

Boosted HCCI - Controlling Pressure-Rise Rates for Performance Improvements using Partial Fuel Stratification with Conventional Gasoline

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ABSTRACT

This study investigates the potential of partial fuel stratification for reducing the knocking propensity of intake-boosted HCCI engines operating on conventional gasoline. Although intake boosting can substantially increase the high-load capability of HCCI, these engines would be more production-viable if the knock/stability load limit could be extended to allow higher loads at a given boost and/or to provide even higher thermal efficiencies. A technique termed partial fuel stratification (PFS) has recently been shown to greatly reduce the combustion-induced pressure-rise rate (PRR), and therefore the knocking propensity of naturally aspirated HCCI, when the engine is fueled with a ϕ -sensitive, two-stage-ignition fuel. The current work explores the potential of applying PFS to boosted HCCI operation using conventional gasoline, which does not typically show two-stage ignition. Experiments were conducted in a single-cylinder HCCI research engine (0.98 liters) at 1200 rpm. The engine was equipped with a compression-ratio 14 piston, and combustion phasing was controlled by EGR addition.

PFS is produced by premixing the majority of the fuel and then directly injecting the remainder (up to about 20%) in the latter part of the compression stroke. For PFS to be effective, the fuel's autoignition chemistry must vary with the local equivalence ratio (ϕ) to produce a staged combustion event. Accordingly, tests were conducted to determine the ϕ -sensitivity of gasoline. They show that at naturally aspirated conditions ($P_{in} = 1$ bar), gasoline is not ϕ -sensitive, and PFS is not effective for reducing the PRR. However, with sufficient intake boost (e.g. $P_{in} = 2$ bar), gasoline is found to become highly ϕ -sensitive, and PFS very effectively reduces the PRR. Varying the amount of PFS, by adjusting either the

timing or amount of DI fuel, allows control of the PRR reduction. Applying PFS to high loads at $P_{in} = 2$ bar substantially shifts the knock/stability limit and increases the maximum IMEP_g from 11.7 (premixed) to 13.0 bar (PFS). Maximum load improvements with PFS are also seen for other intake pressures ranging from 1.6 to 2.4 bar. Finally, because it allows more advanced combustion phasing without knock, PFS is also effective for increasing the thermal efficiency of boosted HCCI over a range of loads for each P_{in} , yielding typical fuel economy improvements of 2 - 2.5%. Overall, PFS has a strong potential for improving gasoline-fueled boosted HCCI operation.

INTRODUCTION

Concerns about limited petroleum supplies, global warming, and urban air pollution are driving the demand for engines that are both highly fuel efficient and have low levels of toxic emissions. At the same time, significant market penetration requires that cost of these improvements be minimal. Homogeneous Charge Compression Ignition (HCCI) and HCCI-like combustion has been shown to be capable of providing high thermal efficiencies and ultra-low NOx and particulate matter emissions, using relatively inexpensive gasoline spark-ignition (SI) engine fuel injectors. These features make HCCI well suited for future transportation engines; however, the limited power output of these engines still remains a barrier to their widespread implementation.

The load limitation typically occurs because the maximum cylinder-pressure rise rate (PRR) increases with fueling rate, eventually causing engine knock. To mitigate these high PRRs with increased fueling, the combustion phasing can be retarded, but the amount of allowable retard is limited by

excessive cycle-to-cycle variation in the power output, and eventually misfire [1,2]. As the fueling is increased, it can become increasingly difficult to maintain sufficiently retarded combustion to prevent run-away knock [3] without having poor combustion stability that can drift into misfire [1,2,4]. Closed-loop control systems can help extend this knock/stability limit [1], but the maximum load for naturally aspirated HCCI remains low relative to traditional spark-ignition or turbo-charged diesel engines.

INTAKE-BOOSTED HCCI

Intake-pressure boosting is often used to increase the power output of internal combustion engines, and for HCCI engines, it provides a means of increasing the power while still maintaining a sufficiently dilute charge to prevent NO_x formation. However, early efforts with intake-boosted HCCI met with limited success due to difficulties in controlling engine knock. The knocking propensity of HCCI increases with boost because the greater charge mass with boost results in a greater pressure rise with combustion, and because boost enhances the autoignition process causing the combustion phasing to advance. As a result, most early attempts to achieve high loads with boosted HCCI used special fuels and/or reduced compression ratios [5, 6, 7, 8].

Recently, however, the authors demonstrated that substantial increases in the high-load limit of HCCI could be achieved with intake pressure boosting using conventional gasoline and a compression ratio of 14 [9]. In this previous work, the maximum load attainable increased substantially with boost, up to the highest load attempted, a gross indicated mean effective pressure (IMEP_g)¹ = 16.3 bar at P_{in} = 3.25 bar. For all conditions, fully premixed fueling was used, NO_x emissions were very low, soot was not detectable, and PRRs were kept sufficiently low to prevent engine knock. Controlling the PRR under highly boosted conditions required substantial combustion-phasing retard. Fortunately, for boosted operation with gasoline fueling, good combustion stability could be maintained at much later combustion phasings than was possible for naturally aspirated operation, due to an enhancement of the intermediate-temperature reactions that occur prior to full hot ignition, as discussed in detail in Ref. [9]. To achieve the required retard despite the pressure-induced enhancement of autoignition, a combination of reduced intake temperature and cooled EGR was used.

Although these previous results [9] show the strong potential of boosted HCCI with gasoline fueling, there are still several aspects where improvements are desirable. Three key areas are: 1) For boost levels below P_{in} ≈ 2.6 bar, the load is

limited by the knock/stability limit.² If the combustion PRR could be reduced by some means other than timing retard, the reduced knocking propensity could shift the knock/stability limit or remove it altogether, thus allowing higher fueling rates at these boost levels. 2) Exhaust enthalpies, although adequate, are lower than desirable for driving a turbo-charger. Increasing the load for a given boost would increase exhaust temperatures, easing turbo-charger design. 3) Although the thermal efficiencies reported in [9] are high, even higher thermal efficiencies could be obtained if less timing retard was required to control knock.

PARTIAL FUEL STRATIFICATION

One promising method for controlling the combustion heat release rate (HRR) and hence, the combustion PRR in HCCI engines to use partial fuel stratification. However, for fuel stratification to reduce the HRR, autoignition times must be sensitive to variations in the local fuel/air equivalence ratio (ϕ), so that the ϕ -stratification produces a staged heat release event [10] as autoignition occurs sequentially down the ϕ gradient. Using a special technique to isolate fuel-chemistry effects from thermal effects, Dec and Sjöberg [11] showed that the autoignition chemistry of some fuels is sensitive to variations in ϕ (ϕ -sensitive) while for other fuels variations in ϕ have little or no effect on the autoignition delay times. They showed that PRF80, which exhibited two-stage ignition at the conditions studied, showed significant ϕ -sensitivity, while iso-octane and gasoline, which exhibited single-stage ignition were not ϕ -sensitive at these naturally aspirated conditions. Subsequently, Sjöberg and Dec [10] demonstrated that the ϕ -sensitivity of PRF83, which showed two-stage ignition at the conditions studied, could be exploited to reduce the HRR of a naturally aspirated HCCI engine. To accomplish this, they used a mixture preparation technique dubbed partial fuel stratification (PFS), in which the majority of the fuel is fully premixed and a fraction (up to about 20%) is directly injected during the latter part of the compression stroke using a gasoline-type direct injector (GDI) to provide the mixture stratification. As discussed in Ref. [10], this PFS technique was chosen because computations indicated that it could provide more than enough fuel stratification, and the premixed fuel maintains a sufficiently high ϕ in the leanest parts of the charge to obtain good combustion efficiency [12]. The large premixed portion also assures good air utilization throughout the combustion chamber.

Later, the current authors conducted a more in-depth study of the potential of PFS to reduce HRRs of HCCI [13]. This work showed that a two-stage-ignition fuel with stronger pre-ignition reactions (*i.e.* reactions that occur at low and intermediate temperatures prior to hot ignition), PRF73, was

¹IMEP_g refers to the IMEP over the compression and expansion strokes only, while IMEP_{net} refers to the IMEP over the entire cycle, including the gas-exchange strokes.

²For P_{in} ≥ 2.6 bar, the maximum load is limited by oxygen availability because of the high EGR levels required to control combustion phasing, so reductions in the PRR will not allow higher loads unless they also reduce EGR requirements.

very effective at reducing the HRR at a high-load operating point while maintaining clean (low NO_x), efficient, and stable HCCI combustion. This work also showed that a single-stage-ignition fuel, iso-octane, showed no tendency to reduce the HRR. Finally, in a work concurrent with the study presented here, the authors showed that a distillate fuel with two-stage ignition and an overall HCCI autoignition reactivity similar to PRF73 was also very effective at reducing the HRR with PFS [14]. These results are consistent with other works in the literature. Dahl *et al.* [15] showed that the PRR of a two-stage fuel (RON ~ 50) was reduced with mixture stratification more effectively than with a RON = 100 gasoline in a high residual engine utilizing negative valve overlap. Wada and Senda [16] found that mixture stratification produced a larger reduction in PRR with a fuel showing a strong low-temperature heat release (LTHR), PRF45, than did a fuel with weak LTHR, PRF80. All of these previous studies were at naturally aspirated conditions.

These previous works consistently show that mixture stratification is effective for reducing the HRR for fuels having two-stage ignition, and it is ineffective for fuels with single-stage ignition. Additionally, they indicate that two-stage fuels having stronger LTHR are more effective than those with less. Generalizing, fuels that have greater HCCI autoignition reactivity (*i.e.* that autoignite more easily under HCCI conditions) are more likely to have a higher ϕ -sensitivity and to be more effective for use with mixture stratification than those with lower autoignition reactivity.

Gasoline is designed to have a low autoignition reactivity (high octane number), and it has been shown to be a single-stage ignition fuel at typical naturally aspirated HCCI operating conditions [9,17], which suggests that it will not be effective with mixture stratification. However, intake-pressure boosting enhances autoignition reactivity of gasoline significantly [9]. Most of this enhancement of the pre-ignition reactions is in the form of increased intermediate-temperature heat release (ITHR) with significant LTHR being evident only at the lowest intake temperature (T_{in}) tested, 45°C. Although it is strictly speaking still a single-stage fuel (or nearly single-stage) at most of the boosted conditions presented in our previous study of boosted HCCI [9], the substantial increase in autoignition reactivity suggests that gasoline may still become ϕ -sensitive under boosted conditions, allowing the use of PFS to reduce the heat release rate.

OBJECTIVES

The objective of the current work is to examine the potential for using mixture stratification to improve boosted HCCI operation with conventional gasoline. The results of this study are presented in four parts:

1. Examination of the ϕ -sensitivity of gasoline for naturally aspirated and boosted HCCI operation.

2. Investigation of the effects of PFS on the PRR and HRR at a selected high-load fueling rate for naturally aspirated (1 bar) and boosted (2 bar) operation. To determine how the effectiveness of PFS changes with the amount of stratification, sweeps of both the direct-injection (DI) timing and the DI fraction are conducted for both intake pressures.

3. Application of PFS to determine the extent to which the high-load limit of boosted HCCI can be extended for a range of boost levels from 1.6 - 2.4 bar absolute.

4. Investigation of the potential of using PFS to improve the efficiency of boosted HCCI by allowing more advanced combustion phasing without knock.

EXPERIMENTAL SETUP & DATA ACQUISITION

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. Figure 1 shows a schematic of the engine, which has been converted for single-cylinder operation by deactivating cylinders 1-5. The engine specifications and operating conditions are listed in Table 1. The configuration of the engine and facility are identical to those used our previous study of intake pressure boost [9]. Additional information on the facility may be found in Ref. [3,12].

The active HCCI cylinder is fitted with a compression-ratio (CR) = 14 custom piston as shown in Fig. 1b. This piston provides an open combustion chamber with a large squish clearance and a quasi-hemispherical bowl. Comparison tests showed minimal differences in performance and emissions compared to the CR = 14 broad shallower-bowl piston used in many earlier studies *e.g.* [4,18]. Both pistons provide a small topland ring crevice, amounting to only 2.1% of the top-dead-center (TDC) volume. A more complete discussion of the comparison between pistons may be found in Ref. [3].

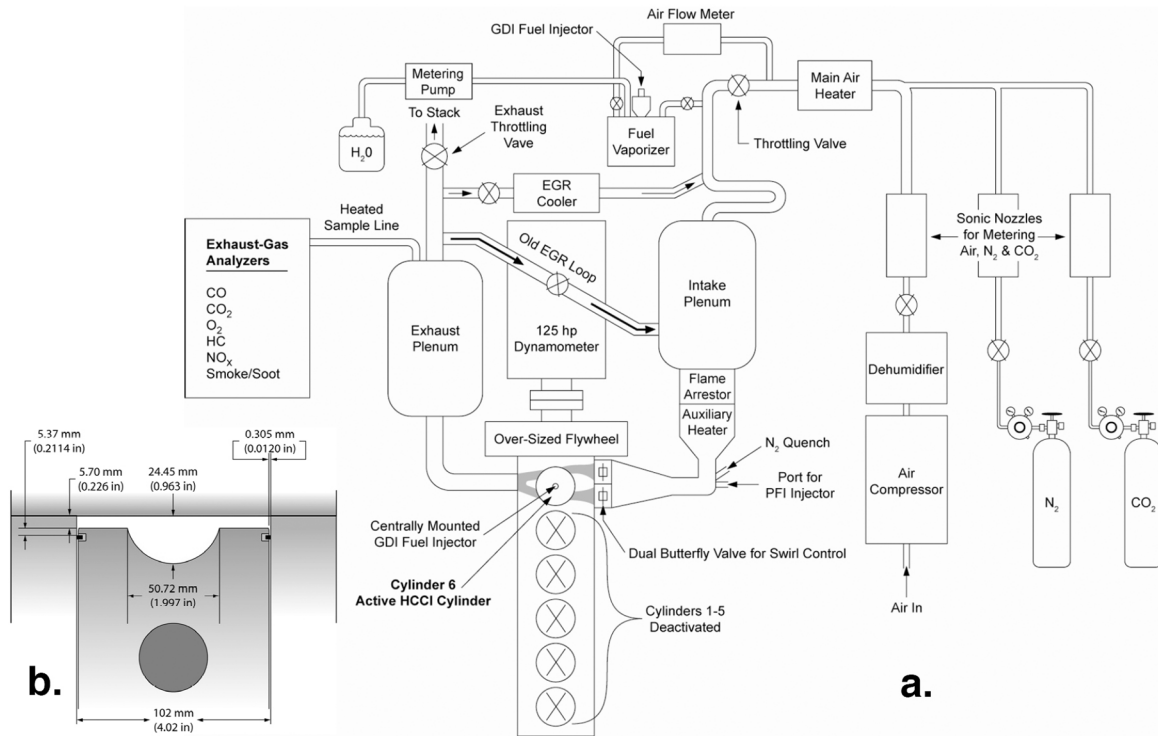


Figure 1. Schematic of the HCCI engine facility (a), and combustion chamber geometry of the CR = 14 piston, at TDC (b).

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
No. of Valves	4
IVO	0° CA*
IVC	202° CA*
EVO	482° CA*
EVC	8° CA*
Swirl Ratio	0.9
Fueling system	Fully Premixed / 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
Intake Temperature	60 – 143°C
Intake Pressure (abs.)	1.0 – 2.4 bar
Coolant Temperature	100°C

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

As shown in Fig. 1a, the engine facility is equipped to use real EGR or to simulate EGR by adding N₂, CO₂ and H₂O in proportions that would result from complete combustion of

the fuel (here called complete stoichiometric products or CSP). In this article, the term EGR refers only to real EGR, while the term CSP is used to refer to the simulated EGR. As shown in the figure, the EGR loop is equipped with a cooler (a gas-to-water heat exchanger), and 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 or air flow plus CSP was adjusted to achieve the desired intake pressure, as measured by a pressure transducer on the intake runner. Intake pressures varied from 1.0 bar (simulating naturally aspirated conditions) to 2.4 bar for the current study. All pressures given are absolute. For operation with CSP, 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 and introduced into the intake flow near the location where the EGR is introduced as shown in Fig. 1a. 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. The EGR fraction was varied by adjusting the amount of supplied air, and then adjusting the exhaust throttle to maintain the desired intake pressure.

To achieve partial fuel stratification, a combination of premixed and DI fueling was used. The premixed fueling system, shown at the top of the schematic in Fig. 1a, consists of a 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. The DI fueling is accomplished using a second GDI injector mounted centrally in the cylinder head. A positive displacement fuel flow meter was used to determine the amount of fuel supplied. The fuel was a research-grade gasoline supplied by Chevron-Phillips Chemical Co., and its specifications are listed in Table 2. Note that this fuel meets the same basic specifications as the fuel used in our previous study of boosted HCCI [9], but it is from a different batch, and there are small difference in composition and the distillation curve.

TABLE 2. Chevron-Phillips Research-Grade Gasoline*

Antiknock Index (R+M)/2.....	87.6
RON	91.7
MON	83.4
Specific gravity	0.753
Carbon, wt%.....	86.27
Hydrogen, wt%.....	13.73
Oxygen, wt%	<0.05
A/F Stoichiometric	14.62
Lower Heating Value, gas-phase fuel (MJ/kg)	42.94
LHV for stoichiometric charge (MJ/kg)	2.749
Hydrocarbon Type, vol%	
Aromatics.....	25.7
Olefins	5.0
Saturates	69.3
Distillation, °C	
5%	56
10%	63
30%	81
50%	101
70%	118
90%	147
95%	161

*Based on analysis provided by Chevron-Phillips.

Similar to some of our other recent works [13,14], 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

$$\phi_m = \frac{(F/C)}{(F/A)_{stoich}}$$
, 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 ϕ is < 1 for all conditions presented, so combustion is never oxygen limited.

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. 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° - 143°C. All data were taken at an engine speed of 1200 rpm

DATA ACQUISITION

Cylinder pressure measurements were made with a transducer (AVL QC33C) mounted in the cylinder head approximately 42 mm off center. The pressure transducer signals were digitized and recorded at $\frac{1}{4}^\circ$ 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 temperatures were monitored using thermocouples mounted in the two intake runners close to the cylinder head. Firedeck temperatures were monitored with a 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 [13]. For all data presented, 0° crank angle (CA) is defined as TDC intake (so TDC compression is at 360°). This eliminates the need to use negative crank angles or combined bTDC, aTDC notation.

The crank angle of the 50% burn point (CA50) was used to monitor the combustion phasing, and the 10% burn point

*Based on analysis provided by Chevron-Phillips.

(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 [12]). Computations were performed for each individual cycle, disregarding heat transfer and assuming a constant ratio of specific heats [19]. The average of 100 consecutive individual-cycle CA10 or CA50 values were then used to monitor CA10 or CA50 during engine operation and for the values reported. The reported PRRs and ringing intensities 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 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 ringing intensity developed by Eng [20] 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 PRR (in real time), pressure, and temperature, respectively, γ is the ratio of specific heats (c_p/c_v), and R is the gas constant. 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 obvious ripples on the pressure trace, a ringing criterion of 5 MW/m² was selected as the ringing limit for operation without knock. This corresponds to about 8 bar/ $^\circ\text{CA}$ at 1200 rpm, naturally aspirated. It should be noted that as P_{in} is increased above 1.0 bar, $dP/d\theta$ can exceed 8 bar/ $^\circ\text{CA}$ since the corresponding increased value of $P_{\max}$ in the denominator of Eq. 1 reduces the ringing intensity for a given PRR. At all boost levels 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 of the effects of PFS presented in the second part of the results. Here, the heat release was computed in a more refined way from the ensemble-averaged pressure trace (with the 2.5 kHz low-pass filter applied), using the Woschni correlation for heat transfer [19]. Using

the ensemble-averaged pressure trace has benefits from the standpoint of reduced noise on the heat-release traces. On the other hand, it can lead to overestimated burn durations if the cycle-to-cycle variations are large. However, for the condition where this was applied, the phasing was fairly stable with the standard deviation of CA50 over 100 fired cycles averaging about 1.2 $^\circ$ 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 $^\circ$ - 25 $^\circ$ CA, the relatively small cycle-to-cycle variation in CA50 will have little effect.

Exhaust emissions data were also acquired, with the sample being drawn from 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. In addition, a second CO₂ meter monitored the intake gases just prior to induction into the engine. This allowed the EGR fraction of the intake gases to be computed from the ratio of the intake and exhaust CO₂ concentrations. Smoke measurements were also made with an automated smoke meter.

RESULTS AND DISCUSSION

Φ -SENSITIVITY OF GASOLINE

As discussed in the introduction, the successful application of partial fuel stratification to reduce the HRR requires that the autoignition timing be sensitive to variations in the local ϕ_m . To determine the ϕ -sensitivity, the change in fuel-chemistry effects on autoignition must be isolated from the change in thermal effects (wall and residual-gas temperatures) and the effects of residual-gas composition as the global ϕ_m is varied. To accomplish this, an alternate-firing method is used, as in [10,11,14], where 19 cycles fire at a baseline ϕ_m , followed by 1 cycle fired at the ϕ_m of interest, for which the data are acquired. In this manner, the wall temperature remains nearly constant as the ϕ_m of cycle 20 is varied, since heating is dominated by the 19 cycles whose ϕ_m remains constant [11]. Additionally, the temperature and composition of the residuals in the charge of cycle 20 remain constant since they always come from the 19th cycle fueled at the same baseline ϕ_m .

Changing the fueling rate from cycle to cycle for this alternate-firing technique, requires the use of direct fuel injection. However, DI fueling can potentially change the in-cylinder thermal conditions and mixture homogeneity, even for early injection [21]. To minimize this, most of fuel is supplied with the premixed fueling system, and only a small portion is directly injected to reach the required ϕ_m . For example, for a baseline ϕ_m of 0.42, 80% of the baseline fuel

(corresponding to $\phi_m = 0.34$) is premixed. DI is then used to add $\phi_m = 0.08$ to reach the baseline ϕ_m . For a test with $\phi_m < 0.42$, the DI ϕ_m of cycle 20 is reduced below 0.08, while the premixed fueling remains constant. Similarly, for a test $\phi_m > 0.42$, the DI ϕ_m of cycle 20 is increased above 0.08. Note that the test ϕ_m cannot be lower than 0.34 in this case. Also, in order to generate as homogeneous of a mixture as possible, an early DI timing, 40°CA, is used. This timing was determined from a preliminary DI timing sweep at a base ϕ_m to ensure good mixture homogeneity and no piston wetting by comparing the performance with the fully premixed case.

To determine the ϕ -sensitivity, CA10 of the base- ϕ_m condition (*i.e.* all 20 cycles fired at the baseline ϕ_m) is first adjusted to a selected value (368° CA for the data in Fig. 2a) by adjusting T_{in} and the amount of CSP (simulated EGR) supplied. These values are then held constant while the ϕ_m of cycle 20 is varied above and below the baseline ϕ_m value, and the change in CA10 is recorded (ensemble averaged over 20 ϕ_m -of-interest cycles).

Figure 2a shows the ϕ -sensitivity at $P_{in} = 1$ bar (naturally aspirated) of gasoline, compared to previous data at this same intake pressure for PRF73 and iso-octane from Ref. [13]. At these conditions, PRF73 is a two-stage-ignition fuel, while gasoline and iso-octane are single-stage ignition fuels. The ϕ -sensitivity is shown in terms of CA10, which is indicative of the hot-ignition timing. PRF73 shows significant advancement of CA10 with increasing ϕ_m as a result of the strong dependence of its robust pre-ignition reactions (reactions responsible for the LTHR and ITHR) on the fuel concentration. Thus, PRF73 is quite ϕ -sensitive, and a range of autoignition times can be expected from an in-cylinder ϕ_m distribution produced by PFS, with a commensurate reduction in the HRR as shown in [13]. In contrast, CA10 for gasoline ($P_{in} = 1$ bar) is moderately retarded with increasing ϕ_m . This behavior is similar to iso-octane, which has autoignition characteristics similar to gasoline at these conditions [22]. Both of these fuels have relatively weak pre-ignition reactions that are not sensitive to changes in fuel concentration. As a result, their CA10 behavior is dominated by the decrease in γ (c_p/c_v) with increased fueling, which reduces the compressed-gas temperature, causing CA10 to become retarded. Gasoline is retarded slightly less than iso-octane, indicating that its pre-ignition chemistry is slightly enhanced by increasing ϕ_m , but not enough to overcome the γ effect. Note that this same cooling effect also occurs for PRF73, but its ϕ -sensitivity is sufficiently strong to overcome it and still advance CA10 significantly. These findings are in good agreement with a comparison of the ϕ -sensitivity of gasoline, iso-octane, and PRF80 at a similar operating

condition [11]. This lack of ϕ -sensitivity for gasoline at $P_{in} = 1$ bar suggests that PFS will not be effective for reducing the HRR for naturally aspirated operation.

However, intake pressure boosting significantly enhances the autoignition reactivity of gasoline as discussed in the Introduction, and in greater depth in Ref. [9]. As shown in our previous work [9], the ITHR increases substantially as boost levels are increased from 1 to 2 bar. To prevent this enhancement from causing overly advanced combustion timing, T_{in} was decreased from around 140°C to 60°C and significant amounts of EGR were added. The 60°C low-temperature limit was chosen to prevent water condensation in the cooled EGR gases. With $T_{in} = 60^\circ\text{C}$ significant LTHR was not observed. Only a small hint of LTHR could be seen on a highly amplified scale for P_{in} of 1.8 bar and higher. Thus, gasoline remained largely a single-stage fuel over the boost range studied in [9], 1 - 3.25 bar. Despite this single-stage behavior, the large increase in pre-ignition reactivity in terms of increased ITHR, suggests that gasoline may become ϕ -sensitive under boosted operation.

To test this hypothesis, ϕ -sensitivity sweeps were conducted for $P_{in} = 2$ and 1.6 bar. Because the pressure rise with combustion is greater for boosted operation, more timing retard is required to prevent excessive ringing (knock). Therefore, it was not possible to match both the base $\phi_m = 0.42$ and CA10 of the naturally aspirated data in Fig. 2a. Accordingly, two ϕ -sensitivity sweeps were conducted for $P_{in} = 2$ bar: one matching the base- $\phi_m = 0.42$ in Fig. 2a but with CA10 retarded by about 5° CA, and another matching the CA10 = 368° CA in Fig. 2a, but with a base- $\phi_m = 0.345$. A ϕ -sensitivity sweep was also conducted for $P_{in} = 1.6$ bar matching the CA10 in Fig. 2a with a base- $\phi_m = 0.367$.

Figure 2b presents the results of these boosted gasoline sweeps along with the $P_{in} = 1$ bar gasoline and PRF73 data from Fig. 2a. As can be seen, gasoline has a very strong ϕ -sensitivity at $P_{in} = 2$ bar for both ϕ_m ranges, and a moderate ϕ -sensitivity at $P_{in} = 1.6$ bar. In fact, the ϕ -sensitivity of gasoline at $P_{in} = 2$ bar is even stronger than that of PRF73 as evident by its steeper slope in Fig. 2b. At $P_{in} = 1.6$ bar, the sensitivity is somewhat less than PRF73, but still significant. These results suggest that PFS should be effective for reducing the HRR of boosted HCCI which could allow higher fueling rates without excessive ringing (knock) and/or more advanced combustion phasing for higher efficiency. Additionally, they suggest that the ITHR pre-ignition reactions are as important, or perhaps more important, for determining ϕ -sensitivity than the LTHR reactions previously thought to be key [10,11,13].³

³In a concurrent work [14], the authors also conclude that ITHR is quite important for ϕ -sensitivity, based on other comparisons between fuels.