

Boosted HCCI Combustion Using Low-Octane Gasoline with Fully Premixed and Partially Stratified Charges

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ABSTRACT

High-load HCCI combustion has recently been demonstrated with conventional gasoline using intake pressure boosting. The key is to control the high combustion heat release rates (HRR) by using combustion timing retard and mixture stratification. However, at naturally aspirated and moderately boosted conditions, these techniques did not work well due to the low autoignition reactivity of conventional gasoline at these conditions. This work studies a low-octane distillate fuel with similar volatility to gasoline, termed Hydrobate, for its potential in HCCI engine combustion at naturally aspirated and low-range boosted conditions. The HCCI combustion with fully premixed and partially stratified charges was examined at intake pressures (P_{in}) from 100 to 180 kPa and constant intake temperature (60°C) and engine speed (1200 rpm).

First, the key parameters for high-load HCCI operation were investigated, including 1) intermediate temperature heat release (ITHR), which determines the potential of using combustion retard to control the HRR, and 2) ϕ -sensitivity, which determines the effectiveness of mixture stratification for controlling the HRR. It is found that consistent with the low octane number, Hydrobate shows strong ITHR and ϕ -sensitivity even at $P_{in} = 100$ kPa. However, the ITHR of Hydrobate is not enhanced by increasing intake pressure, which is different from gasoline at $P_{in} = 100 - 180$ kPa, but similar to gasoline at $P_{in} \geq 180$ kPa. For high-load operation, it is found that with a fully premixed charge, Hydrobate allowed higher loads than conventional gasoline for $P_{in} < 160$ kPa, mainly due to the lower intake temperature allowing a higher fueling rate. However, for $P_{in} \geq 160$ kPa, the maximum load reached by Hydrobate was lower than gasoline due to the higher level of EGR required, which limited the oxygen available for combustion. On the other hand, higher thermal efficiency was consistently observed with Hydrobate at all P_{in} tested, mainly due to the lower intake temperature and more advanced combustion phasing. Applying partial fuel stratification to Hydrobate combustion further improved thermal efficiency compared to the fully premixed case, since it allowed even more advanced combustion phasing; however, it only slightly extended the high-load limit, again due to the limited oxygen availability. This work indicates that low-octane gasoline is a superior HCCI fuel for naturally aspirated and modestly boosted operations.

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INTRODUCTION

Homogeneous charge compression ignition (HCCI) engines possess a potential for 30% improvement in engine efficiency compared to modern gasoline spark-ignition (SI) engines, and they produce nearly zero NO_x and soot emissions that plague modern diesel engines. However, major technical problems remain to be solved for the wide implementation of HCCI engines. One of them is the overly high heat release rates (HRR) at high-loads, which causes

combustion noise and potentially engine damage. This problem prevents the benefits in engine efficiency and emissions of HCCI engines from being realized over the entire engine operating map, and it requires switching to conventional combustion mode (SI or diesel combustion) for high-load operation.

Two main approaches have been used to control the high HRR of HCCI combustion, combustion timing retard and mixture stratification. Retarding combustion timing has been

found very effective for reducing the HRR when combustion occurs in the early expansion stroke [1, 2]. This is because the expanding cylinder volume counteracts the pressure rise resulting from combustion, and this effect becomes increasingly prominent farther from top dead center (TDC). However, excessive timing retard can cause combustion to become unstable [1, 2], and eventually misfire. Sjöberg and Dec [3] pointed out that this is primarily because the positive temperature rise rate required for successful autoignition cannot be sustained with overly retarded combustion timing. They also found that fuels with high autoignition reactivity¹, in this case, fuels showing two-stage ignition, can maintain stable combustion at more retarded timings than fuels with low autoignition reactivity, e.g. fuels showing single-stage ignition [3]. They attributed the difference to the stronger heat release prior to hot ignition of the high reactivity fuels, in particular, the intermediate-temperature heat release (ITHR) which occurs between the low-temperature heat release (LTHR) and the hot ignition.

Mixture stratification has also been used to control the HRR of high-load HCCI operations [4,5,6,7,8]. The idea is that by stratifying the charge to create a gradient in fuel concentration, sequential autoignition and staged combustion can be resulted due to high- ϕ zones autoigniting sooner and low- ϕ zones autoigniting later. However, in order for this mechanism to work, fuel pre-ignition chemistry, more precisely, the rates of the reactions leading to hot ignition, must be sensitive to the variations in the local equivalence ratio. This requires the fuel to have considerable autoignition reactivity. For example, previous studies [5, 7, 9] found that two-stage ignition fuels show significantly advanced autoignition timing (higher reaction rates) with increasing equivalence ratio, i.e. they are strongly ϕ -sensitive. Consequently, given suitable mixture stratification, these fuels could drastically reduce the HRR. On the other hand, for fuels exhibiting low autoignition reactivity, such as iso-octane and gasoline at naturally aspirated conditions, autoignition timings were not advanced with increasing equivalence ratios. As a result, these fuels showed no effect for reducing combustion HRR and associated pressure rise rate (PRR) with similar mixture stratification to that used with two-stage fuels². It should also be noted that a similar approach has been used in the recent dual-fuel studies [12,13,14,15] which use two fuels with drastically different autoignition reactivities, e.g. gasoline and diesel fuel. This technique exploits both ϕ -gradients and reactivity gradients within the charge to control the combustion HRR, and it is properly named as reactivity controlled compression ignition (RCCI).

The two strategies for controlling HCCI HRR, i.e. combustion retard and mixture stratification, provide significant potential for high-load HCCI operation without engine knock. This has been demonstrated with conventional gasoline using high intake-pressure boost in recent studies [10, 16] and achieved high loads comparable to those of conventional IC engines. However, at naturally aspirated and moderately boosted pressures, these studies found that the combustion-retard strategy did not work as effectively as at higher pressures, and the mixture-stratification strategy did not work at all. This is because the conventional gasoline did not have the required autoignition reactivity at these conditions for the two strategies to work. Also, due to the low autoignition reactivity of gasoline, considerable intake heating is required for the HCCI combustion at naturally aspirated and low-boost pressures, which reduces thermal efficiency and limits the highest loads that can be reached at these conditions.

Based on these observations, fuels with higher autoignition reactivity (lower octane rating) than conventional gasoline would be favored at naturally aspirated and moderately boosted conditions for reaching higher loads with combustion retard and/or mixture stratification. Also, from fuel-production perspective, low-octane gasoline fuels generally require less processing at the refinery, and thus are cheaper to produce and cause less CO₂ emissions [17]. These factors make these non-conventional fuels attractive for advanced engine combustion.

Numerous studies have been conducted with low-octane gasoline fuels, both distillate and surrogate, in HCCI engine combustion (for example in Ref. [18,19,20,21,22,23,24,25]). However, few studies have focused on using low-octane fuels for high-load HCCI operations [20, 25], in particular, under boosted intake conditions.

In this work, we study a low-octane gasoline for high-load HCCI engine combustion with fully premixed and partially stratified charges, under both naturally aspirated and boosted intake conditions. The objective of this work is to investigate:

1. Fundamentals of boosted HCCI combustion with two-stage ignition fuels, including the variation of ITHR and ϕ -sensitivity with intake pressure.
2. High-load limits of fully premixed HCCI for a low-octane gasoline at naturally aspirated and moderately boosted conditions.
3. High-load limits of partially stratified HCCI using the low-octane gasoline.

¹Autoignition reactivity in this work is characterized by the magnitude of pre-ignition heat release, in particular ITHR (as discussed in the following), and LTHR if it exists. Note that autoignition reactivity can be characterized in various ways, such as combustion phasing under a given intake condition (as described in [11]).

²However, ϕ -sensitivity can change with engine operating conditions. For example, intake boosting significantly enhances the ϕ -sensitivity of gasoline [10], while intake heating drastically reduces the ϕ -sensitivity of PRF73 (iso-octane/n-heptane = 73:27 mixture by volume) [11].

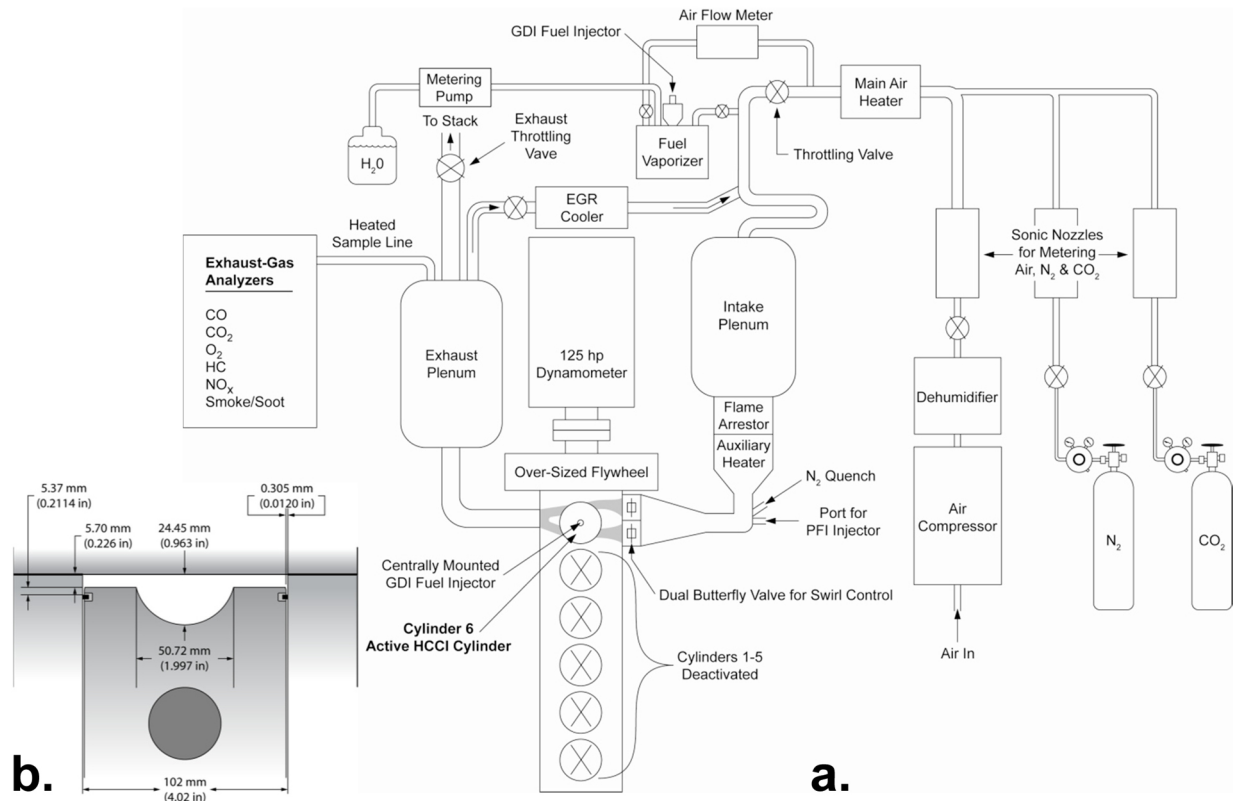


Fig. 1. (a) Schematic of the HCCI engine facility and (b) combustion chamber geometry of the CR = 14 piston at TDC.

4. Comparison of the low-octane gasoline and a conventional gasoline for high-load HCCI performance.

EXPERIMENTAL

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 Figure 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 custom piston, as shown in Fig. 1b, which 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 is identical to previous studies with gasoline [10, 16] and low-octane fuels [7, 11].

The engine is set up to allow both premixed fueling and direct injection (DI) fueling. 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 fuel-vaporizing chamber and appropriate plumbing to ensure thorough premixing with the air and EGR upstream of the intake plenum. This system supplies all the fuel for the

fully premixed operation and ~ 90% of fuel for the partially stratified operation. The DI fueling is via another GDI injector (Bosch, 8-hole) mounted centrally in the cylinder head, which is used only to supply ~10% of fuel for partially stratified operation. A positive displacement fuel flow meter was used to determine the total amount of fuel supplied.

The intake air was supplied by an air compressor and precisely metered by a sonic nozzle. Real EGR and simulated EGR were both used in this work. As shown Fig. 1a, the cooled real EGR is introduced well upstream of the intake plenum to ensure that the intake charge is well mixed. This configuration requires the exhaust pressure to be greater than the intake pressure (typically 2 kPa in this work) 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. Simulated EGR containing N₂, CO₂ and H₂O in proportions that would result from complete combustion of the fuel (here called complete stoichiometric products or CSP) was also used in this work, in particular, for ϕ -sensitivity measurements where the amount of combustion products needs to be carefully controlled. Nitrogen and carbon dioxide were supplied from gas bottles and metered with sonic nozzles. Water was supplied via a variable-speed rotary pump while the mass flow rate was determined by continuously monitoring the weight of the supply bottle on an electronic scale. The water was vaporized in the same chamber as the fuel. These EGR components were mixed

with the air and fuel upstream the intake plenum to ensure sufficient mixing.

Table 1. Engine Specifications

Displacement (single-cylinder).....	0.981 liters
Bore	102 mm
Stroke	120 mm
Connecting Rod Length	192 mm
Geometric Compression Ratio	14:1
No. of Valves	4
IVO	0° CA ^a
IVC	202° CA ^a
EVO	482° CA ^a
EVC	8° CA ^a
Swirl Ratio	0.9
Fueling System	Fully Premixed / GDI
DI Injector.....	Bosch, 8-hole
Included Angle.....	70°
Hole Size.....	Stepped-hole, min. hole dia. = 0.125 mm
Injection Pressure.....	150 bar
Coolant/Oil Temperature	100°C

^a0° CA is taken to be TDC intake. The valve-event timings correspond to 0.1 mm lift.

In this work, an equivalence ratio based on total charge mass, rather than air alone, is used, which is called charge-mass equivalence ratio (ϕ_m) and 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 a 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 the fuel) for operating conditions with different fuels and different EGR levels. Note that ϕ_m is the same as conventional air-based ϕ when no EGR is used. It should also be noted that the air-based ϕ is ≤ 1 for all conditions presented, so combustion is never fuel rich overall.

The intake tank and plumbing were preheated to 50 - 60°C to avoid condensation of the fuel or water from the EGR gases. The intake temperature was precisely controlled by an auxiliary heater mounted close to the engine. The cooling water and lubricating oil were maintained at 100°C during the tests.

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 recorded at 1/4° crank angle (CA) increments for 100 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 thermocouples mounted in the two intake runners close to the cylinder head.

For all data presented, 0°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 on the fly during the experiment. CA50 was determined from the cumulative apparent heat-release rate (AHRR), computed from the cylinder-pressure data (after applying a 2.5 kHz low-pass filter [26]). 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 CA50 of main combustion event although LTHR is excluded from burn-duration calculation. Computations of CA50 were performed for each individual cycle, disregarding heat transfer and assuming a constant ratio of specific heats ($\gamma = c_p/c_v$) [27]. The average of 100 consecutive individual-cycle CA50 values was then used to monitor CA50 and for the CA50 values reported. The pressure-rise rates and ringing intensities (detailed below) are computed from the same low-pass-filtered pressure data. For each cycle, the maximum pressure-rise rate 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 θ is a variable representing °CA). However, this does not correctly reflect the potential for knock under boosted conditions where the cylinder pressure increases significantly. In this work, the correlation for ringing intensity developed by Eng [28] is used as a measure of engine knock.

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

Where $(dP/dt)_{\max}$, $P_{\max}$, and $T_{\max}$ are the maximum values of cylinder pressure rise rate (in real time), cylinder pressure, and charge temperature, respectively, γ is the ratio of specific heats (c_p/c_v), 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 was selected as the ringing limit for operation without knock for this engine. At 1200 rpm, this corresponds to about 8 bar/°CA at naturally aspirated pressure, and higher $dP/d\theta$ at boosted pressures. At all pressures tested, perceived engine knock correlated well with the ringing intensity rising above 5 MW/m², giving confidence in this correlation.

A second method of computing the HRR was used for detailed HRR-curve analysis, for example, those shown in Fig. 2. 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 [27] and a variable γ which updates as a function of burn fraction. In using the Woschni correlation, the main coefficient, C , and a coefficient in the gas velocity term, C_2 , were adjusted for each combustion condition so that the heat release rate is zero before and after the combustion (similar coefficients were used for comparable operating conditions). Using the ensemble-averaged pressure trace has benefits from the standpoint of reduced noise on the heat-release traces. On the other hand, it can lead to overestimated burn durations if the cycle-to-cycle variations are large. However, for the condition where this was applied, the combustion phasing was fairly stable with the standard deviation of CA50 over 100 fired cycles averaging $< 1.2^\circ\text{CA}$. Moreover, the results of this detailed HRR analysis are mainly used for comparisons of the early part of the heat release, leading up to hot ignition. Since these early autoignition reactions persist for $15^\circ\text{--}25^\circ\text{CA}$, the relatively small cycle-to-cycle variation in CA50 will have little effect.

Although these detailed HRR curves appear quite smooth when plotted on a scale showing the entire combustion event, small quasi-periodic fluctuations were evident when viewed on the highly amplified scale used for comparisons of the early heat release. This is believed to be an artifact of the HRR being based on the derivative of the pressure trace (which amplifies noise in the pressure data) and the low-pass filtering process [26]. To mitigate these fluctuations, the HRR data were further filtered using a piecewise smoothing function, as described in Ref. [16], with care taken to ensure that the filtering does not alter the HRR trends.

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, CO₂, HC, NO_x, and O₂ levels were measured using standard exhaust-gas analysis equipment. Smoke measurements were also made with an automated smoke meter.

FUEL PROPERTIES AND TEST CONDITIONS

This work studies a low-octane distillate fuel, Hydrobate, which is essentially the same as the one used in a recent study [11]. Table 2 lists the fuel properties of Hydrobate. As mentioned in that study, n-heptane was added to the base Hydrobate (supplied by Chevron) to enhance the HCCI reactivity [11]. However, due to batch-to-batch variation in the composition of the base Hydrobate, less nheptane was added in this work (11 - 12 vol%, compared to 15% in the previous work) in order to match a reference operating point at naturally aspirated condition.

The HCCI combustion of Hydrobate was studied at $P_{in} = 100, 130, 160$ and 180 kPa. ϕ -sensitivity, which measures the variation of 10% burn point (CA10) with ϕ_m , and high-load limit at each pressure were determined with fully premixed fueling and partial stratified fueling. ϕ -sensitivity was tested

with a specially design technique, as detailed below. High-load limit was approached by first operating the engine at a relatively stable condition, and then gradually increasing the fueling rate to the knock/stability limit or stoichiometric limit, whichever came first. During this process, T_{in} was fixed at 60°C , and combustion HRR was controlled by adjusting CA50 with EGR. A constant ringing intensity, 5 MW/m^2 , was used in most cases. However, lower ringing intensities were used in several cases where this produced a more stable combustion phasing. In addition, a separate set of experiments was performed to investigate the variation of ITHR with P_{in} for this two-stage ignition fuel. For this sweep, ϕ_m was held constant while P_{in} was varied, and the CA10 was kept constant by varying EGR. All data reported in this work were obtained at an engine speed of 1200 rpm.

Table 2. Properties of Hydrobate^{a,b}

Formula.....	C _{7.67} H _{16.2} ^c
Density at 20°C, g/cm ³	0.738
Carbon, wt. %	85.0
Hydrogen, wt. %	15.0
Lower Heating Value, MJ/kg.....	43.96
RON.....	69.0 ^d
MON.....	66.0 ^d
Antiknock Index	67.5 ^d
<u>Distillation, °C</u>	
10%	88
50%	109
90%	151
<u>Hydrocarbon Composition, wt.%</u>	
Normal Paraffin.....	15.6
Branched Paraffin.....	36.7
Olefin	1.6
Naphthene	26.1
Aromatics	14.1
Oxygenates.....	0
Others	5.8

^aData for the Hydrobate used in [11], which contains 85% base Hydrobate and 15% n-heptane by volume.

^bHydrobates used in this work contains 88-89% base Hydrobate and 11-12% n-heptane due to batch-to-batch variation in base-Hydrobate composition. These mixtures are expected to have properties very similar to those shown here.

^cEstimated from fuel hydrocarbon composition.

^dCalculated based on detailed fuel composition.

The ϕ -sensitivity test is conducted in a carefully designed experiment to isolate fuel-chemistry effect from thermal and residual-gas effects on ignition timing as ϕ is varied. The latter effects primarily result from wall heating and the varying temperature and composition of residual-gas at different ϕ . An alternative firing method is used, as in Ref. [7, 9], where 19 cycles fire at a baseline ϕ , followed by 1 cycle