

Application of Corona Discharge Ignition in a Boosted Direct-Injection Single Cylinder Gasoline Engine: Effects on Combustion Phasing, Fuel Consumption, and Emissions

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ABSTRACT

The downsizing of internal combustion engines to increase fuel economy leads to challenges in both obtaining ignition and stabilizing combustion at boosted intake pressures and high exhaust gas recirculation dilution conditions. The use of non-thermal plasma ignition technologies has shown promise as a means to more reliably ignite dilute charge mixtures at high pressures. Despite progress in fundamental research on this topic, both the capabilities and operation implications of emerging non-thermal plasma ignition technologies in internal combustion engine applications are not yet fully explored. In this work, we document the effects of using a corona discharge ignition system in a single cylinder gasoline direct injection research engine relative to using a traditional inductive spark ignition system under conditions associated with both naturally aspirated (8 bar BMEP) and boosted (20 bar BMEP) loads at moderate (2000 rpm) and high (4000 rpm) engine speeds. Analysis of experimental results shows that relative to optimum load-speed equivalent baseline operation, using the corona discharge ignition system improves fuel economy by (1) reducing cycle-to-cycle variability, (2) promoting more complete combustion, and (3) enabling combustion phasing advancement at high loads by extending the knock limit. Additionally, the system reduces emissions by extending the practical exhaust gas recirculation limits of stable operation.

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INTRODUCTION AND BACKGROUND

Spark-Ignition (SI) internal combustion engines are increasingly operating with boosted intake pressures in order to maintain power output while simultaneously reducing engine size and weight [1,2]. In addition to the use of cooled Exhaust-Gas-Recirculation (EGR), fuel stratification - stratifying the charge such that ignition occurs in locally rich regions and propagates to leaner regions - can improve combustion stability in boosted conditions because it can reduce the likelihood of end-gas auto-ignition [3]. Direct Injection (DI) methods [1,4] have accomplished fuel stratification in boosted gasoline engines. However, while fuel stratification via Gasoline DI (GDI) in conjunction with cooled EGR helps stabilize combustion at boosted intake pressures, ignition is more difficult. As the charge gas density increases, spark plug voltage and current must also increase to arc in the denser gas, resulting in higher rates of electrode ablation. Decreasing the spark plug gap can mitigate the increased voltage/current requirements, but has the effect of increasing heat losses to the developing flame kernel, which can increase the number of

misfires and increase cycle-to-cycle variability to an unacceptable level [5]. Thus, traditional inductive spark ignition systems will face challenges in the downsized, boosted, high-EGR SI engines of the future, and alternative ignition strategies must be explored.

Plasma-Assisted-Ignition (PAI) - the generation or enhancement of plasma in a combustion environment through the use of microwaves, radio frequency waves, dielectric barrier discharges, nanosecond pulsed discharges, and other electric discharges - can reduce ignition delay times and increase flame speeds when implemented in alternative ignition technologies [6, 7, 8, 9, 10]. Most importantly, the PAI techniques that utilize non-thermal (non-equilibrium) plasma can ignite mixtures without generating a high-temperature arc, instead generating enough radical and excited-state species to ignite mixtures chemically rather than thermally. As a result, non-thermal plasma ignition approaches avoid the aforementioned challenges of inductive spark ignition systems at high charge densities.

The non-thermal PAI technique explored in this investigation is corona discharge ignition, in which a single electrode in a combustion chamber receives a large and fast negative voltage pulse or series of pulses controlled by an alternating current (AC) electric field. The resulting large voltage at the electrode increases the reduced electric field in the immediate vicinity of the electrode, pushing electrons outward, which impact neutral molecules and generate more electrons via electron-impact ionization reactions [9, 10, 11]. Electrons can also impact diatomic molecules and dissociate them. One example is the dissociation of molecular oxygen into atomic oxygen, which can more readily participate in reactions that lead to ignition [6, 7, 8, 9]. Depending on the geometry of the electrode, the discharge volume can be many orders of magnitude larger than the discharge volume of a traditional spark plug and the ignition can occur more quickly and more robustly due to the increased radical population in the larger volume [12,13]. This effect diminishes at higher engine loads [11] or high pressures in general [13] since the increased density of the gas in these conditions inhibits electron mobility, ultimately reducing the frequency of both electron-impact ionization and electron-impact dissociation reactions [9].

Researchers have been investigating ignition by corona discharge for some time, and two main methods for generating and/or sustaining the corona discharge have emerged: (1) sending high voltage direct current (DC) to the electrode and (2) using radio frequency (RF) electromagnetic waves generated by resonant circuitry that includes the electrode (henceforth referred to as “RF corona discharge” ignition (RFCDI)). Early experiments utilizing high voltage DC corona discharges showed promise, but power limitations of available technology restricted significant improvements in engine performance to low loads [11]. The community has subsequently made much progress in RFCDI technology through investigations in constant volume non-engine laboratory experiments [12,14, 15, 16], with the concept finally demonstrating an improvement in engine performance [16, 17, 18, 19, 20]. Other researchers have also advanced high voltage DC corona discharge ignition technologies, a notably emergent technology being in the form of “pulsed corona” discharges [9,21, 22, 23]. This concept has informed current nanosecond pulsed discharge ignition technologies, which, like the RFCDI technologies, demonstrate faster ignition times in constant volume chamber experiments [23, 24, 25] and improvement in DI engine performance using both gasoline [26] and E85 [27]. Thus, regardless of the corona discharge generation method under consideration, the technologies have matured to the point where researchers can characterize their potential for implementation in production engines by examining PAI in engines at high loads.

Researchers have investigated RFCDI in two-stroke SI engines [28] as well as GDI four-stroke engines at naturally aspirated [16, 17, 18, 19,29] and boosted [18] loads. Schenk and coworkers [18] extended the EGR limits from 22% to 24% at the highest boosted loading attempted (2000 rpm/14 bar BMEP), though the greatest EGR limit extension (3% to 14%) was at a loading corresponding to naturally aspirated conditions (2000 rpm/2 bar BMEP). For the boosted loading, they extended the lean limit for gasoline-air combustion from $\lambda \approx 1.6$ to $\lambda \approx 1.78$ while for the naturally aspirated loads they extended it from $\lambda \approx 1.17$ to $\lambda \approx 1.3$.

In this paper, we examine the extension of practical operating EGR limits at naturally aspirated (8 bar BMEP) and boosted (20 bar BMEP) loads with moderate (2000 rpm) and high (4000 rpm) speeds using RFCDI relative to corresponding optimum conditions for which inductive SI engine operation is practical. Specific attention is paid to the changes in brake-specific fuel consumption (BSFC) and changes in brake-specific emissions. For this investigation, the “optimum baseline spark ignition operation” for a given engine speed and load is a specific set of valve, injection, and ignition timings at a specific EGR rate that minimizes BSFC. We present BSFC and emission data as relative changes from this optimum SI baseline because we used a proprietary engine and fueling architecture for the tests in this study.

We first discuss the setup of the experiment and the operating procedure. Then we present the results, discussion, and conclusions. We divide the results into three separate sections for different “key test points” corresponding to engine speed and loading: Moderate speeds with naturally aspirated loads (2000 rpm, 8 bar BMEP), moderate speeds with boosted loads (2000 rpm, 20 bar BMEP), and high speeds with boosted loads (4000 rpm, 20 bar BMEP). Within each of these conditions, we further divide the results into three sections. The first section briefly overviews the operating parameters for an EGR sweep conducted for the specified speed and loading. The second section discusses and compares the combustion phasing of some operating points selected from the overview for both inductive SI operation and RFCDI operation. The last section compares processed summary data including cycle-to-cycle variation, brake-specific fuel consumption and brake-specific emissions for the same selected operating points.

EXPERIMENTAL SETUP AND PROCEDURE

Equipment and Measurement

Piston-Cylinder Assembly and Manifolds

For this investigation, we used a 4-stroke direct injection single cylinder research engine (SCRE, Ricardo “Hydra” crankcase). Table 1 lists the engine specifications.

Table 1. Engine specifications for Ricardo “Hydra” SCRE.

Displaced volume	500 cc
Stroke	90 mm
Bore	84 mm
Compression ratio	10.3:1
Number of Valves	4 (Dual, Variable Valve Timing)

The piston had a bowl to enhance fuel-air mixing, and both the fuel injector and ignitor (traditional inductive spark or corona ignitor) were located near the center of the top of the cylinder angled toward one another, between the four overhead valves (two inlet, two exhaust) with variable valve timing. Figure 1 shows a representative diagram of the piston-cylinder assembly with the RF corona ignitor and fuel injector, adapted from Osbourne et al [1] and King and Böcker [4].

A Kistler 6043Asp pressure transducer recorded the instantaneous in-cylinder pressure and corresponding crank angle degree (0.5°CA resolution) with an uncertainty of 0.2 bar (specified by the manufacturer) and an A&D Phoenix RT combustion analysis system encoded the signals. Thermocouples placed immediately ahead of and after the intake and exhaust valves recorded the mean intake and exhaust manifold temperatures, respectively, with a manufacturer-specified accuracy of 3°C . Mean fuel flow to the fuel injector was controlled by an AVL 733S Fuel Balance and measured with a Micro Motion Coriolis meter with a specified accuracy of 0.8% of the flow rate. Mean air and exhaust flow to and from the manifolds was measured using an $8''\text{H}_2\text{O}$ differential pressure transducer, with a specified accuracy of 0.5% of the flow rate.

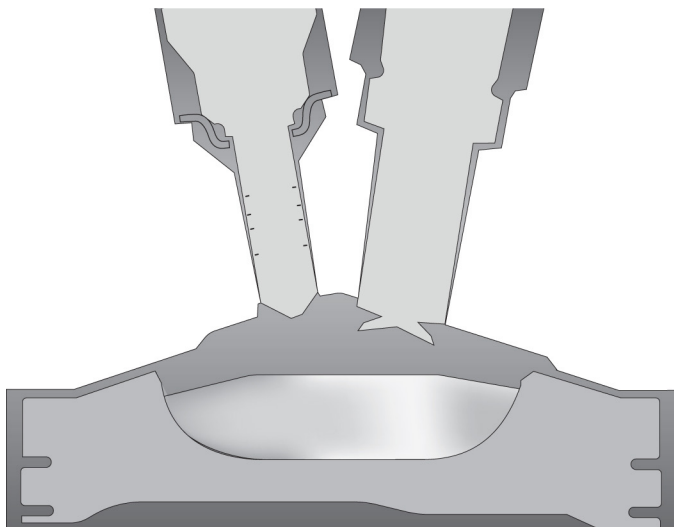


Figure 1. Representative cross-section of piston/cylinder assembly through fuel injector (left) and four-pronged corona ignitor (right) adapted from Osbourne et al [1] and King & Böcker [4].

Dynamometer Assembly

The engine was connected to an AC dynamometer (McClure HS255) via a drive shaft (Bailey Morris), and the torque was measured (HBM T40B) with an accuracy of 0.05% of the measured value. Corrected brake torque, brake power, and BMEP were calculated according to SAE J1349 [30].

Emissions

Mean emissions were measured with a Horiba MEXA-7100 DEGR Raw Gas Bench and sampled after the exhaust split for EGR. We calculate the uncertainties in the emissions measurements based on the manufacturer's error estimate as a percentage of the full scale of the range used for measurement, and we provide the details for the individual species uncertainties in the [Appendix](#). As mentioned, we present the brake specific emission data as relative changes because we used a proprietary engine and fueling architecture in these tests.

Ignition System

A traditional inductive spark ignition system was used for the baseline tests presented in this study to provide a basis of comparison for the RFCDI system. The inductive spark ignition system

comprised a standard J-hook spark plug with a gap of 0.7 mm and a coil-on-plug type ignition coil. The electrode of the RF corona ignitor (Federal Mogul, Inc.) had four sharp points symmetrically spaced about its principal axis. [Figure 2](#) provides a representative diagram.

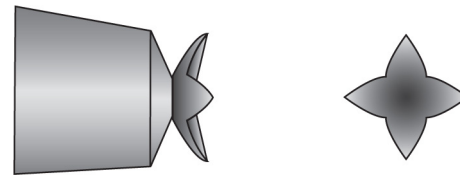


Figure 2. Representative diagram of the four-pronged corona ignitor used in this investigation.

These four points generated the corona discharge in the chamber. The RFCDI system functioned similarly to others that have been developed [16,17,20], both in using radio frequency waves to drive electrons in the charge and in that the circuitry of the system limits the overall current to the electrode, inhibiting the generation of an arc (spark). Instantaneous voltage and current measurements (0.5°CA resolution) were recorded between the DC power supply of the corona igniter and the DC-to-AC transformer that generated the RF electric field. An example plot of an average of these measurements is given in [Figure 3](#). We use these voltage and current measurements to estimate the total primary energy ("from the wall") required for the ignitor to generate and sustain the discharge for each cycle. While the resolution of these measurements is not sufficient for highly accurate calculations, it is good enough for comparing the energy requirements for different engine operating conditions.

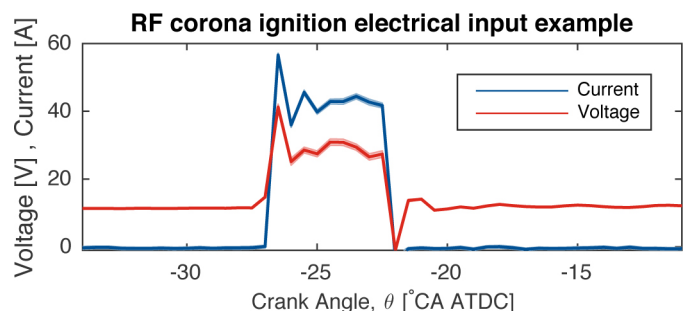


Figure 3. Example plot of average voltage and current measurements recorded from the DC power supply to the RFCDI system. The operating condition shown is 2000 rpm/8 bar BMEP/25% EGR, with an ignition timing of $-26.5^\circ\text{CA ATDC}$. Shaded regions indicate 95% confidence intervals based on 300 engine cycles. Average maximum power was calculated to be 2.4 kW.

Testing Conditions

All experiments were performed using certification gasoline with a Research Octane Number (RON) of 98. For all the tests, the oil temperature was regulated to $90 \pm 3^\circ\text{C}$, the fuel temperature was regulated to $30 \pm 3^\circ\text{C}$, and the intake manifold temperature was regulated to $45 \pm 3^\circ\text{C}$. The intake air came from a temperature and pressure controlled air supply drawn from outdoors of the laboratory. The pressure was regulated with a compressor to meet boosting requirements, the temperature was regulated to $25 \pm 3^\circ\text{C}$, and the relative humidity was regulated to $30 \pm 3\%$.

Experimental Procedure

A series of baseline experiments with traditional inductive SI was completed such that a fair comparison to the RFCDI system performance could be made. We selected three “key test points” at both moderate and high speeds with low and high loading. [Table 2](#) shows the operating parameters for these points. These three operating points represent the minimum BSFC practically possible with SI operation for the three engine speeds and loadings considered in this investigation.

Table 2. Operating parameters and uncertainties for baseline key testing points we use for comparison of the inductive SI and the RFCDI results. The °CA convention assumed in this paper uses Top Dead Center (TDC) of the compression stroke as 0 °CA. We discuss all of these operating points in more detail in several plots in this paper, in which they are highlighted in black. Details regarding the determination of the uncertainties not specified by manufacturer accuracy are provided in the [Appendix](#).

Key Test Point		1	2	3
EGR _{optimum}	[%]	16 ± 1	10 ± 1	7.5 ± 1
Speed	[rpm]	2000 ± 2	2000 ± 2	4000 ± 2
BMEP	[bar]	8.03 ± 0.5	19.78 ± 1.2	20.42 ± 1.2
NMEP	[bar]	8.76 ± 0.5	20.39 ± 1.2	21.39 ± 1.2
$\theta_{cam,intake}$	°CA	-241.2 ± 0.5	-271.4 ± 0.5	-252.6 ± 0.5
$\theta_{cam,exhaust}$	°CA	283.8 ± 0.5	255.7 ± 0.5	260.7 ± 0.5
$\theta_{injection}$	°CA	334 ± 0.5	280 ± 0.5	300 ± 0.5
Fuel Pressure	[bar]	140 ± 2	200 ± 1	200 ± 4
Intake Pressure	[bar]	1.01 ± 0.01	1.96 ± 0.01	1.97 ± 0.01
$\theta_{ignition}$	°CA	-18.11 ± 0.5	1.98 ± 0.5	2.16 ± 0.5
Normalized Air-Fuel Ratio	λ	0.99 ± 0.05	1.00 ± 0.05	0.81 ± 0.05
CA50	°CA	6.85 ± 0.51	24.25 ± 0.51	23.22 ± 0.51

For each operating point, the valve, injection, and ignition timings were optimized successively with regards to BSFC: first, five different intake cam timings were tested to determine the optimum timing while holding exhaust valve timing, injection timing, and ignition timing constant. Then, five different exhaust cam timings were tested with the optimum intake cam timing and other timings constant. After the optimization of the exhaust cam timing, these optimum settings were used similarly to determine the optimum injection timing, and then again this optimization procedure informed the optimization of the ignition timing. [Table 2](#) shows these timings. For the naturally aspirated load testing point, the ignition timings correspond to the maximum brake torque (MBT). For the boosted load points, the timings were advanced to the knock limit.

We calculate the net heat release rate (HRR) using the equation [3]:

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

where we neglect crevice and other heat transfer terms, and all variables are functions of θ . We calculate the ratio of specific heats γ as a function of θ for each cycle (once for compression, γ_{comp} , and once for expansion, γ_{exp}) by calculating a least-squares linear fit of the natural log of P versus the natural log of V for the regions in which $\ln(P)$ vs. $\ln(V)$ is linear. For the part of the cycle during combustion

with nonlinear behavior, we make a linear interpolation between γ_{comp} and γ_{exp} . We then obtain the total net cumulative heat release by integrating HRR over θ .

We provide the resulting CA50 values from these operating parameters at the bottom of [Table 2](#). In this paper, we define CA50 as the crank angle location in the cycle where 50% of the cumulative net heat release over the cycle is achieved. Likewise, we define CA10 and CA90 as the crank angle locations where 10% and 90% of the total cumulative heat release is achieved, respectively. Additionally, in this paper, we define the flame development angle $\Delta\theta_d$ as the number of crank angles between ignition and CA10, and the burning angle $\Delta\theta_b$ as the number of crank angles between CA10 and CA90. We refer to the sum of these as the overall burning angle, $\Delta\theta_o$. We discuss these values in more detail in the combustion phasing part of the Results section.

We estimate the peak in-cylinder temperature for each cycle through the ideal gas relation:

$$T_{peak,est} = \frac{PV}{m_{ch}R} \quad (2)$$

where m_{ch} is the mass of the charge in the cylinder calculated from the measured air, fuel, and EGR mass flow rates and R is the specific gas constant for the mixture. This estimation does not factor in any crevice mass losses through blow-by and so the primary usefulness of this calculation is for comparison only.

For the RFCDI tests, corona ignitor parameters were also optimized for each engine operating condition; these parameters consisted of the corona duration and base voltage supply. Typically, the duration is longer at higher EGR rates, and the supply voltage is higher for higher loading. The optimization resulted in effectively increasing the ignition energy as much as possible without excessive arcing—that is, the percentage of cycles with arc events detected was limited to below 10%. Each experiment consisted of recording the data described for 300 consecutive engine cycles at steady-state.

The following equation provides the cooled EGR rate in this study:

$$EGR(\%) = \frac{X_{CO_2,intake} - X_{CO_2,ambient}}{X_{CO_2,exhaust} - X_{CO_2,ambient}} \quad (3)$$

where $X_{CO_2,i}$ represents the mole fraction of CO_2 in the intake and exhaust mixtures, and the ambient CO_2 concentration was 330 ppm. The EGR rate in [Table 2](#) corresponds to the optimum EGR rate for that engine speed and loading—that is, the EGR rate (with optimized timings) which minimizes fuel consumption for inductive SI operation, and for which the coefficient of variation of indicated mean effective pressure (COV_{IMEP}) is less than 3% during engine operation [3]:

$$COV_{IMEP} = 100 \times \frac{\sigma_{IMEP}}{\mu_{IMEP}} < 3\%$$

For this investigation, μ_{IMEP} is the mean IMEP over the 300 consecutive cycles, and σ_{IMEP} is the standard deviation of IMEP over the same 300 cycles.

RESULTS

In these results, we discuss the combustion phasing of certain baseline SI and RFCDI operating points in more detail. In the tables and plots that follow, we highlight these points with specific colors for clarity. The black open markers/lines correspond to the optimum baseline SI operating points shown in Table 1. The red open markers/lines correspond to RFCDI operation at the same EGR rate as that optimum baseline SI operation. The blue open markers and lines correspond to the optimum RFCDI operating point determined in this study. We define the optimum RFCDI operating point as the EGR rate which minimizes BSFC while keeping $COV_{IMEP} < 3\%$.

Additionally, we present plots of brake-specific fuel consumption, estimated peak in-cylinder pressure, and brake-specific emissions alongside each other. While we provide some cursory explanation to relate the three in this section to facilitate reading and draw attention to the plots, we present the bulk of the analysis for all of the results in the Discussion section of this paper.

We report the BSFC and emissions results for the RFCDI operation as percentage change from the optimum baseline SI operating point:

$$\Delta\%(EGR) = 100 \times \frac{RFCDI(EGR) - SI(EGR_{opt})}{SI(EGR_{opt})} \quad (5)$$

where the SI operation at EGR_{opt} corresponds to the operating point in Table 2. For all reported values with uncertainties, we give the details of the uncertainty calculations in the Appendix.

Moderate Speeds and Naturally Aspirated Loading Conditions (2000 rpm/8 bar BMEP)

This section presents results for moderate engine speeds and loads associated with naturally aspirated operation.

Overview of Operating Parameters for EGR Sweep

Table 3 shows an overview of the operating parameters similar to those in Table 1 for the EGR sweep of the moderate speed and naturally aspirated loading tests.

To facilitate understanding of results that follow, we also provide the percentage change in fuel mass flow relative to Key Test Point 1 (2000 rpm/8 bar BMEP/16% EGR) in Table 2. Most of the parameters were unchanged across EGR rates, with the significant exceptions being the ignition timing θ_{ign} , intake pressure P_{int} , and fuel mass flow, $\dot{m}_{fuel}$. For an EGR rate of 16%, the ignition timing was retarded slightly (by 2.2 ± 1.0 °CA) from the 16% EGR baseline SI operation in Table 1 to obtain MBT. At higher EGR rates, the ignition timing required advancement (by 7.8 ± 1.0 °CA) from the 16% EGR baseline SI operation to obtain MBT. The CA50 values remained relatively unchanged for most of the EGR rates relative to the baseline optimum SI case.

Table 3. Operating parameters for the 2000 rpm/8 bar BMEP RFCDI tests for different EGR rates examined. Uncertainties are the same as for Key Test Point 1 in Table 2 but are omitted to save space, except for the CA50 values. Selected points that are discussed in more detail throughout this paper are bolded and colored: 16% EGR (red) and 25% EGR (blue).

EGR	[%]	16.3	19.6	24.9	28.0
Speed	[rpm]	2000	2000	2000	2000
BMEP	[bar]	7.86	7.82	7.84	7.75
NMEP	[bar]	8.75	8.71	8.79	8.70
θ_{int}	°CA	-241.3	-241.3	-241.4	-241.3
θ_{exh}	°CA	283.6	283.6	283.5	283.5
θ_{inj}	°CA	334	334	334	334
P_{fuel}	[bar]	140	140	140	140
P_{int}	[bar]	1.07	1.11	1.13	1.16
θ_{ign}	°CA	-16.58	-19.66	-26.50	-34.54
AFR	λ	0.99	0.99	0.99	0.98
CA50	°CA	7.0±0.5	6.5±0.5	6.1±0.5	6.6±0.6
$\dot{m}_{fuel}$	[%]	-0.97 %	-1.81 %	-3.27 %	-3.60 %

Combustion Phasing

For moderate speeds and naturally aspirated loads (2000 rpm, 8 bar BMEP), the RFCDI system enabled high EGR (up to 25%) operation with an earlier pressure rise, shown in Figure 4 alongside the net HRR and cumulative net heat release.

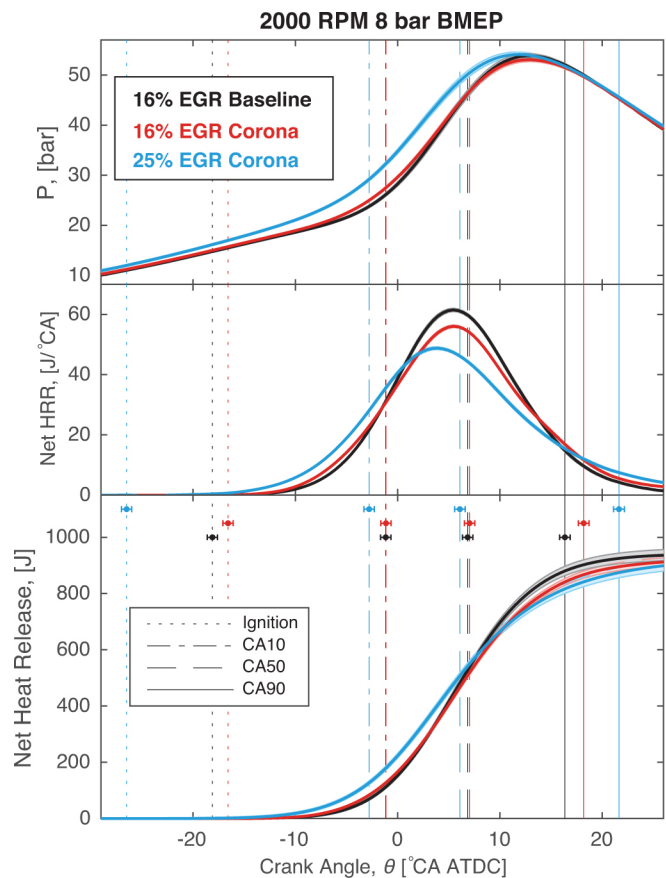


Figure 4. Combustion phasing plots shown for selected EGR conditions for the 2000 rpm/8 bar BMEP tests as a function of crank angle degree: Pressure (top), net HRR (middle), and cumulative net heat release (bottom). $\lambda = 1 \pm 0.05$ for all operating points shown. The shaded regions depict the uncertainty in the values plotted. EGR conditions shown are: 16% EGR with traditional inductive SI (black), 16% EGR with RFCDI (red), and 25% EGR with RFCDI (blue). Crank angle locations of both ignition and friction net heat release are the vertical lines in the figure, with uncertainties as shown as horizontal error bars.

We observed lower HRR for the 25% EGR condition in part due to the reduced fueling shown in Table 3, and on average this led to less cumulative net heat release, though uncertainties preclude a definitive conclusion based on these data alone. Though the RFCDI system lowered the flame development angle $\Delta\theta_d$ for the 16% EGR operating point, the ignition timing required substantial advancement at 25% EGR. The CA50 did not significantly advance (by 0.7 ± 1.0 °CA) relative to the 16% EGR baseline SI case.

Figure 5 plots burn duration angles against EGR rate for the 2000 rpm/8 bar BMEP tests for RFCDI operation and baseline SI operation. In general, the RFCDI system reduced the overall burning angle $\Delta\theta_o$ for each EGR rate, but more significantly in the flame development angle $\Delta\theta_d$ than the burning angle $\Delta\theta_b$. At 16% EGR, $\Delta\theta_b$ from the RFCDI system was actually larger than $\Delta\theta_b$ from the optimum baseline spark ignition operation, but this was the exception rather than the rule for the 2000 rpm/8 bar BMEP tests. For the optimum RFCDI operating point at 25% EGR, $\Delta\theta_d$ reduced by 5.6 ± 1.0 °CA compared to traditional SI, and $\Delta\theta_b$ reduced by 2.7 ± 1.0 °CA.

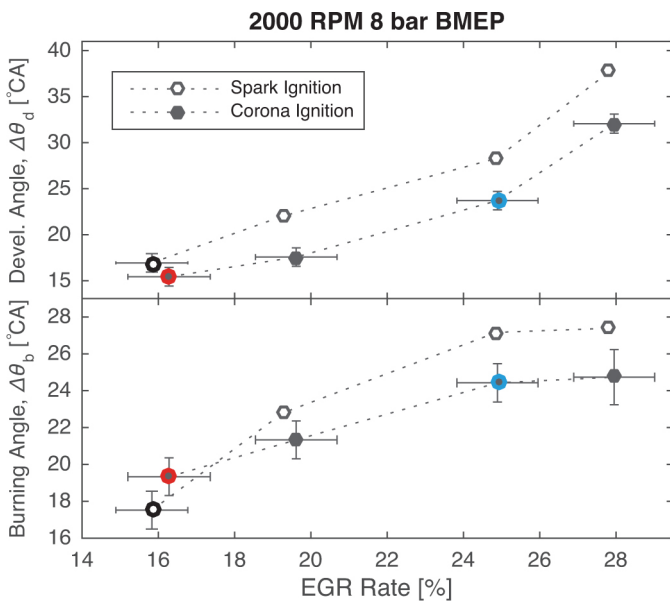


Figure 5. Plots of burn duration for the 2000 rpm/8 bar BMEP tests. Baseline SI operation is shown with open markers and RFCDI operation is shown with filled markers. $\lambda = 1 \pm 0.05$ for all operating points shown. Flame development angle $\Delta\theta_d$ (top) and burning angle $\Delta\theta_b$ (bottom). The operating points shown in Figure 4 are highlighted with colored outlines: 16% EGR inductive SI (black), 16% EGR RFCDI (red), and 25% EGR RFCDI (blue).

Cycle to Cycle Variability, Brake-Specific Fuel Consumption and Emissions

Figure 6 shows a plot of COV_{IMEP} for RFCDI operation across the range of EGR rates for the 2000 rpm/8 bar BMEP tests along with the values for the corresponding baseline inductive SI operation, shown with open markers for comparison. Overall, using the RFCDI system did not have a significant effect on the COV_{IMEP} for all EGR rates tested except for the highest EGR rate (28%), indicating that any

combustion stabilization resulting from using the RFCDI system is more nuanced than can be determined from COV_{IMEP} alone. We provide more detail for this in the Discussion section.

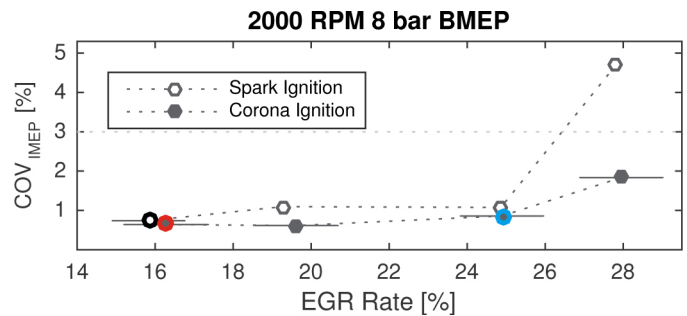


Figure 6. Plot of COV_{IMEP} for the 2000 rpm / 8 bar BMEP tests. $\lambda = 1 \pm 0.05$ for all operating points shown. The operating points shown in Figure 4 are highlighted with colored circles: 16% EGR inductive SI operation (black), 16% EGR RFCDI operation (red), and 25% EGR RFCDI operation (blue).

Figure 7 shows plots of changes in BSFC, averages of estimated peak in-cylinder temperatures $T_{peak,est}$, and changes in brake-specific emissions (relative to the 16% EGR baseline SI test) for the RFCDI 2000 rpm/8 bar BMEP tests.

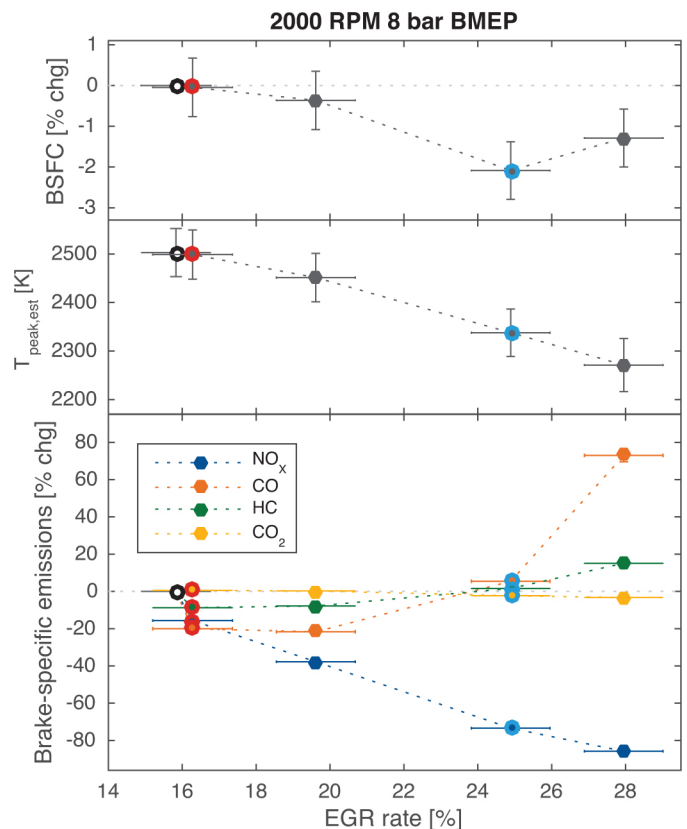


Figure 7. Changes in brake-specific fuel consumption (top), average of estimated peak in-cylinder temperatures (middle), and changes in brake-specific emissions (bottom) for the 2000 rpm/8 bar BMEP tests. $\lambda = 1 \pm 0.05$ for all operating points shown. For BSFC and emissions, all RFCDI points are plotted as percentage change from the optimum baseline SI operation marked with black circles. The operating points shown in Figure 4 are highlighted with colored circles: 16% EGR RFCDI operation (red), and 25% EGR RFCDI operation (blue).