

NO_x-Reduction by Injection-Timing Retard in a Stratified-Charge DISI Engine using Gasoline and E85

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ABSTRACT

The lean-burn stratified-charge DISI engine has a strong potential for increased thermal efficiency compared to the traditional throttled SI engine. This experimental study of a spray-guided stratified-charge combustion system compares the engine response to injection-timing retard for gasoline and E85. Focus is on engine-out NO and soot, and combustion stability. The results show that for either fuel, injection-timing retard lowers the engine-out NO emissions. This is partly attributed to a combination of lower peak-combustion temperatures and shorter residence time at high temperatures, largely caused by a more retarded combustion phasing.

However, for the current conditions using a single-injection strategy, the potential of NO reduction with gasoline is limited by both elevated soot emissions and the occurrence of misfire cycles. In strong contrast, when E85 fuel is used, the combustion system responds very well to injection-timing retard. By using near-TDC injection of E85 ultra-low emissions of NO can be achieved. Despite the low NO emissions, both combustion efficiency and stability remain relatively high, but a mitigation of partial-burn cycles is desirable. One striking aspect of engine operation with near-TDC injection of E85 is the requirement and ability to ignite the head of one of the penetrating fuel jets. This early ignition results in a relatively short time delay between the end of the injection event and CA50. The closely-coupled injection and ignition lead to a combustion event that visually appears to have a finer structure, possibly indicative of a higher turbulence level during combustion. Furthermore, for gasoline it was not possible to operate with near-TDC injection to achieve very low NO emissions since gasoline required spark ignition at the end of the injection event. Modeling of fuel vaporization and mixing suggests that the gas-phase equivalence ratio is too high for gasoline to allow stable ignition of the head of one of the fuel jets. For E85, the model shows that the high heat of vaporization prevents the formation of ultra-rich gas-phase mixtures, which together with the high oxygen content of the fuel might explain the ability to ignite the E85 fuel jet during the injection event. Flame modeling in CHEMKIN shows that the strong vaporization cooling of ethanol suppresses the flame speed in rich areas; together with the high oxygen content of the fuel, suppressed flame activity in rich zones could be an important contributor to the low soot emissions that are realized despite highly stratified and mixing-controlled combustion for near-TDC fuel injection.

Direct comparison of stratified combustion with nearly identical heat-release rates for gasoline and E85 reveals 43% lower exhaust NO emissions for E85. This reduction of NO is attributed to two factors: 1) A reduction of the combustion-product temperatures due to strong vaporization cooling of ethanol in E85. 2) Increased post-combustion mixing rates caused by a more closely coupled injection and combustion event, and 52% more fuel injected due to the lower heating value of E85.

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INTRODUCTION

This study focuses on direct-injection spark-ignited (DISI), reciprocating engine operation with a spray-guided

(SG), stratified-charge (SC) combustion system, fueled with E85. The promise of this lean-burn technology is to achieve Diesel-engine fuel efficiency by unthrottled load control and low-temperature combustion, but the challenge is to meet

legislated emissions levels. Low engine-out NO_x and soot emissions must be achieved to avoid expensive aftertreatment technology. Therefore, this study focuses on the engine-out NO and soot, but also considers the need for high combustion efficiency (η_{comb}) in order to achieve high thermal efficiency. The concepts, benefits, and road blocks to implementation of SG-SC operation in a DISI engine have been reviewed by Ando and Arcoumanis [1]. Experimental and modeling techniques for the development of SG-SC and other advanced engines have been summarized by Drake and Haworth [2]. In this study, E85 (85% ethanol and 15 % gasoline) is of interest because it has been introduced to the market as a practical means to displace petroleum with a single-stream high-content biofuel, yet enabling cold starting not achievable with pure ethanol [3]. Compared to gasoline, ethanol has a lower heating value, higher latent heat, and high O_2 content. Thus it requires considerably more fuel mass injected (thereby changing stratification), cools the charge more, and has different reaction chemistry. Ethanol blends can have strong and beneficial knock-suppression effects that allow homogeneous operation with high intake boost pressure, especially for DI engines [3,4]. Effects of ethanol on stratified combustion have been investigated by various researchers, e.g. [5,6]. Studying a SG DISI engine, de Francqueville [7] found that E85 could achieve 35% reduction of NO_x emissions, although these emission values remained very high, supposedly because no EGR dilution was used. Stronger NO_x reduction using E85 was demonstrated by Oh *et al.* [8]. They showed that the use of E85 fuel allowed more fuel-injection retard, and this strongly reduced the NO_x level down to 180 ppm at an IMEP of 360 kPa.

In a recent study by the current authors [9], operation with ultra-low NO and soot emissions was demonstrated using E85. With a start of injection (SOI) of -6°CA and $\text{IMEP}_n = 260$ kPa, it was possible to achieve mandated engine out NO and soot levels, simultaneously. For such a late SOI, E85 required a spark timing before fuel injection for highest combustion stability. High-speed imaging showed strong interactions between the spark plasma and one of the developing fuel jets, creating large areas of weak flame luminosity prior to onset of the main heat release. Using gasoline, it was not possible to operate with near-TDC injection and achieve ultra-low NO levels. This was attributed to gasoline's inability to accept a spark before or during the injection event, but reasons for this were unclear. The study raised multiple questions related to gasoline versus E85 when using near-TDC fuel injection to achieve stratified charge combustion:

A. How is it possible to simultaneously achieve low NO and soot levels with E85, despite the non-premixed nature of the combustion event?

B. Why does E85 allow/require the start of spark discharge to occur before fuel injection for the late-combustion phasing, low NO operation?

C. Why does gasoline require ignition after the passage of the liquid spray and not allow the use of near-TDC fuel injection to achieve ultra-low NO emissions?

D. What are the causes for occurrence of partial-burn cycles for operation with late injection?

This follow-on study will not be able to provide complete answers to all of these questions, but aims to provide more insights than Ref. [9] using systematic changes in operation, empirical combustion analysis, and thermal and chemical modeling. The objective of this paper is to systematically examine the combined effects of injection-timing retard, spark timing, and N_2 dilution on emissions and combustion stability for gasoline and E85. The results are divided into five main parts:

1. An examination of the required spark timing (ST) relative to the fuel-injection event for gasoline and E85, both at a typical SOI of -31°CA .

2. The effects of SOI-retard for both gasoline and E85. This is the main section of the paper.

3. A comparison of the flame structure for E85 operation with various degrees of stratification, using high-speed flame imaging.

4. A discussion of some of the factors, like mixture stoichiometry and vaporization cooling in the fuel jet, that contribute to enable clean and stable operation with near-TDC injection of E85.

5. An examination of the EGR-induced trade-off between NO emissions and combustion stability for late SOI of E85.

Preceding the results are descriptions of the experimental setups, data acquisition and analysis. This is followed by a description of the models used for fuel-jet and flame-speed calculations.

EXPERIMENTAL SETUP

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 piston and head 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. Figure 1 shows a cross section of the combustion chamber and Fig. 2 shows a picture of the piston top. The piston bowl has a slight cut-out on one side to facilitate a better view into the bowl when the pent-roof side-window on the right-hand side in Fig. 1 is used. Engine data and operating conditions are given in Table 1. The engine was fueled using an 8-hole injector and the distance from the injector tip to spark-plug gap was 13 mm. The injector was oriented such that two of the fuel sprays straddled the spark-plug gap. For all data, except for the images of well-mixed operation in Figs. 16a and one of the

intake valves was deactivated to increase in-cylinder swirl and tumble levels, see [Table 1](#).

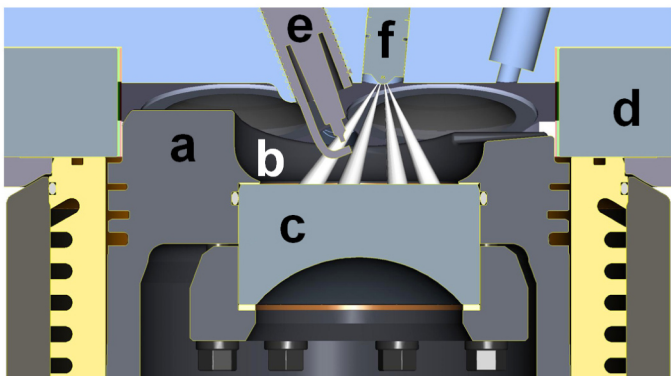


Figure 1. CAD rendering of cross-section of combustion chamber at TDC with a - piston, b - piston bowl, c - piston-bowl window, d - pent-roof window, e - spark plug, and f - fuel injector.



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

N₂ dilution was used to lower the intake mole fraction of oxygen [O₂] to simulate the dilution effects of EGR. This charge dilution is important for controlling NO_x emissions. The lowest intake [O₂] being used here (16.5%) is equivalent to that with 40% external exhaust gas recirculation, but of course does not include the fluid properties of real EGR. Air and N₂ were metered with sonic nozzles. Exhaust gas emissions were measured using a Horiba MEXA-584L emission analyzer. NO₂ was not measured, and its contribution to engine-out NO_x has been neglected. The NO sensor was calibrated at 70 or 1000 ppm, depending on the exhaust concentration to be measured. Exhaust smoke was measured with an AVL 415S smoke meter, and reported as

soot mass concentration using its internal correlation between paper blackening and soot concentration.

Table 1. Engine Specifications and Operating Conditions

Displacement.....	0.552 liters
Bore / Stroke	86.0 / 95.1 mm
Geometric Compression Ratio	12:1
Swirl / Tumble Index.....	0 / 0.27
Swirl / Tumble Index (one valve deactivated)	2.7 / 0.62
Fuel Injector.....	Bosch 8-hole w/ 60° included angle
Hole Size	Stepped-hole, min. dia. = 0.125 mm
Injection Pressure	170 bar
Spark Coil / Energy	Bosch, ZS-L 1x1E / 106mJ
Engine Speed.....	1000 rpm
Intake Temperature	26-28°C
Intake Pressure (abs.).....	95 kPa
Coolant Temperature	60°C

For each operating point with the all-metal engine configuration, the in-cylinder pressure, intake and exhaust pressure, and fuel pressure were acquired for 500 consecutive cycles. For some conditions, also spark current was measured. The apparent heat-release rate (AHRR) was computed from the in-cylinder pressure for each individual cycle using a constant ratio of specific heats (γ) following [10].

For optical diagnostics, the engine was equipped with a quartz window in the piston bowl, see [Fig. 1c](#). This window had a concave ($R = 25\text{mm}$) lower surface to enable a wide-angle view into the combustion chamber. High-speed imaging through the piston-bowl window was accomplished using a monochrome Phantom v710 camera via a 45° Bowditch mirror and a 180mm camera lens, collecting broadband luminosity¹. Liquid fuel was visualized using a high-intensity LED with 2 or 5 μs pulse length², with the light entering via a quartz window installed in the end of the pent-roof section closest to the injector, see [Fig. 1d](#). In this optical configuration, the engine was operated identically to the pre-established all-metal operating points, but with a shorter run duration using a Fire3-Skip9 sequence to achieve a 25% duty cycle and thereby reduce the heat load on the non-cooled quartz windows. All images were acquired on the 3rd fired cycle of each sequence to ensure proper temperature and composition of the residual gases. To limit run times, typically only 100 cycles were acquired for each optical operating point.

Fuels

This study uses gasoline and E85. The gasoline is Chevron-Phillips research-grade gasoline with an antiknock index of 87. It is a typical gasoline and its specifications are given in [Table 2](#). The E85 fuel was prepared in-house by

¹Only for the well-mixed operation in [Fig. 16a](#), a short-wavelength pass filter with a cut-off wavelength of 650 nm was used to suppress thermal emission from O₂ and H₂O.

²An LED pulse length of 2 μs was used for all SOI = -31°C imaging, and 5 μs for all SOI = -6°C imaging.

blending the research-grade gasoline with anhydrous high-purity grade ethanol in 15%/85% proportions by volume.

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

* Based on analysis provided by Paragon Labs, Inc.

MODELING SETUP

Fuel-Jet Model

Following the works by Naber and Siebers [11,12] and assuming adiabatic mixing-controlled fuel vaporization, supporting computations of fuel/air mixing and vaporization were performed [13]. The analytical model describes penetration based on conservation of mass and momentum for a steady fuel injection rate into a developing control volume. For simplicity, the velocity and mass distributions within a conical envelope for the jet were assumed radially uniform at any axial distance from the injector, X . These computations used ethanol as a surrogate for E85 and iso-octane as a surrogate for gasoline, since the thermodynamic properties of gasoline were not accurately known. Following a fluid parcel from the injector tip to the surrounding ambient, the model first considers fuel vaporization and the mixing of liquid fuel, vaporized fuel, and ambient gas. No consideration is given in the spray model to atomization processes or droplets. Instead, this idealized model treats the spray as a locally homogeneous flow where the transport rates between phases at droplet surfaces are fast relative to the transport rates as a result of the mixing processes in the spray. After the location where all liquid has vaporized, the mixing process is purely in gas-phase and continues without combustion until the fluid parcel equals the ambient gas.

Flame-Speed Model

For better understanding of E85 combustion, supporting calculations of the laminar flame speed were performed using CHEMKIN-PRO [14]. The model assumes adiabatic conditions and neglects radiative heat transfer. The E85 fuel was approximated with ethanol, for which a validated and detailed reaction mechanism was available, as described in Ref. [15]. The air/ N_2 dilution-sweep was modeled by varying the proportions of O_2 and N_2 of the reactants while not including CO_2 and H_2O . The temperatures of the reactants at various fuel/air-equivalence ratios (ϕ) were extracted from the fuel-jet model to accurately account for vaporization cooling effects.

RESULTS

For all data in this study, the engine was operated in a manner that provides relatively low in-cylinder gas temperatures. First, the temperature of the fresh gases was close to room temperature. Second, the valve timings provide low residual levels (≈ 5 -6% by mass). Third, a somewhat late intake valve closing makes the effective CR ≈ 11 . Fourth, a coolant temperature of 60°C was selected for the all-metal tests. This lower-than-normal coolant temperature allows the use of elevated coolant temperature for the skip-fired optical tests to achieve similar in-cylinder surface temperature as for the all-metal tests.

For the data presented below, the spark timing (ST) was adjusted for each operating point to maximize engine stability (minimize the standard deviation of IMEP_n), unless explicitly noted otherwise. The in-cylinder high-swirl flow, accomplished by deactivating one of the intake valves, reduced the cyclic variability of IMEP. This is presumed to be the result of a more consistent flow structure from cycle to cycle thanks to the more directed intake flow [16].

In this study, the crank angles are referenced as after top dead center of the combustion stroke, aTDC. The start of injection (SOI) is the crank angle when the first liquid enters the combustion chamber, as determined by Mie-imaging via either the piston-bowl window or the pent-roof side window. Likewise, end-of-injection EOI is when the last liquid leaves the injector, as determined by Mie-scatter imaging via the side window. Furthermore, to help the interpretation of spark-spray interactions it should be noted that, at the current engine speed of 1000 rpm, the liquid fuel tip requires 1°CA to travel from the injector to the spark-gap area 13 mm downstream. Hence, if the results show that the SOI-to-ST delay = 1°CA, this means that the spark discharge starts exactly when the first liquid fuel arrives near the spark gap. It should be noted that the duration of spark discharge is about 6°CA, as discussed further below.

In the following, the effects of injection-timing retard will be systematically examined. As will be clear in the following sections, the ST relative to the fuel-injection event is critically important for stable operation, in particular for

operation with near-TDC injection. Therefore, the spark timing effects are examined first.

Spark-Timing Window for E85 and Gasoline

This section compares the required ST for both gasoline and E85 at a common SOI of -31°CA , where the engine would operate well with both fuels near 375 kPa IMEP_n. Figure 3 shows how IMEP_n and the corrected standard deviation of IMEP_n (CSD of IMEP_n) varies with ST relative SOI.

The corrected standard deviation of IMEP_n is defined as:

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

Here, the standard deviation (σ) of IMEP_n for fired operation has been corrected for the heat transfer losses of a motored cycle by being normalized by the difference in average IMEP_n for fired and motored operation. The difference in IMEP_n for fired and motored operation equals the IMEP_n rise caused by the combustion event. This definition allows the combustion stability to be consistently evaluated even for operation with IMEP_n near zero or negative, (e.g. idle operation or vehicle coasting.)

Figure 3 shows that the spark-timing window with high IMEP_n and low CSD of IMEP_n is roughly 3°CA wide for gasoline, and 4°CA wide for E85. The width of the spark-timing window is of course dependent on the amount of N_2 dilution used to control NO emissions. Since the dilution levels were somewhat different for the two fuels, as required for the most stable engine operation, the IMEP_n variations in Fig. 3 are not due to ST alone. However, for each fuel, the location of the ST window does not change much with dilution level, so Fig. 3 exemplifies how the window of stable ST shifts with fuel type. The x-axis shows the delay of ST relative to SOI. Recall that a delay of 1°CA is coincident with the arrival of the fuel at the spark-plug gap and that the spark-plasma duration is substantial, as shown below.

Figure 4 gives one example of the spark-current duration, for ST = -27°CA . The spark-current transition from negative to positive (truncated at -400mA) corresponds to the beginning of the plasma discharge. The plasma breakdown occurs within $40\mu\text{s}$ (0.24°CA) of ST, but is not resolved here; only the arc and glow phases (regions with positive current in Fig. 4) are resolved. Figure 4 compares the ensemble-averaged spark current for operation with and without fuel injection. Without fuel, the spark current duration is more than 12°CA long. However, with fuel injection the spark discharge is substantially shorter ($\sim 6^\circ\text{CA}$) and has a higher peak current.

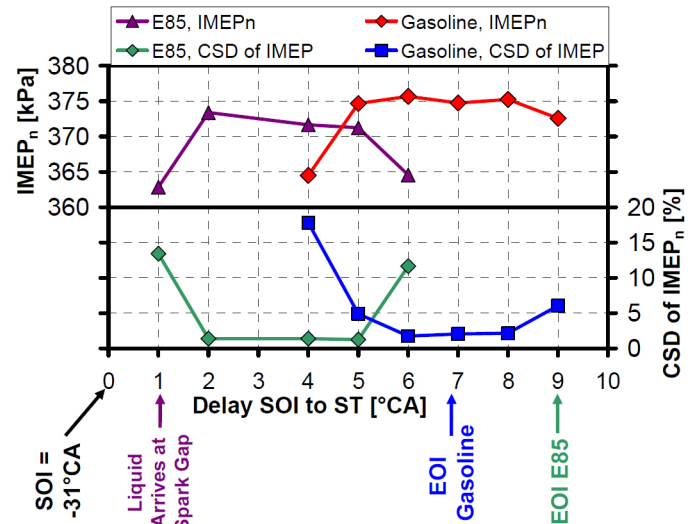


Figure 3. Results from spark sweeps for SOI = -31°CA . Intake $[\text{O}_2]$ = 16.5% for gasoline, and 18% for E85.

Two of the vertical arrows in Fig. 3 represent the end of injection (EOI) for gasoline and E85, thus revealing the timing of the early portion of the spark plasma relative to the beginning and end of the liquid spray near the spark plug gap. For gasoline, the most stable combustion is achieved when ST is near EOI. Figure 5a shows an example of the relatively small cyclic variations of IMEP_n for the optimal ST. However, if the ST is advanced from the optimum, misfire cycles start to appear, as shown in Fig. 5b. Apparently, for some cycles, the combination of ϕ and flow conditions is such that no robust flame kernels are produced. If the ST is retarded to after EOI, Fig. 5c shows that both partial burn and misfire cycles appear.

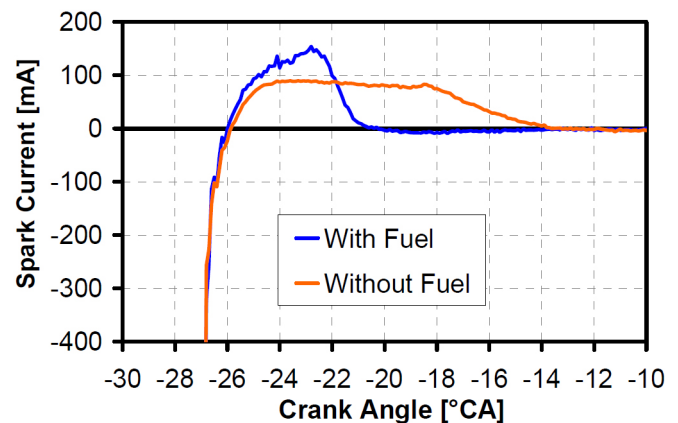


Figure 4. Example of spark current with and without fuel injection, ensemble averaged over 114 cycles. ST = -27°CA , E85 with SOI = -31°CA corresponding to Fig. 16b.

These results for gasoline are consistent with those of Drake *et al.* [17], who observed that very early spark resulted in misfire cycles, and that late spark resulted in both partial burns and misfire cycles. The spark timing relative to EOI for which partial-burn cycles start to occur is nearly identical to that observed by Peterson *et al.* [18] in a similar spray-guided DISI stratified charge configuration. In that high-speed imaging study, it was shown that for operation with late ST a flame kernel was formed even for partial-burn cycles, but it failed to develop properly to consume the fuel in the piston bowl.

For E85, operation with too-early ST also results in misfire cycles, as Fig. 6b shows. The same is true if ST is late, but note that no partial burn cycles are detected. This indicates that the ignition and flame-kernel propagation is different between gasoline and E85. However, a stronger indicator of differences between the fuels is the shift in optimal ST window relative to SOI, as Fig. 3 shows. Note that the injection duration is longer for E85 in order to maintain a similar IMEP_n for both fuels. This happens because the heating value of E85 is substantially lower than gasoline, therefore requiring 52% higher fuel mass. Factoring in the longer injection duration, Fig. 3 shows that the most stable E85 combustion is achieved when the ST is in the first half or in the middle of the injection event. In contrast, gasoline requires ST near EOI as discussed above and in Refs. [17,18]. The effects of using such early ST with E85 will be discussed further below. Figure 3 serves as an example of the available ST window for a given SOI. As SOI is retarded, the relative ST window changes progressively. Therefore, for the data acquired at various SOI, ST had to be adjusted in order to maintain lowest possible cyclic variability of IMEP_n, and to avoid occurrence of misfire cycles.

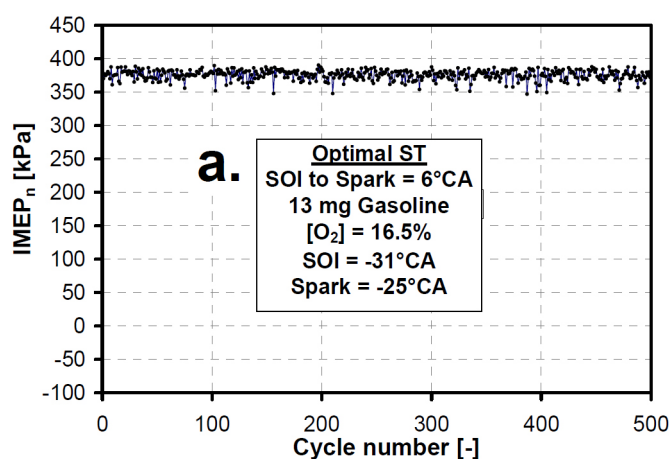


Figure 5.

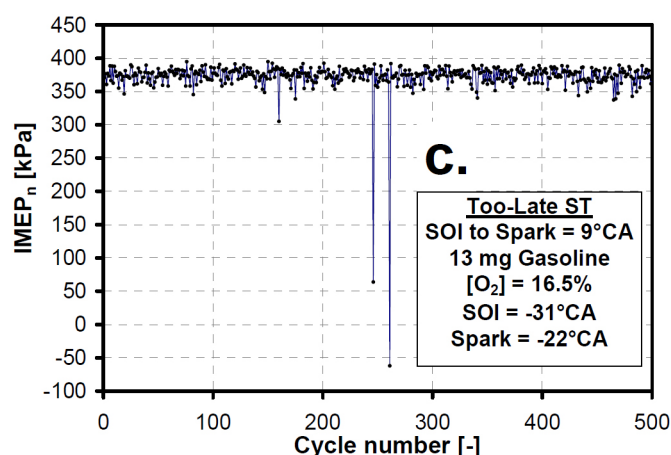
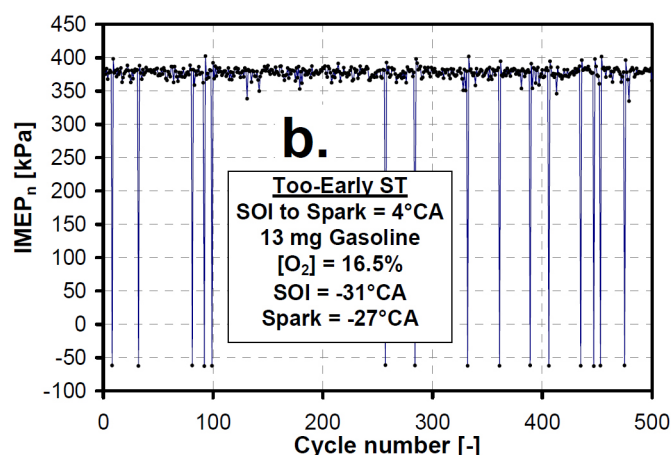


Figure 5. (cont.) IMEP variations for gasoline operation with a) optimal spark timings relative SOI, b) too-early spark, and c) too-late spark.

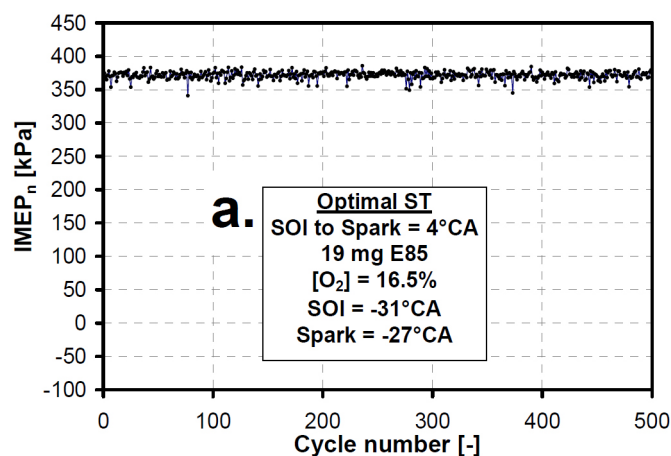


Figure 6. IMEP variations for E85 operation with a) optimal spark timings relative to SOI, b) too-early spark, and c) too-late spark.