

Increasing the Load Range, Load-to-Boost Ratio, and Efficiency of Low-Temperature Gasoline Combustion (LTGC) Engines

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ABSTRACT

Low-temperature gasoline combustion (LTGC) has the potential to provide gasoline-fueled engines with efficiencies at or above those of diesel engines and extremely low NO_x and particulate emissions. Three key performance goals for LTGC are to obtain high loads, reduce the boost levels required for these loads, and achieve high thermal efficiencies (TEs). This paper reports the results of an experimental investigation into the use of partial fuel stratification, produced using early direct fuel injection (Early-DI PFS), and an increased compression ratio (CR) to achieve significant improvements in these performance characteristics. The experiments were conducted in a 0.98-liter single-cylinder research engine. Increasing the CR from 14:1 to 16:1 produced a nominal increase in the TE of about one TE percentage unit for both premixed and Early-DI PFS operation. Compared to fully premixed fueling, Early-DI PFS allowed significant CA50 advancement, resulting in higher TEs for both CRs, with peak gross-indicated TEs of 48.4% and 49.8% for CRs of 14:1 and 16:1, respectively. Early-DI PFS was also found to be very effective for increasing the high-load limit over a range of intake boost pressures for both CRs. For CR = 14:1, a maximum load of 19.4 bar IMEP_g (gross indicated mean effective pressure) was achieved with an intake pressure (P_{in}) of 3.0 bar absolute using Early-DI PFS, compared to only 17.2 bar IMEP_g for premixed fueling at this P_{in} . Perhaps more importantly, $P_{in} = 3$ bar is substantially less than the $P_{in} = 3.45$ bar required to reach this load for premixed fueling, which also required the use of a less reactive E20 gasoline. For $P_{in} > 2.5$ bar, the maximum load was lower with CR = 16:1; however, for $1.6 \leq P_{in} \leq 2.5$ bar, higher loads were achieved with CR = 16:1.

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INTRODUCTION

Low-temperature gasoline combustion (LTGC), based on the compression ignition of a premixed or partially premixed dilute charge, can provide gasoline-fueled engines with efficiencies at or above those of diesel engines. As such, LTGC offers the potential for substantial reductions in fuel consumption and CO₂ emissions compared to gasoline spark-ignition engines. Also, by providing a means of producing high-efficiency engines that operate on light distillates, LTGC complements diesel engines which use middle distillates, for more effective overall use of crude-oil supplies and lower total CO₂ production. Furthermore, engine-out NO_x and particulate emissions can be extremely low for well-characterized LTGC combustion, reducing aftertreatment requirements, and therefore, the cost of the overall package. However, several technical challenges must be overcome before LTGC engines are commercially viable.

Related to these challenges, three key performance goals for LTGC are to obtain high loads, reduce the intake-pressure boost levels required for these high loads, and achieve high thermal efficiencies (TEs). Early work on LTGC focused mainly on well-premixed operation termed homogeneous charge compression ignition (HCCI) [1, 2, 3, 4, 5, 6, 7, 8, 9, 10, 11], and the majority of this work involved studies at naturally aspirated conditions, which limited the maximum load to about 5 bar indicated mean effective pressure, gross (IMEP_g),

due to the knock/stability limit of LTGC [7,11,12].¹ Initial efforts to apply intake boosting to LTGC/HCCI for higher loads while maintaining high dilution levels resulted in only limited success due to overly rapid combustion and engine knock [14,15]. Following this, various groups pushed boosted HCCI-like low-temperature combustion (LTC) to loads of about 16 bar IMEP_g, using fuels like natural gas and ethanol [15,16], or using diesel-type direct injectors (DI) with both gasoline-like or diesel-like fuels [17,18]. However, for the latter, NO_x and smoke emissions were problematic (above US 2010 limits) particularly at higher loads. More recently, loads of 16 and 18 bar IMEP_g were achieved with ultra-low NO_x and particulate emissions using well-mixed HCCI with a regular (87 AKI) E0 gasoline and an E10 blend comparable to mid-grade pump gasoline (90.5 AKI), respectively [19,20]. In these works, the combustion heat-release rate (HRR) was controlled to prevent knock by retarding combustion phasing through a combination of reduced intake temperatures and well-controlled EGR addition.

¹ The classic knock/stability limit of LTGC/HCCI occurs when further increases in fueling would cause knock, but the knock cannot be controlled by further combustion-phasing retard because combustion becomes unstable [7,11,12]. At these high-load limit conditions, when the combustion becomes unstable, it either (a) advances to a full misfire or goes to uncontrollable runaway knock [7,8,13], so the definition of an allowable stability limit (e.g. 3% COV of IMEP_g) is generally not meaningful.

Using this same approach with the fuel ethanol content increased to 20% (E20) (93 AKI) allowed loads up to 20 bar IMEP_g [21]. For these high-load well-mixed studies, NOx and particulate emissions were more than factor of 10 below US 2010 limits for all boosted conditions.

These high-load limits achieved with typical gasoline-like fuels (16 – 20 bar IMEP_g) for well-mixed LTGC are comparable to those of turbocharged diesel engines with a similar cylinder-head pressure limits (155 bar). However, the boost pressures required to achieve them were fairly high, 3.2 – 3.6 bar absolute, for 16 and 20 bar IMEP_g, respectively. Diesel-like high loads have also been reached with the DI diesel approach to LTGC, but similar high boost levels were typically required [22, 23, 24, 25]. In practice, it could be very difficult for a turbocharger to produce these boost levels at the stated loads for a high-efficiency engine that has a relatively low-enthalpy exhaust stream.

To overcome this problem and achieve high loads with less boost requires that the charge be less dilute, so the engine operates with essentially the same fueling rates, but at lower boost pressures. Two features of gasoline autoignition can help facilitate this. First, increased boost pressure enhances the autoignition reactivity of gasoline, reducing the requirement for intake heating or retained hot residuals. This extends the load increase 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 TE as discussed below. Second, as boost levels increase, the autoignition reactivity of typical gasoline fuels becomes sensitive to the local equivalence ratio (ϕ) (i.e. the fuel becomes ϕ -sensitive) [26]. Under these conditions, regions with higher ϕ autoignite faster than regions of lower ϕ , so if the charge mixture contains a distribution of equivalence ratios, the charge will autoignite sequentially, reducing the peak heat-release rate (HRR) [26,27].

This ϕ -sensitivity of gasoline under boosted conditions can be exploited to extend the high-load limit and/or achieve higher efficiencies through a technique known as partial fuel stratification (PFS). With this technique, direct fuel injection is used to produce a non-uniform fuel distribution in the cylinder that autoignites sequentially. 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, which increases the TE, or a combination of increased load and efficiency improvements can be made [11,26,27]. In these previous works a technique subsequently termed standard PFS (std-PFS) was used in which the majority of the fuel (80 – 90%) was premixed, and the remainder was supplied by DI in the latter part of the compression stroke to produce the PFS.

For premixed fueling, as used for the high loads (16 – 20 bar IMEP_g) mentioned above, or for std-PFS, the intake temperature (T_{in}) must be kept sufficiently high to prevent fuel condensation in the intake system. Accordingly, a minimum T_{in} of 60°C was used by the authors for these previous studies [19, 20, 21]. This T_{in} is well below those required to obtain autoignition for naturally aspirated operation, but with sufficient intake boost, the charge becomes overly reactive with

this T_{in} , and EGR is required to prevent overly advanced combustion timing. The amount of EGR required increases with boost pressure and with fuel load for a given boost. As a result, at higher boost pressures, the amount of EGR can become so high that it limits the oxygen available for combustion and thus the maximum load that can be achieved at a given P_{in} [19–21,26,].

DI fueling eliminates concerns about fuel condensation in the intake system, which allows the use of T_{in} s down to near ambient values, ~30°C, or those representative of turbo-charged engines with intercoolers, 40°C, as demonstrated in recent works by the authors [20,21].² Moreover, these works showed that even for DI fueling early in the intake stroke (60° CA after top dead center [TDC] intake) the charge mixture is not completely uniform by the time combustion occurs (see images in [20]), and the resulting mixture distribution works well for PFS, even though the amount of stratification has not been optimized. With these features, this technique, termed early-DI PFS (E-DI PFS), has several advantages for increasing the load for a given intake boost, i.e. increasing the load-to-boost ratio, compared to premixed fueling. First, reducing T_{in} from 60°C to 30 or 40°C increases the charge mass inducted into the cylinder. Second, with the lower initial charge temperature, less EGR is required to control the increase in autoignition reactivity with boost, which leaves more air available for combustion, so fueling rates can be increased for a given boost pressure for conditions where the maximum load was oxygen limited for premixed fueling. Third, the PFS effect reduces the timing retard required to control knock, which tends to improve combustion stability, allowing higher loads for conditions that are not oxygen limited (i.e. conditions that are knock/stability limited [7,19]). Finally, TEs are typically higher for E-DI PFS than for premixed fueling, which gives an additional modest increase in the load-to-boost ratio.

Many of these features of E-DI PFS with reduced T_{in} (30 or 40°C) act to increase the TE as well as increasing the high-load limit for a given intake pressure (P_{in}). First, lower T_{in} s result in lower combustion temperatures for the same load. This occurs both because combustion temperatures are lower when initial temperatures are lower, and because, with the charge density increase due to the lower T_{in} s, the same load (i.e. IMEP_g) can be produced with a more dilute charge, which also reduces combustion temperatures. Second, lower T_{in} s significantly reduce the amount of EGR required to retard the combustion timing sufficiently to prevent knock. Both the lower combustion temperatures and the reduced EGR act to increase the specific-heat ratio ($\gamma = c_p/c_v$) of the combustion products allowing more work to be extracted during the expansion stroke, increasing the TE [19,28]. The lower temperatures also act to reduce heat transfer losses. Finally, with PFS, less timing retard is required to control knock, which increases the effective expansion ratio, and therefore, the TE.

The TE can also be increased by increasing the geometric compression ratio (CR) of the engine. In addition to directly increasing the expansion ratio, increasing the CR reduces the temperature required for autoignition because it results in higher pressures at the end of the compression stroke. These higher

² These temperatures, 30 or 40°C, were selected because they were the lowest values that could be reliably obtained with the air supply system in the low-boiling-point study [28] showed that the TE increased with T_{in} reduction down to 40°C, but that reducing T_{in} to 30°C produced no additional increase in TE.

pressures accelerate the autoignition reactions, so the same combustion timing can be achieved with lower near-TDC temperatures. This reduction in the autoignition temperature and the greater temperature increase during the compression stroke act together to reduce the required T_{in} for naturally aspirated and low-boost conditions. The reduced T_{in} increases the charge density, which acts to increase both the TE and maximum load at a given P_{in} , as discussed above. However, for higher boost levels, T_{in} is reduced to the minimum allowable level (60°C for premixed and 30–40°C for DI) and EGR is added to control combustion timing. For these cases, the higher CR requires more EGR, which can cause the maximum load to become oxygen limited at a lower P_{in} . This effect acts to limit the maximum load at the cylinder head pressure limit to a lower value for the higher CR.

The objective of the current work is two-fold:

1. To explore the potential of E-DI PFS for improving the performance of intake-boosted LTGC engines in terms of obtaining high loads, reducing the boost pressure required for these loads, and for improving the TE.
2. To determine the amount that the TE can be improved by increasing the CR from 14:1 to 16:1, and the effect of this CR increase on the high-load limits and load-to-boost ratio for both premixed and E-DI fueling.

Experiments were conducted in a single-cylinder LTGC engine that is described in the next section along with the fuel composition and data-acquisition techniques. Then, the results are presented in four

parts: 1) the ability of E-DI PFS to increase the high-load limit of boosted LTGC compared to premixed fueling at CR = 14:1, 2) the effect of increasing the compression ratio from 14:1 to 16:1 on the thermal efficiency for both naturally aspirated and boosted conditions, 3) detailed comparison of the changes in TE with load for E-DI and premixed fueling at a typical boosted condition ($P_{in} = 240$ kPa) for CR = 16:1, and 4) the high-load limits at CR = 16:1 for E-DI and premixed fueling over a range of intake pressures compared to the load limits at CR = 14:1. 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 1a 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 the same as those used in previous studies of intake pressure boost [19, 20, 21, 28], except that a CR = 16:1 piston is also included for this study. Additional information on the facility may be found in Ref. [6].

The active LTGC cylinder is fitted with either a CR = 14:1 or 16:1 custom piston as shown in Fig. 1b. Both pistons provide an open combustion chamber with a large squish clearance.

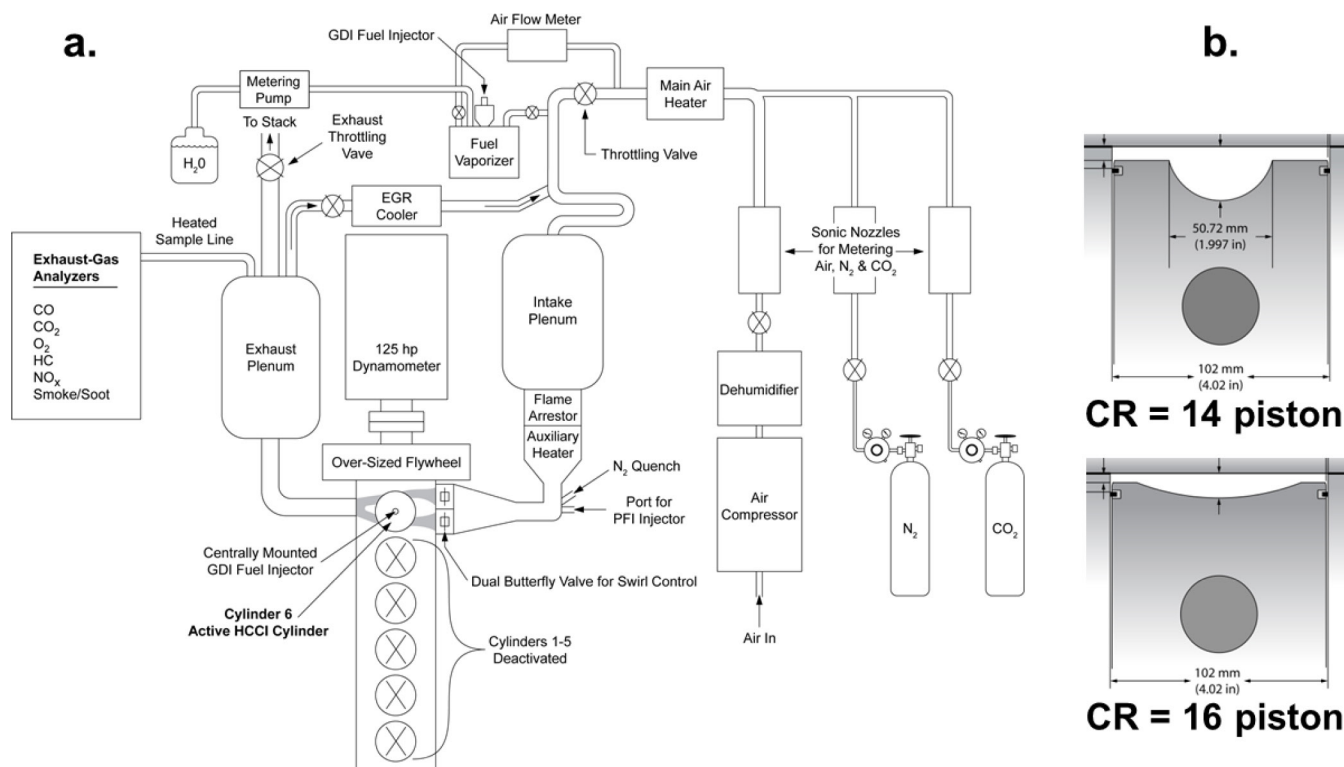


Figure 1. a) Schematic of the LTGC/HCCI engine facility, and b) combustion chamber geometry for the CR = 14:1 and 16:1 pistons at TDC

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 ratios	14:1 and 16:1
Number of valves	4
IVO	0°CA *
IVC	202°CA *
EVO	482°CA *
EVC	8°CA *
Swirl ratios	0.9 (Head #1) and 0.7 (Head #2)
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°CA is taken to be TDC intake. The valve-event timings are for 0.1 mm lift.

The CR = 16:1 piston has a broad shallow bowl, whereas the CR = 14:1 piston has a narrower, quasi-hemispherical bowl. However, comparison tests showed minimal differences in performance and emissions between this narrower-bowl piston and a CR = 14:1 piston with a broad shallow bowl similar to the CR = 16:1 piston shown in Fig. 1b. A more complete discussion of the comparison between these CR = 14:1 pistons may be found in Ref. [8]. Both the CR = 14:1 and 16:1 pistons in Fig. 1b provide a small topland ring crevice, amounting to only 2.1 and 2.4% of the TDC volume, respectively.

For almost all the data presented in this study, the engine was equipped with the same cylinder head used in all our previous works, Head #1, which gave a swirl ratio of 0.9. However, insufficient data were acquired in this original dataset to clearly show the changes in TE between CR = 14:1 and 16:1 as P_{in} was increased from 1.0 to 2.4 bar. Accordingly, one additional dataset was acquired after the engine had been switched to a different cylinder, Head #2. These data are presented in Figs. 9 and 10, and they are the only data in this article acquired with Head #2. Head #2 has an identical valve configuration to Head #1, and it also gives a similar low swirl ratio of 0.7, but the intake-port geometry is not identical. Head #2 was installed for reasons unrelated to the current study, and the differences in performance for the data presented in this article (Figs. 9 and 10) are small.

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.

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. For the current study, intake pressures varied from 1.0 bar (simulating naturally aspirated conditions) to 3.0 bar. All pressures given in this article are absolute unless specifically specified otherwise. 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. 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 [19].

Because brake torque measurements are not reliable for a single-cylinder conversion from a six-cylinder engine, as used for this study (due to high friction losses), all loads and TEs presented in this article are indicated values.

In this study both premixed and 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 [19,21,26,28], 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 Document Sharing Hub

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 in the cylinder for complete combustion.

Fuel Specifications

For all the data acquired using Head #1, the engine was fueled with a standard Tier II certification gasoline with an AKI = 93 and zero ethanol supplied by Haltermann Solutions. It will be referred to as CF-E0 (certification fuel, zero ethanol). For the data acquired using Head #2 (Figs. 9 and 10 only), the fuel was a research-quality, regular-grade E10 gasoline called RD5-87. This fuel was designed to be representative of current regular-grade E10 fuels sold in the U.S. The specifications for both fuels are listed in Table 2.

Table 2. Fuel Properties for the Certification Gasoline (CF-E0), and the research quality regular-grade gasoline (RD5-87).

	CF-E0, Haltermann Certification Gasoline*	RD5-87, Research-Quality, Regular-Grade, E10 Gasoline
Antiknock Index (R+M)/2	92.7	88.4**
RON	96.6	92.1
MON	88.7	84.8
Specific gravity	0.743	0.749
Carbon [wt%]	86.49	82.26
Hydrogen [wt%]	13.34	13.98
Oxygen [wt%]	None detected	3.67
A/F Stoichiometric	14.56	14.05
Lower Heating Value, gas-phase [MJ/kg]	42.97	41.639
Hydrocarbon type [vol%]		
Aromatics	34.3	22.12
Olefins	0.12	6.51
Saturates	64.0	60.09
Ethanol	0	9.97
Distillation [°C]		
5%	43.3	54.5
10%	52.8	57.5
30%	80.6	64.8
50%	106.1	99.8
70%	117.8	125.3
90%	159.4	155.9
95%	172.8	171.2

* Based on analysis provided by Haltermann

** Regular-grade gasolines sold in the U.S. that contain 10% ethanol commonly have AKIs ~ 88, which is above the minimum value of 87 shown on the pump. The 88.4 AKI of this RD5-87 is well within the range of current, regular-grade pump gasolines, especially considering that RON or MON values can vary up to 0.7 octane units between tests done at different laboratories.

In addition to the two fuels listed in Table 2 that were used for the current study, some data are presented from earlier studies [20,21] as a review and lead-in to the current study. These review data were acquired using a zero ethanol (E0), regular-grade gasoline and with this E0 gasoline blended with 10 or 20% ethanol by volume. These

three fuels are referred to as E0, E10, and E20. This E0 gasoline was designed to be typical of U.S. regular-grade (87 pump octane number) gasoline prior to the use of ethanol. It had a measured AKI = 86.9, and the E10 and E20 had AKIs of 90.5 and 92.8, respectively, as estimated from the non-linear blending curves of Anderson *et al.* [29]. Additional information on these E0, E10, and E20 fuels may be found in [21]. CF-E0 was also used for some of the review data.

Data Acquisition

Cylinder pressure measurements for Head #1 were made with two very similar transducers (AVL QC33C and AVL QC34C) mounted in the same port in the cylinder head approximately 42 mm off center. The QC33C was used for most of the data presented; however, this transducer failed toward the end of the study, and it was replaced with the QC34. The QC34C was used only for the CR = 16:1 data in Figs. 11 and 12. For Head #2 an AVL GH15D pressure transducer was used, and it was mounted in approximately the same location as the transducers for Head #1.

The pressure transducer signals were digitized and recorded at $\frac{1}{4}^\circ$ 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. CA50 was determined from the cumulative apparent heat-release rate (AHRR), computed from the cylinder-pressure data [30] (after applying a 2.5 kHz low-pass filter [6]). Computations were performed for each individual cycle, disregarding heat transfer and assuming a constant ratio of specific heats [19].³ The average of 100 consecutive individual-cycle CA50 values were then used to monitor 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.

LTGC/HCCI combustion is often limited by high PPRRs, which cause acoustic oscillations in the charge gas resulting in audible engine knock [7,12,31, 32, 33]. 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 θ , where θ is a variable representing $^\circ$ CA). However, this does not correctly reflect

³ The specific heat ratio (γ) used to compute the AHRR at each condition 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 thermodynamic properties over the cycle and a Woschni heat transfer correlation [19, 21] shows good agreement, with differences typically being less than one or two tenths of a degree.

the potential for knock with changes in intake boost or engine speed. In this work, the correlation for ringing intensity (RI) developed by Eng [32] 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 $\text{RI} = 5 \text{ MW/m}^2$ was selected as the ringing limit for operation without knock. This is the same limit used in our previous works, for example [7,19,21,34], and a more complete discussion of the selection of this value may be found in Refs. [7,34]. This value, 5 MW/m^2 , corresponds to about $8 \text{ bar/}^\circ\text{CA}$ 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/m^2 , giving confidence in this correlation.

Exhaust emissions data were also acquired, with the sample being drawn from the exhaust plenum using a heated sample line (see Fig. 2a). CO , CO_2 , HC , NO_x , and O_2 levels were measured using standard exhaust-gas analysis equipment. In addition, a second CO_2 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 CO_2 concentrations. Smoke measurements were also made with an automated smoke meter. However, smoke was not detected for any premixed operating points, and for all E-DI and PFS data reported, it was more than an order of magnitude or more below the US-2010 heavy-duty on-road emission standards. Similarly, NO_x emissions were well below US-2010 limits, and for all intake-boosted data they were more than an order of magnitude below these limits.

Engine Operation

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 .

To control combustion timing (CA50) a combination of intake temperature adjustment and EGR addition was used, depending on the amount of intake boost. For naturally aspirated operation ($P_{\text{in}} = 1 \text{ bar}$), intake heating was required to achieve autoignition for both CRs, so no EGR was used. Increasing the intake pressure

significantly enhanced the autoignition reactivity, so T_{in} was reduced with increased P_{in} until reaching to the lowest acceptable temperature (60°C for premixed and 30 or 40°C for E-DI fueling). For further increases in boost, T_{in} was held constant at these lowest values, and EGR was added to control combustion timing. Thus, comparisons of T_{in} provide a measure of autoignition reactivity for naturally aspirated and low-boost conditions, and the amount of EGR, or intake oxygen (O_2) provide a measure of reactivity for higher boost pressures. This transition of CA50 control from T_{in} to EGR typically occurred between $P_{\text{in}} = 1.6$ and 1.8 bar for $\text{CR} = 14:1$ and between 1.4 and 1.6 bar for $\text{CR} = 16:1$.

For all data presented, either the intake temperature or EGR level was adjusted to retard CA50 sufficiently to maintain a $\text{RI} \leq 5 \text{ MW/m}^2$ to prevent knock. Most intake-boosted data are acquired at $\text{RI} = 5 \text{ MW/m}^2$, which gives the most advanced CA50, and hence highest thermal efficiency, without knock. However, for naturally aspirated operation, $\text{RI} = 3 \text{ MW/m}^2$ was used for most data because it gives better combustion stability, i.e. less likely to lead to runaway knock or misfire [8]. The exception to this is the naturally aspirated data in Figs. 6 and 7, which was acquired at $\text{RI} = 5 \text{ MW/m}^2$, except as noted.

RESULTS AND DISCUSSION

High-Load Limits – $\text{CR} = 14:1$

Review of Premixed Fueling Results

Figure 2a shows the maximum loads achieved for LTGC/HCCI with premixed fueling for a $\text{CR} = 14:1$ in this laboratory, as reported in a recent paper by the authors [21]. All boosted data points were acquired with $\text{RI} = 5 \text{ MW/m}^2$ to maximize TE while avoiding knock. The four fuels used for this earlier work are described in the Fuel Specifications section above. For each fuel except E20, the maximum IMEP_g increases in approximately a linear manner with increasing boost pressure, up to a point, beyond which the rate of increase of IMEP_g with P_{in} slows, producing a curve in the plot. The zero ethanol (E0) 87 AKI fuel is the most reactive for intake boosted operation, and EGR addition begins with the $P_{\text{in}} = 1.8 \text{ bar}$ data point, as evident from the sharp drop in exhaust O_2 from $P_{\text{in}} = 1.6$ to 1.8 bar , as shown in Fig. 2b. As P_{in} is increased above 1.8 bar , the amount of EGR required to prevent overly advanced combustion timing also increases, as evident from the drop in exhaust O_2 from $P_{\text{in}} = 1.8$ to 2.6 bar . For this fuel, the exhaust O_2 is nearly zero at $P_{\text{in}} = 2.6 \text{ bar}$, meaning that the charge is stoichiometric with a very high level of dilution from the EGR ($\phi = 1.0$, $\phi_m = 0.47$). To overcome the additional autoignition enhancement for $P_{\text{in}} > 2.6 \text{ bar}$, an even higher EGR fraction is needed. This requires that the fraction of fresh reactants (i.e. fuel and air) in the charge be decreased, resulting in a reduction in the slope of the IMEP_g curve mentioned above (Fig. 2a). Thus, the maximum load for E0 is oxygen limited for $P_{\text{in}} \geq 2.6 \text{ bar}$, whereas it is knock/stability limited for $P_{\text{in}} < 2.6 \text{ bar}$. Despite being oxygen limited, a maximum load of 16.3 bar IMEP_g was reached with $P_{\text{in}} = 3.2 \text{ bar}$ at the cylinder-head pressure limit ($\phi = 1.0$, $\phi_m = 0.38$). A more complete discussion of this intake boost sweep, including cylinder pressure, mass-averaged temperature and HRR traces may be found in [19].

