

Gasoline-Like Fuel Effects on High-Load, Boosted HCCI Combustion Employing Negative Valve Overlap Strategy

Vickey B. Kalaskar, Derek A. Splitter, and James P. Szybist
Oak Ridge National Lab.

ABSTRACT

In recent years a number of studies have demonstrated that boosted operation combined with external EGR is a path forward for expanding the high load limit of homogeneous charge compression ignition (HCCI) operation with the negative valve overlap (NVO) valve strategy. However, the effects of fuel composition with this strategy have not been fully explored. In this study boosted HCCI combustion is investigated in a single-cylinder research engine equipped with direct injection (DI) fueling, cooled external exhaust gas recirculation (EGR), laboratory pressurized intake air, and a fully-variable hydraulic valve actuation (HVA) valve train. Three fuels with significant compositional differences are investigated: regular grade gasoline (RON = 90.2), 30% ethanol-gasoline blend (E30, RON = 100.3), and 24% iso-butanol-gasoline blend (IB24, RON = 96.6). Results include engine loads from 350 to 800 kPa IMEP_g for all fuels at three engine speeds 1600, 2000, and 2500 rpm. All operating conditions achieved thermal efficiency (gross indicated efficiency) between 38 and 47%, low NO_x emissions (≤ 0.1 g/kWh), and high combustion efficiency ($\geq 96.5\%$). Detailed sweeps of intake manifold pressure (atmospheric to 250 kPa_a), EGR (0 - 25% EGR), and injection timing are conducted to identify fuel-specific effects. The major finding of this study is that while significant fuel compositional differences exist, in boosted HCCI operation only minor changes in operational conditions are required to achieve comparable operation for all fuels. In boosted HCCI operation all fuels were able to achieve matched load-speed operation, whereas in conventional SI operation the fuel-specific knock differences resulted in significant differences in the operable load-speed space. Although all fuels were operable in boosted HCCI, the respective air handling requirements are also discussed, including an analysis of the demanded turbocharger efficiency.

CITATION: Kalaskar, V., Splitter, D., and Szybist, J., "Gasoline-Like Fuel Effects on High-Load, Boosted HCCI Combustion Employing Negative Valve Overlap Strategy," *SAE Int. J. Fuels Lubr.* 7(1):2014, doi:10.4271/2014-01-1271.

INTRODUCTION

Despite the fact that HCCI has not realized its potential on production engines in the more than 3 decades since it was originally reported [1, 2, 3], it remains a promising mode of combustion to simultaneously achieve high efficiency and low NO_x emissions. While there has been a substantial amount of research and development on HCCI combustion, there remains a significant number of implementation barriers.

Some of the barriers associated with HCCI have been overcome, or substantially reduced. Specifically, sensors for feedback control, engine controllers, valve trains, and DI fueling systems have all developed at a rapid rate, providing a greater means to control HCCI combustion [4]. However, additional substantial barriers still remain, that limit the current production viability of HCCI. Specifically, under naturally-aspirated conditions, HCCI combustion is limited to a relatively light operating load of approximately 400 kPa net indicated mean effective pressure (IMEP_{net}¹). While this has always been

a limitation of naturally aspirated HCCI combustion, it is becoming even more critical given the trends of engine downsizing and hybridization for fuel economy improvements [4]. Downsized engines utilize turbo-machinery for higher specific power, allowing the engine to operate at higher loads with higher efficiency than a larger displacement naturally aspirated engine. Hybrid electric technologies aim to minimize the use of the engine at the lightest load conditions where efficiency is lowest. In both cases the engine duty cycle is shifted to higher specific power conditions, thereby reducing the relevance of naturally aspirated HCCI combustion.

Using a boosted air handling system has been shown as a way to provide additional dilution has been solidified as a path for higher load HCCI combustion [5, 6, 7, 8, 9, 10]. The present study builds on the cited works by employing NVO fueling. The engine facility used at ORNL, has been previously reported in [7], where engine loads in excess of 650 kPa IMEP_{net} were demonstrated, while also characterizing the NVO engine control authority and sensitivity. The results of this study are

¹ Defined as the indicated mean effective pressure for the entire 4-stroke process, i.e., 720°CA

that NVO duration was a coarse means of controlling combustion phasing, and fuel injection timing was a fine control variable [7].

Although these recent works have demonstrated load expansion and control authority of boosted HCCI, an additional HCCI production barrier is that HCCI and other advanced combustion modes can be highly sensitive to the fuel composition. Viable production engines require fuel robustness as current market-available fuel composition is not constant on either a seasonal or vendor basis, and the fuel landscape in the United States is undergoing a rapid change. In 2007, the Energy Independence and Security Act (EISA) was signed into law [11]. This law established a new renewable fuel standard requiring the nation use 36 billion gallons of renewable fuels per year by 2022.

Because of this legislation, fuel ethanol doubled from 5.48 billion gpy in 2006 to 12.95 billion gpy in 2012 [12]. Ethanol is by far the largest volume renewable fuel for the transportation sector, and will likely continue to be an essential element of meeting the EISA target. This volume currently puts the United States at the E10 blend wall, meaning that no additional ethanol can be consumed at a blend of 10 vol% and there have been a number of recent studies showing that mid-level blends of ethanol, in the 15-30% range, could enable higher efficiency in a purpose-built engine because of the high octane number and high latent heat of vaporization [13, 14, 15, 16, 17, 18]. Such a fuel could increase the ethanol blend wall and assist OEMs in achieving the fuel economy standards.

In addition to ethanol, there has also been a significant amount of interest in using iso-butanol as a renewable blend stock for spark ignition engines. Iso-butanol is more compatible with the current fueling infrastructure than ethanol, including being compatible with pipelines because it does not separate into an aqueous phase in the presence of water [19]. Iso-butanol offers advantageous fuel properties in that its volumetric energy density approaches that of gasoline, it is a high octane number blending component, and it has a high latent heat of vaporization, though neither the octane number nor the latent heat of vaporization are as high as ethanol.

While there have been a significant number of previous studies investigating the effects of fuel properties on HCCI combustion [20, 21, 22, 23, 24, 25, 26, 27], the present investigation explores if the rapidly changing fuel composition in the United States could affect the high load limits of boosted HCCI operation in a manner similar to SI operation. Specifically, in this investigation we investigate a baseline fuel, market available 87AKI E0 gasoline. Additionally, splash blends of 30 vol% ethanol and 24 vol% iso-butanol are made with the baseline fuel. Using these fuels, boosted HCCI experiments are conducted in a single-cylinder research engine at conditions relevant to high load expansion.

EXPERIMENTAL

Single Cylinder Engine

The engine used in this study is a highly modified 2.0 L GM Ecotec engine. This engine uses a side-mounted direct injection (DI) fueling system. Engine geometry details are presented in Table 1. Three of the four cylinders on this engine are disabled and the stock 9.2:1 compression ratio (r_c) piston has been replaced by an 11.85:1 r_c piston.

Table 1. Engine geometry.

Bore x Stroke [mm]	86 x 86
Conrod length [mm]	145.5
Compression ratio [-]	11.85:1
Fuel injection system	Direct injection, side-mounted

The cylinder head is modified to accommodate a fully-variable hydraulic valve actuation (HVA) system from Sturman Industries. This system uses custom valves with longer valve stems than the stock configuration, but otherwise the valve geometry is unchanged. The cylinder pressure is measured using a piezoelectric pressure transducer from Kistler (6125C). In production, this engine uses a cam-driven fuel pump; however, the HVA system required substantial cylinder head modifications, disabling the production fuel pump. Instead, fuel is supplied to the rail using an external pump. A picture of the setup is shown in Figure 1.

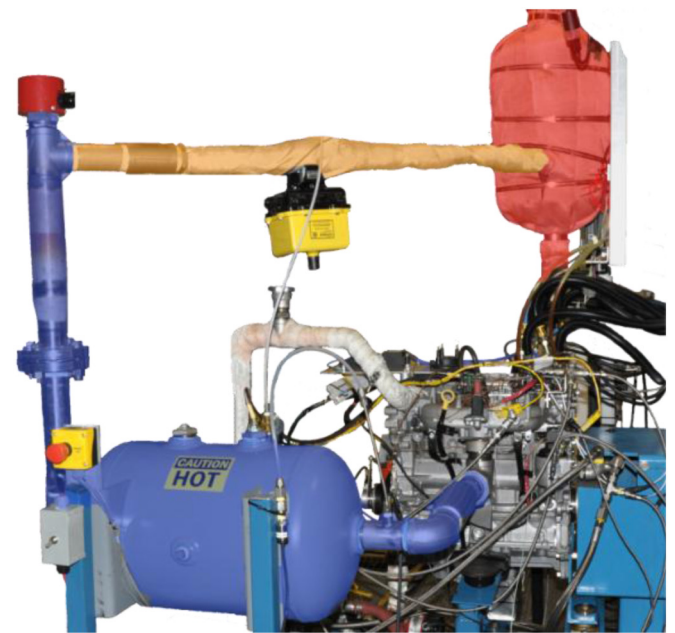


Figure 1. Single cylinder engine equipped with HVA valve train and laboratory air handling system.

The HVA system allows fully independent control of each of the four valves. Valve lift, duration, and timing can be controlled including an option to disable a valve using the system. Although there is great system flexibility, the valve lift profile is

more abrupt and square wave-like, as compared to a cam based profile. A comparison of Sturman and cam-based profiles was made while investigating the low load limit of HCCI with this engine in [28].

To make the present study as relevant as possible, the valve lifts are held constant at 4mm and valve duration is held at 98°CA constant (measured at a valve lift of 0.8 mm). The intake and exhaust valve events are then symmetrically phased with respect to fired TDC. A total of 32°CA of cam phasing authority is used to span the range of conditions investigated. Figure 2 shows the cylinder pressure and intake and exhaust valve profiles for a light-load and high-load condition to illustrate the investigated range of conditions.

DRIVEN Combustion Analysis Toolkit (DCAT) is used to acquire crank-angle resolved data and analyze combustion performance. These measurements include the cylinder pressure, valve lifts, and fuel injection command signal. The resolution of this system is 0.2 CA deg.

A schematic of the experimental engine setup is shown in Figure 3. Pressurized and dried facility air that has less than 5% relative humidity is metered to the engine using a mass air flow controller. The engine is equipped with separate electro-mechanical valves for backpressure and external exhaust gas recirculation (EGR), enabling independent control of intake and exhaust manifold pressures and EGR rate. Cooled EGR mixes with fresh air upstream of an air heater, followed by the intake plenum and then the intake manifold. EGR is measured using a non-intrusive method that utilizes pressure-compensated wideband oxygen sensors in both the intake and exhaust. The engine operating strategy follows the recipe in [7], where a

constant pressure differential of 10 kPa is maintained between intake and exhaust manifolds except for a small subset of points that investigate turbocharger efficiency requirements.

The details of the engine operating system are presented in Table 2. Fuel flow is measured using a coriolis-based fuel flow meter, and the fuel rail pressure is maintained at a pressure of 10000 kPa. Engine coolant temperature is maintained at 90°C. Intake manifold temperature is maintained at a temperature of 52°C, regardless of external EGR rate. The combustion emissions are monitored using a standard array of analyzers. NO_x is measured using a heated chemiluminescence analyzer,

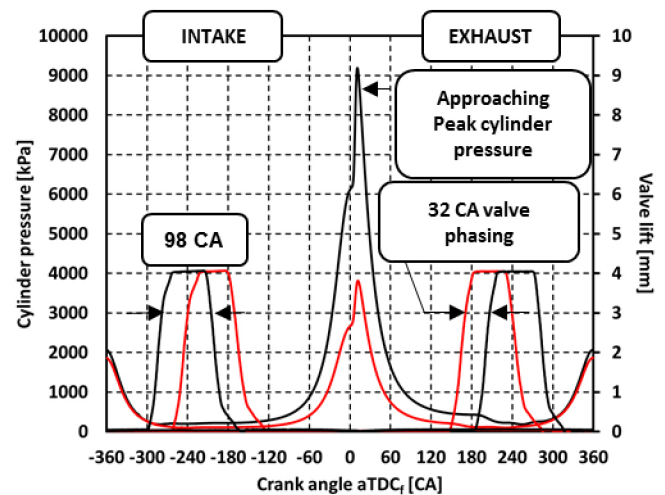


Figure 2. Range of valve timing used in this study and corresponding differences in fired HCCI pressure traces (low-load represented in red, and high-load represented in black).

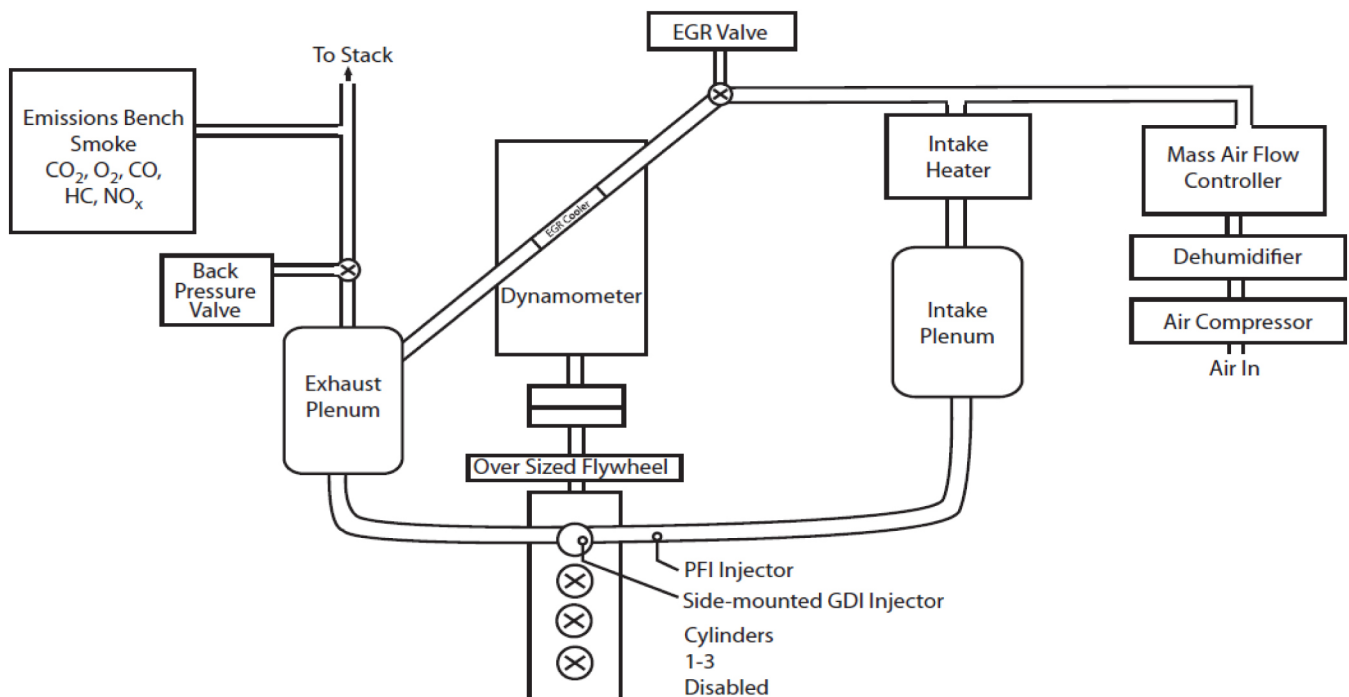


Figure 3. Schematic of experimental engine setup.

CO and CO₂ are measured using infrared analyzers, oxygen is measured using a paramagnetic analyzer, and unburned hydrocarbon emissions (HC) are measured with a heated flame ionization detector. Finally, combustion noise is measured using the AVL 4050 combustion noise meter.

Table 2. Engine operating conditions.

Fuel injection pressure [kPa]	10000
Coolant temperature [°C]	90
Intake manifold temperature [°C]	52

Fuels

Three fuels are tested. Two fuels are alcohol-gasoline blends and the third was an unblended gasoline. All fuels are based on commercially obtained 87 AKI E0 “regular” pump fuel sourced directly from a distribution terminal. The two alcohol blended fuels are splash blended on site with either 24% neat iso-butanol or 30% neat-ethanol, with alcohol and gasoline volume fractions being measured in un-blended fractions and then combined. Splash blending these alcohols with finished market E0 gasoline is likely not what would be done at an industrial scale if the tested blend ratios were to be market sold. It is more likely that a blendstock for oxygenate blends (BOB) fuel would be used, which tend to have lower octane numbers. The blending ratios used are based on the following: The 24% iso-butanol is selected as it has nearly identical oxygen content as E15, which the EPA has currently regulated as the maximum fuel oxygen content [29]. The 30% ethanol blend is selected as the EPA recently stated in tier III notification of proposed rulemaking [30] that there is no present foreseeable issue with higher ethanol-gasoline blends, citing blends as high as E30 are being considered. These alcohols are of interest as blendstocks because they exhibit nearly identical MON values and similar RON values [29,30]. However, the energy density of iso-butanol is higher than that of ethanol on both volume and mass bases, thus making its energy density closer to a gasoline. Additionally, the lower water solubility of iso-butanol as compared to ethanol offers advantages in certain markets such as marine environments where humidity and water contact are more prone. Interestingly, research by Stein *et al.* [15] demonstrates that the Reid vapor pressure of intermediate ethanol blends are similar to that of gasoline, making such fuels attractive to regulatory bodies, as indicated in the recent EPA Tier III notification of proposed rulemaking [30].

In the present study, the neat alcohols used in blending are of non-denatured reagent grade purity and obtained directly from suppliers. Sigma-Aldrich supplied the iso-butanol at a purity of >99%, and Decon Labs supplied the non-denatured ethanol. After splash blending the fuels on-site, the fuels are sent for independent analysis at an ASTM certified laboratory, and key fuel properties from the analysis are presented in Table 3.

As a result of these differences, specifically the octane number and latent heat of vaporization, the fuels have significantly different anti-knock properties leading to significantly different behavior during conventional SI combustion using the identical experimental facility and in [17,18]. The SI results demonstrate that the improved anti-knock characteristics of the E30 resulted in an approximate doubling of the peak torque throughout the investigated engine speed range with this engine and hardware, as shown in Figure 4.

RESULTS

The results are presented in four subsections. The first section discusses the operating range of this engine under HCCI operation followed by the general trends observed throughout the campaign. In the second, the effects of engine operation variables such as EGR, intake manifold pressure (boost pressure), intake and exhaust valve timing, and speed on engine operation are illustrated. The third discusses fuel-specific effects. Finally, the theoretical turbocharger efficiency requirements based on the intake and exhaust manifold conditions, and its impact on pumping loop work and net thermal efficiency are discussed in the fourth section.

Table 3. Fuel properties

FUEL	E0	IB24	E30
Oxygenates ASTM D5599 [%]	<0.1 any	23.64 iso-butanol	30.65 ethanol
HoV [kJ/kg] ²	352 [29,31]	442.7 [29,31,32]	529.4 [29,31,32]
HoV with gasoline energy eq. [kJ/kg] ³	352 [29,31]	470.3 [29,31,32]	598.9 [29,31,32]
Reid vapor pressure, ASTM D5191 [psi]	13.13	12.29	13.28
RON, ASTM D2699 [-]	90.2	96.6	100.3
MON, ASTM D2700 [-]	83.9	86.8	88.8
LHV, ASTM D240 [MJ/kg]	43.454	40.846	38.105
$\lambda=1$, AFR [-]	14.70	14.13	12.85
C, ASTM D5391 wt. [%]	86.49	80.63	74.4
H, ASTM D5391 wt. [%]	14.06	13.89	13.73
O, ASTM D5599 wt [%]	<0.1	3.71	11.34
Specific gravity, ASTMD4052 [-]	0.7289	0.7426	0.7448

2 Calculated through a linear combination of neat alcohol and neat gasoline HoV.

3 Calculated on a gasoline-equivalent energy basis, i.e., required for matched load.