

Effects of Gasoline Reactivity and Ethanol Content on Boosted, Premixed and Partially Stratified Low-Temperature Gasoline Combustion (LTGC)

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ABSTRACT

Low-temperature gasoline combustion (LTGC), based on the compression ignition of a premixed or partially premixed dilute charge, can provide thermal efficiencies (TE) and maximum loads comparable to those of turbo-charged diesel engines, and ultra-low NO_x and particulate emissions. Intake boosting is key to achieving high loads with dilute combustion, and it also enhances the fuel's autoignition reactivity, reducing the required intake heating or hot residuals. These effects have the advantages of increasing TE and charge density, allowing greater timing retard with good stability, and making the fuel ϕ -sensitive so that partial fuel stratification (PFS) can be applied for higher loads and further TE improvements. However, at high boost the autoignition reactivity enhancement can become excessive, and substantial amounts of EGR are required to prevent overly advanced combustion. Accordingly, an experimental investigation has been conducted to determine how the tradeoff between the effects of intake boost varies with fuel-type and its impact on load range and TE. Five fuels are investigated: a conventional AKI=87 petroleum-based gasoline (E0), and blends of 10 and 20% ethanol with this gasoline to reduce its reactivity enhancement with boost (E10 and E20). A second zero-ethanol gasoline with AKI=93 (matching that of E20) was also investigated (CF-E0), and some neat ethanol data are also reported.

Results show that ethanol content has little effect on LTGC autoignition reactivity for naturally aspirated operation, but it produces a large effect for boosted operation, with the reactivity enhancement with boost being reduced by an amount that correlates with ethanol content. In contrast, CF-E0 showed a reactivity enhancement with boost similar to E0. Related to this autoignition enhancement, the effect of fuel-type on the increase in ITHR with boost was also investigated since it correlates with the ability to retard CA50 with good stability for higher loads without knock and to apply PFS effectively. The study showed that by adding ethanol, less EGR is required with boost, leaving more oxygen available for combustion. As a result, the high-load limit could be increased from 16.3 to 18.1 to 20.0 bar IMEP_g for E0, E10, and E20, respectively, and to 17.7 bar for the high-AKI gasoline. TE vs. load curves for the various fuels at typical boosted conditions are also presented and discussed. At boosted conditions, PFS was found to be very effective for increasing the TE, with the peak TE increasing from 47.8% for premixed fueling to 48.4% with PFS, and TE improvements up to 2.8 %-units were achieved at higher loads.

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INTRODUCTION

Meeting the global demand for increased ground transportation while constrained by limited petroleum supplies and the need to reduce CO₂ emissions, will require significant increases in engine efficiency. Moreover, it is important that both gasoline and diesel engines be highly efficient to minimize total CO₂ emissions and maximize the benefit from crude oil supplies. With this in mind, increasing the efficiency of gasoline engines offers the biggest potential for improvement since current gasoline spark-ignition (SI) engines are considerably less fuel-efficient than diesels.

Low-temperature gasoline combustion (LTGC), based on the compression ignition of a premixed or partially premixed dilute charge, has been shown to be capable of providing thermal efficiencies and maximum loads comparable to those of turbocharged diesel engines. At the same time, it can give engine-out NO_x and particulate emissions well below mandated standards, eliminating the need for expensive aftertreatment systems for these emissions. LTGC was initially investigated mainly in the form of homogenous-charge compression ignition (HCCI) [1, 2, 3, 4, 5, 6, 7, 8, 9]. Subsequent improvements have been demonstrated through the use of various

forms of charge stratification, spark assist, and intake-pressure boosting. For example, low-load combustion efficiency has been substantially increased through the use of charge stratification [8, 9, 10, 11, 12], and combining charge stratification with spark assist has allowed operation down to idle without external heating [13,14]. Also, the use of intake-pressure boost has extended the high-load limit above 18 bar gross indicated mean effective pressure (IMEP_g) with good combustion stability, no engine knock, and ultra-low NO_x and particulate emissions [15].

An important characteristic of intake boosting for LTGC is that the increased pressure enhances the autoignition reactivity of gasoline, reducing the requirement for intake heating or retained hot residuals. This can further extend the increase in load with boost because the initial charge density increases with the lower temperatures, increasing the charge-mass inducted at a given intake pressure. These lower temperatures also increase the efficiency because they result in lower combustion temperatures, which is expected to reduce heat transfer losses, and the reduced temperatures increase the specific heat ratio ($\gamma = c_p/c_v$) allowing more work to be extracted for a given expansion ratio [15].

Another feature of boosted LTGC is that the autoignition reactivity of typical gasoline fuels becomes sensitive to the local equivalence ratio (ϕ) as boost levels increase [16]. For ϕ -sensitive fuels, or fuels that become ϕ -sensitive with boost, regions with higher ϕ autoignite faster than regions of lower ϕ . Thus, if the charge mixture contains a distribution of equivalence ratios, the charge will autoignite sequentially, reducing the peak heat-release rate (HRR) [16,17]. This effect can be exploited to extend the high-load limit and/or achieve higher efficiencies through a technique known as partial fuel stratification (PFS), whereby direct injection is used to produce a non-uniform fuel distribution. The reduced peak HRR with PFS means that more fuel can be added before reaching a peak pressure rise rate (PPRR) sufficient to induce knock. Alternatively, for the same fueling, the reduced peak HRR means that less combustion-phasing retard is required to control knock, so combustion phasing can be advanced for higher thermal efficiencies, or a combination of increased load and efficiency improvements can be made [8,15,16].

With the advantages of a reduced initial charge temperature and the ability to apply PFS, intake boosting can provide significantly higher loads for LTGC than would be expected from the increase in intake-pressure alone. Additionally, these same two advantages of intake boosting can improve the thermal efficiency above the already-high values associated with naturally aspirated LTGC/HCCI [1,4,7,18,19].

However, as intake-boost pressures continue to be increased to obtain further increases in load, the autoignition enhancement can become so great that combustion phasing becomes overly advanced even with no intake heat or heating from retained hot residuals. To overcome this problem, exhaust gas recirculation (EGR) can be applied to slow the autoignition reactions [20] and counteract the enhancement caused by the increased boost. Eventually, however, EGR levels reach the point where all the excess air in the dilute charge is replaced with EGR gases, so the mixture is stoichiometric, although still dilute with EGR. For boost pressures above this point, the fresh-reactant fraction of the charge must be reduced to accommodate the additional

EGR fraction required to control combustion phasing. Thus, the maximum load is limited by the lack of available oxygen rather than by the knock/stability limit encountered at lower intake pressures [21]. Since the charge contains proportionally less reactants and more EGR, the increase in load with increased boost becomes progressively smaller above the stoichiometric point. The high-load limit occurs when either the increase in load is no longer sufficient to warrant the cost of the increased boost, or the cylinder-head pressure limit is reached.

Decreasing the autoignition reactivity of the fuel, or the fuel's reactivity enhancement with boost, offers a means of reducing the EGR requirement so more air is available to allow higher loads for a given boost pressure. It is well known that adding ethanol to conventional gasoline raises its anti-knock index, AKI = (RON + MON)/2, which corresponds to reduced autoignition reactivity of the end-gas in an SI engine operating at the RON- and MON-test conditions [22]. Alternatively, the composition of the base gasoline can be changed to increase the AKI without ethanol addition. Increasing the AKI of the fuel suggests that it will be less reactive under boosted LTGC/HCCI conditions. However, it is important to note that the autoignition conditions in an LTGC/HCCI engine are not the same as those used to determine the AKI, so a direct correlation between AKI and LTGC autoignition reactivity should not be expected [23,24]. Nevertheless, for lack of better information, the change in AKI is used here as a guide for selecting the fuels to be investigated.

In addition to providing a means of decreasing the autoignition reactivity of gasoline for boosted LTGC, blending ethanol produced from renewable sources into conventional gasoline reduces petroleum consumption. It can also reduce the net increase in atmospheric CO₂ caused by gasoline engines since the CO₂ from the ethanol combustion is recycled by the plants used for ethanol production.¹ For these reasons, and for historical reasons related to reducing carbon monoxide (CO) emissions, most gasoline sold in the United States (U.S.) is currently required to contain approximately 10% ethanol. Moreover, efforts are underway to increase the allowable ethanol content to 15 or even 20%.

The objective of current work is to investigate how decreasing the autoignition reactivity (*i.e.* increasing the AKI) of the gasoline affects the performance of boosted LTGC for both fully premixed operation (*i.e.* HCCI) and operation with PFS. For this study, the base-line performance was obtained using a conventional petroleum-based, 87-AKI gasoline containing no ethanol (E0). This base gasoline corresponds to a regular-grade U.S. gasoline, but with no ethanol. The lower-reactivity (higher AKI) ethanol-blends were produced by mixing this same gasoline with 10 and 20% ethanol to form E10 and E20, respectively. This approach eliminates variations in performance due to changes in the composition of the base fuel, as the AKI is increased by the ethanol. A second gasoline with a higher AKI of 93 and no ethanol, a certification fuel (CFE0), was also investigated to understand how decreasing the fuel autoignition reactivity without ethanol compares to ethanol addition. Finally, the performance with neat ethanol (E100) was investigated for some conditions.

1. This assumes that the CO₂ produced by burning the ethanol from biomass is sufficiently low.

Fully premixed operation is evaluated over a range of intake pressures (P_{in}) from atmospheric, $P_{in} = 1$ bar, to the highest practical boost pressure for each fuel, which is above 3.0 bar for all fuels except E100. An emphasis is placed on understanding how changes in fuel composition affect the autoignition reactivity, EGR requirements, stability, and maximum load over the range of intake pressures studied. Operation with PFS is also investigated to determine how the various fuel types affect the ability of PFS to provide the potential benefits for increased efficiency and load, as discussed above.

The following section provides a description of the experimental setup, fuel compositions, and data-acquisition techniques. This is followed by the presentation of the results in four parts: 1) the effects of fuel composition on the overall autoignition reactivity for both naturally aspirated and boosted operation, 2) combustion stability with combustion-phasing retard (necessary to control the PPRR at high loads) is investigated and related to the magnitude of the intermediate temperature heat release (ITHR) of the various fuels for boosted and non-boosted operation, 3) premixed LTGC performance in terms of the achievable thermal efficiency and high-load limits, and 4) the application of PFS to increase the thermal efficiency, along with some discussion of the effects of PFS on maximum load. Finally, the last section summarizes the study, and conclusions are drawn.

EXPERIMENT, FUELS, AND DATA ACQUISITION

Engine Facility

The LTGC/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 1 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 studies of intake pressure boost [15,16,21]. Additional information on the facility may be found in Ref. [9].

The active LTGC cylinder is fitted with a compression-ratio (CR) = 14 custom piston as shown in Fig. 1b. 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 some earlier studies *e.g.* [20]. 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. 1a, 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 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 below their dew point. 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. 1a.

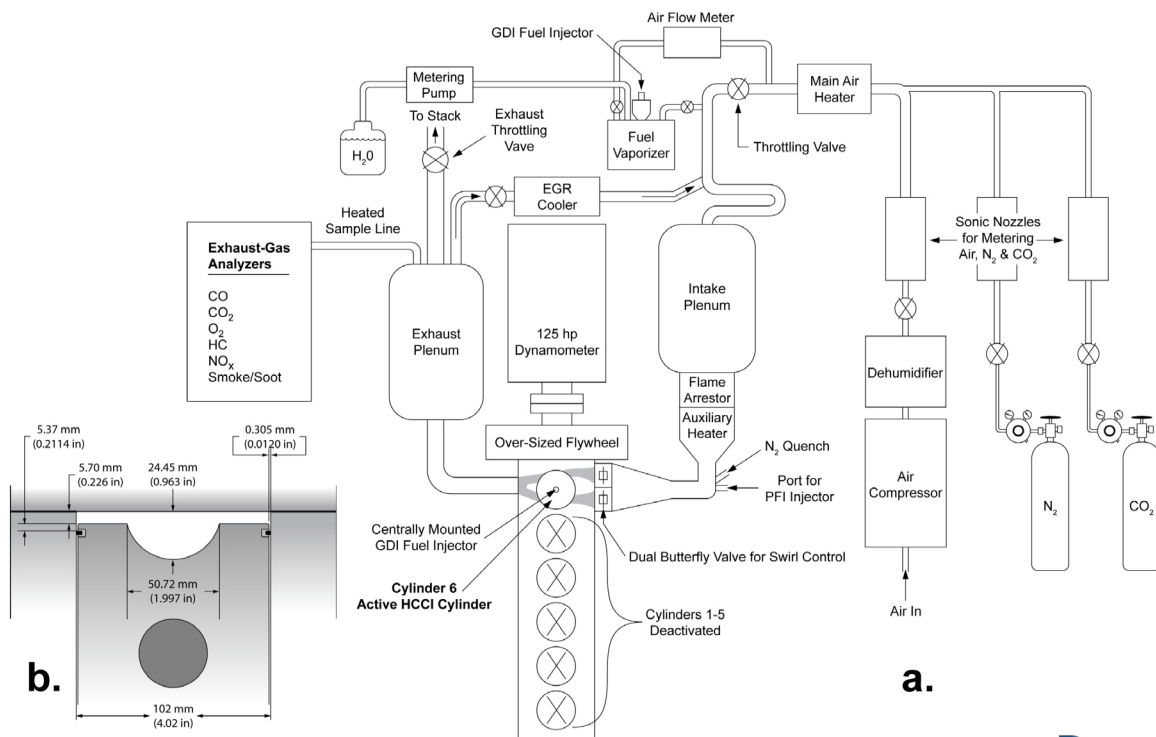


Figure 1. a) Schematic of the LTGC/HCCI engine facility, and b) combustion chamber geometry of the CR=14 piston, at TDC.

Intake air was supplied by an air compressor and precisely metered by a sonic nozzle as shown in Fig. 1a. 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 3.6 bar for the current study. Note that all pressures given in this article 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.²

Table 1. Engine Specifications and Operating Conditions.

Displaced (single cylinder)	0.981 liters
Bore	102 mm
Stroke	120 mm
Connecting rod length	192 mm
Geometric compression ratio	14:1
Number of valves	4
IVO	0°C A *
IVC	202°C A *
EVO	482°C A *
EVC	8°C A *
Swirl ratio	0.9
Fueling system	Fully premixed and GDI
GDI injector	Bosch, 8-hole
Included angle	70°
Hole size	Stepped-hole, min. dia. = 0.125 mm
Injection pressure	120 bar
Engine speed	1200 rpm
Centerline intake temperature	30 to 160°C
Intake pressure (abs.)	1.0 to 3.6 bar
Coolant temperature	100°C

* 0°C A is taken to be TDC intake. The valve-event timings are for 0.1 mm lift.

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. 1a, 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 some conditions, a combination of premixed and DI fueling was used. For all conditions using the premixed fueling system, the intake temperature was kept at 60°C or higher to prevent

condensation of the pre-vaporized fuel in the intake. A positive displacement fuel flow meter was used to determine the amount of fuel supplied.

Similar to some of our other recent works [15,16,25], 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 the conventional air-based ϕ when no EGR is used. It should also be mentioned that the air-based ϕ averaged over the cylinder contents is ≤ 1 for all conditions presented, so there is always sufficient oxygen for complete combustion.

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 to 160°C. All data were taken at an engine speed of 1200 rpm.

Fuel Specifications

Five fuels were used for the current study:

1. **E0:** 87-AKI gasoline supplied by Chevron-Phillips Chemical Co., specifications are listed in Table 2.
2. **E10:** 10% (liquid vol.) neat ethanol mixed with the 87-AKI Chevron-Phillips gasoline.
3. **E20:** 20% (liquid vol.) neat ethanol mixed with the 87-AKI Chevron-Phillips gasoline.
4. **E100:** neat ethanol (anhydrous).
5. **CF-E0:** 93-AKI certification gasoline supplied by Haltermann Solutions, specifications are listed in Table 2.

Note that the 87-AKI gasoline used here is the same fuel used in one of our most recent studies [15], and it meets the same specifications as the fuels used in our other recent studies of boosted LTGC/HCCI [21] and LTGC with PFS [16], but it is from a different batch, and there are small differences in composition and the distillation curve.

As discussed in the introduction, blending ethanol with the 87-AKI gasoline increases its AKI. Ethanol has an AKI of 99.5, and for lack of better information, it has sometimes been assumed that the AKI of gasoline/ethanol blends could be computed by linearly combining the AKIs of the constituents. However, a recent work by Anderson et al. [26] has shown that adding ethanol has a quite non-linear effect on

2. This exhaust pressure is much lower than would be required to drive an actual turbocharger. However, the current work focuses on changes in IMEP_g which is hardly affected by using a higher, more-realistic exhaust pressure, as was shown in a previous study [21]. In this earlier work, a realistic back pressure (based on a 40% combined turbo and compressor efficiency) for a $P_{in} = 2$ bar condition reduced the IMEP_g by < 0.2% (or about a 0.1%-unit decrease in thermal efficiency), which is in the noise of the data. (Net IMEP was reduced by about 4%.) Thus, not considering the higher exhaust pressure is not expected to significantly affect the results presented here.

the autoignition reactivity, with the AKI increasing much more rapidly for blends of up to 30% ethanol than for higher ethanol concentrations. Table 3 gives the RON, MON, and AKI for all the fuels tested, with the values for E10 and E20 estimated from the blending curves in [26]. The AKIs for E10 and E20 based on the linear blending assumption are also given for reference.

Table 2. Fuel Properties.

	Chevron-Phillips Research-Grade Gasoline *	Haltermann Solutions Certification Gasoline
Antiknock Index (R+M)/2	86.9	92.7
RON	91.0	96.6
MON	82.7	88.7
Specific gravity	0.746	0.743
Carbon [wt%]	86.37	86.49
Hydrogen [wt%]	13.55	13.34
Oxygen [wt%]	0.08	None detected
A/F Stoichiometric	14.60	14.56
Lower Heating Value, gas-phase [MJ/kg]	43.10	42.97
LHV for stoichiometric charge [MJ/kg]	2.763	2.762
Hydrocarbon type [vol%]		
Aromatics	24.5	34.3
Olefins	5.1	0.12
Saturates	70.1	64.0
Distillation [°C]		
5%	56.9	43.3
10%	63.4	52.8
30%	78.9	80.6
50%	96.1	106.1
70%	114.1	117.8
90%	147.3	159.4
95%	164.9	172.8

* Based on analysis provided by Paragon Labs, Inc.

Table 3. Fuel Octane Ratings: RON, MON, and AKI

	RON	MON	AKI	AKI linear blending
E0	91.0	82.7	86.9	N/A
E10	95	86	90.5	88.2
E20	98	87.5	92.8	89.4
E100	109	90	99.5	N/A
CF-E0	96.6	88.7	92.7	N/A

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 1/4° 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 [7]). Computations were performed for each individual cycle, disregarding heat transfer and assuming a constant ratio of specific heats [22].³ 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 PPRRs and ringing intensities are computed from the same low-pass-filtered pressure data. For each cycle, the maximum PRR (*i.e.* PPRR) 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 discussed in numerous previous works (e.g., [2,27, 28, 29, 30, 31]), LTGC/HCCI combustion is often limited by high PPRRs, 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 LTGC 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 (RI) developed by Eng [29] is used as a measure of the propensity for engine knock.

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

Where $(dP/dt)_{\max}$, $P_{\max}$, and $T_{\max}$ are the maximum values of PRR in real time (*i.e.* the PPRR), 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 $RI = 5 \text{ MW/m}^2$ was selected as the ringing limit for operation without knock. This is the same limit used in our previous

3. This 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 crank angle window of interest shows good agreement, with differences typically being less than one or two tenths of a degree.

works, for example [15,21,30], and a more complete discussion of the selection of this value may be found in Ref. [30]. This value, 5 MW/m², corresponds to about 8 bar/°CA at 1200 rpm, naturally aspirated. However, it should be noted that the allowable PPRR 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 RI rising above 5 MW/m², giving confidence in this correlation.

A second method of computing the HRR was used for detailed HRR-curve analysis of the ITHR presented in the second part of the results. Here, the heat release was computed in a more refined way from the ensemble-averaged pressure trace (with the 2.5 kHz low-pass filter applied), using the Woschni correlation for heat transfer [22]. Using the ensemble-averaged pressure trace has benefits from the standpoint of reduced noise on the heat-release traces. On the other hand, it can lead to overestimated burn durations if the cycle-to-cycle variations are large. However, for the condition where this was applied, the phasing was fairly stable with the standard deviation of CA50 over 100 fired cycles averaging about 1.2° CA. Moreover, the results of this detailed HRR analysis are mainly used for comparisons of the early part of the heat release, leading up to hot ignition. Since these early autoignition reactions persist for 15° - 25° CA, the relatively small cycle-to-cycle variation in CA50 will have little effect. Finally, when viewing the early part of the HRR, a highly amplified scale is required, and small quasi-periodic fluctuations were evident. These are thought to be an artifact of the HRR being based on the derivative of the pressure trace and the low-pass filtering of the pressure trace discussed above. To mitigate these effects for a more clear comparison of the data for different fuels, a piecewise smoothing filter was applied to the HRR traces as discussed in detail in Ref. [21].

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 below) US-2010 limits and is not reported.

RESULTS AND DISCUSSION

Premixed LTGC (HCCI) Autoignition Reactivity

Experiments were conducted at both naturally aspirated and boosted conditions to determine how the changes in AKI between the various fuels (cf. Table 3) affect autoignition reactivity under conditions of interest for LTGC. The premixed fueling system was used to ensure that the charge was well mixed (HCCI conditions) to eliminate any potential variation due to differences in the ϕ -sensitivity of the fuels. For all tests, a constant $\phi_m = 0.4$ was used, and combustion phasing was adjusted to maintain a constant RI of 5 MW/m².

For naturally aspirated operation ($P_{in} = 1$ bar), all fuels required intake heating to achieve autoignition, so no EGR was used, and the amount of intake heat required provides a measure of the relative autoignition reactivity of the fuels. Figure 2a. shows the results for E0, E20, E100, and CF-E0. Both the directly measured intake temperature (T_{in}) and a bottom dead center temperature (T_{BDC}) determined by the method described in [34] are presented for each fuel. It should be noted that although T_{in} was adjusted to obtain a constant ringing intensity, CA50 was also nearly constant for all the fuels, $365.53 \pm 0.27^\circ$ CA. As the figure shows, E0, E20, and E100 require almost the same temperature for autoignition, $T_{in} \sim 150^\circ\text{C}$, which corresponds to a T_{BDC} of $\sim 163^\circ\text{C}$. E10 data were not acquired, but they can be expected to be similar to E0 and E20. Thus, this 87-AKI gasoline and ethanol have very similar autoignition reactivities at naturally aspirated conditions, despite the large difference in their AKIs (86.9 vs. 99.5). As might be expected, the E20 blend of these two constituents also has a similar reactivity. In contrast, the CF-E0 is less reactive at these conditions, requiring a T_{BDC} of 169°C , which is 6 - 7°C hotter than for the other fuels. Thus, the difference in LTGC autoignition reactivity between the two zero-ethanol gasolines shifts in the direction that would be expected from the difference in their AKIs.

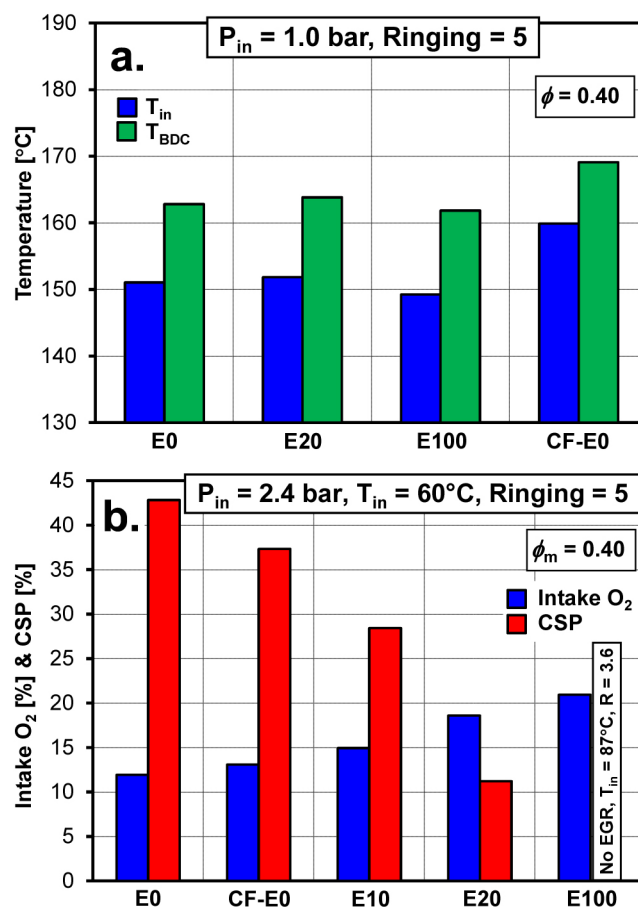


Figure 2. Comparison of autoignition reactivity of the fuels tested at (a) naturally aspirated ($P_{in} = 1.0$ bar) and (b) typical boosted ($P_{in} = 2.4$ bar) conditions.

For boosted operation, however, the relative reactivities of the fuels are quite different. Figure 2b shows the LTGC autoignition reactivities for all five fuels at a typical intake-pressure boosted

condition of $P_{in} = 2.4$ bar. Intake boosting significantly enhances the autoignition reactivities of all the fuels, except for E100 which is only moderately enhanced.

Therefore, T_{in} was reduced to 60°C for all fuels except E100, and a variable amount of EGR was added to further reduce the reactivity. The amount of EGR was adjusted for each fuel to obtain a moderately retarded combustion phasing that gave a RI of 5 MW/m² (CA50 = 374.0 to 376.3° CA). For combustion that is lean overall, as it is here, the EGR may be considered to be composed of a mixture of unreacted air and the products of complete stoichiometric combustion (CSP). Thus, the relative autoignition reactivities of the various fuels may be determined by comparing either the amount of CSP (provided by the EGR) or by the intake oxygen concentration required to obtain RI = 5 MW/m².

As can be seen in Fig. 2b, E0 is the most reactive fuel, requiring the most CSP and the lowest intake oxygen (O_2) concentration. Adding 10 and 20% ethanol to this fuel progressively reduces the LTGC autoignition reactivity as measured by either the reduction in CSP or the increase in intake O_2 . The change in intake CSP/ O_2 is not quite linear with the ethanol addition with the change from E0 to E10 requiring only about 0.82 of the change in CSP/ O_2 required for the change in fuels from E10 to E20. This decrease in autoignition reactivity with ethanol addition corresponds to the increase in AKI shown in Table 3. However, Table 3 shows a larger increase in AKI or RON for the change from E0 to E10 than for the change from E10 to E20, rather than inverse found for the relative changes in CSP/ O_2 as noted above. Shifting the blend all the way to 100% ethanol, significantly reduces the autoignition reactivity beyond E20, requiring a T_{in} of 87°C and no EGR/CSP.

Interestingly, CF-E0 is the second most reactive fuel for the boosted conditions shown in Fig. 2b, despite its AKI being almost the same as E20 (*cf.* Table 3). In fact, the required change in CSP/ O_2 as fueling in shifted from E0 to CF-E0 is less than half that required for a shift in fuels from E0 to E10. Thus, under boosted conditions, CF-E0 is a little less reactive than E0, similar to the difference in reactivity of these two fuels at naturally aspirated ($P_{in} = 1$ bar) conditions.

In summary, adding ethanol to this E0 base gasoline has almost no effect on its autoignition propensity for naturally aspirated conditions, but it has a large effect for boosted operation. As a result, blending ethanol with gasoline at the E10 or E20 levels appears to be advantageous for LTGC/HCCI combustion. It requires essentially no change in intake heating at naturally aspirated conditions, so there is no operational penalty in this respect. More importantly, it significantly reduces autoignition-reactivity enhancement with boost, which is expected to be advantageous for extending the high-load limit since less EGR is required for CA50 control, leaving more oxygen available for combustion. In contrast, decreasing the gasoline's reactivity by changing the base-fuel composition requires additional intake heat for non-boosted conditions, and it provides a relatively small decrease (compared to ethanol blending) in reactivity for boosted operation. Finally, operation with E100 has no penalty for non-boosted operation, but the higher T_{in} requirements for boosted operation are detrimental since they will act to reduce thermal efficiency and to limit the charge-mass inducted, reducing the load

compared to the other fuels, as discussed in the Introduction and with data presented later. With these findings in mind, additional factors affecting high-load, boosted operation with these fuels will be examined, and boosted performance will be evaluated, in the following sections.

ITHR, Stability and CA50 Retard

Background

The ability to retard CA50 to reduce the peak HRR [35] with good combustion stability is critical for achieving high loads without engine knock for boosted premixed LTGC (HCCI) [16]. Previous works have shown that for typical gasoline and gasoline-like fuels, the stability with CA50 retard is related to the temperature-rise rate (TRR) leading up to the onset of the main combustion (*i.e.* just prior to hot ignition) [36] (an example will be presented below in Fig. 3). Maintaining a strong TRR as combustion is retarded after TDC, requires that the early autoignition reactions provide sufficient heat release to overcome the expansion cooling produced by the piston motion. This required heat release, termed the intermediate temperature heat release (ITHR), occurs at temperatures below those of hot ignition but above those of the more well-known low-temperature heat release (which typically occurs 10 - 20° CA before TDC) [21,37,38,39,40,41].⁴ Previous work [21] has shown that for E0,⁵ the ITHR is significantly enhanced as the intake boost is increased and T_{in} is reduced to compensate for the enhanced autoignition reactivity caused by the boost. This is fortuitous because it allows CA50 to be more retarded for boosted operation, which compensates for the higher PPRR that typically occur with boost due to the greater mass of the combusting charge. Moreover, this enhancement of ITHR with boost appears to be typical of most gasolines, since the performance data (presented later in this article) indicate that it also occurs for CF-E0, which has a significantly different composition from E0 (*cf.* Table 2), and it is also observed for a commercial regular-grade gasoline [33]. This boost enhancement of ITHR has also been observed in nheptane/ethanol mixtures [41].

Figure 3, adapted from [21], shows an example of the increase in ITHR (Fig. 3b) with boost, and the resulting increase in TRR (Fig. 3a). These data are for the maximum load point at each boost level, and for $P_{in} \leq 2.4$ bar, they show the most retarded CA50 that could be achieved with good stability at that boost level. It can be seen that the knee in the temperature traces, which marks the onset of hot ignition, shifts to later crank angles as P_{in} is progressively increased from 1.0 to 2.0 bar. This greater retard allows higher loads without knock, and it is possible because of the increased ITHR, as discussed above. It should be noted that this enhancement of ITHR with boost appears to saturate at about $P_{in} = 1.8 - 2.0$ bar which corresponds to the curves with the most retarded combustion phasing in Fig. 3a. For $P_{in} \geq 2.6$ bar, the required EGR levels are so high that the mixture becomes stoichiometric, and ϕ_m must be reduced as P_{in} is increased above 2.6 bar, so that the EGR fraction can be increased, as discussed in the

4. The dominant reaction pathways for the ITHR have been investigated using detailed chemical-kinetic mechanisms for a gasoline surrogate mixture [40] and ethanol/n-heptane mixtures [41] and found to differ from those of the low-temperature heat release (LTHR).

5. The E0 gasoline used in this previous study meets the same specifications as the E0 used for the current study, but it is from a different batch, so there are small differences in composition and the distillation curve.