

Combined Effects of Multi-Pulse Transient Plasma Ignition and Intake Heating on Lean Limits of Well-Mixed E85 DISI Engine Operation

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ABSTRACT

Well-mixed lean SI engine operation can provide improvements of the fuel economy relative to that of traditional well-mixed stoichiometric SI operation. This work examines the use of two methods for improving the stability of lean operation, namely multi-pulse transient plasma ignition and intake air preheating. These two methods are compared to standard SI operation using a conventional high-energy inductive ignition system without intake air preheating. E85 is the fuel chosen for this study.

The multi-pulse transient plasma ignition system utilizes custom electronics to generate 10 kHz bursts of 10 ultra-short (12ns), high-amplitude pulses (200 A). These pulses were applied to a custom spark plug with a semi-open ignition cavity. High-speed imaging reveals that ignition in this cavity generates a turbulent jet-like early flame spread that speeds up the transition from ignition to the main combustion event.

Performance testing shows that lean operation with heated intake air enables a 17% improvement of fuel economy at $\phi = 0.59$ for both ignition systems, relative to that of stoichiometric operation. Moreover, multi-pulse transient plasma ignition offers more stable ultra-lean operation, with IMEP_n variability less than 5% down to $\phi = 0.49$. The ability to operate stably at such lean conditions is attributed to a more stable flame initiation offered by both the increased charge temperature and the multi-pulse transient plasma ignition that allows a later spark timing due to the very fast transition to fully turbulent deflagration.

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INTRODUCTION

In the light of constrained petroleum supply and anthropogenic climate change [1], increased engine efficiency and alternative fuels are of interest. Because of its widespread use, this study deals with combustion in a direct-injection spark-ignited (DISI) engine, examining the potential for efficiency improvements using lean operation. The dominating combustion mode for gasoline SI engines used for automotive applications is well-mixed stoichiometric operation. This allows the application of a three-way catalyst for cost-effective reduction of NO_x and oxidation of CO and HC [2]. However, there are several reasons why stoichiometric operation limits the observed engine efficiency, especially for low and intermediate loads. First, the required intake throttling results in pumping losses. Second, high combustion temperatures lead to both high heat-transfer losses and unfavorable thermodynamic properties of the combustion products. The latter manifests itself as high specific heat capacity, reducing both the ratio of specific heats (γ) and the work-extraction efficiency of the

expansion stroke [2]. A third factor is the inability of the stoichiometric combustion to fully complete near TDC due to dissociation of CO₂ in the hot O₂-depleted gases [3,4].

Lean operation can improve engine efficiency by mitigating all of these effects. However, the challenge is to maintain stable and efficient combustion despite a reduction of flame speeds [5]. The reduction of flame speeds becomes a particular problem from the perspective of ignition and effective flame spread throughout the charge. In this work, multi-pulse (MP) transient plasma ignition is compared against regular inductive spark ignition. For each ignition system, the effects of intake gas heating are also examined. The purpose is to systematically determine effective paths for stable and efficient lean SI operation. Lean SI operation has been studied many times previously [2,5, 6, 7, 8, 9, 10, 11]. The current study offers new insights by comparing engine operation with two

different ignition system at two intake temperatures, encompassing all-metal performance tests, high-speed plasma and flame imaging, and exhaust emissions measurements.

Furthermore, it is of interest to study how these processes are affected by the fuel type. E85 is the fuel chosen for the current study since it represents the extreme point encountered in the fuel tanks of flex-fuel vehicles in the US. Future studies using the same engine geometry will contrast these results with those of pure gasoline and intermediate blends.

EXPERIMENTAL SETUP

Research Engine

The engine used for the experiments is a single-cylinder four-valve research engine that can be set up in two configurations: an all-metal configuration for continuously fired performance testing, or in a nearly identical optical configuration for imaging studies. All tests here were conducted with a water-cooled metal cylinder and oil-lubricated metal piston rings. The piston has a moderately deep piston bowl to aid the stratification of fuel for spray-guided stratified-charge operation, e.g. Refs. [12,13]. However, for these tests the fuel was injected during the intake stroke to generate a well-mixed charge. Figure 1 shows a cross section of the combustion chamber and Fig. 2 shows a picture of the piston top. The piston-bowl rim has a slight cut-out on one side to facilitate a better view into the bowl via the pent-roof side-window on the right-hand side in Fig. 1.

Engine data are given in Table 1. The engine was fueled using an 8-hole injector. The injector was oriented such that two of the fuel sprays straddled the spark-plug gap of the commercial long-reach spark plug. The same injector orientation was used for the custom spark plug described below. During fuel injection, it is possible that limited fuel wetting of the custom spark plug occurred on the exterior of the two ground electrodes closest to the injector. However, high-speed flame imaging has not shown any evidence of rich and sooting combustion near the ground electrodes.

For all data, the phasings of the cam shafts were selected to provide both low residual levels and high volumetric efficiency. The resulting valve lifts are plotted in Fig. 3. It should be noted that intake valve #1 (shown on the right-hand side in Fig. 1) was deactivated to increase in-cylinder swirl and tumble levels, see Table 1. In this study, the crank angles (CA) are referenced as after top dead center of the combustion stroke, aTDC.

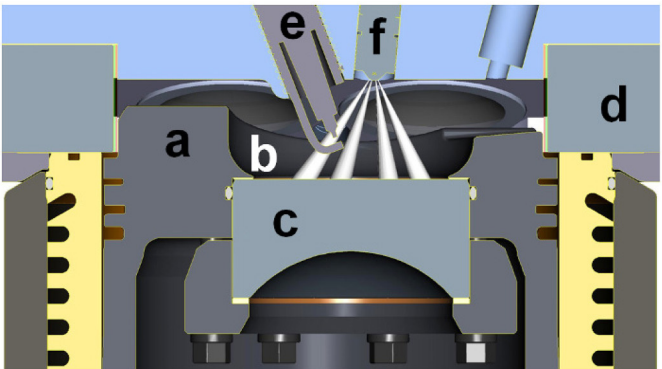


Figure 1. CAD rendering of cross-section of combustion chamber at TDC with a - piston, b - piston bowl, c - piston-bowl window, d - pent-roof window, e - spark plug, and f - fuel injector.

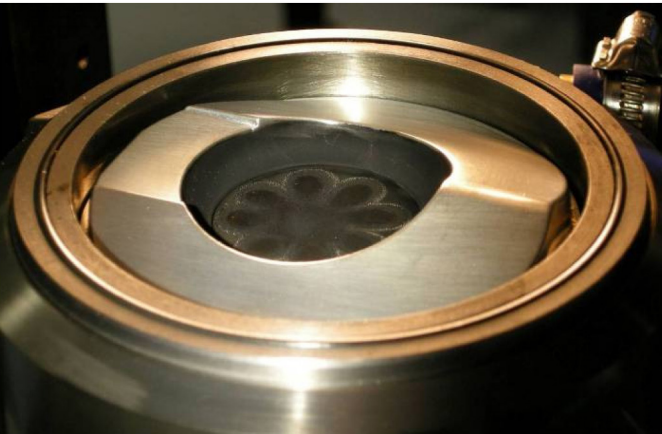


Figure 2. Picture of all-metal piston and cylinder in its dropped-down position.

TABLE 1. Engine Specifications

Displacement	0.552 liters
Bore	86.0 mm
Stroke	95.1 mm
Connecting Rod Length	166.7 mm
Geometric Compression Ratio.....	12:1
Intake Valve Diameter	35.1 mm
Intake Valve Angle Relative Cylinder Axis	18°
Exhaust Valve Diameter	30.1 mm
Exhaust Valve Angle Relative Cylinder Axis.....	16°
Swirl / Tumble Index (one intake valve deactivated)....	2.7 / 0.62
Fuel Injector.....	Bosch 8-hole solenoid-type
Hole Orientation.....	Symmetric with 60° included angle
Hole Size	Stepped-hole, min. dia. = 0.125 mm

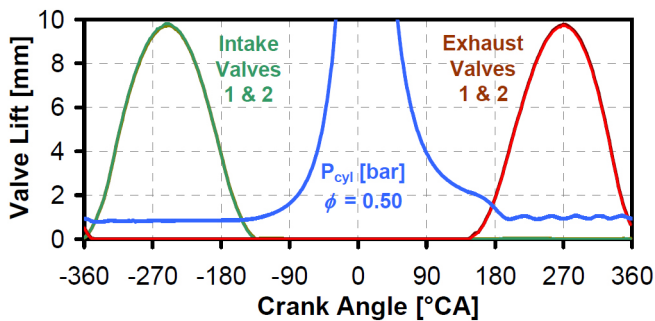


Figure 3. Valve lift profiles measured dynamically using high-speed imaging combined with automatic image processing. An example of a pressure trace is plotted as well.

No EGR was used for these tests. However, the exhaust gas flow was diluted with extra air using nozzles at the entry point of the exhaust tank. This was done to limit the exhaust gas temperature in the tank to avoid overheating the steel. Both the air to the intake tank and the dilution air to the exhaust tank were metered with sonic-flow orifices. This allows the measured exhaust composition to be back-corrected to engine-out mole fractions. A Horiba MEXA-584L emission analyzer was used to measure CO, CO₂ and HC. O₂ was measured using a CAI 600PP gas analyzer. NO_x was measured with a CAI 600CLD gas analyzer. Exhaust smoke was measured with an AVL 415S smoke meter.

Ignition Systems

Two ignition systems were used as described in the following.

Regular Inductive System

The inductive system was supplied by Bosch. The spark coil (model ZS-L 1x1E) can provide relatively high spark energy. For all the tests reported here, the charging time of the primary coil was set to 5 ms. According to measurements by Bosch, this results in a spark energy of 106 mJ. The secondary coil was connected to an NGK spark plug featuring a long-reach single-point J-gap. The spark gap width was adjusted to 1.0 mm prior to the engine tests. A photo of the spark plug is provided in Fig. 4b.

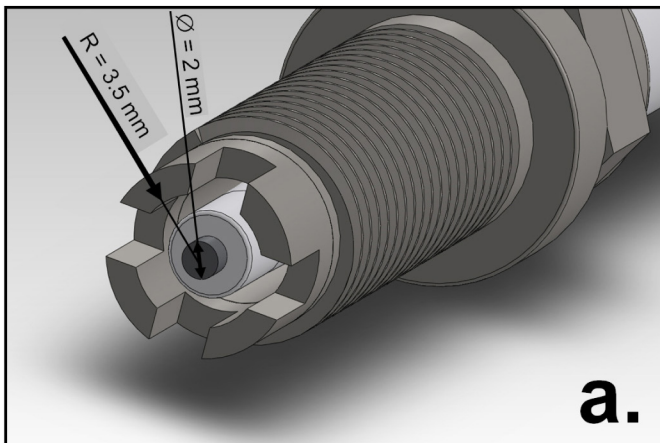


Fig. 4.

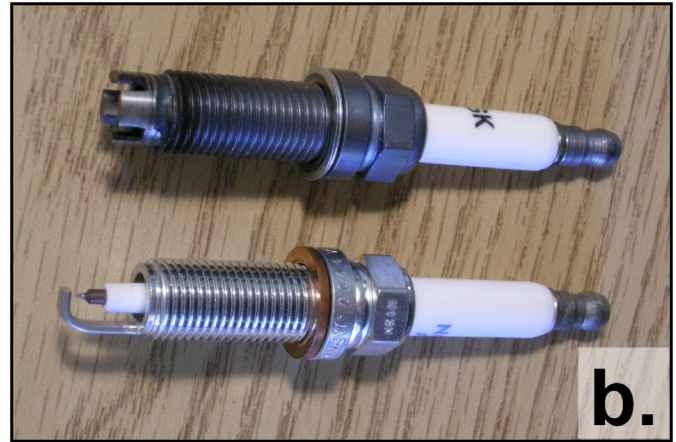


Fig. 4. (cont.) a) CAD rendering of the custom spark plug with dimensions of the semi-open cavity. b) Photo of the commercial long-reach spark plug (bottom) and the in-house modified spark plug (top).

Advanced Ignition

A research-grade advanced ignition system capable of multiple ultra-short pulses was used together with an in-house developed spark plug featuring a semi-open ignition cavity. Figure 4b shows a photo of the custom spark plug in comparison with the commercially available spark plug. Figure 4a provides two of the key dimensions of the semi-open ignition cavity. The gap distance from the central positive anode to any of the four wide ground cathodes is 2.5 mm. Figure 5 shows the custom spark plug mounted in the cylinder head.

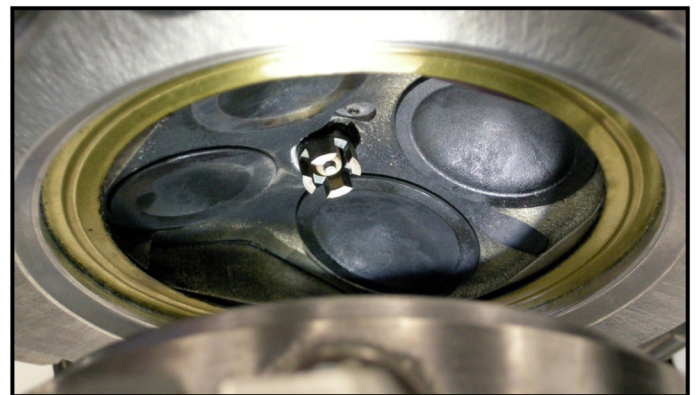


Fig. 5. Photo of the custom spark plug mounted centrally in the four-valve pent-roof cylinder head. (Drop-down cylinder in the foreground is partially blocking the view.)

The pulse generator used in this experiment is based on solid-state electronics and uses magnetic pulse compression to shape the output pulse. It is capable of delivering a 40 kV 12 ns (FWHM) pulse into a 200 Ω load. The voltage amplitude is scalable while maintaining the same pulse width down to an output voltage of 15 kV, and the maximum pulse energy is 40 mJ. The pulse generator was connected to the spark plug using a standard solid core spark plug wire and boot. Both the spark plug wire and the boot were shielded with a custom wire-mesh sheathe that was connected to ground of both the

cylinder head and the pulse generator. The pulse generator was operated in burst mode, in which 10 pulses were delivered at a repetition rate of 10 kHz. The timing for the first of the 10 pulses was controlled by custom electronics coupled to the shaft encoder of the engine. After the first pulse was triggered, the remaining 9 pulses were triggered in the time domain.

Experimental Procedure

For each operating point with the all-metal engine configuration, the engine was allowed to run for several minutes until all measured parameters were stable, at which point the data were acquired. Fuel-flow rate and thermocouple readouts were averaged over one minute. The in-cylinder pressure, spark current for the regular inductive system, intake and exhaust pressure, and fuel pressure were acquired for 500 consecutive cycles using 0.1°CA resolution. For the in-cylinder pressure, an uncooled Kistler 6125C piezoelectric sensor was used in combination with a Kistler 5010B charge amplifier. Day-to-day repeatability was very good, with variations of average indicated mean effective pressure (IMEP) of less than 2%. The apparent heat-release rate (AHRR) is computed from the in-cylinder pressure for each individual cycle using a constant ratio of specific heats ($\gamma = 1.33$) following [4]. For computing combustion-phasing metrics like the 50% burn point (CA50), the AHRR is integrated over the crank-angle range for which AHRR is positive. Unless noted explicitly, this computation of burn points is performed in a conventional way, whereby the integral of AHRR is scaled so that it rises from 0 to 100% for every cycle, irrespective of the actual combustion efficiency (η_{comb}). To quantify IMEP_n instability, the standard deviation (SD) of IMEP is first computed for all 500 cycles. As a measure of the *relative* IMEP_n instability, the corrected standard deviation (CSD) of IMEP is computed:

$$\text{CSD of IMEP [\%]} = 100 \cdot \frac{SD_{\text{IMEP}_{\text{fired}}}}{(\text{IMEP}_{\text{fired}} - \text{IMEP}_{\text{motored}})} \quad (1)$$

This is identical to the traditionally used coefficient of variation (COV), except that the denominator has been corrected for the heat-transfer and blow-by losses of a motored cycle. The difference in IMEP for fired and motored operation equals the IMEP rise caused by the combustion event. This definition allows the relative IMEP_n instability to be consistently evaluated even for operation with IMEP near zero or negative, see Ref. [14] (e.g. idle operation or vehicle coasting.) For operation at the moderate loads of this study, CSD values are only slightly lower than COV values. Furthermore, the value of CSD of IMEP does not depend on the definition of IMEP (i.e. compression-expansion for gross, or all four strokes for net), provided that the gas-exchange strokes are repeatable. (In this study, the gas-exchange strokes are very repeatable as a result of the selected constant valve timings.) In the following, CSD of IMEP_n will be referred to as IMEP_n instability.

Presented values of peak-charge temperatures are computed using the ideal-gas law in combination with the measured pressure (ensemble-averaged over 500 cycles), the known cylinder volume, and the trapped mass. The trapped mass equals the sum of the supplied air, fuel, and residuals. The average molecular weight corresponds to that the measured exhaust composition. As such, the computed temperatures correspond to the mass-averaged temperature of a single-zone model.

Optical Diagnostics Setup

High-speed imaging is presented in this paper to aid the interpretation of the all-metal results. For this imaging, the engine was equipped with a quartz window in the piston bowl, see Fig. 1c. This window has a concave ($R = 25\text{mm}$) lower surface to enable a wide-angle view into the combustion chamber. High-speed plasma and flame imaging through the piston-bowl window was accomplished using a Phantom v7.3 camera via a 45° Bowditch mirror and a camera lens with a focal length of 180 mm, see Fig. 6a. Fully open, the f-stop of this lens is 2.8. The acquired image resolution was 352×352 pixels, but only the central 224×224 pixels were reproduced in the results below and in the example in Fig. 6b.

Side-view imaging was performed with a Phantom v311 camera and a camera lens with a focal length of 72 mm. Fully open, the f-stop of this lens is 1.4. The large aperture in combination with a very short distance between the lens and the side window ($\sim 100\text{ mm}$) provides a high light-collection efficiency. Therefore, most of the presented images of the weak flame luminosity of lean combustion are from this side view. For orientation Fig. 6c shows an example image. The region of interest (ROI) is marked with a rectangle. This is the area presented in the image sequences below using 512×256 pixels. The blurry bright object below the ROI is the piston-bowl window, through which the light entered for this reference image.

The cameras were synchronized to the crank-shaft encoder, simultaneously acquiring images each 0.3°CA, equivalent of 50μs between successive frames at an engine speed of 1000 rpm. This corresponds to a frame rate of 20 kHz. Exposure duration was set to 40μs. Hence, there is a 10μs period for each 0.3°CA when the cameras do not collect light. Hence, roughly 20% of the bright, but very short, transient plasma pulses from the MP ignition system occur during this 10μs inter-frame period. This circumstance makes it possible to present images of low-luminosity flames that develop during the 10-pulse burst of the MP plasma ignition system.

Both monochrome cameras are sensitive in the 350 - 900 nm range, so broadband luminosity was collected from both the ignition plasma and flames. In this optical configuration, the engine was operated similarly to the all-metal operating points, but with a shorter run duration using a Fire3-Skip9 sequence. This 25% duty cycle reduces the heat load on the quartz windows to lower the risk of window failure due to

thermal stresses. All images presented here were acquired on the 3rd fired cycle of each sequence to ensure proper temperature and composition of the residual gases. (The 1st fired cycle has cold residuals containing only air. The 2nd fired cycle has hot residuals but with an O₂ level that is higher than steady-state operation.) Nonetheless, because of the lower surface temperatures of this skip-fired operation, earlier spark timings had to be used in order to achieve the same combustion phasing as the continuously fired all-metal tests. Because the operating conditions are not identical for all-metal and optical operation, the flame imaging is solely used to illustrate fundamental effects of air dilution and multi-pulse ignition on the flame spread. To limit run times, imaging was performed for only 66 combustion cycles for each optical operating point.

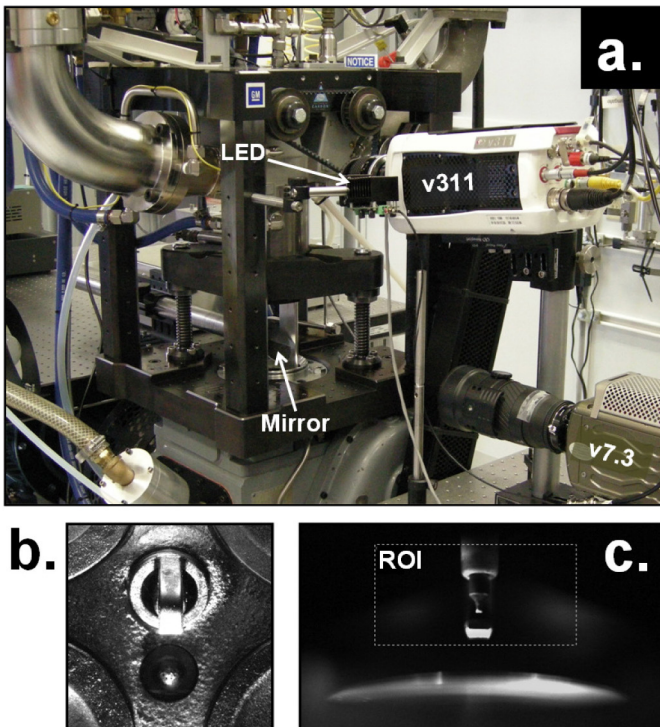


Figure 6. a) Picture of single-cylinder research engine in its optical configuration. b) Bottom-view image at -20°C. c) Side-view image at -20°C, with region of interest (ROI) marked.

Fuel

E85 is used in this study. It was prepared in-house by blending research-grade gasoline with anhydrous high-purity ethanol in 15%/85% proportions by volume. The gasoline fuel is a Chevron-Phillips research-grade gasoline with an antiknock index of 87. It is a typical gasoline and its specifications are given in Table 2. With the assumption of a linear variation of properties upon blending, some estimated properties of the E85 fuel are given in Table 3.

TABLE 2. Chevron-Phillips Research-Grade Gasoline

Antiknock Index (R+M)/2	86.8
RON	91.0
MON	82.7
Specific gravity	0.7456
Carbon, wt%	86.37
Hydrogen, wt%	13.55
Oxygen, wt%	0.08
A/F Stoichiometric	14.60
Lower Heating Value, gas-phase fuel (MJ/kg)	43.10
LHV for stoichiometric charge (MJ/kg)	2.763
Hydrocarbon Type, vol%	
Aromatics	24.5
Olefins	5.1
Saturates	70.1
Distillation, °C	
5%	56.9
10%	63.4
30%	78.9
50%	96.1
70%	114.1
90%	147.3
95%	164.9

TABLE 3. Estimated Properties of E85 Fuel

Specific gravity	0.7826
A/F Stoichiometric	9.808
Lower Heating Value, gas-phase fuel (MJ/kg)	29.23
LHV for stoichiometric charge (MJ/kg)	2.704
Hydrocarbon Type, vol%	
Aromatics	3.7
Olefins	0.8
Saturates	10.5
Oxygenates	.85

RESULTS

These tests employed swirling in-cylinder flow for convenience and for consistency with previous studies (e.g. Refs. [12,13]). The swirling flow was accomplished by deactivating one of the intake valves, as noted previously in conjunction with Table 1.

Table 4 shows the operating conditions that are common to all data sets covered in this study. An early fuel-injection timing was used to create a well-mixed charge of fuel, air and residuals. It should be noted that the selected valve timings result in low residual levels ($\approx 4\text{--}6\%$ by mass). The fueling rate was held constant at 21.6 mg/cycle to within $\pm 0.8\%$. As the charge is leaned out by metering more air into the intake system, the intake pressure rises. For the leanest cases, the engine operation approaches the equivalent of unthrottled operation with an intake pressure (P_{in}) of almost 100 kPa. The targeted combustion phasing, measured as CA50, is 5°CA aTDC. Ideally, the combustion phasing should be selected

on-the-fly to maximize IMEP or BMEP for each combination of ϕ / P_{in} . However, this would have resulted in much longer engine run times and was not practical because the experimental campaign started with optical diagnostics of the MP plasma ignition system. Because of this, CA50 = 5°CA was selected as a compromise.

TABLE 4. Operating Conditions

Fuel	E85
Engine Speed	1000 rpm
Injection Pressure	170 bar
Start of Injection	-300°CA
Fueling Rate	21.6 mg/cycle
Fuel/Air Equivalence Ratio [ϕ]	0.91 to 0.45
Intake Tank Pressure	47 to 96 kPa
Target CA50	5°CA
Coolant Temperature	75°C

Table 5. Available data sets where ϕ was varied.

Data set	Ignition Type	Intake Air Temperature	Imaging
RS	Inductive	30°C	✓
RS Heated	Inductive	100°C	
MP	Multi-pulse	30°C	✓
MP Heated	Multi-pulse	100°C	

Table 5 lists the four data sets acquired with the all-metal configuration of the engine. RS stands for 'regular spark', and MP for 'multi-pulse transient plasma'. For each ignition system, two different intake-air temperatures were used. In addition, the table shows that high-speed imaging results are available for unheated operation for each ignition system.

Combustion Stability Comparison

Figure 7 shows how the IMEP_n instability varies with ϕ for the four data sets. For all data sets, cycle-to-cycle variations of IMEP_n increase with decreasing ϕ to the point of being unacceptable. For this comparison, a 5% limit was chosen.

Figure 7 shows that both intake-air heating and multi-pulse ignition are effective at shifting the stability limit to leaner operation. The combination of intake heat and multi-pulse ignition achieves the leanest stable operation. These results motivate a detailed examination of the data sets to understand the physics responsible for these trends.

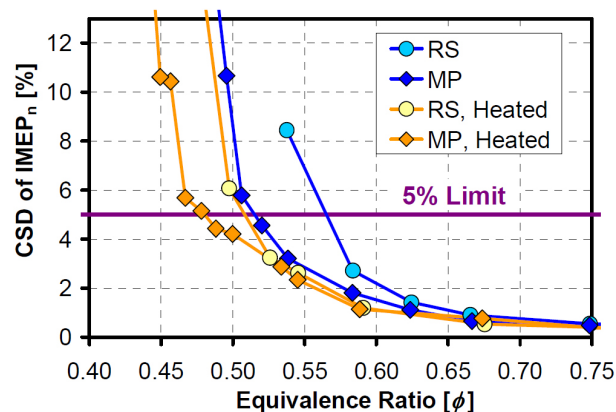


Figure 7. IMEP_n instability as a function of ϕ for the four data sets.

Regular Spark, Unheated

Figure 8b shows how IMEP_n varies with ϕ for operation with regular spark and unheated intake air. IMEP_n rises with a reduction of ϕ in the 0.90 to 0.67 range. It should be noted that no data exist for stoichiometric operation with $\phi = 1$, despite the intent to acquire such data. This is the case because of unintended back-flow of a newly installed by-pass valve, as explained in detail in Appendix A. Because of this, IMEP_n for stoichiometric operation was estimated using extrapolation of the three data points closest to stoichiometric operation, as illustrated in Fig. 8b.

Figure 8 shows that the highest IMEP_n is achieved for $\phi = 0.67$, corresponding to a 12% increase of both IMEP_n and fuel economy relative to stoichiometric operation. (More recently acquired data are presented in Appendix B to show that the extrapolation used for the current data sets leads to a conservative estimate of the fuel-economy improvement.) For even lower ϕ , IMEP_n drops, partly because of the onset of high IMEP_n variability.

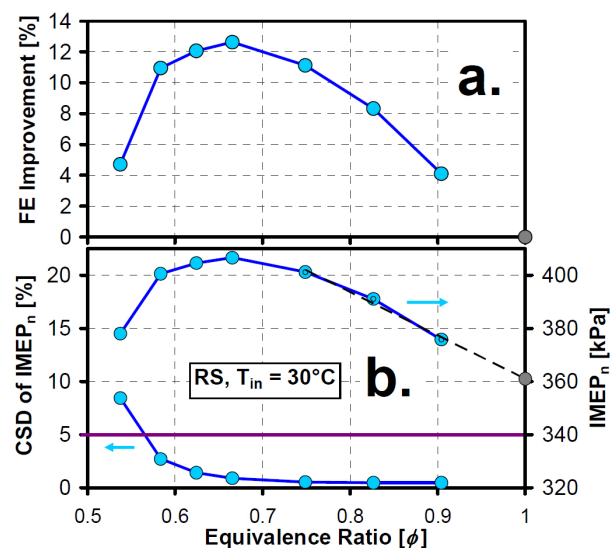


Figure 8. Effect of ϕ on IMEP_n, IMEP_n instability, and estimated fuel-economy improvement for operation with regular spark and unheated intake air.

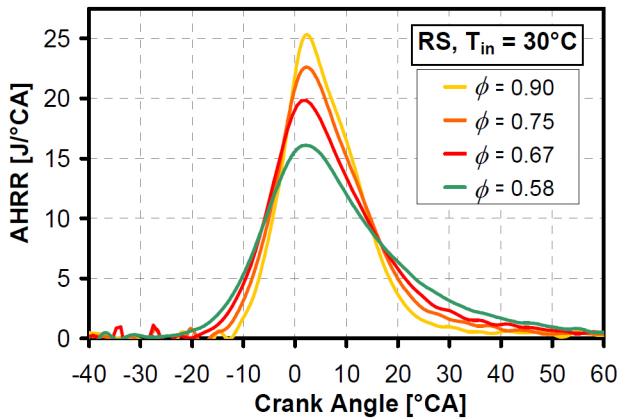


Figure 9. Ensemble-averaged AHRR for selected ϕ for operation with regular spark and unheated intake air.

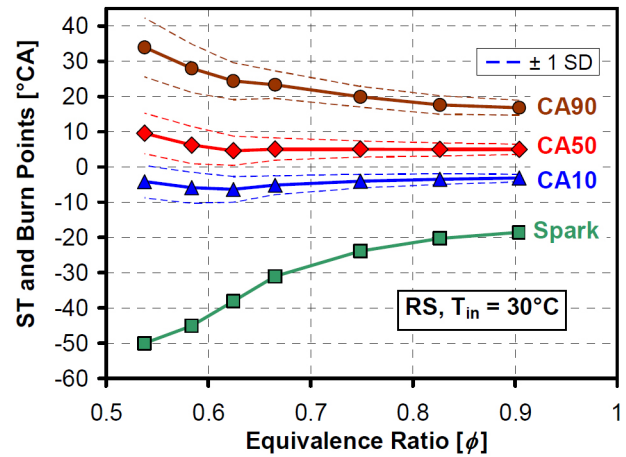


Figure 10. Spark timing and burn points as functions of ϕ for operation with regular spark and unheated intake air.

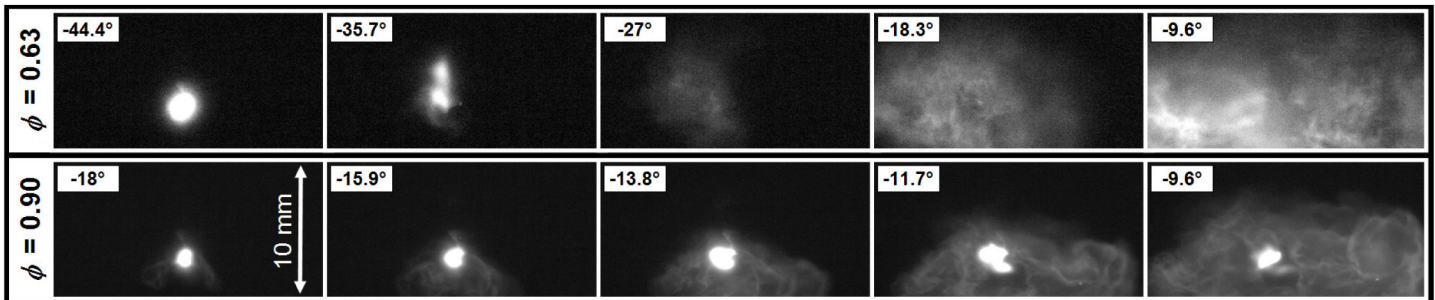


Figure 11. Flame imaging for operation with regular spark and unheated intake air. Camera lens f-stop = 1.4. $\phi = 0.63$: ST = -47.7°CA , relative image gain = 384. $\phi = 0.90$: ST = -21.0°CA , relative image gain = 128.

Figure 9 shows the ensemble-averaged AHRR for a selection of ϕ values. Peak AHRR decreases, and burn duration increases with a reduction of ϕ . The longer 10-90% burn duration is also evident in Fig. 10. However, even more distinct is the large increase of time required from spark to CA10. For $\phi = 0.90$, it takes 14°CA from start of spark discharge until CA10 has been reached. However, at $\phi = 0.63$, for example, the spark-to-CA10 duration is 32°CA . This large increase of spark-to-CA10 duration can be explained by the compounding effects of lower laminar flame speed at lower ϕ , and the lower compressed-gas temperature at earlier crank angles. (The latter point will be discussed further in the next section.) In Fig. 11, flame imaging illustrates this.

Figure 11 shows high-speed plasma and flame imaging at selected crank angles for operation at $\phi = 0.90$ and 0.63 . As noted in the caption, the effective image gain is higher for the leaner case to allow its weaker flame to be reproduced. As a result, the spark plasma shown at -44.4°CA appears brighter and more saturated. The spark timings noted in the caption are earlier than those plotted in Fig. 10 since the lower surface temperatures of skip-fired optical operation requires the use of an earlier spark to achieve the target CA50 = 5°CA . For $\phi =$

0.90 , the first image shown in Fig. 11 is for -18.0°CA . This is only 3.0°CA after the start of spark discharge. Even so, luminous flames have already spread at least 3 mm away from the spark-gap location. In contrast, the image at -35.7°CA for the leaner case shows that it takes 12°CA to achieve a similar flame-spread distance. Furthermore, Fig. 11 shows that the flame structure for $\phi = 0.90$ has distinct features, partly as a result of flame wrinkling due to turbulence. In contrast, the flame of the leaner case has a much more diffuse appearance, indicative of thicker flamelets at lower Damköhler numbers [15].

The very slow early flame spread eventually becomes a problem as the charge is leaned out, as evidenced by the increase of IMEP_n instability in Fig. 8b. This can be understood by considering that there are stochastic variations of the temperature, fuel, flow and turbulence fields. If the flame is weak, small deviations can lead to large changes in phasing of the main combustion.

Intake Air Heating

As shown above, lean operation can lead to very slow early flame spread. One potential way to speed up the transition from ignition to main combustion is to increase the charge