

# Effect of Ignition Improvers on the Combustion Performance of Regular-Grade E10 Gasoline in an HCCI Engine

Chunsheng Ji, John Dec, and Jeremie Dernotte  
Sandia National Labs.

William Cannella  
Chevron USA Inc.

## ABSTRACT

This study explores the use of two conventional ignition improvers, 2-ethylhexyl nitrate (EHN) and di-tert-butyl peroxide (DTBP), to enhance the autoignition of the regular gasoline in an homogeneous charge compression ignition (HCCI) engine at naturally aspirated and moderately boosted conditions (up to 180 kPa absolute) with a constant engine speed of 1200 rpm. The results showed that both EHN and DTBP are very effective for reducing the intake temperature ( $T_{in}$ ) required for autoignition and for enhancing stability to allow a higher charge-mass fuel/air equivalence ratio ( $\phi_m$ ). On the other hand, the addition of these additives can also make the gasoline too reactive at some conditions, so significant exhaust gas recirculation (EGR) is required at these conditions to maintain the desired combustion phasing. Thus, there is a trade-off between improving stability and reducing the oxygen available for combustion when using ignition improvers to extend the high-load limit. Because previous works have shown that partial fuel stratification (PFS) can be applied with more reactive fuels to reduce the heat release rate to allow higher loads or more advanced combustion timing without knock, the potential of the ignition improvers to allow effective PFS was also explored over the same range of intake pressures. The effect of the additives on NOx emissions was also studied. The results showed that NOx emissions increase with increased EHN concentration but are not affected by DTBP. This work indicates that conventional ignition improvers can effectively enhance the HCCI autoignition reactivity of conventional gasoline at naturally aspirated and modestly boosted operations, offering significant benefits for HCCI engines.

**KEYWORDS:** HCCI, Autoignition Reactivity, Ignition Improvers, ITHR, PFS

**CITATION:** Ji, C., Dec, J., Dernotte, J., and Cannella, W., "Effect of Ignition Improvers on the Combustion Performance of Regular-Grade E10 Gasoline in an HCCI Engine," *SAE Int. J. Engines* 7(2):2014, doi:10.4271/2014-01-1282.

## INTRODUCTION

Homogeneous charge compression ignition (HCCI) sometimes called controlled auto ignition (CAI) engines are considered as an alternative to conventional Spark Ignition (SI) and Compression Ignition (CI) engines. The HCCI combustion process is well known to provide much lower NOx and particulate matter emissions compared with conventional engines. Meanwhile, the HCCI engines can have a 30% improvement in thermal efficiency compared with SI engines, which is typically comparable to diesel engines with similar displacement. Recent studies [1,2,3] have further revealed that the thermal efficiencies of HCCI engines can be further improved by applying Partial Fuel Stratification (PFS) in combination with intake boost.

However, significant challenges still remain to be solved with HCCI engines before their widespread application. One significant challenge for HCCI is to control the ignition timing.

Unlike conventional engines in which the ignition is either controlled by using a spark (SI combustion) or by controlling the fuel injection (diesel or conventional CI combustion), the ignition in HCCI is controlled by chemical kinetics, which are sensitive to many factors including fuel properties, oxygen and exhaust gas recirculation (EGR) concentration, temperature, and pressure. For example, although HCCI engines can operate on regular gasoline at naturally aspirated conditions, a large amount of hot residuals or a high intake temperature is necessary for autoignition due to gasoline's low HCCI autoignition reactivity [4]. As a result, charge densities are much lower than those of typical engines, which is an important factor limiting the loads of HCCI engines. Previous work [3] has shown that these problems can be substantially reduced by the use of a low-octane gasoline. Moreover, the combustion phasing for a more reactive fuel can be retarded farther with good stability, which allows higher charge-mass fuel/air equivalence ratio ( $\phi_m$ ) without knock. The combination of

higher  $\phi_m$  and higher charge density with low-octane fuels has been shown to have the potential to significantly increase the load limit compared to regular gasoline. Although low-octane gasoline has shown to be more suitable for HCCI engines compared with conventional gasoline at naturally aspirated condition, these fuels are not readily available in the market. Additionally, the high reactivity of low-octane gasoline can turn into a hindrance for intake-boosted operation above a particular boost level. This is because the fuel/air mixture becomes so reactive that a large amount of EGR is required to prevent serious knock. Under such circumstances, the maximum load can become limited by a lack of oxygen [3].

To find an ideal fuel for HCCI engines, a number of studies have been conducted to investigate the effect of fuel type on the HCCI performance [e.g., 3,4,5,6,7,8,9,10,11,12,13]. To overcome the low reactivity of conventional gasoline, mixtures of primary reference fuels (PRF) with octane numbers (ON) lower than regular gasoline's are also commonly used in the studies of HCCI engines [e.g., 14,15,16]. Alternatively, some conventional ignition improvers, such as 2-ethylhexyl nitrate (EHN) and di-tert-butyl peroxide (DTBP) which are used to improve the cetane number (CN) of diesel fuels, have attracted researchers' interests as a means to enhance the autoignition of regular gasoline in HCCI engines. Both EHN and DTBP have shown to improve the CN of diesel fuel effectively by using a very small amount (<1% by vol.) as reported by the EPA [17]. Similar effects of these additives on the enhancement of autoignition have also been found in the HCCI engines. Reitz and co-workers [18, 19] found that EHN can enhance the autoignition reactivity of low reactivity fuels such as gasoline and E10 (i.e., 10% ethanol in gasoline). Hosseini et al. [20] indicated that a small amount of EHN additive advanced low temperature heat release (LTHR) and high EGR rates were required to retard the combustion phasing. Eng et al. [21] showed that adding DTBP to a fully-blended gasoline both advances the location of maximum energy release and lowers the minimum fueling level for stable HCCI combustion at low loads. They also found that the effect of DTBP in reducing the minimum fueling level was nonlinear. More specifically, DTBP addition levels up to 2% produced a large change in ignition timing, while adding greater amounts resulted in only a small change in ignition timing. Mack et al. [22] observed that with the addition of DTBP, a lower inlet temperature was required but it didn't make later combustion timings easier to obtain. Moreover, the addition of DTBP was found to have different effects on different fuel types. Cernansky and co-workers [23] also indicated that the effect of DTBP is primarily chemical for high ON PRF blends but thermal for low ON PRF blends. Subsequently, authors from the same research group further numerically studied the effect of DTBP on PRF fuels. They [24] concluded that the addition of DTBP amplifies the LTHR and reduces the ON.

Despite the lower cost and greater enhancement in CN of EHN compared with DTBP, there is a concern that the nitrate group in EHN may increase the NOx emissions. Thompson et al. [25] have shown that the addition of EHN has almost no effect on

NOx emissions in conventional diesel operation. However, the EHN doped fuels have been found to increase the NOx emissions in low-temperature diesel combustion as reported by Ickes et al. [26] and Bunting et al. [27]. More specifically, Ickes et al. [26] indicated that 30% of the nitrate group in EHN tends to convert to NOx and the NOx emissions increase proportionally with the amount of EHN. Reitz and co-workers [18, 19] reported that the nitrogen in EHN results in a small increase in NOx emissions from the light-duty reactivity controlled compression ignition (RCCI) combustion. Hosseini et al. [20] showed that EHN produced higher NOx emissions on a relative basis, but the absolute emissions were still low. Comparatively, DTBP was shown to have either minor or no detectable effects on NOx emissions [28].

In summary, the previous works show that the conventional ignition improvers can enhance the autoignition of the regular gasoline in HCCI engines with the potential for a small increase in NOx emissions for EHN. However, there is a dearth of information on understanding the effects of such additives on the operating limits at different boost levels. The effects of conventional ignition improvers on the HCCI operation need to be further validated and understood over a wide range of conditions.

The objective of the current work is to examine the potential for using conventional ignition improvers to enhance the autoignition of regular gasoline in an HCCI engine at naturally aspirated and moderately boosted conditions (up to 180 kPa absolute). The effects of EHN and DTBP on the high-load limit at each intake pressure ( $P_{in}$ ) were studied and compared. Additionally, because previous works have shown that PFS can be applied with more reactive fuels to reduce the heat release rate to allow higher loads or more advanced combustion timing without knock [3, 29, 30], the potential of the ignition improvers to allow effective PFS was also explored over the same range of intake pressure. Finally, the effect of the additives on NOx emissions was investigated.

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 four parts. First, the effects of EHN and DTBP on the autoignition enhancement of regular gasoline in the HCCI engine are presented and discussed. The second part shows that the high-load limits at various  $P_{in}$ s are strongly dependent on the additive concentration. The third part further reveals the potential of PFS to improve the thermal efficiency for the additized fuels. The last part compares the NOx emissions between the base fuel and additized fuels. Finally, the present work is summarized, and conclusions are drawn in the last section.

## EXPERIMENTAL SETUP

### Engine Facility

The HCCI research engine used for this study was derived from a Cummins B-series six-cylinder diesel engine, which is a typical medium-duty diesel engine with a displacement of 0.98 liters/cylinder. As shown in Fig. 1a, the engine has been converted for single-cylinder operation by deactivating cylinders 1-5. The active HCCI cylinder is fitted with a compression-ratio (CR) = 14:1 custom piston as shown in Fig. 1b. This piston provides an open combustion chamber with a large squish clearance and a quasi-hemispherical bowl. 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 gasoline [2, 4] and low-octane fuels [3, 29, 30].

A combination of premixed and direct injection (DI) fueling was used as shown in Fig. 1. The premixed fueling system, shown at the top of the schematic in Fig. 1a, consists of a gasoline direct injection (GDI) fuel 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. The DI fueling is accomplished using a

second GDI injector mounted centrally in the cylinder head. A positive displacement fuel flow meter (Max P002), with a reported accuracy of  $\pm 0.2\%$ , was used to determine the amount of fuel supplied.

The intake air was supplied by an air compressor and precisely metered by a sonic nozzle with an uncertainty of  $\pm 0.25\%$ . Both real and simulated EGR were 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. Simulated EGR consisting of  $N_2$  and  $CO_2$  was also used in this work for some measurements where the amount of combustion products needed to be carefully controlled. Unlike the complete stoichiometric products (or CSP) used for EGR simulation in some previous works from our laboratory [e.g., 3, 4], the simulated EGR used in the current work did not include  $H_2O$ . In order to compensate for this, the amounts of  $N_2$  and  $CO_2$  were adjusted to provide the same total moles and the same specific heat capacity ( $c_p$ ) as CSP containing  $H_2O$ . The amounts of  $N_2$  and  $CO_2$  were precisely controlled using sonic nozzles as indicated in Fig. 1a. This simulated EGR is hereafter called CSPd standing for dry CSP.

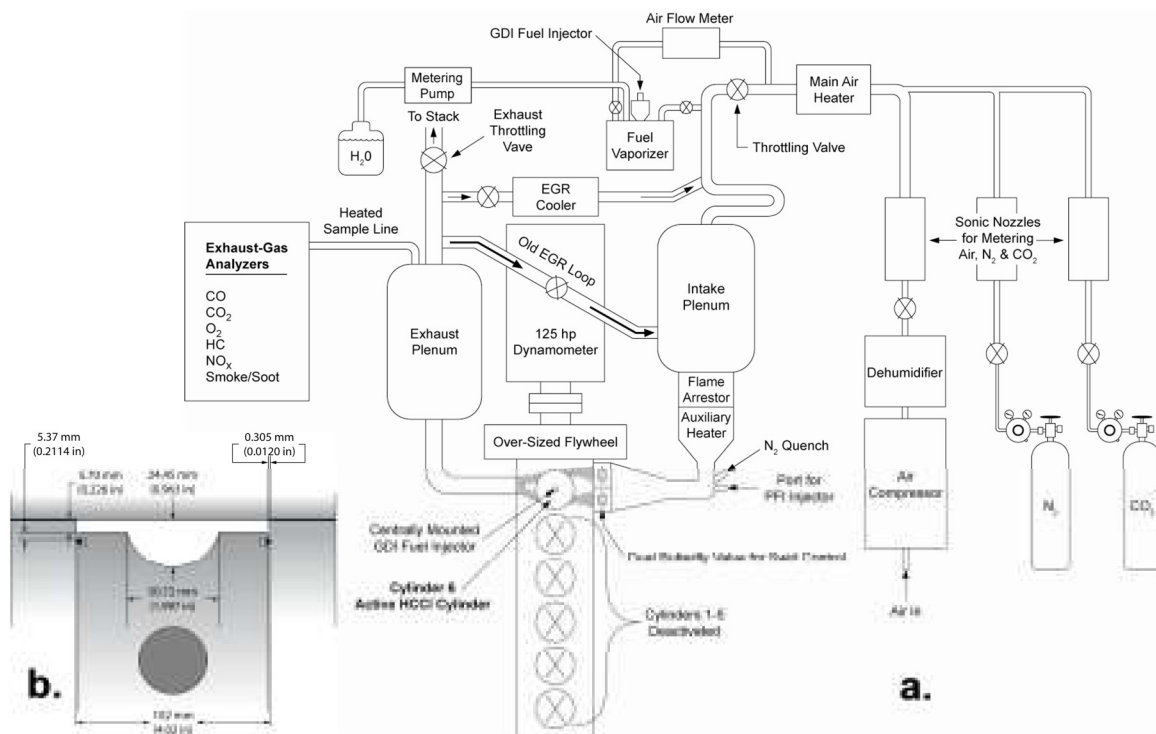


Figure 1. Schematic of HCCI engine facility (a) and combustion chamber geometry of the CR = 14:1 piston, at TDC (b).

Table 1. Engine specifications and operating conditions.

|                                |                      |
|--------------------------------|----------------------|
| Displacement (single-cylinder) | 0.981 liters         |
| Bore                           | 102 mm               |
| Stroke                         | 120 mm               |
| Connecting Rod Length          | 192 mm               |
| Geometric Compression Ratio    | 14:1                 |
| Number of Valves               | 4                    |
| IVO                            | 0° CA*               |
| IVC                            | 202° CA*             |
| EVO                            | 482° CA*             |
| EVC                            | 8° CA*               |
| Swirl Ratio                    | 0.9                  |
| Fueling system                 | Fully Premixed/GD    |
| DI Injector                    | Bosch, 8-hole        |
| Included Angle                 | 70°                  |
| Hole Size (Stepped-hole)       | min. dia. = 0.125 mm |
| Max. Injection Pressure        | 135 bar              |
| Engine Speed                   | 1200 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.

An equivalence ratio based on the total charge-mass, rather than air alone, is used in the current work. This equivalence ratio, referred to as charge-mass equivalence ratio ( $\phi_m$ ) 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 or CSPd), 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  $\phi_m$  is the same as conventional air-based  $\phi$  when no EGR/CSPd is used. It should also be noted that the air-based  $\phi$  is  $\leq 1$  for all conditions presented, so combustion is never fuel rich overall. The overall uncertainty of  $\phi_m$  is calculated as  $\pm 0.32\%$  based on the uncertainties in the air and fuel flow rate measurements.

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 was 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 were 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. All the temperatures were measured with  $K$ -type thermocouples with an uncertainty of  $\pm 0.5^\circ\text{C}$ . All data were taken at an engine speed of 1200 rpm.

## Data Acquisition

Cylinder pressure was measured with a transducer (AVL QC33C) mounted in the cylinder head approximately 42 mm off center. The pressure transducer signals were digitized and recorded at 0.25° Crank Angle (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 temperatures were monitored using  $K$ -type thermocouples mounted in the two intake runners close to the cylinder head. Firedeck temperatures were monitored with a  $K$ -type thermocouple embedded in the cylinder head so that its junction was about 44 mm off the cylinder center and 2.5 mm beneath the surface. Surface temperatures were estimated by extrapolating the thermocouple reading to the surface, using the thickness of the firedeck and assuming that its back surface was at the 100°C cooling-water temperature [29]. 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) 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 [30, 31]). 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 burn-duration calculation. Computations of CA10 and CA50 were performed for each individual cycle, disregarding heat transfer and assuming a constant ratio of specific heats ( $\gamma = c_p/c_v$ ) [32]. The average of 100 consecutive individual-cycle CA10 or CA50 values were then used to monitor CA10 or CA50 during operation and for the values reported. The standard deviation of these 100 cycles are varied from 0.2°CA to 1°CA, or 0.05~0.25% of CA10 or CA50. The reported pressure rise rates (PRRs) and ringing intensities (RIs) (see Eq. 2) are computed from the same low-pass-filtered pressure data. For each cycle, the maximum PRR was analyzed separately with a linear fit over a moving  $\pm 0.5^\circ$  CA window. Similar to CA50, these individual-cycle values were then averaged over the 100-cycle data set.

The acceptable knock limit for HCCI engines is often defined in terms of a maximum allowable PRR ( $dP/d\theta$ , where  $\theta$  is a variable representing °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 ringing intensity (RI) developed by Eng [33] is used as a measure of 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), pressure, and temperature, respectively,  $\gamma$  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/m<sup>2</sup> was selected as the ringing limit for operation without knock. This corresponds to about 8 bar/°CA at 1200 rpm at naturally aspirated for operating conditions typical of those used in this study. At all boost levels tested, perceived engine knock correlated well with the RI rising above 5 MW/m<sup>2</sup>, giving confidence in this correlation.

A second method of computing the heat-release rate (HRR) was used for detailed HRR-curve analysis. 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 [32]. 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.

Exhaust emissions data were also acquired, with the sample being drawn just downstream of the exhaust plenum using a heated sample line (See Fig. 1a). CO, CO<sub>2</sub>, HC, NO<sub>x</sub>, and O<sub>2</sub> levels were measured using standard exhaust-gas analysis equipment as used in our previous studies [e.g., 34]. The NO<sub>x</sub> emissions measurement includes both NO and NO<sub>2</sub>. Soot emissions were monitored during the application of PFS using an AVL smoke meter, and were found to be below the detectability limit for all data reported. In addition, a second CO<sub>2</sub> 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 CO<sub>2</sub> concentrations. The uncertainty in the exhaust gas measurement is reported to be less than ±1%.

### Fuel/Additives Properties and Test Conditions

A regular Chevron commercial gasoline containing 10 vol.% ethanol (supplied by Chevron), hereafter referred as CCG-E10, is used in the current study. Table 2 lists the fuel properties of this fuel. Multiple batches of fuels were used in this work and the batch-to-batch variation was found negligible. Compared

with the research-grade conventional gasoline with no ethanol (Antiknock Index, AKI=87) used in our previous studies [1, 2, 4], CCG-E10 shows higher RON and MON.

EHN with a purity of 97% and DTBP with a purity of 95% are used in the current work (most of the impurities are isomers). The additives were mixed well with CCG-E10 on a volume percentage basis, and the physical properties of fuel mixtures were calculated carefully based on the known specifications of CCG-E10 and additives.

Table 2. Fuel properties of CCG-E10.

|                          |                                                        |
|--------------------------|--------------------------------------------------------|
| Formula                  | C <sub>5.76</sub> H <sub>11.98</sub> O <sub>0.22</sub> |
| Specific gravity (15 °C) | 0.7238                                                 |
| Net Heating Value, MJ/kg | 41.74                                                  |
| Carbon, wt, %            | 81.67                                                  |
| Hydrogen, wt, %          | 4.06                                                   |
| RON                      | 92.5                                                   |
| MON                      | 84.6                                                   |
| Antiknock Index (R+M)/2  | 88.6                                                   |

The effect of EHN and DTBP addition on the CN improvement is shown in Fig. 2. The CNs presented are derived cetane numbers determined using an Ignition Quality Tester (IQT). Both additives are shown to effectively increase CN of CCG-E10. The improvement however is nonlinear for both additives. A greater increase in CN is found for a low percentage of additive (<0.25 vol.%), and the effect becomes more moderate and nearly linear over the range of 0.25-1.0 vol.% additive. Furthermore, EHN displays a greater increase in CN for a given additive percentage than DTBP does, over the whole range tested.

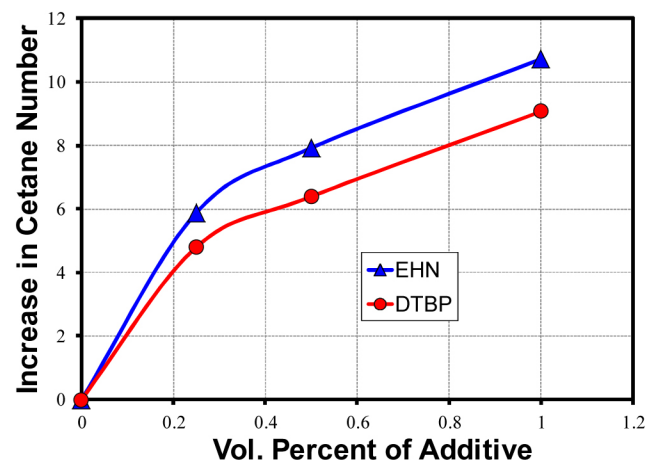


Figure 2. Cetane improvement of CCG-E10 using EHN and DTBP.

For each condition tested, the high-load limit was approached by first operating the engine at a relatively stable condition, and then gradually increasing the fueling rate to either the knock/stability limit or oxygen/stoichiometric limit (these terms will be defined later in the section on high-load limits), whichever came first. During this process, constant RI, 5 MW/m<sup>2</sup>, was

used in most cases. However, lower RI ( $\sim 3$  MW/m<sup>2</sup>) were used in several cases where it produced a more stable combustion phasing. Previous studies [1, 4] have revealed that fuels with high autoignition reactivity show stronger intermediate temperature heat release (ITHR) which occurs after the LTFR and leads into the hot ignition (see Fig. 5 for detailed illustration of LTFR and ITHR). Thus, a separate set of experiments was performed to investigate the variation of ITHR at various  $P_{in}$ s and additive concentrations. For this sweep,  $\phi_m$  was held constant and the CA10 was kept constant by varying EGR or intake temperature ( $T_{in}$ ).

To obtain the benefits of PFS, the fuel must be “ $\phi$ -sensitive” so that autoignition occurs sequentially from richer to leaner regions [6, 29,30]. Strong  $\phi$ -sensitivity means that the autoignition timing of a fuel is strongly dependent on  $\phi_m$ , i.e., the zones with high  $\phi_m$  autoignite sooner and low- $\phi_m$  zones autoignite later. Because of this effect, mixture stratification is effective for slowing down the combustion process for  $\phi$ -sensitive fuels [35]. A  $\phi$ -sensitivity test of the additized fuel was conducted before applying PFS. For this test, an alternative firing method was used in order to isolate the fuel-chemistry effect that produced  $\phi$ -sensitivity from thermal and residual-gas effects on ignition timing as  $\phi_m$  is varied [6]. Similar methodology has also been used and validated in previous studies [e.g., 1, 30, 35]

Throughout the current work, care was taken to make sure 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<sub>g</sub>  $\leq 1\%$ ), and not knocking (RI  $\leq 5$  MW/m<sup>2</sup>), unless otherwise noted.

## RESULTS AND DISCUSSION

### HCCI Autoignition Reactivity

HCCI combustion is extremely sensitive to initial thermodynamic conditions. Usually high  $T_{in}$  ( $\sim 140^\circ\text{C}$ ) is required for fuels with low HCCI reactivity, such as regular gasoline and E10 [2] at naturally aspirated conditions. A lower  $T_{in}$  can be achieved by either increasing the intake pressure [1] or using a more reactive fuel such as hydrobate [3]. Both methods enhance the rate of chemical reactions in the early stages before the hot ignition, reducing the required  $T_{in}$ . Thus, it is of great interest to find out whether adding EHN and DTBP to the CCG-E10 can affect the required  $T_{in}$  in a manner similar to a high-reactivity fuel. Figure 3 shows the effect of EHN and DTBP addition to the CCG-E10 fuel on the  $T_{in}$  required to maintain the same CA10 (366.3°CA) for a constant  $\phi_m = 0.4$  at  $P_{in} = 100$  kPa. By adding small amount of EHN (0.15%) or DTBP (0.35%), the  $T_{in}$  drops drastically from  $144^\circ\text{C}$  to  $95^\circ\text{C}$ .  $T_{in}$  can be further reduced to  $60^\circ\text{C}$  using 0.4%EHN (Fig. 3).

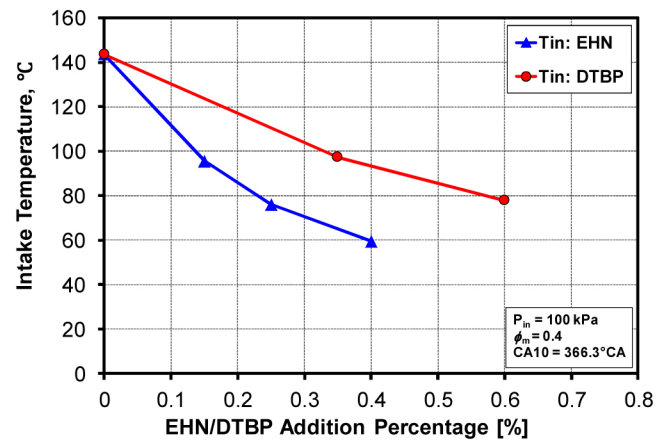


Figure 3. Effect of EHN and DTBP addition on intake temperature ( $T_{in}$ ) at naturally aspirated conditions ( $P_{in} = 100$  kPa),  $\phi_m = 0.4$ , and CA10 = 366.3°CA.

Compared with EHN, DTBP has a smaller effect on  $T_{in}$  for a given percentage added as shown in Fig. 3. Moreover, a nonlinear decrease in  $T_{in}$  is found for both EHN and DTBP; the decrease of  $T_{in}$  is shown to be greater at low additive concentration ( $< 0.25\%$ ) and the effect becomes more moderate as the additive percentage increases. The observations are consistent with those for the CN increase with additive concentration (Fig. 2).

As shown in Fig. 3, more than double the amount of DTBP is required to achieve the same effect as EHN, at least for the test fuel and test range studied in the present work. To further illustrate this point,  $P_{in}$  and  $\phi_m$  are varied for the cases of 0.25%EHN and 0.6% DTBP (which require nearly the same  $T_{in}$ ) to compare their effects on HCCI combustion. The results are shown in Fig. 4.

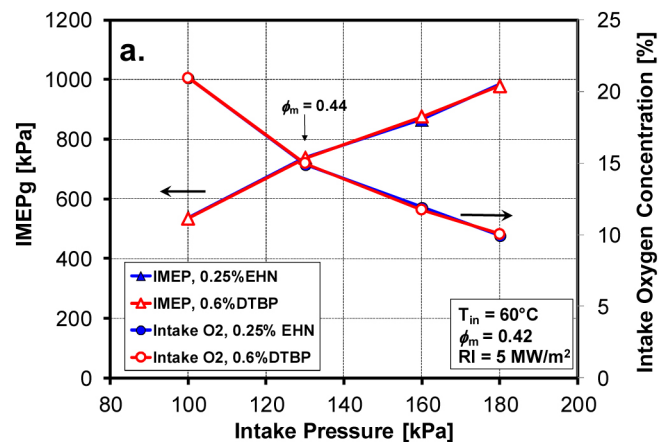


Figure 4. Gross indicated mean effective pressure (IMEPg) and intake oxygen concentration at various  $P_{in}$  (a) and  $\phi_m$  (b) for 0.25%EHN and 0.6%DTBP fuel mixture. For Fig. 4a,  $\phi_m = 0.42$  at all  $P_{in}$ s except for  $P_{in} = 130$  kPa where  $\phi_m = 0.44$ . In all the results shown above,  $T_{in}$  was kept constant at  $60^\circ\text{C}$  and RI = 5 MW/m<sup>2</sup>. EGR was used to adjust CA50 to keep the same ringing.