

Combined Effects of Fuel and Dilution Type on Efficiency Gains of Lean Well-Mixed DISI Engine Operation with Enhanced Ignition and Intake Heating for Enabling Mixed-Mode Combustion

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ABSTRACT

Well-mixed lean or dilute SI engine operation can provide efficiency improvements relative to that of traditional well-mixed stoichiometric SI operation. However, the realized gains depend on the ability to ensure stable, complete and fast combustion. In this work, the influence of fuel type is examined for gasoline, E30 and E85. Several enabling techniques are compared. For enhanced ignition stability, a multi-pulse (MP) transient plasma ignition system is compared to a conventional high-energy inductive spark ignition system. Combined effects of fuel type and intake-gas preheating are examined. Also, the effects of dilution type (air or N₂-simulated EGR) on lean efficiency gains and stability limits are clarified. The largest efficiency improvement is found for lean gasoline operation using intake preheating, showing the equivalent of a 20% fuel-economy gain relative to traditional non-dilute stoichiometric operation. The reason for gasoline's larger efficiency improvement is its lower octane number which facilitates the use of end-gas autoignition to produce mixed-mode combustion. For these conditions, such mixed-mode combustion is required for rapid completion of the inherently slow lean combustion event prior to piston expansion. The fuel-economy gains are somewhat smaller for both E30 and E85 because of higher resistance to end-gas autoignition under lean conditions.

To avoid knocking cycles when mixed-mode combustion is used, the deflagration-based combustion must be very repeatable to ensure consistent compression of the end-gas reactants. Multi-pulse transient plasma ignition is used beneficially to stabilize the combustion, especially for dilute operation which suffers from low flame speeds. However, even with an enhanced ignition system, the best fuel-economy gains of dilute stoichiometric operation with mixed-mode combustion are on the order of 11-12%, which is substantially less than for lean operation.

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INTRODUCTION

Climate change [1] and the need to secure energy supplies are two reasons for a growing interest in engine efficiency and alternative fuels. Well-mixed stoichiometric operation is the dominating combustion mode for gasoline SI engines used for automotive applications. It allows cost-effective compliance with stringent exhaust emissions regulations by the application of a three-way catalyst for reduction of NO_x and oxidation of CO and HC [2]. However, there are several reasons why stoichiometric operation (without exhaust-gas recirculation, EGR) 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 the ratio of specific heats (γ) which leads to a reduction of 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].

Fuel-lean operation can improve engine efficiency by reducing all of these effects. However, the challenge is to maintain stable and efficient combustion despite a reduction of flame speeds in fuel-lean mixtures [5]. The reduction of flame speeds becomes a particular problem from the perspective of ignition and effective flame spread throughout the charge. A recent study demonstrates the combined benefits of intake gas heating and multi-pulse (MP) transient plasma-ignition on lean stability limits for a blended fuel of gasoline with 85% ethanol by volume (E85) [6]. Even a moderate 40K increase of the in-cylinder charge temperature at intake valve closing (IVC) provided a significant extension of the lean-stability limit from a fuel/air equivalence ratio (ϕ) of 0.56 down to 0.51. Using the MP ignition system, the lean-stability limit was extended further down to

$\phi = 0.49$. High-speed plasma and flame imaging revealed that the MP ignition system provided a much faster transition from ignition to fully developed turbulent combustion [6].

Although previous studies have provided useful information on lean-stability limits [2,5, 6, 7, 8, 9, 10, 11], systematic experiments among fuels and fuel blends in the same facility are needed to provide quantitative comparisons. Furthermore, stoichiometric operation with dilution is also of interest, because it can provide fuel-economy benefits while allowing the use of a cost-effective three-way catalyst for NO_x reduction. This study extends previous work [6], examining new data acquired with both 100% gasoline and a blend of gasoline with 30% ethanol by volume (E30), while including comparisons with existing data for E85. In addition to lean operation, the current study also examines N₂-diluted stoichiometric operation that simulates operation with EGR. The current study also offers new insights by systematically comparing engine operation with two different fuels at two intake temperatures and with two dilution types. In addition, the MP ignition system is used to probe the limits and characteristics of lean and dilute SI combustion beyond what is possible with conventional inductive spark systems.

There are two main objectives. First, to reveal effects of fuel type and octane number on the efficiency of lean or dilute SI operation. Second, to determine viable paths for stable and more efficient lean SI operation.

EXPERIMENTAL SETUP

Research Engine Facility

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 the all-metal configuration, using 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. Ref. [12]. 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 with the piston at TDC. The long-reach spark plug allows the spark plasma to develop in a central location of the combustion chamber, which should be particularly beneficial for lean operation. This long-reach spark plug is used with a high-energy inductive ignition system, as described below.

Engine data are given in Table 1. The engine was fueled using a Bosch 8-hole step-hole VCO injector. The injector was oriented such that two of the fuel sprays straddled the spark-plug gap of the commercial long-reach spark plug. For all data, the phasings of the cam shafts relative to the crank shaft were maintained constant. The cam shafts were phased to provide both low residual levels and high volumetric efficiency. A graphical representation of the valve lifts can

be found in Ref. [6]. It should be noted that one intake valve 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.

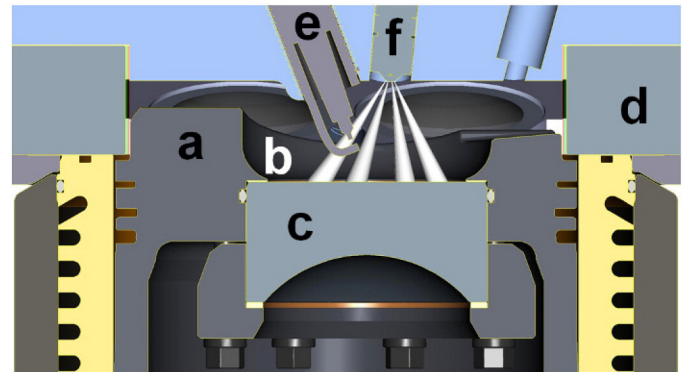


Figure 1. Cross-section of combustion chamber at TDC: a - piston, b - piston bowl, c - piston-bowl window, d - pent-roof window, e - spark plug, and f - fuel injector. Metal window blanks were used in this study.

The air was metered into the intake system using a sonic-flow nozzle. For the stoichiometric dilute test cases, N₂ was also metered with another sonic-flow nozzle to simulate the reduction of oxygen concentration associated with the use of EGR.

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

In discussions that involve only lean operating points, the traditional fuel/air-equivalence ratio (ϕ) is used to quantify the stoichiometry of the supplied reactants. To aid comparisons of lean and dilute data sets, a mass-based equivalence ratio, ϕ_m , is used:

$$\phi_m \equiv \frac{(F/C)_{Actual}}{(F/A)_{Stoichiometric}} \quad (1)$$

where F = Fuel mass, A = Air mass, and C = gas Charge mass.

In this study with simulated EGR, the gas charge mass equals that of air and N_2 , ($C = A + N_2$ mass). ϕ_m is a measure of the chemical energy per reactant mass, regardless of type of diluent. Hence, the use of ϕ_m allows plotting results from both lean and dilute stoichiometric data sets against the same x-axis. It should be noted that when only air is used as a diluent, $\phi = \phi_m$.

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 and to lower the dew point of the exhaust gases to avoid loss of unburned hydrocarbons (HC) and water due to condensation in the unheated sample lines. With precisely known exhaust dilution, the measured exhaust composition was back-corrected to engine-out mole fractions. A Horiba MEXA-584L emission analyzer measured CO , CO_2 and HC . O_2 was measured with 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.

To compute a metric of combustion efficiency (η_{comb}), the measured mole fractions of CO and HC were converted to $mg/cycle$ and multiplied with the respective specific lower heating values (LHV):

$$\eta_{comb} [\%] = 100 \cdot \frac{(m_{\text{Injected-Fuel}} - m_{HC}) \cdot LHV_{\text{Fuel}} - m_{CO} \cdot LHV_{CO}}{m_{\text{Injected-Fuel}} \cdot LHV_{\text{Fuel}}} \quad (2)$$

Here, it is assumed that the LHV of the unburned HC equals that of the injected fuel. Further, the contribution from H_2 to combustion inefficiency is neglected since the level of H_2 in the exhaust was not measured. However, the error introduced is small since H_2 emissions only become significant for rich combustion [4]. In this study, focus is on stoichiometric or lean operation.

Ignition Systems

Two ignition systems were used as described in the following.

Regular Inductive System

The conventional 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 nominally 106 mJ being delivered from the secondary high-voltage coil. 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. 2b (bottom).

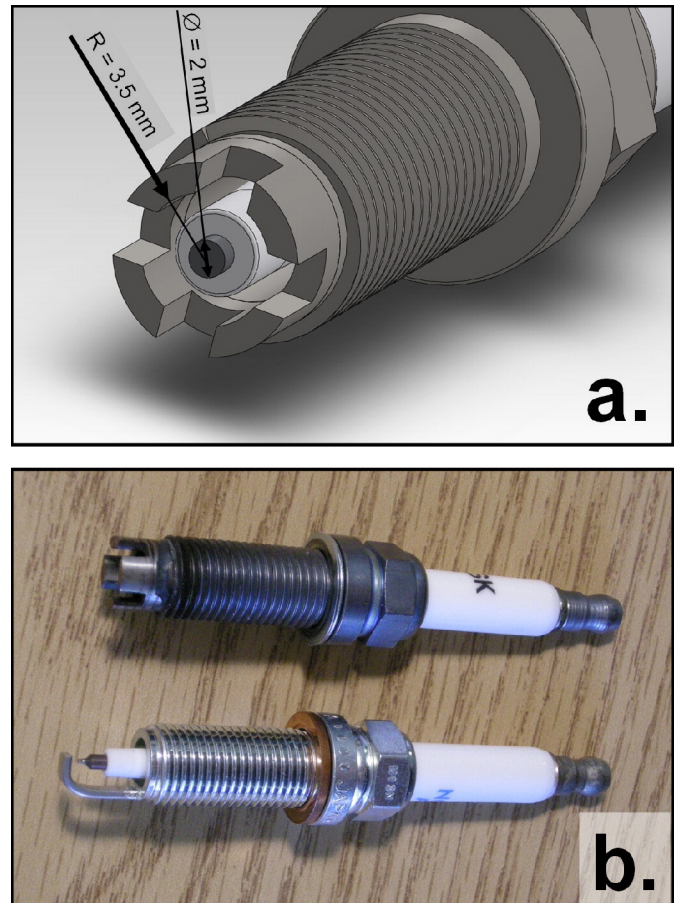


Figure 2. a) Rendering of the custom spark plug with dimensions of the semi-open cavity. b) Photo of the commercial long-reach spark plug (bottom) and the in-house modified spark plug (top).

Enhanced Ignition System

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 2b shows a photo of the custom spark plug (top) in comparison with the commercially available long-reach spark plug (bottom). Figure 2a 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.

The pulse generator used in this experiment is nearly identical to the one used in recent work with E85 fuel [6]. It is based on solid-state electronics and uses magnetic pulse compression to shape the output pulse. Based on current and voltage measurements across a 200 Ω resistor, it is capable of delivering a 40 kV 12 ns (FWHM) pulse with a maximum energy of 40 mJ per pulse. The voltage amplitude is scalable while maintaining the same pulse width down to an output voltage of 15 kV. If the amplitude is sufficiently low for a given gas density, only primary corona streamers are generated [13]. However, to provide effective ignition for lean conditions, higher-amplitude pulses had to be used, but still with a 12ns FWHM duration. The higher amplitude leads to breakdown of the gas and formation of a

spark with a very short time duration. Since the spark gap is a more complex load than a resistor, the exact amount of energy delivered to the spark plug is not known for these engine experiments.

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 time domain, with the 10-pulse burst spanning 0.9 ms in total. Because of the very short pulse duration, each arc is terminated before it enters its glow phase. Therefore, the name “multi-pulse (MP) transient plasma ignition” was introduced in the recent lean-burn study [6].

Experimental Procedure and Data Analysis

For each operating point with the all-metal engine configuration, the engine is allowed to run for several minutes until all measured parameters are stable, at which point data are acquired. Fuel-flow rate and thermocouple readouts are averaged over one minute. The in-cylinder pressure, spark current for the regular inductive system, intake and exhaust pressure, and fuel pressure are acquired for 500 consecutive cycles using 0.1°C resolution. For the in-cylinder pressure, an uncooled Kistler 6125C piezoelectric sensor is used in combination with a Kistler 5010B charge amplifier. Day-to-day repeatability is 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. For all data presented, except for the graphs in Figs. 17 & 20, 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 η_{comb} . For the burn-point-based AHRR profiles in Figs. 17 & 20, the final burn point for each cycle varies according to an estimate of the combustion efficiency for each cycle. The latter is computed from the integral of the AHRR and the average η_{comb} derived from exhaust-emissions measurements. This refined technique allows examining on a per-cycle basis how changes to the AHRR profile affect the combustion efficiency.

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:

$$\text{CSD of IMEP [\%]} = 100 \cdot \frac{SD_{\text{IMEP}_{\text{fired}}}}{(\text{IMEP}_{\text{fired}} - \text{IMEP}_{\text{motored}})} \quad (3)$$

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 mass-averaged temperature at the time of spark are computed using the ideal-gas law in combination with the measured pressure (averaged over 500 cycles), the known cylinder volume, and the trapped mass. The trapped mass equals the sum of the supplied air, N₂, fuel, and residuals. The average molecular weight used in the ideal-gas law corresponds to the mixture of fresh charge and residuals. As such, the computed temperatures correspond to the mass-averaged temperature of a single-zone model, ignoring potential thermal stratification in the combustion chamber. To guide the understanding of end-gas autoignition, estimated end-gas reactant temperature ($T_{\text{end-gas}}$) is reported as well. It is computed using Eq. 4, with the assumption of isentropic (no heat release) and adiabatic (no heat transfer) compression by the combustion-induced pressure rise that occurs after the spark timing (ST).

$$T_{\text{end-gas}} = T_{\text{avg@ST}} \cdot \left(\frac{P_{\text{actual}}}{P_{\text{ST}}} \right)^{\left(\frac{\gamma - 0.07 - 1}{\gamma - 0.07} \right)} \quad (4)$$

$T_{\text{avg@ST}}$ and P_{ST} correspond to the mass-averaged gas temperature and pressure at the time of spark, respectively. (Note that ST is quite early, -57 to -38°C, for the operating points where $T_{\text{end-gas}}$ is plotted. Therefore, heat transfer does not affect $T_{\text{avg@ST}}$ substantially for those operating points.) P_{actual} refers to the pressure at any crank angle after ST. γ represents the ratio of specific heats. For each operating point, γ is determined from the measured rise of pressure and computed rise of mass-averaged gas temperature that occur between IVC and ST. The subtraction of 0.07 from γ in Eq. 4 is performed to compensate for the reduction of γ that occurs as the gas temperature keeps rising after ST. To allow estimating the accuracy of this method, Appendix A shows a comparison with a more elaborate method to compute the end-gas compression heating based on the thermodynamic library built into CHEMKIN-PRO. It should be noted that heat release due to autoignition reactions will cause the actual temperature of the end-gas reactants to be higher than $T_{\text{end-gas}}$, but a treatment of this is the beyond the scope of the current study.

Fuels

This study is focused on a comparison of gasoline and E30. It also includes one data point using E85. [Table 2](#) provides a comparison of octane numbers.

[Table 2.](#) Measured octane numbers for gasoline. Octane numbers for E30 and E85 are estimated based on ethanol-blending trends in Ref. [15].

Fuel	Gasoline	E30	E85
AKI	92.7	98	100
RON	96.6	104	108
MON	88.7	91	91

The gasoline was supplied by Haltermann Solutions as an exhaust-emissions and fuel-economy certification fuel for the US market. Key properties are listed in [Table 3](#).

[Table 3.](#) Specifications of Haltermann certification gasoline

Density [kg/l]	0.743
Carbon, wt%	86.7
Hydrogen, wt%	13.3
Oxygen, wt%	none detected
A/F Stoichiometric	14.54
Lower Heating Value, gas-phase fuel [MJ/kg]	42.57
LHV for stoichiometric charge [MJ/kg]	2.738
Hydrocarbon Type, vol%	
Aromatics	32.7
Branched Alkanes	55.0
Linear Alkanes	7.5
Cyclic Alkanes	2.9
Alkenes	0.07
Not Classified	1.9
Distillation, °C	
5%	43
10%	51
30%	75
50%	103
70%	116
90%	158
95%	171

The E30 fuel was prepared in-house by blending the Haltermann certification gasoline with anhydrous high-purity ethanol in 70%/30% proportions by volume. With the assumption of a linear variation of properties upon blending, some estimated properties of the E30 fuel are given in [Table 4](#).

The E85 fuel was also prepared in-house, but in 15%/85% proportions by volume. However, the gasoline fuel used for the E85 blending is a Chevron-Phillips research-grade gasoline, which has a slightly lower antiknock index of 87 [6].

[Table 4.](#) Estimated properties of E30 fuel blended from the high-octane certification gasoline.

Specific gravity	0.757
A/F Stoichiometric	12.81
Lower Heating Value, gas-phase fuel (MJ/kg)	37.9
LHV for stoichiometric charge (MJ/kg)	2.747
Hydrocarbon Type, vol%	
Oxygenates	30.0
Aromatics	22.9
Branched Alkanes	38.5
Linear Alkanes	5.2
Cyclic Alkanes	2.0
Alkenes	0.05
Not Classified	1.4

RESULTS

These tests employ swirling in-cylinder flow for consistency with a recent lean-burn study in the same engine [6]. The swirling flow is accomplished by deactivating one of the intake valves, as noted previously in conjunction with [Table 1](#).

[Table 5](#) shows the operating conditions that are common to all ϕ_m -sweeps that use the regular spark (RS) system. Fuel was injected during the intake stroke using three injections of equal duration to create a well-mixed charge of fuel, air, N₂ and residuals. Moderate adjustments to the injection timing between data points were performed for E30 to achieve low soot emissions while maximizing the available time for fuel-air mixing. It was found that nearstoichiometric data points for E30 required SOI = -300°C A to keep the smoke levels near zero. On the other hand, the lean-stability limits improved with an earlier SOI = -330°C A. In contrast, gasoline operation did not require such SOI adjustments to maintain low smoke, so SOI = -330°C A was used throughout. All highly lean or dilute operating points were operated with SOI = -330°C A, so the lean and dilute stability limits can be compared directly between fuels.

The selected valve timings result in low residual levels (≈ 46 kPa 45 kPa 4-6% by mass). For each fuel, the fuel mass per cycle was first established for stoichiometric operation without N₂ dilution. The target was to achieve a net indicated mean effective pressure (IMEP_n) of 370 kPa. For subsequent data points at other ϕ_m , this fuel mass was held constant. Stoichiometric operation is highly throttled due to the relatively low load, with an absolute intake pressure (P_{in}) of only 45-46 kPa. As the charge is leaned out by metering more air into the intake system, the intake pressure rises. For the leanest operating points, P_{in} reaches 93 kPa.

Effect of Combustion Phasing for Lean E30 Operation

The selection of spark timing (ST) and combustion phasing are critically important for the engine performance. This section discusses how spark-timing and combustion-phasing targets shift with changes of ϕ .

Table 5. Operating conditions for ϕ_m -sweeps with regular spark system.

Injection Pressure	170 bar	
# of Fuel Injections	3	
Δ CA between SOIs of Fuel Injections	15°CA	
Spark Energy	106 mJ	
Coolant Temperature	75°C	
IMEPn @ $\phi = 1$ without dilution	370 kPa	
Fuel	Gasoline	E30
Fuel per Cycle	15.6 mg	17.8 mg
Intake Pressure @ $\phi = 1$ (unheated)	46 kPa	45 kPa
Intake Pressure Range	37 - 93 kPa	45 - 89 kPa
Start of First Injection	-330°CA	-330° to -300°CA
Spark Timing Criteria	1) Avoid misfires, 2) Avoid strong audible knock, 3) CA50 = 5°CA	1) Avoid misfires, 2) Avoid strong audible knock, 3) For maximum IMEPn (MBT)

Figure 3d shows how the net indicated thermal efficiency (η_{th}) varies with CA50 for two different operating points. For the unheated stoichiometric case, highest η_{th} is encountered for CA50 $\approx$ 10°CA. (For consistency with common literature descriptions of spark timing for maximum brake torque, this data point is labeled MBT. It was determined by sweeping ST in small steps, but only two stoichiometric data points were acquired for Fig. 3.) In contrast, the lean heated operation with $\phi = 0.55$ requires a much earlier CA50 to attain MBT phasing. For this lean operating point, there are three main contributors to the improved η_{th} with advancing combustion phasing in the CA50 = 17 - 4°CA range,:

1. Higher combustion efficiency (η_{comb}), see Fig. 3a.
2. Shorter combustion duration (Fig. 3b).
3. Better combustion phasing with less AHRR during the expansion stroke (Fig. 4), *i.e.* larger fraction of constant-volume combustion.

The higher combustion efficiency is coupled to the disappearance of partial-burn cycles that plague the IMEPn stability for lean operation with late combustion phasing, see Fig. 3c. On the other hand, Fig. 3c shows that the IMEPn instability increases rapidly for overly advanced CA50. This is caused by the appearance of complete misfire cycles. Figure 5a shows that the AHRR for operation with ST = -57°CA has substantial cycle-to-cycle variations, but the combustion proceeds without misfires.

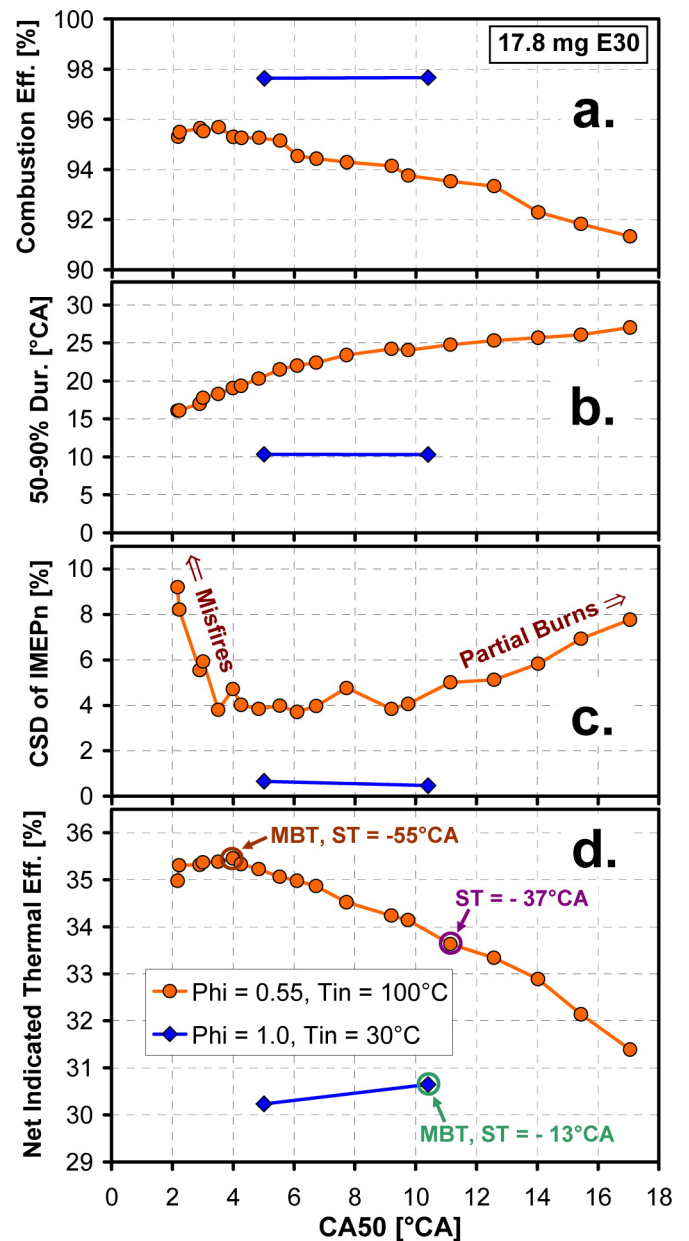


Figure 3. Effect of combustion phasing on a) combustion efficiency and b) 50-90% burn duration, c) IMEP instability, and d) thermal efficiency for both stoichiometric and lean heated operation with E30 fuel.

When ST is advanced to -65°CA, Fig. 5b shows that 2 out of 250 cycles misfire completely; see example cycle without positive AHRR plotted in blue. The misfires occur partly because of the low compressed-reactant temperature at the time of spark. Analysis of this spark-timing sweep with $\phi = 0.55$ indicates that a reactant temperature of 570K is required to ensure successful inflammation for all cycles. Below this temperature, the flame speed becomes too low to ensure robust flame development for this low ϕ , as demonstrated in Ref. [6]. (For higher ϕ , robust flame development can be achieved even if the temperature is lower than 570K.) The importance of the reactant temperature at the time of spark will be discussed more in the section that compares the RS system with the MP ignition system. The exact reason for the appearance of the two misfires in Fig. 5b is not known. Examination of the spark current