

Improving Efficiency and Using E10 for Higher Loads in Boosted HCCI Engines

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ABSTRACT

This study systematically investigates the effects of various engine operating parameters on the thermal efficiency of a boosted HCCI engine, and the potential of E10 to extend the high-load limit beyond that obtained with conventional gasoline. Understanding how these parameters can be adjusted and the trade-offs involved is critical for optimizing engine operation and for determining the highest efficiencies for a given engine geometry. Data were acquired in a 0.98 liter, single-cylinder HCCI research engine with a compression-ratio of 14:1, and the engine facility was configured to allow precise control over the relevant operating parameters. The study focuses on boosted operation with intake pressures (P_{in}) ≥ 2 bar, but some data for $P_{in} < 2$ bar are also presented. Two fuels are considered: 1) an 87-octane gasoline, and 2) E10 (10% ethanol in this same gasoline) which has a lower autoignition reactivity for boosted operation.

This study considers several engine operating parameters, including: intake temperature, fueling rate, engine speed, fuel type, and the effect of various amounts of mixture stratification using three fueling methods: fully premixed, early-DI, and premixed + late-DI (termed partial fuel stratification, PFS). The effects of these operating parameters on the factors affecting thermal efficiency, such as combustion phasing (CA50), amount of EGR required, ringing intensity, combustion efficiency, $\gamma = c_p/c_v$, and heat transfer are also explored and discussed. The study showed that in general, thermal efficiency improves when parameters are adjusted for lower intake temperatures, less CA50 retard, and less EGR, as long as the ringing intensity is ≤ 5 MW/m² to prevent knock, and combustion efficiency is good (*i.e.* $\geq$ about 96%). Additionally, applying a small amount of mixture stratification (using PFS or early-DI fueling) improves efficiency by allowing more CA50 advance when boost levels are sufficient for these fuels to be ϕ -sensitive. E10 gives a small increase in thermal efficiency because EGR requirements are reduced. E10 is also effective for increasing the maximum load for $P_{in} \geq 2.4$ bar, and increasing the high-load limit to IMEPg = 18.1 bar, with no engine knock and ultra-low NOx and soot emissions, compared to IMEPg = 16.3 bar for gasoline. Overall, this study showed that the efficiencies for boosted HCCI can be increased above their already good baseline values. For our engine configuration, improvements of 3 - 5 thermal-efficiency percentage units were achieved corresponding to a reduction in fuel consumption of 7 - 11%.

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INTRODUCTION

Improving the efficiency of internal combustion engines is critical for meeting the global need to reduce petroleum consumption and CO₂ emissions. Homogeneous charge compression ignition (HCCI) engines have a strong potential for contributing to these goals since they have high efficiencies, and they provide ultra-low NOx and particulate emissions without the expensive aftertreatment required by most other high efficiency concepts (*e.g.* stratified-charge spark and diesel engines). Thermal efficiencies of HCCI

engines are well above those of spark-ignition (SI) engines, typically comparable to diesel engines with similar displacement. Moreover, recent research has shown that HCCI efficiencies are higher for intake-boosted operation [1], and by applying partial fuel stratification in combination with intake boost, efficiencies can be increased even further [2].

In addition to providing gains in thermal efficiency, intake pressure boosting has been shown to be effective for overcoming the problem of limited power output that has historically been a challenge for HCCI. Loads up to 16.3 bar

IMEPg¹ (gross indicated mean effective pressure) were achieved at an intake pressure (P_{in}) of 3.25 bar absolute for well-premixed operation using conventional gasoline with an anti-knock index (AKI)² = 87. This load limit for P_{in} = 3.25 bar was the result of limited oxygen availability due to the high levels of EGR (exhaust gas recirculation) required to prevent overly advanced combustion phasing and excessive pressure rise rates (PRR).³ At more moderate boost levels, where the load for well-premixed operation was limited by the more-typical knock/stability limit [1,3,4] and excess oxygen was available, partial fuel stratification (PFS) has been shown to be very effective for reducing the maximum PRR to allow both higher loads and/or increased efficiency [2]. For example, at P_{in} = 2 bar, the maximum load was increased from IMEPg = 11.7 bar for fully premixed fueling to IMEPg = 13.0 bar with PFS, which is comparable to the maximum load of an emissions-compliant diesel engine at this boost pressure. The reduction in PRR produced by PFS also allows more advanced combustion phasing without knock for significant gains in thermal efficiency.

These increased high-load capabilities of boosted HCCI (and HCCI with PFS) combined with the recently demonstrated extension of HCCI-like operation down to idle [5,6] suggest the possibility of an engine operating with HCCI-like combustion over the operating map. Moreover, the recent improvements in thermal efficiency with intake boost and PFS discussed above suggest that even higher efficiencies may be possible with HCCI-like engines, perhaps exceeding typical diesel-engine efficiencies with sufficient development, as has recently been demonstrated for other low-temperature combustion concepts [7,8,9,10]. With these performance characteristics and their ultra-low engine-out NOx and soot emission, HCCI-like engines operating on gasoline or a gasoline-like fuel are strong candidates for future transportation engines.

To realize the full potential of boosted HCCI, it is important to understand how the operating parameters affect the thermal efficiency of these engines. To achieve the highest efficiencies for a given engine geometry it is critical to know how the operating parameters can be adjusted to better optimize the fundamental factors (such as combustion phasing) that control thermal efficiency, and the trade-offs involved. Another key issue is to further increase the high-load limit to utilize high-efficiency HCCI combustion over the widest possible portion, perhaps all, of the operating map. One of the main factors affecting the high-load limit is the limited oxygen availability due to high EGR levels required to control autoignition with boost. High EGR levels can also be detrimental to efficiency due to the lower ratio of specific heats ($\gamma = c_p/c_v$) of the EGR gases, which reduces the work

that can be extracted for a given expansion ratio [11]. One way to reduce the EGR requirements is to change the fuel composition. In particular, ethanol autoignition is much less enhanced by intake boost than gasoline, thus little or no EGR is required with boost [12]. Accordingly, gasoline/ethanol blends such as E10 (10% ethanol in gasoline), a blend that is typical of the majority of gasoline sold in the United States, may be beneficial for reducing the EGR required for boosted HCCI operation.

Toward these goals of increased efficiency and higher loads, the current study has two main objectives:

1. Conduct a systematic study of the engine operating parameters affecting the thermal efficiency of an intake-boosted HCCI engine, and to the extent possible, determine how adjusting these parameters changes the fundamental factors (e.g. combustion phasing, γ , combustion efficiency, and heat transfer) controlling thermal efficiency. Then, apply this information to seek the highest efficiencies for our current engine configuration.
2. Determine how E10 affects HCCI operation and thermal efficiencies compared to straight gasoline, and investigate the potential of E10 for efficiency and high-load improvements.

BACKGROUND

Intake-pressure boosting is often used to increase the power output of internal combustion engines, and for HCCI engines, it provides a means of increasing the power while still maintaining a sufficiently dilute charge to prevent NOx formation. Although most early attempts to achieve high loads with boosted HCCI used special fuels and/or reduced compression ratios [13,14,15,16], the authors recently showed that intake boosting can be applied using conventional gasoline (AKI = 87, no ethanol) to achieve high loads without engine knock and with ultra-low NOx and soot [1]. In this previous work, intake pressure was systematically increased, and the maximum IMEPg determined for each P_{in} . As shown in Fig. 1a, the maximum load increases progressively with P_{in} , with the highest value being IMEPg = 16.3 bar at P_{in} = 3.25 bar. As also shown in Fig. 1a, the ringing intensity (discussed in the next section) was held to a maximum of 5 MW/m², thus preventing any significant acoustic oscillations on the pressure trace or any audible noise indicative of engine knock. This was accomplished by reducing the intake temperature as low as 60°C and adding cooled EGR as necessary to retard the combustion phasing [17] sufficiently to obtain an acceptable ringing intensity (i.e. an acceptable PRR) [4].

Figure 1b shows the gross indicated thermal efficiencies corresponding to these high-load points. The thermal

¹IMEPg refers to the IMEP over the compression and expansion strokes only, while IMEP_{net} refers to the IMEP over the entire cycle, including the gas-exchange strokes.

²Anti-knock index, AKI = (RON + MON)/2.

³Higher loads could also have been achieved with greater boost despite the limited oxygen, but this would have resulted in excessive peak cylinder pressures.

efficiency is about 44% for most of the intake pressures tested, but it increases to 46.8% at the highest $P_{in} = 3.25$ bar. For this previous dataset, the objective was to determine the highest possible loads at each boost, and thermal efficiency was not considered. However, the fortuitous, rather substantial increase in efficiency at the highest P_{in} suggests that significantly higher thermal efficiencies may be possible with boosted HCCI if operating parameters are appropriately optimized.

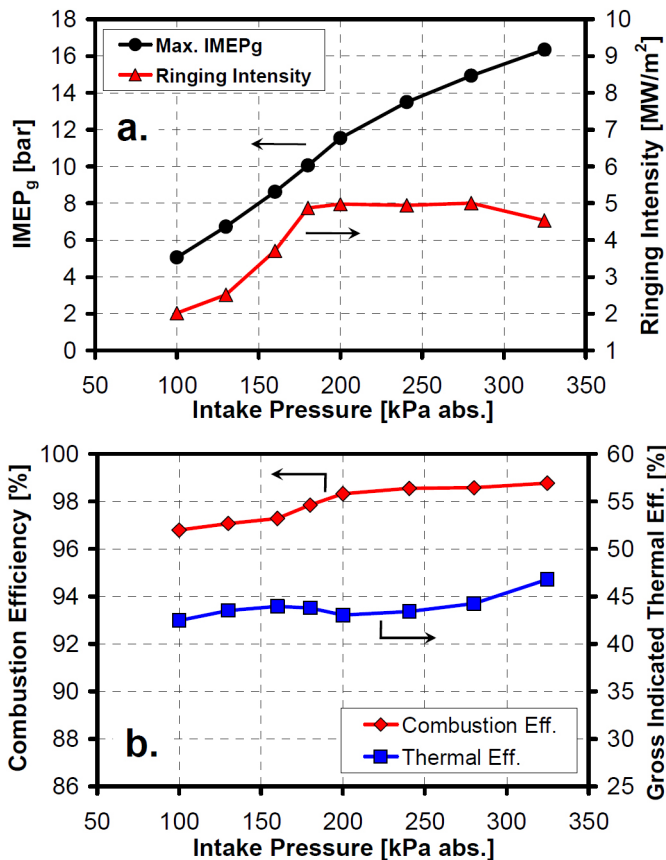


Figure 1. (a) Maximum load (IMEPg) and ringing intensity as a function of intake pressure; (b) combustion and thermal efficiencies for these maximum-load points. Reproduced from Ref. [1].

Combustion efficiency is one factor affecting thermal efficiency, and as shown in Fig. 1b, it increases from about 97 to 99% over the range of P_{in} tested. However, the magnitude of this combustion efficiency increase is too small to explain the observed increase in thermal efficiency, and the shape of the combustion efficiency curve is quite different from that of the thermal efficiency. Therefore, other factors must be more strongly affecting the thermal efficiency. These factors need to be understood in order to push HCCI to its full efficiency potential, which is a main objective of the current study.

EXPERIMENTAL SETUP & DATA ACQUISITION

ENGINE FACILITY

The HCCI research engine used for this study was derived from a Cummins B-series six-cylinder diesel engine, which is a typical medium-duty diesel engine with a displacement of 0.98 liters/cylinder. Figure 2 shows a schematic of the engine, which has been converted for single-cylinder operation by deactivating cylinders 1-5. The engine specifications and operating conditions are listed in Table 1. The configuration of the engine and facility are identical to those used in our previous study of intake pressure boost [1]. Additional information on the facility may be found in Ref. [18].

TABLE 1. Engine Specifications and Operating Conditions

Displacement (single-cylinder)	0.981 liters
Bore	102 mm
Stroke	120 mm
Connecting Rod Length	192 mm
Geometric Compression Ratio	14:1
No. of Valves	4
IVO	0° CA*
IVC	202° CA*
EVO	482° CA*
EVC	8° CA*
Swirl Ratio	0.9
Fueling system	Fully Premixed / GDI
GDI Injector	Bosch, 8-hole
Included Angle	70°
Hole Size	Stepped-hole, min. dia. = 0.125 mm
Injection Pressure	120 and 150 bar
Engine Speed	1200 rpm, except for Figs. 7 & 8
Intake Temperature	30 – 90°C
Intake Pressure (abs.)	1.0 – 3.4 bar
Coolant Temperature	100°C

*0° CA is taken to be TDC intake. The valve-event timings correspond to 0.1 mm lift.

The active HCCI cylinder is fitted with a compression-ratio (CR) = 14 custom piston as shown in Fig. 2b. This piston provides an open combustion chamber with a large squish clearance and a quasi-hemispherical bowl. Comparison tests showed minimal differences in performance and emissions compared to the CR = 14 broad shallower-bowl piston used in many earlier studies e.g. [17]. Both pistons provide a small topland ring crevice, amounting to only 2.1% of the top-dead-center (TDC) volume. A more complete discussion of the comparison between pistons may be found in Ref. [19].

As shown in Fig. 2a, the engine facility is equipped to allow the use of cooled EGR. The flow rate of cooling water can be adjusted to control the temperature of the EGR gases

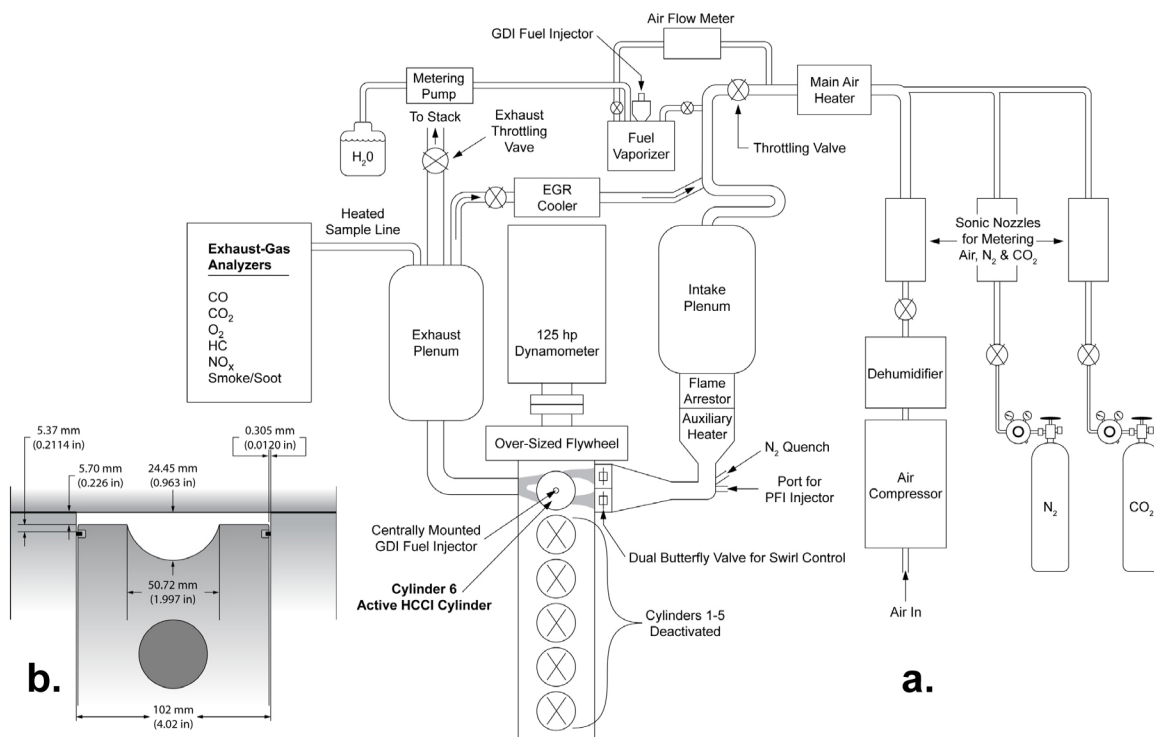


Figure 2. Schematic of the HCCI engine facility (a), and combustion chamber geometry of the CR = 14 piston, at TDC (b).

down to temperatures as low as 30°C. A water trap downstream of the EGR cooler removes the water that condenses if the EGR gases are cooled to these low temperatures. The EGR is introduced well upstream of the intake plenum, so the EGR travels with the intake air through a series of bends before reaching the intake plenum, to insure that the intake charge is well mixed. With this configuration, the exhaust pressure must be greater than the intake pressure for EGR to flow into the intake. The required back pressure was achieved by throttling the exhaust flow using the valve shown in Fig. 2a.

Intake air was supplied by an air compressor and precisely metered by a sonic nozzle as shown in Fig. 2a. For operation without EGR, the air flow was adjusted to achieve the desired intake pressure, as measured by a pressure transducer on the intake runner. Intake pressures varied from 1.0 bar (simulating naturally aspirated conditions) to 3.4 bar for the current study. All pressures given are absolute. For operation with EGR, the air flow was reduced from the amount required to achieve the desired intake pressure with air alone, and the valve on the EGR line was opened. The exhaust back-pressure throttle valve was then adjusted to produce enough EGR flow to reach the desired intake pressure. This typically resulted in the exhaust pressure being about 2 kPa greater than the intake pressure. For consistency, the back pressure was maintained at 2 kPa above the intake pressure, even when EGR was not used.

In this study both premixed and direct injection (DI) fueling were used. The premixed fueling system, shown at the top of the schematic in Fig. 2a, consists of a gasoline-type direct injector (GDI) injector mounted in an electrically heated fuel-vaporizing chamber and appropriate plumbing to ensure thorough premixing with the air and EGR upstream of the intake plenum. DI fueling was accomplished using a second GDI injector mounted centrally in the cylinder head as shown. For PFS, a combination of premixed and DI fueling was used. A positive displacement fuel flow meter was used to determine the amount of fuel supplied. Two fuels were used: 1) a research-grade gasoline supplied by Chevron-Phillips Chemical Co., whose specifications are listed in Table 2; and 2) E10 which consists of 10% (vol.) neat ethanol mixed with this same Chevron-Phillips research-grade gasoline. Based on AKI = 99.5 for ethanol (RON = 109, MON = 90) [20], the E10 has an AKI = 88.1, which falls between regular and mid-grade pump gasolines sold in the U.S. Also, all grades of pump gasoline sold in the U.S. commonly contain 10% ethanol.

Note that the research-grade gasoline meets the same basic specifications as the fuel used in our previous studies of boosted HCCI [1] and HCCI with PFS [2], but it is from a different batch, and there are small differences in composition and the distillation curve.

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.

Similar to some of our other recent works [2,21], this work uses an equivalence ratio based on total charge mass, rather than air alone. This equivalence ratio, referred to as the charge-mass equivalence ratio (ϕ_m), is defined as:

$$\phi_m = \frac{(F/C)}{(F/A)_{stoich}} \quad (1)$$

where F/C is the mass ratio of fuel and total inducted charge gas (*i.e.* fresh air and EGR), and $(F/A)_{stoich}$ is the mass ratio of stoichiometric fuel/air mixture for complete combustion. This provides a convenient and consistent way to compare data with the same supplied energy content per unit charge mass (*i.e.*, the same dilution level) for operating conditions with different fuels and different EGR levels. Note that ϕ_m is the same as conventional air-based ϕ when no EGR is used. It should also be noted that the air-based ϕ is ≤ 1 for all conditions presented, so in general there is always sufficient oxygen.

Prior to starting the experiments, the engine was fully preheated to 100°C by means of electrical heaters on the “cooling” water and lubricating-oil circulation systems. An auxiliary heater mounted close to the engine provided precise control of the intake temperature, which ranged from 30° -

90°C. All data were taken at an engine speed of 1200 rpm except for the speed sweeps in Figs. 7 and 8.

DATA ACQUISITION

Cylinder pressure measurements were made with a transducer (AVL QC33C) mounted in the cylinder head approximately 42 mm off center. The pressure transducer signals were digitized and recorded at ¼° CA increments for one hundred consecutive cycles. The cylinder-pressure transducer was pegged to the intake pressure near bottom dead center (BDC) where the cylinder pressure reading was virtually constant for several degrees. Intake-air temperatures were monitored using thermocouples mounted in the two intake runners close to the cylinder head. For all data presented, 0° crank angle (CA) is defined as TDC intake (so TDC compression is at 360°). This eliminates the need to use negative crank angles or combined bTDC, aTDC notation.

The crank angle of the 50% burn point (CA50) was used to monitor the combustion phasing, and the 10% burn point (CA10) was used as a representative marker for the hot-ignition point. CA10 and CA50 were determined from the cumulative apparent heat-release rate (AHRR), computed from the cylinder-pressure data (after applying a 2.5 kHz low-pass filter [18]). Computations were performed for each individual cycle, disregarding heat transfer and assuming a constant ratio of specific heats [22].⁴ The average of 100 consecutive individual-cycle CA10 or CA50 values were then used to monitor CA10 or CA50 during engine operation and for the values reported. The reported PRRs and ringing intensities are computed from the same low-pass-filtered pressure data. For each cycle, the maximum PRR was analyzed separately with a linear fit over a moving $\pm 0.5^\circ$ CA window. Similar to CA50, these individual-cycle values were then averaged over the 100-cycle data set.

As mentioned in the Introduction and Background sections, and discussed in numerous previous works (*e.g.*, [4,23,24,25,26,27]), HCCI combustion is often limited by high PRRs, which cause acoustic oscillations in the charge gas resulting in audible engine knock. If this phenomenon is not controlled, it can result in unacceptable noise levels and potentially, engine damage. The acceptable knock limit for HCCI engines is often defined in terms of a maximum allowable PRR ($dP/d\theta$, where θ is a variable representing °CA). However, this does not correctly reflect the potential for knock with changes in engine speed, or under boosted conditions where the cylinder pressure changes significantly. In this work, the correlation for ringing intensity developed by Eng [26] is used as a measure of the propensity for engine knock.

⁴This specific heat ratio (γ) is determined from a fit to the actual pressure data to account for differences in gas temperature or EGR levels between operating conditions. Comparison of CA50 values computed in this manner with those computed by detailed calculations using real-gas properties that vary over the cycle and a Woschni heat transfer correction shows good agreement, with differences typically being less than one or two tenths of a degree.

$$\text{Ringing Intensity} \approx \frac{1}{2\gamma} \cdot \frac{\left(0.05 \cdot \left(\frac{dP}{dt}\right)_{\max}\right)^2}{P_{\max}} \cdot \sqrt{\gamma R T_{\max}} \quad (2)$$

Where $(dP/dt)_{\max}$, $P_{\max}$, and $T_{\max}$ are the maximum values of PRR (in real time), pressure, and temperature, respectively, γ is the ratio of specific heats (c_p/c_v), R is the gas constant, and 0.05 is an empirical correlation constant introduced by Eng which has units of milliseconds. The ringing is a measure of the acoustic energy of the resonating pressure wave that creates the sharp sound commonly known as engine knock. Based on the onset of an audible knocking sound and the appearance of strong ripples on the pressure trace, a ringing criterion of 5 MW/m² was selected as the ringing limit for operation without knock. This is the same limit used in our previous works [1,2,4], and a more complete discussion of the selection of this value may be found in Ref. [4]. This value, 5 MW/m², corresponds to about 8 bar/°CA at 1200 rpm, naturally aspirated. However, it should be noted that the allowable PRR increases with boost due to the increased value of $P_{\max}$ in the denominator. At all boost levels tested, audible engine knock correlated well with the ringing intensity rising above 5 MW/m², giving confidence in this correlation. Similarly, Eng [26] shows that Eq. 2 gives the same ringing value for different PRR limits selected by Lund University for operation at two different intake pressures.

Exhaust emissions data were also acquired, with the sample being drawn from the exhaust plenum using a heated sample line (see Fig. 2a). CO, CO₂, HC, NO_x, and O₂ levels were measured using standard exhaust-gas analysis equipment. In addition, a second CO₂ meter monitored the intake gases just prior to induction into the engine. This allowed the EGR fraction of the intake gases to be computed from the ratio of the intake and exhaust CO₂ concentrations. Smoke measurements were also made with an automated smoke meter. However, smoke was not detected for any premixed operating points, and for all early-DI and PFS data reported, it was an order of magnitude or more below the US-2010 heavy-duty on-road emission standards, so smoke is not reported. Similarly, NO_x was well below (typically more than an order of magnitude) US-2010 limits and is not reported.

RESULTS AND DISCUSSION

FACTORS AFFECTING THERMAL EFFICIENCY

Many factors affect the thermal efficiency of HCCI and partially stratified HCCI engines. As mentioned in the introduction, the current study focuses on those that can be controlled by varying the operating parameters for our current engine configuration. Engine configuration parameters such as compression ratio, valve timing, and in-cylinder swirl can also affect engine efficiency, but they are beyond the scope of the current study.

All thermal efficiencies reported in this article are based on the gross indicated work, as determined from the IMEP_g, and are therefore, indicated thermal efficiencies. For brevity, they will typically be referred to simply as “thermal efficiencies” and are defined in the classical manner as:

$$\text{Thermal Efficiency (T-E)} = \frac{\text{Gross Indicated Work}}{\text{LHV} * m_f} \quad (3)$$

where LHV is the lower heating value of the fuel supplied in the gas phase, and m_f is the mass of fuel.⁵ For some conditions, combustion efficiencies are also presented, which are defined as:

$$\text{Combustion Efficiency (C-E)} = \frac{\text{HR}}{\text{LHV} * m_f} \quad (4)$$

where HR is the heat released, which is estimated by correcting (LHV * m_f) for the unburned CO and HC from the emissions measurements.

Changing various engine operating parameters can affect several factors that in turn affect the thermal efficiency. However, investigating the effects in a systematic manner is challenging since it is difficult to change only one factor at a time. Table 3 lists the operating parameters considered in this study and the main factors they affect related to changes in thermal efficiency. The order of the parameters in Table 3 is approximately the order that they are addressed in this article, with the exception of EGR. EGR is listed first since it is used in combination with changes in most other parameters to provide combustion-phasing control [1,2,17]. However, EGR is not directly investigated in a systematic manner in this article since it has been well investigated in previous works [11,28]. As indicated by the factors listed in Table 3, and the short explanation following each listing of a factor, there are many interactions between these factors, and adjusting engine parameters to change one factor often requires compensation

⁵Note that this classical definition of thermal efficiency is the same as the “fuel conversion efficiency” used by Heywood [22]. It should not be confused with Heywood's “thermal conversion efficiency” which is equal to T-E/C-E, with T-E and C-E defined by Eqs. 3 and 4. T-E/C-E is often referred to as a thermodynamic efficiency [11,28]. Thermodynamic efficiencies are not presented in this article.

TABLE 3. Parameters Considered and the Main Factors They Affect

<u>Engine operating parameter</u>	<u>Main factors affected</u>
EGR*	CA50 – EGR is used to retard CA50 to prevent excessive ringing.** γ^{***} – Higher EGR rates decrease γ (Charge-gas composition effect). Combustion efficiency – For high levels of EGR, the reburn of CO and unburned HC can significantly affect combustion efficiency [1].
Temperature at BDC Intake (T_{BDC}) Intake Temperature (T_{in}) was varied in the current study as a means of varying T_{BDC} .	CA50 – T_{BDC} affects autoignition timing and therefore CA50. γ – Changes in temperature directly affect γ and can also affect the amount of EGR required to maintain CA50. Heat transfer – Higher combustion temperatures increase heat transfer.
Amount of Fuel Supplied (ϕ_m)	CA50 – The amount of fuel affects the PRR and therefore, the CA50 required to control ringing. γ – Higher ϕ_m reduces γ due to gas-composition and temperature effects. Changes in ϕ_m can also modify EGR or T_{in} requirements to control CA50. Heat transfer – Higher ϕ_m increases combustion temperature and therefore heat transfer. Combustion efficiency – Higher ϕ_m increases combustion temperature and therefore combustion efficiency, assuming that the air-based $\phi \leq 1$.
Engine Speed	CA50 – Engine speed affects real-time PRR (dP/dt) and therefore, the CA50 required to control ringing. γ – Changes in engine speed can modify EGR or T_{in} requirements to control CA50. Heat transfer – Heat transfer losses scale with engine speed.
Fuel Type	CA50 – 1) Autoignition quality of the fuel affects the CA50. 2) ϕ-sensitivity of the fuel affects the potential for using PFS or early-DI fueling strategies to reduce PRR, and therefore, the required CA50 retard. γ – Differences in autoignition quality can change the amount of EGR required to maintain CA50.
Intake Pressure (P_{in})	CA50 – Increased P_{in} enhances autoignition of gasoline and E10, which will advance CA50. γ – Changes in P_{in} affect EGR or T_{in} requirements to control CA50. Heat transfer – Heat transfer losses per unit-charge mass should decrease, based on standard Woschni heat-transfer correlation [22].
Fueling Strategy (Premixed vs. PFS or Early-DI)	CA50 – The fueling strategy affects the HRR and PRR, and therefore, the CA50 required to control ringing. γ – Changes in fueling strategy can affect EGR or T_{in} requirements to control CA50. They can also affect γ via composition (amount of fuel in the compression stroke) and/or temperature (cooling effect of direct injection).

*Influence of EGR was not studied in a systematic manner in this article, but EGR was used in each parametric investigation as a tuning parameter to control CA50.

**In addition to excessive noise and potential engine damage, high ringing can decrease thermal efficiency by increasing heat transfer due to the onset of acoustic oscillations, as shown by Tsurushima et al. [25]

*** $\gamma = c_p/c_v$ directly influences the thermal efficiency by affecting the amount of work that can be extracted for a given expansion ratio.

with another parameter to isolate individual effects to the greatest extent possible. An example of this is given in the next subsection which addresses the effect of intake temperature on thermal efficiency. In this case, the amount of EGR was varied with the intake temperature to maintain a constant CA50, since changes in CA50 can have a large impact on efficiency whether they are caused by changes in intake temperature or some other parameter.

EFFECTS OF T_{IN} (T_{BDC}) ON EFFICIENCY

The first parameter considered in this study is the charge temperature at bottom dead center (T_{BDC}) at the start of the

compression stroke, which subsequently affects the compressed-gas and combustion temperatures. In our research engine, T_{BDC} was varied by varying the intake temperature (T_{in}), but it should be recognized that in an HCCI engine using retained hot residuals to control autoignition, a similar effect on T_{BDC} could be achieved by varying the amount of hot residuals.

An intake pressure (P_{in}) of 2 bar was selected for this study (and other studies presented in succeeding sections) since it gives performance that is representative of intake-boosted operation, and it is near the center of the boost-pressure range investigated in our previous works [1,2]. This

P_{in} is sufficiently high for robust autoignition with intake temperatures as low as 30°C, the lowest T_{in} that could be repeatably obtained in our engine facility. The engine was fueled with gasoline at a constant fueling rate of 55 mg/injection for all temperatures, with the objective of determining the effects of T_{in} on thermal efficiency for an approximately constant load. Over the T_{in} range investigated, this fueling yielded an IMEPg of 11 bar $\pm 3\%$ and corresponded to $0.38 \leq \phi_m \leq 0.45$. This fueling rate was sufficient to produce high combustion efficiencies ($> 97.5\%$) for all conditions studied, eliminating this potential variable from the study.

Constant CA50

As presented in Fig. 3, T_{in} was varied from 90°C down to 30°C, and both fully premixed and early-DI fueling were considered. Although this T_{in} range exceeds that of current turbo-charged engines, temperatures down to 30°C were included to better understand trends and the potential benefits of low T_{in} . These data are also beneficial for comparison with other recent studies of high-efficiency engines done at similar low intake temperatures [8,10]. The start of injection (SOI) for DI fueling was at 40°C (after TDC intake). Premixed data were not acquired for $T_{in} < 50^\circ\text{C}$ since condensation of the pre-vaporized fuel was considered likely. Because changes in combustion phasing can significantly affect thermal efficiency, independent of temperature changes, T_{in} sweeps were first conducted with CA50 held constant at 376.7°C by adjusting the amount of EGR (see annotation in Fig. 3a). These results are shown by the black lines with circular symbols in Fig. 3a. As can be seen, as T_{in} is reduced, efficiency progressively increases from about 44.2% to about 45.2%. This improvement results from the combination of two effects: 1) the γ of the mixture is increased with reduced T_{in} because of the lower temperatures and because less EGR is required to retard the CA50 to the desired value of 376.7°C, and 2) heat transfer losses are reduced since the compressed-gas and combustion temperatures are lower when T_{in} is reduced. Note that the lower combustion temperatures result from a combination of the reduced T_{in} and from the lower ϕ_m that occurs with reduced T_{in} because fueling was held constant. It is also evident that there is not a significant difference between premixed and early-DI fueling when CA50 is held constant.⁶ The slightly higher thermal efficiencies for DI fueling are likely the result of a small reduction in T_{BDC} for the same T_{in} due to the charge cooling resulting from vaporization of the DI fuel.

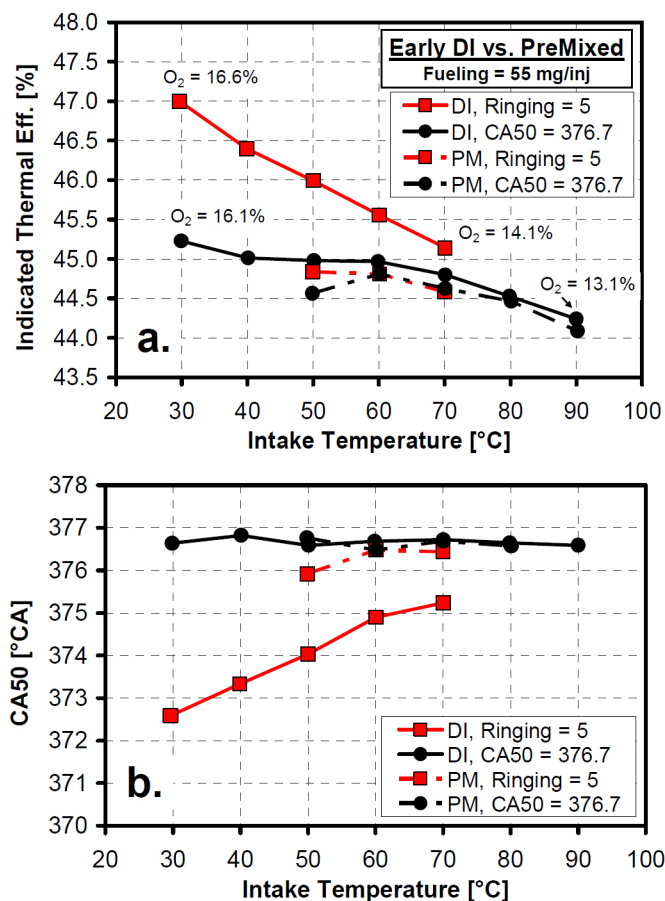


Figure 3. The effect of changes in T_{in} on thermal efficiency (a) and CA50 (b) for fully premixed and early-DI fueling, for both constant CA50 = 376.7°C and for constant Ringing = 5 MW/m². Intake-O₂ molar percentages are shown for the endpoints of the two DI sweeps. In the legend, DI = direct injection and PM = premixed.

Constant Ringing Intensity

When CA50 is retarded well after TDC as it is here, the highest efficiencies are obtained when CA50 is advanced as much as possible while maintaining the ringing intensity ≤ 5 MW/m², to prevent knock. The CA50 of 376.7°C mentioned above was selected because it gives a ringing intensity of about 5 MW/m² for $T_{in} = 60^\circ\text{C}$, the midpoint of the temperature sweep. As T_{in} is varied for the premixed fueling, the ringing intensity varies only a little from this value for $50 \leq T_{in} \leq 70^\circ\text{C}$, so adjusting CA50 to maintain a constant ringing of 5 MW/m² over this T_{in} range gives almost

⁶The small drop in thermal efficiency for the premixed data point at $T_{in} = 50^\circ\text{C}$ with CA50 = 376.6°C is thought to result from a small amount of fuel condensing in the intake system, supporting the decision to not use premixed fueling for $T_{in} < 50^\circ\text{C}$. Note that this apparent reduction in the amount of fuel reaching the engine also reduces the ringing (not shown) for the constant CA50 data point, allowing CA50 to be slightly advanced for the constant ringing = 5 MW/m² data point (see Fig. 3b). This advancement in CA50 results in a slightly higher thermal efficiency compared to the constant CA50 data point (Fig. 3a) even though both should have essentially the same fuel loss due to condensation.