

Boosted Premixed-LTGC / HCCI Combustion of EHN-doped Gasoline for Engine Speeds Up to 2400 rpm

Chunsheng Ji, John Dec, and Jeremie Dernotte
Sandia National Laboratories

William Cannella
Chevron Energy Technology Company

ABSTRACT

Previous work has shown that conventional diesel ignition improvers, 2-ethylhexyl nitrate (EHN) and di-tert-butyl peroxide (DTBP), can be used to enhance the autoignition of a regular-grade E10 gasoline in a well premixed low-temperature gasoline combustion (LTGC) engine, hereafter termed an HCCI engine, at naturally aspirated and moderately boosted conditions (up to 180 kPa absolute) with a constant engine speed of 1200 rpm and a 14:1 compression ratio. In the current work the effect of EHN on boosted HCCI combustion is further investigated with a higher compression ratio (16:1) piston and over a range of engine speeds (up to 2400 rpm). The results show that the higher compression ratio and engine speeds can make the combustion of a regular-grade E10 gasoline somewhat less stable. The addition of EHN improves the combustion stability by allowing combustion phasing to be more advanced for the same ringing intensity. The high-load limits of both the straight (unadditized) and additized fuels are determined, and the additized fuel is found to achieve a higher maximum load at all engine speeds and intake pressures tested, if it is not limited by lack of oxygen. The results reveal that the higher loads with EHN are the result of either reduced intake temperature requirements at naturally aspirated conditions or a reduction in heat release rate at higher intake pressures. Such effects are also found to increase the thermal efficiency, and a maximum indicated thermal efficiency of 50.1% is found for 0.15% EHN additized fuel at 1800 rpm and 180 kPa intake pressure. Similar to previous studies, the nitrogen in EHN increases NO_x emissions, but they remain well below US-2010 standards. Higher engine speeds are found to have slightly lower NO_x emissions for additized fuel at intake boosted conditions.

Keywords: HCCI, LTGC, Autoignition Reactivity, Combustion Stability, Ignition Improvers, 2-Ethylhexyl Nitrate

CITATION: Ji, C., Dec, J., Dernotte, J., and Cannella, W., "Boosted Premixed-LTGC / HCCI Combustion of EHN-doped Gasoline for Engine Speeds Up to 2400 rpm," *SAE Int. J. Engines* 9(4):2016, doi:10.4271/2016-01-2295.

INTRODUCTION

Well premixed low-temperature gasoline combustion (LTGC), also known as homogenous charge compression ignition (HCCI) [1], and partially stratified LTGC (HCCI-like) engines, are considered promising techniques to provide high thermal efficiencies (comparable to diesel engines), ultra-low NO_x emissions, and near zero particulate matter (PM) emissions [2, 3, 4, 5, 6, 7, 8, 9, 10]. High-volatility fuels, like gasoline, are advantageous for LTGC/HCCI, as these fuels are relatively easy to form into a premixed or partially premixed charge. As a result, conventional gasoline with an Anti-Knock Index (AKI) = 87 or above has been widely studied in HCCI and HCCI-like engines [e.g., 4, 5, 6]. Despite its high-volatility, the low autoignition reactivity of conventional gasoline is considered to be a handicap for its application in these engines, especially at naturally aspirated and low-to-intermediate intake boost conditions. For example, at naturally aspirated conditions, a large amount of hot residuals or a high intake temperature (T_{in}) is necessary for autoignition due to gasoline's low autoignition reactivity [4] under

LTGC/HCCI conditions. As a result, charge densities are much lower than those of typical engines, which is an important factor limiting the loads of LTGC/HCCI engines.

Mixtures of primary reference fuels (PRF) with low octane number (ON) have been used in the studies of HCCI engines [e.g., 11, 12, 13] to overcome the low reactivity of conventional gasoline.

Alternatively, low-octane gasoline has been shown to be more suitable for HCCI engines compared with conventional gasoline at naturally aspirated conditions [14]. However, these fuels are not readily available in the market. Recent studies show that some conventional diesel ignition improvers, such as 2-ethylhexyl nitrate (EHN), which is used to improve the Cetane Number (CN) of diesel fuels, can be used to enhance the autoignition of conventional gasoline in HCCI engines [e.g., 15, 16, 17, 18, 19]. Reitz and co-workers [15, 16, 17] found that EHN can enhance the autoignition reactivity of low reactivity fuels such as gasoline (hydrocarbon only, no ethanol), E10 (gasoline containing 10 vol% ethanol), and E10 and methanol. Hosseini et al. [18] indicated that a small amount of EHN

additive advanced the low-temperature heat release (LTHR) and high exhaust gas recirculation (EGR) rates were required to retard the combustion phasing.

A previous study by Ji et al. [19] was conducted to understand the effect of adding EHN to regular-grade (87 AKI) E10 gasoline on combustion in an HCCI engine with compression ratio (CR) 14:1 and 1200 rpm engine speed. The study found that EHN addition was effective for enhancing the HCCI autoignition reactivity of the E10 gasoline. The required T_{in} can be reduced significantly with a very small amount of this additive ($\leq 0.4\%$) at naturally aspirated conditions. The reduction in required T_{in} increased the charge density with a commensurate increase in gross indicated mean effective pressure (IMEP_g). This effect, combined with the greater allowable CA50 retard with good stability for the additized fuel, allowed the high-load limit to be increased substantially at naturally aspirated and low-boost conditions, by 30% at intake pressure, $P_{in} = 100$ kPa and 22% at $P_{in} = 130$ kPa. Ji et al. [19] also found that the EHN produced a small increase in the NO_x emissions for HCCI combustion approximately proportional to the additive concentration. About 30% of the nitrate groups in the EHN were converted to NO_x. This observation is consistent with that reported by Dempsey [17] and Ickes [20].

The objective of the current work is to further investigate the effects of EHN on E10 gasoline combustion in an LTGC engine. Fully premixed fueling, i.e. HCCI combustion is used to eliminate any potential for changes in mixture formation from affecting the results. Compared to our previous work [19] a higher CR = 16:1 is used for most studies, and engine speeds are varied up to 2400 rpm for naturally aspirated and moderately boosted conditions (up to 180 kPa absolute). Engine speed has been shown to have a significant effect on LTHR and intermediate temperature heat release (ITHR), combustion stability, and operating range [21, 22, 23, 24, 25] of HCCI engines. The effects of EHN on the high-load limits at each intake pressure are studied and compared at engine speeds of 1200,

1800, and 2400 rpm. The combustion stability at high engine speed and high CR is investigated and discussed. The LTHR and ITHR, which can affect stability, of both EHN-additized fuel and straight (unadditized) fuel are also compared at various engine speeds. Finally, the effects of EHN on gross indicated thermal efficiency (TE) and NO_x emissions are also investigated at different engine speeds.

The next section describes the experimental facility, data acquisition and analysis techniques, fuel/additive properties, and test conditions. Following this, the results are presented in two parts. The effects of CR = 16:1 compared to CR = 14:1 on the autoignition enhancement of additized and straight fuels in the HCCI engine are first presented and discussed. Then, the second part shows the effects of engine speed on combustion stability and high-load limits of both additized and straight fuels at various P_{in} s. The peak TE and NO_x emissions at various P_{in} s and engine speeds for both fuels are measured and discussed as well. Finally, the study is summarized, and conclusions are drawn in the last section.

EXPERIMENTAL SETUP

Engine Facility

As with our previous study [19], the HCCI research engine 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. As shown in Fig. 1a, the engine has been converted for single-cylinder operation by deactivating cylinders 1-5. Two CR (14:1 and 16:1) custom pistons are used to study the effect of CR on the autoignition reactivity of the additized fuel. The combustion chamber geometry of two pistons at TDC is shown in Fig. 1b. The CR = 14:1 piston provides an open combustion chamber with a large squish clearance and a quasi-hemispherical bowl and the CR = 16:1 piston has a broad shallow bowl. Both pistons provide a small topland ring crevice with less than 0.9% of the TDC volume. The engine specifications and operating conditions are listed in Table 1.

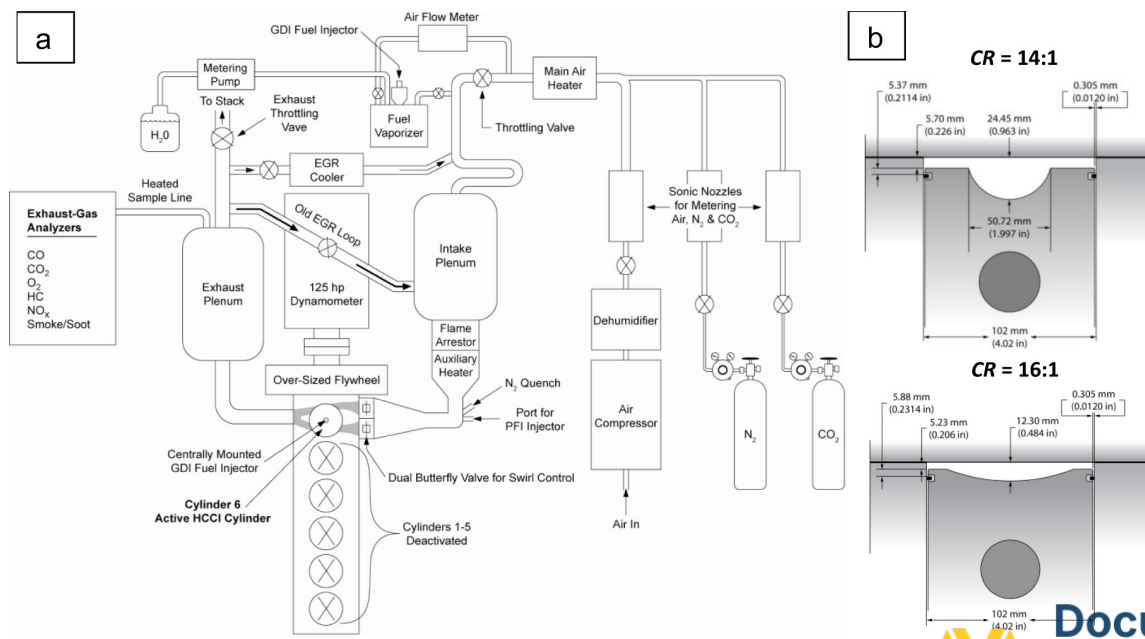


Figure 1. Schematic of HCCI engine facility (a) and combustion chamber geometry of the CR = 14:1 and 16:1 pistons, at TDC (b).

Premixed fueling is used in this study. The premixed fueling system, shown at the top of the schematic in Fig. 1a, consists of a gasoline direct injector (GDI) mounted in an electrically heated fuelvaporizing chamber and appropriate plumbing to ensure thorough premixing of the vaporized fuel with the air and EGR upstream of the intake plenum. A positive displacement fuel flow meter (Max Machinery P002) is used to determine the amount of fuel supplied.

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 and 16:1
No. of Valves	4
IVO	0° CA*
IVC	202° CA*
EVO	482° CA*
EVC	8° CA*
Swirl Ratio	0.9
Fueling system	Fully Premixed
Engine Speed	538-2400 rpm
Intake Pressure (abs.)	100 ~ 180 kPa
Intake Temperature	60 ~ 145°C
Coolant/Oil Temperature	100°C

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

The intake air is supplied by an air compressor and precisely metered by a sonic nozzle. Real EGR is used in this work. The cooled real EGR is introduced well upstream of the intake plenum to ensure that the intake charge is well mixed. The exhaust pressure is controlled by throttling the exhaust flow so that it is higher than the intake pressure (typically 1~3 kPa in this work), which allows the EGR to flow back into the intake. Since the effect of EGR is determined by the amount of combustion product gases delivered back into the engine, the concentration of complete stoichiometric products (CSP) in the intake is calculated from the EGR level at each operating condition, and the CSP and/or the intake oxygen concentration are used for the analysis in this study. Also, an equivalence ratio based on the total charge-mass, rather than air alone, is used in this work. This equivalence ratio, referred to as charge-mass equivalence ratio (ϕ_m) is defined as $\phi_m = \frac{(F/C)}{(F/A)_{stoich}}$, 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 $\phi \leq 1$ for all conditions presented, so combustion is never overall fuel rich.

The intake pressures varied from 100 kPa (simulating naturally aspirated conditions) to 180 kPa for the current study. All pressures given are absolute. The engine is fully preheated and kept at 100°C during all the tests by means of electrical heaters on the cooling water and lubricating-oil circulation systems. In addition, the intake tank and plumbing are preheated to 50 - 60°C to avoid condensation of the fuel or water from the EGR gases. An auxiliary heater mounted close to the engine provided precise control of the intake temperature to maintain the desired combustion phasing. Intake temperatures ranged from 60 - 145°C. The engine was operated with a speed range between 538 to 2400 rpm to study the effect of engine speed on required T_{in} , and LTHR and ITHR. The high-load limits are determined and studied at 1200, 1800, and 2400 rpm.

Data Acquisition

Cylinder pressure is measured with a transducer (AVL QC33C) mounted in the cylinder head approximately 42 mm off center. The pressure transducer signals are digitized and recorded at 0.25° Crank Angle (CA) increments for 100 consecutive cycles for most measurements. The combustion is found to be more unstable with a large coefficient of variation (COV) of IMEP_g near high-load limits at high speeds, and thus more cycles are used near high-load limits. More details about the combustion stability and sampling cycles will be given in the next section. The cylinder-pressure transducer is pegged to the intake pressure near bottom dead center (BDC) where the cylinder pressure reading is virtually constant for several degrees. Intake temperatures are monitored using K-type thermocouples mounted in the two intake runners close to the cylinder head. Firedeck temperatures are monitored with a K-type thermocouple embedded in the cylinder head so that its junction is about 44 mm off the cylinder center and 2.5 mm beneath the surface. Surface temperatures are estimated by extrapolating the thermocouple reading to the surface, using the thickness of the firedeck and assuming that its back surface is at the 100°C cooling-water temperature. For all data presented, 0°CA is defined as TDC (top dead center) intake (so TDC compression is at 360°CA). This eliminates the need to use negative crank angles or combined bTDC, aTDC notation.

The crank angle of the 50% burn point (CA50) is used to monitor the combustion phasing, and the 10% burn point (CA10) is used as a representative marker for the hot-ignition point. CA10 and CA50 are determined from the cumulative apparent heat-release rate (AHRR), computed from the cylinder-pressure data (after applying a 2.5 kHz low-pass filter [10]). The start of heat release is set at the minimum point on the AHRR curve before the main heat release peak. For two-stage ignition cases, this minimum point is located between the LTHR peak and main heat release peak. This method provides a consistent measure to compare CA10 and CA50 of main combustion event although LTHR is excluded from the burn-duration calculation. Computations of CA10 and CA50 are performed for each individual cycle, disregarding heat transfer and assuming a constant ratio of specific heats (gamma, $\gamma = c_p/c_v$) [26]. The average of 100 consecutive individual-cycle CA10 or CA50 values are then used to monitor CA10 or CA50 during operation and for the values reported. The reported pressure rise rates (PRRs) and ringing intensities (RIs, see Eq. 1) are computed from the same low-pass filtered pressure data. For each cycle, the maximum (min.) PRR is analyzed.

separately with a linear fit over a moving $\pm 0.5^\circ$ CA window. Similar to CA50, these individual-cycle values are then averaged over the 100-cycle data set.

The acceptable knock limit for HCCI engines is often defined in terms of a max. allowable PRR ($dP/d\theta$, where θ is a variable representing $^\circ\text{CA}$). However, this does not correctly reflect the potential for knock under boosted conditions where the cylinder pressure changes significantly. In this work, the correlation for RI developed by Eng [27] is used as a measure of the potential for engine knock and to avoid knocking operation:

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

where $(dP/dt)_{\max}$, $P_{\max}$, and T_{peak} are the max. values of PRR (in real time), pressure, and peak temperature, respectively, γ is the ratio of specific heats, and R is the gas constant. Based on the onset of an audible knocking sound and the appearance of obvious ripples on the pressure trace, a ringing criterion of 5 MW/m² is selected as the ringing limit for operation without knock. A previous study by the authors [28] shows that RI is a good criterion for avoiding knocking combustion and its detrimental effects.

A second method of computing the heat-release rate (HRR) is used for detailed HRR-curve analysis. Here, the heat release is 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 [26]. The results of this detailed HRR analysis are used for comparisons of the early part of the heat release, leading up to hot ignition, and for comparisons of the shape of the main, high-temperature heat release rate with and without EHN addition.

Exhaust emissions data are also acquired, with the sample being drawn from the downstream of exhaust plenum using a heated sample line (See Fig. 1(a)). CO, CO₂, HC, NO_x, and O₂ levels are measured using standard exhaust-gas analysis equipment, as in our previous studies [e.g., 19]. In addition, a second CO₂ meter monitored the intake gases just prior to induction into the engine, which allowed the EGR fraction of the intake gases to be computed from the ratio of the intake and exhaust CO₂ concentrations.

Fuel/Additives Properties and Test Conditions

A Chevron commercial regular-grade gasoline containing 10 vol.% ethanol (supplied by Chevron) with nominal AKI = 87, hereafter referred as CCG-E10, is used in the current study. The fuel properties can be found in the previous study [19]. Ignition improver EHN with a purity of 97% is used in the current work (most of the impurities are isomers). The additive is mixed well with CCG-E10 to a 0.15 vol.%, and the physical properties of fuel/additive mixtures (e.g. density) are calculated from the known values for CCG-E10 and the additive, based on this ratio.

For each condition tested, the high-load limit is approached by first operating the engine at a relatively stable condition, and then gradually increasing the fueling rate to the knock/stability limit or oxygen/stoichiometric limit [19], whichever came first. During this process, the RI was held constant at 5 MW/m² for conditions with boosted intake pressure. For naturally aspirated operation, a lower RI of 3 MW/m² was used because it produced more stable combustion. Incomplete burn cycles or partial burn cycles occurred intermittently at high engine speeds when approaching high-load limits, yet combustion was still under control. Such partial burn cycles resulted in more unburned hydrocarbons in the residual gas and would often cause a significantly higher heat release in next cycle, resulting in a high RI and knock, which might damage the engine. The onset of these abnormal combustion cycles increased COV of IMEP_g above 2%. In order to avoid such a circumstance, the high-load limit here is determined as COV of IMEP_g ≤ 2%. This will be discussed in greater detail in the next section.

Throughout the current work, care was taken to ensure that the data were collected under conditions where the HCCI combustion was clean (NO_x < 0.1 g/kWh, near-zero soot), complete (combustion efficiency > 95%), stable (COV of IMEP ≤ 2%), and not knocking (ringing intensity ≤ 5 MW/m²), unless otherwise noted.

RESULTS AND DISCUSSION

Effect of Compression Ratio

Increasing the CR from 14:1 to 16:1 causes both the temperature and pressure to increase more rapidly during the compression stroke, producing higher temperatures and pressures near TDC if the intake conditions are kept constant. Because these increases in both temperature and pressure will enhance the autoignition reactivity, adjustments must be made to prevent overly advanced combustion (CA50), which results in a high peak PRR and engine knock. In the current work, data were acquired at the same intake pressures for both CRs, and either the intake temperature or EGR level was adjusted to retard CA50 sufficiently to maintain a RI ≤ 5 MW/m² to prevent knock. Most intake-boosted data were acquired at RI = 5 MW/m², which gives the most advanced CA50, and hence highest thermal efficiency, without knock. However, for naturally aspirated operation, RI = 3 MW/m² was used because it gives better combustion stability, i.e. less likely to lead to runaway knock or misfire [31].

Straight Unadditized Fuel (no EHN added)

Figure 2 provides a comparison of the pressure and temperature traces prior to hot ignition for CRs of 14:1 and 16:1, for both naturally aspirated ($P_{\text{in}} = 100$ kPa) and boosted ($P_{\text{in}} = 180$ kPa) operation. For both intake pressures, the engine speed is 1200 rpm, and loads are similar for both CRs, with the IMEP_g ≈ 483 and 830 kPa for $P_{\text{in}} = 100$ and 180 kPa, respectively. As evident in Fig. 2a, with the higher CR piston, the cylinder pressure reaches a higher pressure before hot ignition, which enhances the autoignition reactivity. To compensate for this pressure-induced reactivity enhancement and the increased temperature rise during compression with the higher CR, T_{in} is reduced by 35°C (from 135°C for CR =

14:1 to 102°C for CR = 16:1) for the $P_{in} = 100$ kPa data in Fig. 2. For both CRs at medium-to-high intake-boost conditions, the pressure-induced autoignition enhancement becomes so large that T_{in} adjustment alone is no longer sufficient to control CA50. Therefore, T_{in} is reduced to 60°C (the minimum value used in this study so that fuel condensation does not occur in the intake system), and EGR is added to retard CA50 to maintain $RI = 5$ MW/m². As indicated in the legend of Fig. 2, for $P_{in} = 180$ kPa, the amount of EGR must be increased from 27.7% for CR = 14:1 to 46.0% for CR = 16:1 at the same load.

It is interesting to note the differences in charge temperature for the two CRs and the two boost levels, as shown in Fig. 2b. For $P_{in} = 100$ kPa, T_{in} is 33°C cooler for CR = 16:1, but the difference in charge temperature diminishes as the piston moves towards TDC due to the greater temperature rise with compression for CR = 16:1. However, just prior to the onset of combustion (~5° CA before CA10), the temperature is still lower with CR = 16:1, even though CA10 is the same for both cases. This temperature difference indicates the effect of the greater pressure enhancement of autoignition for CR = 16:1. With P_{in} increased to 180 kPa, the pressure enhancement is so large that T_{in} is reduced to 60°C for both CRs, and the charge temperature remains well below that of the $P_{in} = 100$ kPa data up through the time of hot ignition in agreement with the much higher pressures throughout the cycle for $P_{in} = 180$ kPa (Fig. 2a). In addition EGR must be added for $P_{in} = 180$ kPa, and the EGR gases have a relatively low γ , which acts to reduce the temperature rise with compression. The higher EGR levels for CR = 16:1 mitigate the temperature rise with compression relative to CR = 14:1, and the temperature prior to hot autoignition is only slightly higher for CR=16:1 (Fig. 2b).

Figure 2c shows the early part of the HRR profiles (corrected using Woschni equation for heat transfer [26]) for CR = 16:1 and 14:1 at $P_{in} = 100$ and 180 kPa at the same conditions as Fig. 2a and 2b. The HRR is normalized by the total heat release (THR) to remove effects due only to differences in the amount of fuel supplied. At $P_{in} = 100$ kPa, no LTHR is found for either CR because the charge temperature in this region (15~25°CA before CA10) is high (870K~940K) due to the high T_{in} required as shown in Fig. 2b. However, Fig. 2c does indicate a slightly stronger ITHR for CR = 16:1 compared to CR = 14:1 (at $P_{in} = 100$ kPa) as a result of the lower charge temperature and higher pressure (see Fig. 2b and 2a). The ITHR period in Fig. 2c begins at about -15° CA relative to CA10 and continues to hot ignition. At $P_{in} = 180$ kPa, LTHR is present for both CRs, but the amount for CR = 16:1 is about half that for CR = 14:1 even though temperatures are similar as shown in Fig. 2b. Due to the higher in-cylinder pressure, more EGR is required at CR = 16:1 (see values in the legend of Fig. 2), which reduces the intake O_2 and suppresses the LTHR at CR=16:1. The difference in HRR between the two CRs becomes smaller in the ITHR region and eventually vanishes before hot ignition (by -5° CA relative to CA10), as shown in Fig. 2c. As shown in previous studies [e.g., 4, 5, 6, 19], the combustion stability in an HCCI engine is strongly related to the amount of ITHR. Stronger ITHR allows more retarded CA50 with good combustion stability and therefore, an increased high-load limit with an acceptable RI. Due to the opposite effect of CR on the amount of ITHR at $P_{in} = 100$ kPa compared to 180 kPa, it is expected that the effect of CR on stability near the high-load limit will vary with P_{in} .

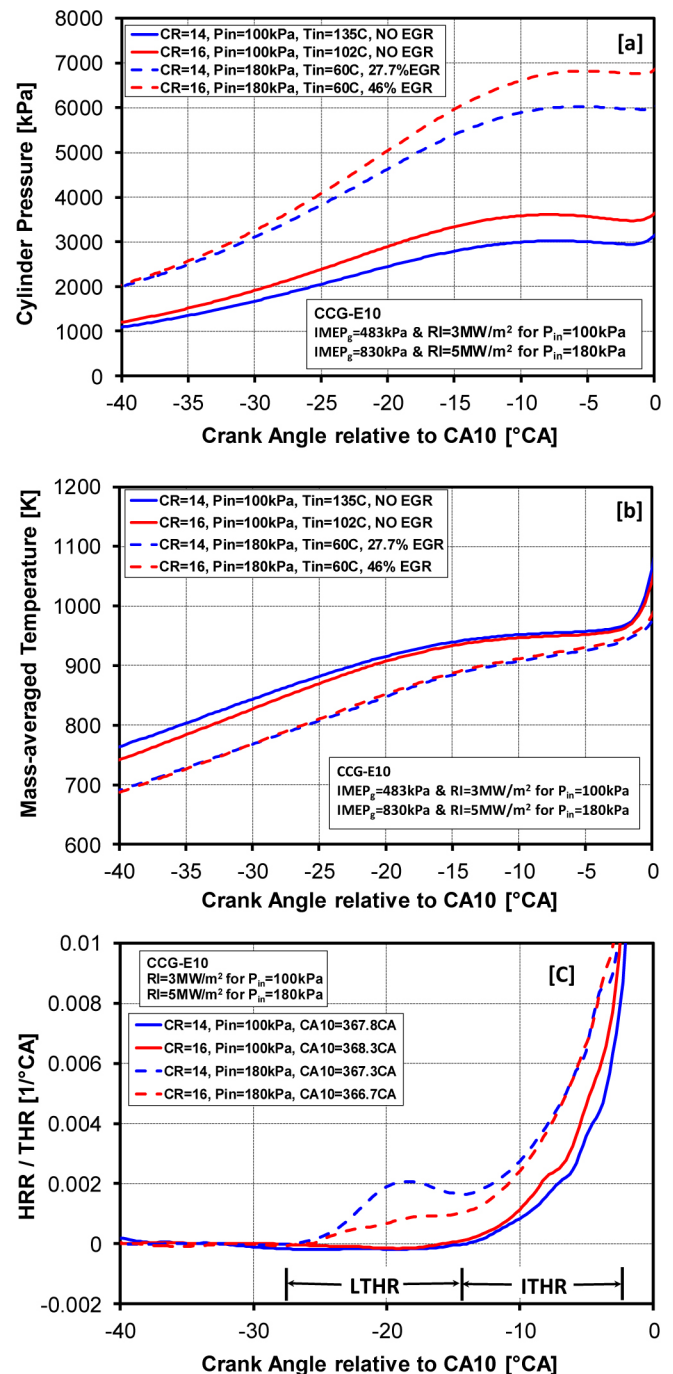


Figure 2. Cylinder pressure (a), mass-averaged charge temperature (b), and normalized HRR (c) of straight fuel at $P_{in} = 100$ with $IMEP_g = 483$ kPa, and 180 kPa with $IMEP_g = 830$ kPa for CR = 14:1 and 16:1. $RI = 3$ MW/m² at $P_{in} = 100$ kPa and 5 MW/m² at $P_{in} = 180$ kPa. Engine speed is 1200 rpm.

Figure 3a shows the high-load limits of straight fuel with no EHN addition at various P_{in} s with CR = 16:1 for an engine speed of 1200 rpm. For comparison, the high-load limits with CR = 14:1 from our previous study [19] are plotted as well. Similar to CR = 14:1, at CR = 16:1, all high-loads are still limited by knock/stability at all P_{in} s. At $P_{in} = 100$ kPa, the required T_{in} for the max. load is reduced from 130°C to 92°C (a difference of 42°C) by increasing the CR from 14:1 to 16:1. As P_{in} is increased from 100 to 180 kPa, autoignition reactivity is enhanced for both CRs, but the reactivity is always greater for CR = 16, so a lower T_{in} and/or more EGR is required than

for CR = 14. For example, Fig. 3a shows that for CR = 14, increasing P_{in} from 100 to 130 kPa, reduces the required T_{in} to 93°C, but for CR = 16, T_{in} must be reduced to 60°C (the minimum value allowed), and EGR is also required to reach the desired CA50 for RI = 5 MW/m², as shown in Fig. 3b. Note that EGR is indicated as percentage of complete stoichiometric products, CSP, in the intake, and it is also shown by the amount of intake O₂. Similarly, at P_{in} = 160 and 180 kPa, T_{in} is reduced to 60°C for both CRs, but more EGR (or CSP) is required and less intake O₂ is available for CR = 16:1 compared with CR = 14:1 as shown in Fig. 3b. With the high EGR levels, the intake O₂ is reduced to 11.3% at P_{in} = 180 kPa for CR = 16.

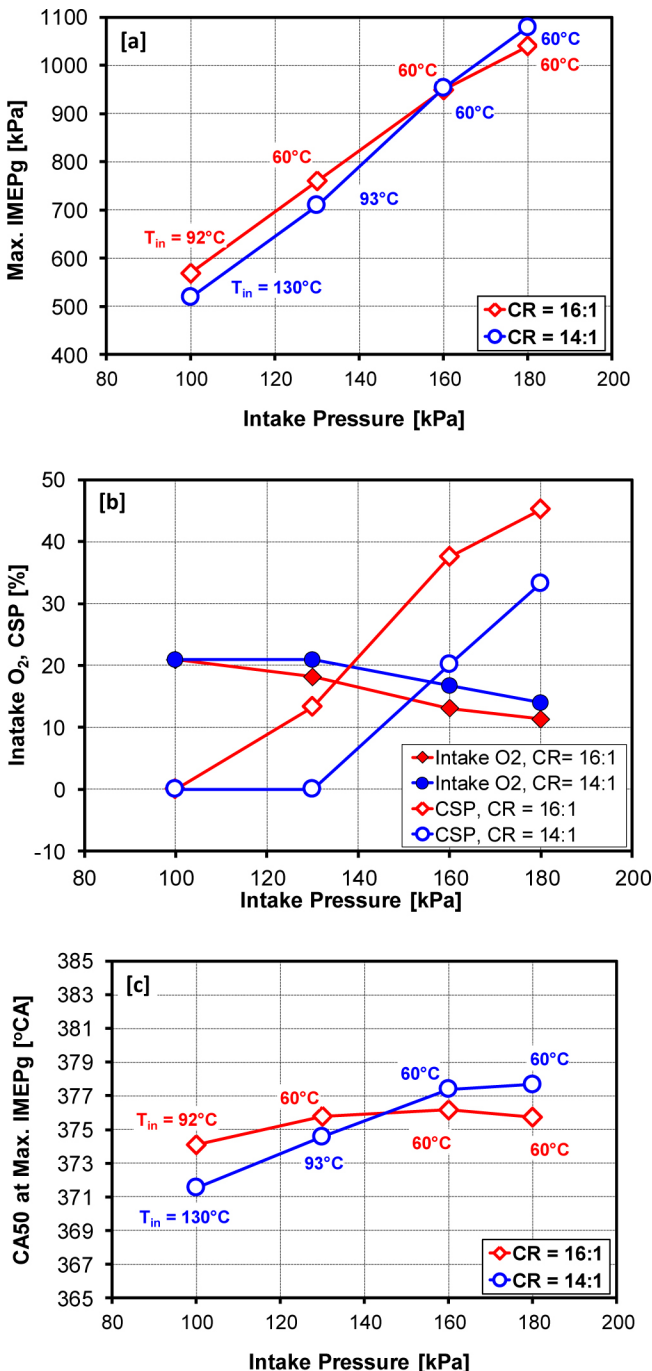


Figure 3. High-load limits of straight fuel at CR = 16:1 and 14:1 with 1200 rpm (a), required O₂ and percentage of CSP in intake for each max. IMEP_g point (b), and CA50 at each max. load (c). Engine speed is 1200 rpm.

The max. IMEP_g changes a little more moderately as a function of P_{in} at CR = 16:1 compared with CR = 14:1, as shown in Fig. 3a. At P_{in} = 100 and 130 kPa, the max. IMEP_gs for CR = 16:1 are respectively, 9.7% and 7.6% higher than those for CR = 14:1, but at P_{in} = 160, the max. IMEP_gs are nearly the same, and at 180 kPa, it is 3.6% lower for CR = 16:1. As discussed earlier, the LTHR and ITHR are weaker for CR = 16:1 compared with CR = 14:1 at high P_{in} s, which reduces combustion stability, decreasing the high-load limit. In addition, the faster expansion rate with CR = 16:1 also contributes to the decreased combustion stability at high-loads. Due to the reduced combustion stability, the CA50s at max. loads for P_{in} = 160 and 180 kPa are expected to be less retarded for CR = 16:1 compared with CR = 14:1, as shown in Fig. 3c. At naturally aspirated and low intake boost conditions (P_{in} = 130 kPa), however, the required T_{in} is lower for the higher CR. This increases the charge density, which increases in-cylinder charge mass, and therefore, results in a higher maximum load, as discussed in [19] and shown in Fig. 3a. The lower T_{in} also favors LTHR and ITHR as shown in Fig. 2c. Such slightly higher ITHR for CR = 16:1 acts to improve stability, which allows a more retarded CA50 now as shown in Fig. 3c.

Figure 4 compares the COV of IMEP_g at CR = 16:1 and 14:1 for fueling (ϕ_{in}) sweeps up to the high-load limit points shown in Fig. 3 for P_{in} = 100 and 180 kPa. The COV is found to be below 1% for all the stable points shown in Fig. 4. The last points with CR = 16:1 at P_{in} = 100 and 180 kPa have the COV > 6%, and it is hard to control combustion or keep it stable. Such loads are considered to have unstable combustion and are not counted for the max. loads shown in Fig. 3. It is noted that for CR = 16:1, at the high-load limit, the COV increases dramatically, from 1% to 6%, for less than a 1% increase in the IMEP_g, as shown in Fig. 4. Similar behavior was also found at CR = 14:1, but the high COV points at slightly high loader loads were not recorded, so are not shown here. This suggests that when approaching the high-load limit, the combustion stability is very sensitive to fueling rate at this engine speed (1200 rpm). Under such circumstances, the combustion is hard to control and easily changes to knocking or misfire runaway. Note that the differences in the lowest load of the sweeps in Fig. 4 are not related to the stable operating range but are simply artifacts of where the data sweeps for the high-load limits were started.

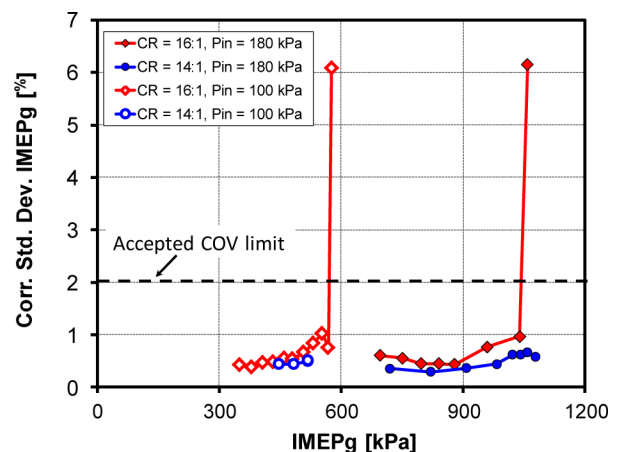


Figure 4. COV of IMEP_g of straight fuel at P_{in} = 100 and 180 kPa with CR = 14:1 and 16:1. Engine speed is 1200 rpm.