channel between the top of the piston and the cylinder head when the piston is near top dead center (TDC). The channel projects the combustion gases from the inner volume to the outer volume as shown by the orange arrow. Hot, high velocity gases from the inner volume then serve to ignite the outer volume without using more of the high reactivity fuel. Bergin et al. [7] show that the 2-zone piston helps control the rate of pressure rise and decreases unburned fuel emissions at high loads (9 - 18 bar BMEP) in both light- and heavy-duty (LD and HD) engine applications.

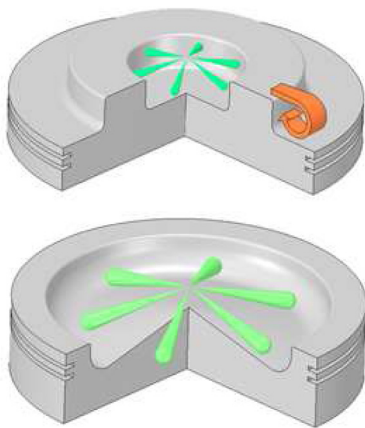


Figure 1. 2-zone geometry (top) and generic conventional diesel geometry (bottom). Green shapes indicate fuel sprays. Orange arrow indicates hot combustion gases from inner volume entering outer volume.

PAPER FOCUS AND STRUCTURE

A GM 1.9L diesel engine [8] with the intake manifold modified for port fuel injection of gasoline was used to experimentally investigate the 2-zone piston design. The operating point used as the basis for the study was derived from the highest loads experienced in the light duty vehicle fuel economy drive cycle [9]. Comparisons between the 2-zone piston geometry and another [10] optimized for RCCI operation were made. Tradeoffs were seen between the piston designs and injection strategies. Of particular interest was the tradeoff between pressure rise rate (PRR) or noise and incomplete combustion products, namely carbon monoxide (CO) and unburned hydrocarbons (UHC).

For the 2-zone piston experiments, the load was increased from the starting point of 7.6 bar BMEP (9.2 bar IMEP) up to 9.2 bar BMEP (12.2 bar IMEP). The performance of the 2-zone piston was examined over a range of start of injection (SOI) timings. Two ranges in the SOI sweep were of particular interest. Early timings ($\sim -55^\circ$ after top dead center (ATDC)) that allow the direct injected fuel more time to premix with the cylinder charge are referred to as RCCI mode. Later timings ($\sim -25^\circ$ to -10° ATDC) where the direct injected fuel mixes primarily with the inner volume are referred to as jet-ignition mode.

A computational model validated with the experimental data was used to highlight important events in the combustion sequence for the RCCI and jet-ignition modes. The differences in temporal and spatial distribution of the DI fuel were used to explain trends in emissions and indicated specific fuel consumption (GISFC). One modeling study examined the effects of air-fuel ratio (AFR), exhaust gas recirculation (EGR) and DI fuel quantity on the ignition delay of the diesel injection and the shape of the heat release rate. A study on the effect of high load operation (~ 14.5 bar IMEP) was conducted to determine the limiting circumstances for RCCI operation with open geometries using gasoline and diesel.

Fluctuations in the measured pressure data were found when conducting tests with the novel 2-zone piston design. An analysis of the pressure fluctuations and their connection to engine noise was performed.

Cycle to cycle variability was seen, particularly at the highest load. The numerical model and experimental results were used to provide possible explanations for the variability.

MULTI-DIMENSIONAL MODEL

Combustion simulations were performed using the KIVA-3V release 2 code [11] coupled with CHEMKIN II for detailed chemical kinetic calculations. Improvements have been made to many physical and chemistry models in the original KIVA-3V code [12, 13, 14, 15]. In this study, *iso*-octane was used as the gasoline surrogate, while *n*-heptane was chosen to represent the diesel fuel chemistry. A reduced primary reference fuel (PRF) mechanism that contains 45 species and 142 reactions was used to simulate both the gasoline (PRF 100) and diesel fuel (PRF 0) chemistry [16]. The physical properties of diesel fuel were modeled as tetradecane. The intake port injected gasoline was modeled as a homogeneous mixture at the beginning of the simulation, which started from intake valve closure (IVC) and ended at exhaust valve opening (EVO). A sector of the combustion chamber was modeled, including a single fuel spray plume by assuming symmetry about the cylinder axis.

Soot was predicted using a phenomenological model based on the approach developed by Hiroyasu [17]. Acetylene was used as an inception species in the soot model. The soot model was coupled to the chemistry solver through the addition of 13 reactions involving acetylene. NO_x emissions were predicted using a reduced NO mechanism that includes 4 species and 12 reactions [18].

The spray model uses the Lagrangian-Drop and Eulerian-Fluid (LDEF) approach. Droplet breakup was modeled using the hybrid Kelvin Helmholtz (KH)-Rayleigh Taylor (RT) model described by Beale and Reitz [13]. The droplet collision model is based on O'Rourke's model, but a radius of influence method was used to determine the possible collision partners to further remove mesh dependency [12]. Interactions between droplets and the wall were considered via a wall film sub-model [19, 20].

EXPERIMENTAL ENGINE & LABORATORY SETUP

Multi-Cylinder Light-Duty Compression Ignition Engine

The engine used for this study was a modified 2007 GM direct injection 1.9L diesel engine. The engine geometric specifications are shown in Table 1.

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

Number of Cylinders	4
Bore [mm]	82.0
Stroke [mm]	90.4
Connecting Rod Length [mm]	145.4
Total Displacement [L]	1.9
DI Injector Nozzle hole number [-]	7
DI Injector Nozzle hole diameter [μm]	141

Prior to the present study, the original equipment manufacturer (OEM) re-entrant bowl geometry [10] was replaced with a shallow/open bowl geometry which reduced the piston crown surface area. Using single-cylinder engine experiments, the modified piston bowl geometry has yielded an increase in combustion efficiency and indicated efficiency for RCCI operation compared to the OEM piston using a variety of fuels [21]. Throughout this study, this piston will be referred to as the “open” geometry. The open pistons were replaced with 2-zone piston geometry. A CAD model of the 2-zone piston and a picture of the installation in the multi-cylinder GM engine block are shown in Figure 2. Throughout this study, this piston geometry will be referred to as “2-zone”.

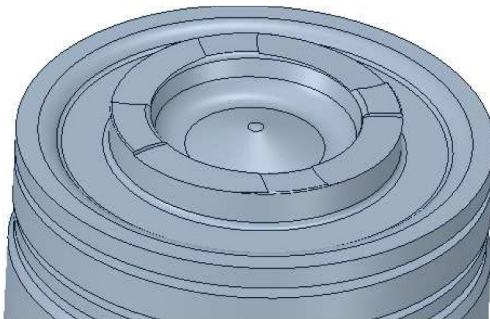


Figure 2. The 2-zone piston bowl geometry in CAD (top) and installed in the multi-cylinder GM 1.9L engine (bottom).

Relevant information for the OEM, open and 2-zone piston designs are included in Table 2. The 2-zone piston has a lower surface area than the OEM bowl. The open geometry has the lowest surface area. Information for the OEM bowl is included for reference only and is not part of the study.

Table 2. General specifications for OEM, open and 2-zone pistons

Piston Geometry	OEM	open	2-zone
Compression Ratio [-]	17.5	15.1	15.5
Crown Surface Area [cm^2]	75.1	56.3	68.1
Inner/Outer volume [%]	N/A	N/A	25/75

The DI fuel system was unmodified from production form. An engine driven Bosch CP1H pump delivers high pressure fuel to a common rail. The direct injectors are Bosch CRI2.2 solenoid driven common rail injectors. The nozzle tips have 7 holes with nominal diameter of 0.14 mm and an included angle of 148 degrees.

The engine is equipped with the stock variable geometry turbocharger (VGT), but an aftermarket charge air cooler (CAC) was installed to be able to control the intake temperature. The intake manifold was modified to incorporate extended tip narrow spray angle port-fuel injectors (PFI), which deliver the low reactivity gasoline for RCCI operation. Figure 3 shows the overall engine and fuel system layout for RCCI operation.

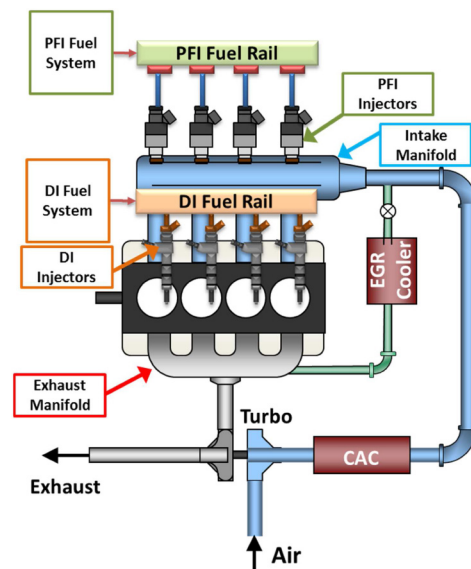


Figure 3. Intake, exhaust, and fuel system schematic.

The stock engine control unit (ECU) was replaced with a full-pass control system from National Instruments - Powertrain Controls (formerly Drivven, Inc.) which 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.

Data Acquisition & Analysis

The DI fuel flowrate and the PFI fuel flowrate were both measured using Micro Motion Coriolis fuel flowmeters (model CMFS010). The intake fresh air flowrate was measured using a laminar flow element. 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 was acquired using Kistler model 6058A pressure transducers installed in glow plug adapters of all 4 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 sampled at a resolution of 0.2 crank angle degrees (CAD). For each operating point 300 consecutive cycles of cylinder pressure data were acquired.

Combustion analysis was conducted using National Instruments - Powertrain Controls software known as the Drivven 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

$$AHRR = \frac{1}{\gamma-1} V \frac{dP}{d\theta} + \frac{\gamma}{\gamma-1} P \frac{dV}{d\theta} \quad (1)$$

where P is the ensemble averaged and filtered in-cylinder pressure, V is the cylinder volume, and γ is the ratio of specific heats. 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 combustion phasing was calculated to be the crank angle at which the accumulated heat release had reached 50% of its maximum, denoted as CA50.

To be consistent with the modeling results, a different method for calculating the experimental heat release was used when comparing experiments and modeling. The apparent heat release rate (AHRR) was calculated using Equation 1 as before, but P is the filtered in-cylinder pressure, V is the cylinder volume, and γ is the ratio of specific heats assumed to have a constant value of 1.33. Due to fluctuations in the measured pressure with the 2-zone piston, AHRR was calculated from ensemble averaged (300 cycles) and filtered pressure. A 4 kHz low pass filter was used.

Combustion noise was calculated using the algorithm from Shahlari et al. [22]. The combustion noise calculation applies two different filters to the raw cylinder pressure signal, one for structural attenuation and the other for the response of human hearing [22]. The combustion noise calculation was only applied to cylinder #1.

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 flowrate for a given command duration. Kinetically controlled combustion strategies like RCCI are especially sensitive to changes in these parameters. This can lead to significant cylinder-to-cylinder imbalances in the combustion phasing (CA50), indicated mean effective pressure (IMEP), PRR, 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 was used to measure total UHC. The UHC emissions will be reported on a C_1 -basis. A heated chemiluminescence analyzer was used to measure total NO_x emissions (NO + NO₂). Both CO and CO₂ were measured using non-dispersive infrared instruments. Intake and exhaust O₂ was measured using a paramagnetic detector. Both intake and exhaust CO₂ were measured to monitor the EGR rate. 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 0.05 or less and will not be reported further. The low-speed data (air flow, fuel flow, boundary conditions, and emissions) were recorded for 180 seconds at ~ 2 Hz.

Engine Operating Conditions and Results

A single direct injection strategy was used throughout this study. The DI fuel was a certification grade 2007 ULS diesel with a cetane number of 46 (Haltermann Fuels HF-00582). The port injected fuel was a certification grade gasoline with Research Octane Number (RON) of 96.6 and Motor Octane Number (MON) of 88.6 (Haltermann Fuels HF-00437).

The open geometry piston was characterized by examining sensitivities to SOI timing, boost, swirl, and pre-mixed gasoline fraction. Gasoline fraction was defined as:

$$\text{gasoline fraction} = \frac{\text{gasoline mass}}{\text{gasoline mass} + \text{diesel mass}} \quad (2)$$

A single case, which is representative of optimal RCCI operation within the capacity of the air handling system was chosen. At the time the open geometry RCCI engine tests were run, EGR was not examined as a parameter. This was due to the relatively low cooling capacity of the EGR cooler. The intake temperature would rise once the EGR rate was greater than 10%. Since pre-mixed, kinetics-driven operating conditions are sensitive to initial temperature conditions, any parametric sweep of EGR rate would also include initial temperature thereby increasing the chances of premature ignition. The late injection timings used with the 2-zone pistons enter the cylinder when the temperatures allow less ignition delay. Thus, EGR was needed to suppress high temperature reactions. Accordingly the EGR cooler capacity was increased before the 2-zone piston tests to

allow almost constant intake temperature operation while increasing the EGR rate. Preliminary modeling studies with the 2-zone piston [7] showed good performance with an AFR of 23.0 and a 30% EGR rate for a 9.2 bar IMEP engine load. Those conditions could not be duplicated exactly with the stock air system on the multi-cylinder engine in the current study, but the experimental conditions were deemed to be sufficiently close.

The experimental campaign presented here was conducted at a fixed engine speed of 2600 rev/min as shown in Table 3. The SOI timings are shown with command and actual timings using an injection delay value of 0.328 ms [23].

The operating condition parameters and test results are shown for three different geometry and injection strategy combinations from left to right in Table 3. The first column shows the open piston geometry with an RCCI injection strategy. In this instance, an RCCI strategy refers to the use of an early ($\sim -55^\circ$ ATDC) SOI timing where the diesel fuel is directed toward the edge of the cylinder and has time to blend with the port-injected gasoline. The second column shows the 2-zone pistons with the RCCI injection strategy. The last five columns show results for the 2-zone piston with the late SOI jet-ignition ($\sim -25^\circ$ to -10° ATDC) strategy. Hereafter, the individual jet-ignition cases will be abbreviated as “jet-ign 1”, “jet-ign 2”, etc. For the 2-zone jet-ignition cases, the load was increased in a stair step fashion from 7.8 bar - 9.2 bar BMEP by increasing the port fuel injection mass flow rate and adjusting the SOI of the diesel fuel to maintain combustion phasing.

Figure 4 shows the pressure and heat release for all cases in Table 3. Two different basic shapes for the heat release can be seen. The RCCI cases exhibit a gradual build up to the peak followed by a quick fall off and finally a short tail. All of the 2-zone jet-ignition cases exhibit the same small initial heat release (inner volume combustion) followed by a larger event when the outer volume ignites. The peak magnitudes of the second jet-ignition heat release event are of a similar magnitude to the RCCI cases except they are phased later. The later phasing of the jet-ignition cases results in similar peak pressures compared to the RCCI open case despite having higher intake manifold pressures. The later heat release phasing did not adversely affect the combustion efficiency (see Table 3). Further analysis on the heat release shape of the jet-ignition cases is included in the *Modeling Results* section.

The injection timings for the jet-ignition cases were significantly later (SOI -12.1° to -10.8° ATDC) than the RCCI cases (SOI $\sim -56.0^\circ$ ATDC). The jet-ignition cases had higher pre-mixed gasoline fractions (~ 0.94) compared to RCCI open (0.87). The amount of DI fuel for the jet-ignition cases stayed constant over the load range. For the boundary conditions in the present study, the combustion phasing was positively correlated with the fuel injection timing. That is, if the injection was advanced then phasing was advanced and vice versa. This is contrary to RCCI operation, where the combustion phasing advances for retarded direct injection timings [24].

Table 3. Engine test operating parameters for open and 2-zone geometries

Injection Strategy	RCCI	RCCI	jet-ign 1	jet-ign 2	jet-ign 3	jet-ign 4	jet-ign 5
Piston Geometry	open	2-zone	2-zone	2-zone	2-zone	2-zone	2-zone
Speed [rev/min]	2600	2600	2600	2600	2600	2600	2600
BMEP [bar]	7.56	8.37	7.75	8.66	8.88	8.95	9.22
Diesel SOI Command, (actual)	-61.0, (-56.0)	-60.0, (-55.0)	-17.0, (-12.0)	-15.6, (-10.6)	-15.0, (-10.0)	-15.1, (-10.1)	-14.3, (-9.3)
GIMEP Cylinder #1 [bar]	9.29	10.96	10.81	11.26	11.64	11.69	12.07
Intake Pressure [bar]	1.50	1.78	1.77	1.82	1.85	1.86	1.88
Intake Temperature [°C]	46.03	43.4	42.1	43.3	44.3	46.1	48.9
EGR [mass based %]	9.15	30.9	28.6	28.5	28.7	30.6	30.9
AFR [-]	32.9	26.6	25.0	24.7	24.3	23.6	22.9
Diesel Rail Press. [bar]	500	500	400	400	400	400	400
Fuel mass [g/cycle/cylinder]	0.0211	0.0257	0.0257	0.0272	0.0279	0.0284	0.0292
Gasoline Fraction [-]	0.866	0.902	0.930	0.932	0.932	0.933	0.935
Combustion Noise* [dB]	98.6	99.2	99.5	99.0	101.0	97.8	99.2
CA 50 [° ATDC]	5.49	7.71	11.53	13.12	12.30	14.7	14.56
COV IMEP [%]	1.60	1.81	2.39	2.38	2.11	3.41	3.32
Combustion Efficiency [%]	95.03	96.92	96.40	96.36	96.35	96.13	96.65

*Combustion Noise calculated using the algorithm from Shahlari et al. [22] applied to cylinder #1.

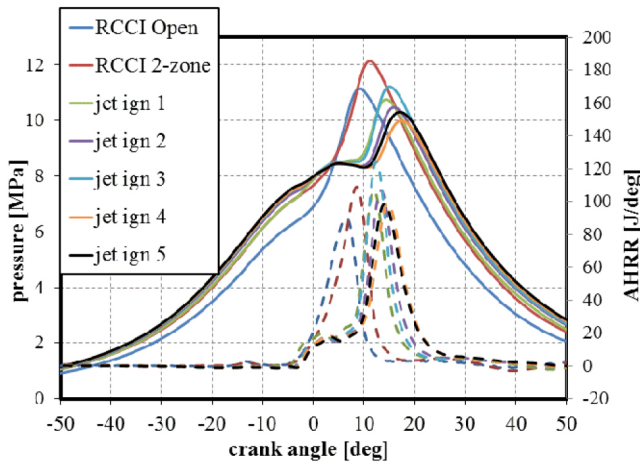


Figure 4. Experimental pressure and AHRR for all cases.

An important detail to note before discussion of the emissions results was that the 2-zone piston had a shorter top ring land height (4 mm vs 7.1 mm) compared to the open piston. With the open piston, the bowl volume is centrally located and the region from the bowl outer diameter to the cylinder wall is flat (see Figure 7 top). The 2-zone design moved the bowl volume outward toward the cylinder wall (see Figure 7 bottom). Accordingly, the crevice height shrunk to accommodate the increased volume. The resulting ring land height was deemed to be adequate for the limited experimental campaign. Based on the crevice height differences only, higher UHC and CO emissions were expected from the open piston. However, combustion phasing, operating condition and piston geometry still play major roles in determining unburned fuel emissions.

Figure 5 shows the CO+UHC versus NOx for all cases. The RCCI 2-zone case had the lowest NOx emissions of any of the cases because of the introduction of a higher EGR rate (31% vs 9.1 %) compared to the RCCI open case. Additionally, operating with an RCCI injection strategy allowed the DI fuel to pre-mix to sufficient levels to decrease fuel rich areas and avoid NOx formation. This case showed that the modifications made to the piston for the 2-zone geometry were, at least, compatible with RCCI operation such that the piston could be used for either jet-ignition or RCCI injection strategies. The RCCI 2-zone case had the highest combustion efficiency.

The RCCI 2-zone case had the best emissions trade-off of any of the cases. The jet-ignition cases had lower CO+UHC (i.e., high combustion efficiency) relative to the open piston case. Generally, however, CO+UHC emissions tend to increase with EGR rate because the oxygen concentration and local combustion temperatures decrease. In this instance, the load also changed, so it was difficult to draw any conclusions from the lower CO+UHC emissions from the 2-zone piston.

As discussed later in the modeling results section, the initial combustion drives the fluid in the outer chamber away from the relatively cool head, liner, and piston. The disturbance of fluid in the boundary layers helps explain why the CO+UHC emissions are lower

for the jet ignition cases despite the fact that 75% of the pre-mixed gasoline does not have any diesel fuel in it to help promote complete combustion unlike in the RCCI cases.

An interesting trend can be seen with the jet-ignition cases in that the NOx emissions decrease as the load is increased. Retarded DI timing allowed less time for NOx formation. Note, as the jet-ignition case number increases, so does the engine load.

Figure 6 shows the brake specific fuel consumption (BSFC) versus NOx for all cases. The RCCI open case has the best BSFC due to lower pumping losses compared to the other cases. The lower pumping losses stem from the lower supplied intake pressure from the VGT and the lower level of high pressure loop EGR.

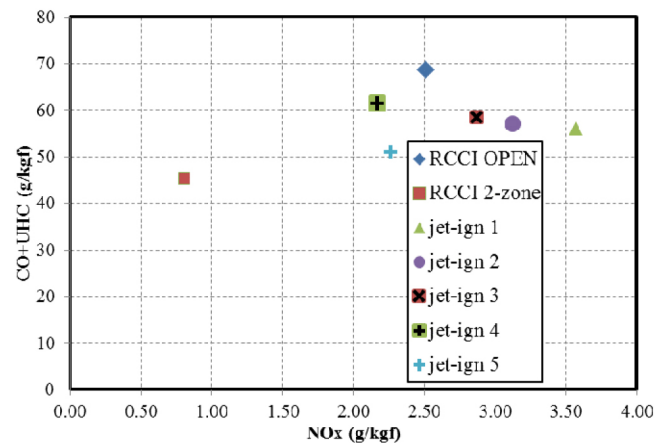


Figure 5. Experimental CO+UHC versus NOx for all cases in Table 3.

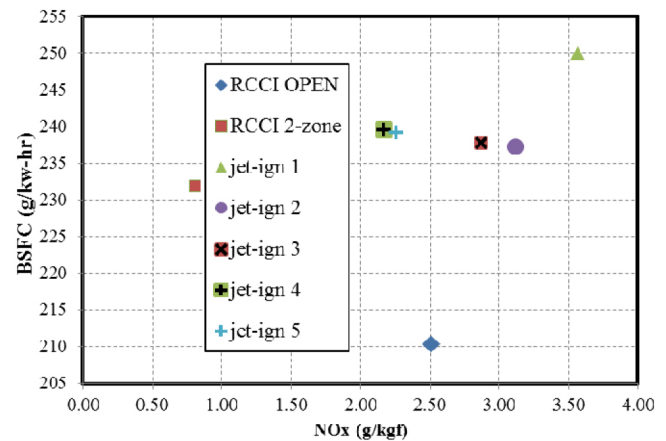


Figure 6. Experimental BSFC versus NOx for all cases in Table 3.

NUMERICAL MODEL VALIDATION

Setup

An axi-symmetric assumption was used for the multi-dimensional modeling study. The meshes are 1/7th of the total volume corresponding to the 7-hole direct injectors. Figure 7 shows the meshes used for the open and 2-zone piston geometries. Note the open geometry includes bumps on the head surface. These are remnants from an earlier study and do not have a significant effect. Later, in the modeling comparison between the open and 2-zone