

Load Expansion of Stoichiometric HCCI Using Spark Assist and Hydraulic Valve Actuation

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ABSTRACT

A spark-assist homogeneous charge compression ignition (SA-HCCI) operating strategy is presented here that allows for stoichiometric combustion from 1000-3000 rpm, and at loads as high as 750 kPa net IMEP. A single cylinder gasoline engine equipped with direct fuel injection and fully variable hydraulic valve actuation (HVA) is used for this experimental study. The HVA system enables negative valve overlap (NVO) valve timing for hot internal EGR. Spark-assist stabilizes combustion over a wide range of engine speeds and loads, and allows for stoichiometric operation at all conditions. Characteristics of both spark-ignited combustion and HCCI are present during the SA-HCCI operating mode, with combustion analysis showing a distinctive spark ignited phase of combustion, followed by a much more rapid HCCI combustion phase. At high load, the maximum cylinder pressure rise rate is controlled by a combination of spark timing and retarding the intake valve closing angle. The latter reduces the effective compression ratio, and therefore the compressive temperatures, allowing the high load limit of the operating range to be expanded. The SA-HCCI operating strategy exhibits improved thermal efficiency at most operating conditions, with fuel consumption improvements up to 9% realized at light engine loads. The SA-HCCI operating strategy presented here does not provide an efficiency advantage at all operating points compared to SI combustion; a decrease was observed at the highest speed and at loads above 500 kPa net IMEP. At light engine loads the majority of the heat release takes place during the HCCI phase of combustion, and as such the NO_x emissions are very low and are similar to levels observed in pure HCCI. At higher loads, a larger portion of the heat release takes place during the spark ignited phase of combustion, which produces NO_x emissions that are much higher than is typically associated with HCCI, but still represent a decrease from conventional SI combustion. By limiting the fuel/air mixture to stoichiometric conditions, the

higher NO_x emissions do not represent an implementation barrier to this strategy because compatibility is maintained with very effective conventional 3-way catalysts.

INTRODUCTION

The advantage offered by advanced combustion strategies, such as homogeneous charge compression ignition (HCCI) for gasoline engines and premixed charge compression ignition (PCCI) for diesel engines, is the ability to simultaneously enable high engine efficiency while producing low engine-out emissions of particulate matter and oxides of nitrogen (NO_x). Reducing engine-out emissions is important to diesel engines as a way to comply with emissions regulations while minimizing the fuel economy penalty associated with regenerating aftertreatment devices. However, modern gasoline engines have very clean tailpipe-out emissions because mature 3-way catalyst aftertreatment systems are highly effective at stoichiometric air-to-fuel ratio conditions.

Thus, the goal of applying advanced combustion techniques on a gasoline engine platform is to improve engine efficiency without increasing tailpipe-out emissions. This can mean either producing sufficiently low engine-out NO_x emissions so that NO_x aftertreatment is not required, in which case the engine can operate fuel-lean, or it means maintaining a stoichiometric air-to-fuel ratio to maintain compatibility with 3-way catalyst aftertreatment systems. In both cases, there will be sufficiently high levels of unburned emissions, carbon monoxide (CO) and unburned hydrocarbons (HC), that some aftertreatment will be required, and the exhaust temperature must be sufficiently high to maintain good catalyst performance.

Improved efficiency has been demonstrated in numerous studies for HCCI combustion, and while the improvements can be substantial, it is important that the efficiency benefit of

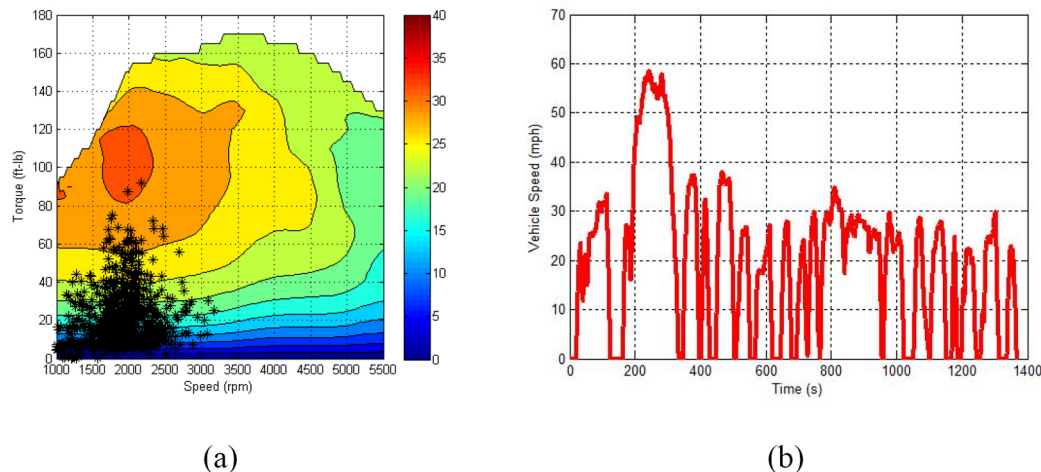


Figure 1. (a) Engine efficiency map of a 2007 Saab Biopower vehicle and (b) vehicle speed trace for an FTP engine drive cycle. Data markers on (a) are operating points from the FTP drive cycle.

advanced combustion techniques translates into a real-world fuel economy benefit. The difficulty of this challenge is illustrated in Figure 1. Figure 1 (a) shows previously unpublished efficiency data from a 2007 Saab Biopower collected at the Oak Ridge National Laboratory (ORNL) chassis laboratory. Figure 1 (b) shows vehicle speed as a function of time for the Federal test protocol (FTP) drive cycle, which is used to calculate city fuel economy.

The goal of the FTP drive cycle is to represent city driving, and is therefore a highly transient test cycle with a large number of accelerations, decelerations, and some time spent at idle conditions. The black data markers overlain on the efficiency contour plot in Figure 1 (a) are operating points collected during the FTP drive cycle. As can be seen, the engine duty cycle is concentrated mainly to speeds below 3000 rpm, and at part-load conditions that are substantially below the peak engine efficiency. Thus, in order for advanced combustion strategies to increase real-world fuel economy, they must provide an efficiency benefit in this portion of the operating map.

The operating range of HCCI combustion is limited to a relatively small portion of the operating map. If the engine load becomes too low, the engine is prone to combustion instability and misfire, and if the engine load becomes too high, the engine is prone to a high rate of in-cylinder pressure rise, similar to knock, resulting in unacceptably loud engine operation and damage to the engine.

In order to expand the HCCI engine operating envelope, several researchers have investigated spark assisted operation for example Koopmans et al, [1] where a spark event was used during the transition into HCCI. Also the study of Persson et al, [2] where spark assisted HCCI was applied at low operating loads to increase the engine operating envelope of gasoline fueled operation. Similarly the use of spark

assisted HCCI was applied at stoichiometric conditions at lower engine speeds in the study of Osborne et al, [3].

The use of spark ignition prior to auto-ignition at increased engine operating loads has also been investigated for example, Urushihara et al, [4] and Yoshizawa et al, [5] where a small quantity of fuel was formed near the spark plug by direct injection which then formed a propagating flame after the spark event. A lean bulk homogeneous charge mixture contained within the cylinder (formed by port injection) was then ignited by this flame propagation event. This approach allowed an increase in engine operating load to be achieved. More recent work has also re-examined spark-assist in HCCI combustion and confirmed that this allows for higher load operation than HCCI alone. For example, in a modeling study by Lavoie et al. [6], it was found that spark assisted HCCI combustion could be used to extend the upper range limit of HCCI combustion by providing additional heat when the charge temperature was insufficient for HCCI auto-ignition.

During optical investigations into the characteristics of spark assisted HCCI, Persson et al [7], it was confirmed that spark ignited flame propagation is present albeit with a reduced flame speed when compared with conventional SI combustion. Auto-ignition is then triggered by the preceding SI combustion event. In a separate study involving optical investigation of spark assisted HCCI, Zigler et al [8], it was indicated that the intake conditions could be important when examining the nature of transition from SI combustion to bulk HCCI and that a richer, higher temperature intake charge tended to magnify the triggering nature of spark combustion. Investigations into the effects of spark assistance along with the timing of direct injection in an HCCI engine, Kalian et al, [9], confirmed that an increase in IMEP could be achieved with the use of spark assist but that the resulting increase in combustion temperature leads to increases in emissions of

oxides of nitrogen, concurring similar findings from other researchers in this field.

In the study of Yun et al. [10], it was found that the upper operating limit of stoichiometric HCCI could be extended with spark-assist as a direct consequence of reduced combustion noise at high load. This could be attributed to the slow initial heat release that resulted during spark ignited flame propagation followed by auto-ignition. The overall burn duration was therefore extended compared to HCCI combustion without spark assist, which led to a reduction in combustion noise. A study performed by Bunting [11], used spark assist during near-stoichiometric HCCI operation in order to increase combustion stability and also suggested the possibility of its use in expanding the engine operating range to higher loads.

The purpose of this paper is to explore the ability of spark-assist to expand the operating load in conjunction with control of intake valve closing under stoichiometric engine conditions. The combustion technique uses a spark-assist to help to initiate and control combustion phasing, and uses control of the intake valve closing angle to adjust the effective compression ratio of the engine. As will be detailed on the following pages, these techniques allow the engine load to be increased while maintaining a moderate rate of in-cylinder pressure rise. The study is also focused solely on stoichiometric conditions as a way to maintain compatibility with conventional 3-way catalyst technology. As will be presented, this combustion strategy does produce substantial levels of NO_x emissions at some operating conditions, and while the levels are lower than for conventional SI combustion, exhaust NO_x aftertreatment will still be required.

EXPERIMENTAL METHODOLOGY

RESEARCH ENGINE

The details of this experiment have been reported previously in reference [12], and for brevity, a condensed description of the experiment is given here. A highly modified GM 2.0L Ecotec engine with direct fuel injection is used for the study with the engine geometry listed in Table 1. Three of the cylinders of the production engine have been disabled to allow single-cylinder operation, a custom piston has been installed to increase compression ratio to 11.85, and the engine is operated naturally aspirated for this study.

Table 1. Engine Geometry

Bore	86 mm
Stroke	86 mm
Connecting Rod	145.5 mm
Fueling	Direct Injection
Compression Ratio	11.85
Valves per Cylinder	4

The engine is equipped with a hydraulic valve actuation (HVA) system to enable fully variable valve actuation. Machining modifications have been made to the cylinder head to accommodate the small research module HVA system from Sturman Industries. Modifications include using custom intake and exhaust valves with longer valve stems. While the valve material is different than the production valves, the combustion chamber geometry is unchanged. Other changes to the engine include a custom exhaust system and a Kistler sparkplug with an integrated piezoelectric pressure transducer, model number 6117BCD25. In the production configuration from the original equipment manufacturer (OEM), the high pressure fuel pump for the gasoline direct injection (GDI) fueling system is driven by the intake cam shaft. However, because the cam shafts are removed, the fuel is supplied by an externally powered fuel pump. A picture of the engine is shown in Figure 2.

A flexible open engine controller from Drivven Inc. is used for engine management. The engine controller uses electronic components from National Instruments as well as Drivven Inc. The engine controller program uses the LabVIEW software programming environment, including the RealTime and the Field Programmable Gate Array (FPGA) add-in modules. The controller is configured to control spark, fuel, and throttle. An added software package in the engine controller, Drivven Combustion Analysis Toolkit (DCAT), performs the crank-angle resolved data acquisition and combustion analysis. These measurements include cylinder pressure, valve lift feedback from each of the four valves and current sent to the fuel injector.

FUEL

The fuel used for this study is UTG-96, an emissions certification gasoline from Chevron-Phillips Chemical Company. The gasoline specifications are listed in Table 2.

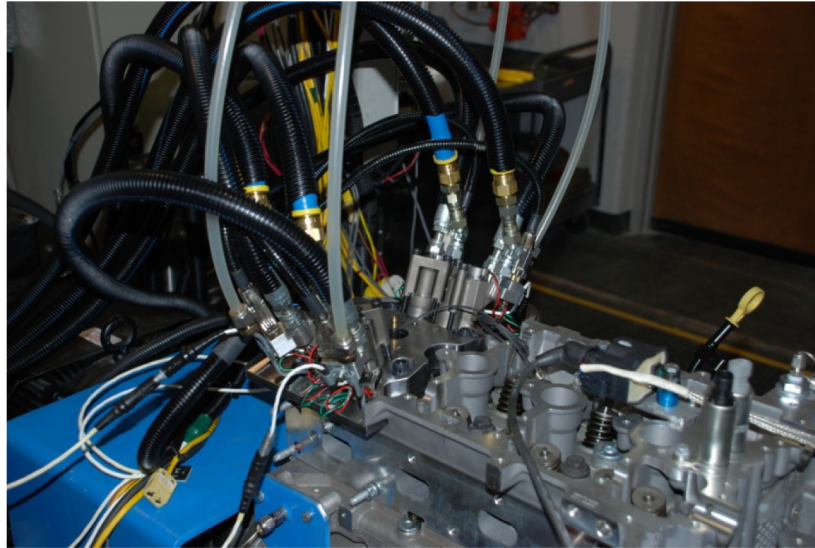


Figure 2. Modified cylinder head with hydraulically actuated valves.

Table 2. Fuel properties of UTG-96 from Chevron-Phillips Chemical Company

Fuel Property	Test Method	UTG-96
Research Octane Number (RON)	ASTM D2699	96.1
Motor Octane Number (MON)	ASTM D2700	87
Antiknock Index (R+M)/2	N/A	91.6
Wt. % C	ASTM D240	86.5
Wt. % H	ASTM D240	13.5
Stoichiometric Air-Fuel Ratio	N/A	14.55
Specific Gravity	ASTM D 4052	0.7400
Lower Heating Value (MJ/kg)	ASTM D240	42.8
Reid Vapor Pressure (psia)	ASTM D5191	9

Table 3. A comparison of the baseline SI combustion and the SA-HCCI engine operating strategies.

	SI Combustion	SA-HCCI
Fuel Rail Pressure (bar)	95	95
Fuel Injection Timing (CA)*	-280	-340
Equivalence Ratio	1.0	1.0
Intake Valve Opening (CA)*	-344	-313 to -234
Intake Valve Closing (CA)*	-180	-180 to -124
Intake Valve Lift (mm)	9	3 to 6
Exhaust Valve Opening (CA)*	180	170
Exhaust Valve Closing (CA)*	349	234 to 313
Exhaust Valve Lift (mm)	9	2 to 3.5
*0 CA refers to combustion TDC		

ENGINE OPERATING STRATEGY

This study compares two engine operating modes: a baseline spark-ignited (SI) combustion mode and the SA-HCCI combustion mode. For both operating modes the engine is operated from 1000 to 3000 rpm in increments of 500 rpm. Engine load sweeps are performed at each speed. For the baseline SI combustion operating mode, the valve events are the same for all speed/load conditions, and engine load is controlled by throttling the intake. For the SA-HCCI combustion mode, the engine is operated unthrottled and a negative valve overlap (NVO) valve timing strategy is used to meter the required air into the engine and to provide in-cylinder exhaust gas recirculation (EGR). A summary of the fuel injection and valve strategies is given in [Table 3](#).

RESULTS AND DISCUSSION

OPERATING RANGE

[Figure 3 \(a\)](#) shows the speed-load operating range of the SA-HCCI combustion strategy compared with the operating range for conventional SI combustion. The lowest load for this combustion strategy is limited either by misfire or unstable combustion (quantified as COV of NMEP > 3%), with misfire being the more frequent cause. The high load for this combustion strategy is limited by the maximum rate of in-cylinder pressure rise, which was not permitted to exceed 700 kPa/CA. As can be seen, the operating range of the SA-HCCI combustion strategy is significantly smaller than for conventional SI combustion, but does include much of the mid-load range of the operating map.

Unlike HCCI combustion, the SA-HCCI operating strategy is very dependent on the spark. With the constraint of stoichiometric operation imposed here, there are only three

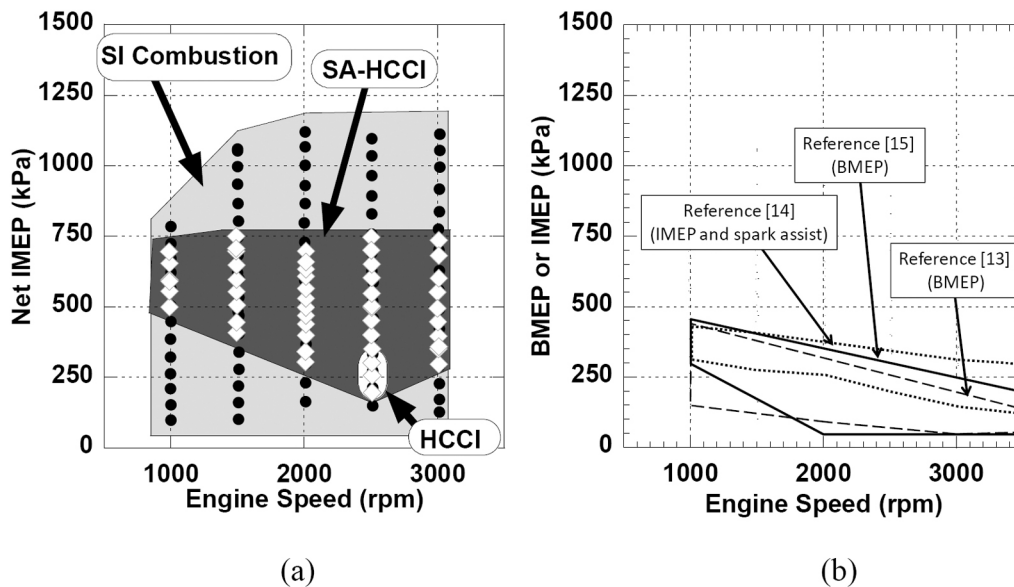


Figure 3. (a) Comparison between the operating ranges of SA-HCCI compared to conventional SI combustion, and (b) operating range of NVO HCCI combustion found in the published literature.

points where the combustion event is not dependent on the spark assist (i.e. HCCI). These points are the three lowest load points at 2500 rpm. Operating the engine under varying lean conditions, as is frequently done with NVO HCCI studies, does enable a broader operating range without spark, but the lean operating range is not considered here.

For comparative purposes, Figure 3 (b) displays the operable HCCI speed-load range from references [13], [14], and [15]. These studies all operate under lean conditions, utilize NVO for internal residuals, and operate with compression ratios from 10.3:1 to 10.5:1 compared to the 11.85:1 used in this study. Persson et al. [14] is differentiated from Zhao et al. [13] and Allen et al. [15] because the results are exhibited as IMEP rather than BMEP, and it includes spark assist at some operating conditions. These studies all exhibit similar low-load operating ranges, and can be considered as representative of the NVO HCCI operation in naturally aspirated engines with gasoline. The portion of the map operable with SA-HCCI in Figure 3 (a), investigated in this study, is considerably larger. SA-HCCI is applicable to higher load conditions, and experiences misfiring in the low speed/load range that HCCI typically operates well in. Thus, the SA-HCCI strategy presented here compliments the commonly reported operating range found in other HCCI studies. It is also worth reiterating that there is a difference in the basis of comparison in Figure 3: Figure 3 (a) shows net IMEP whereas (b) shows both net IMEP and BMEP.

COMBUSTION ANALYSIS

The larger operating range achieved in this study can be understood by analyzing the combustion process. Figure 4 (a) compares the cylinder pressure and valve events at a nominal

load of 2.0 bar net IMEP at 2500 rpm, while Figure 4 (b) compares the heat release rates. For the advanced combustion case, this is one of the few operating conditions that does not require a spark-assist for ignition, and is therefore labeled as HCCI. Under SI conditions at this load the intake manifold pressure is throttled to a significant vacuum, 30 kPa absolute. In contrast, the HCCI condition is unthrottled and uses negative valve overlap (NVO) to meter in only the desired quantity of fresh air for the start of compression. As a result of the different pressures at the start of compression, the peak pressure for HCCI is nearly double the peak pressure for SI combustion.

The heat release rates for the two modes of combustion are shown in Figure 4 (b). As is readily apparent, the HCCI combustion event is of a much shorter duration than the SI combustion event. Combustion analysis shows that the 10-90% combustion duration for the HCCI condition, which is defined as the time between the 10% mass fraction burned and the 90% mass fraction burned, is approximately 6 CA degrees, whereas the 10-90% combustion duration for the SI combustion case is approximately 24 CA. As a result of the longer combustion duration, the peak rate of heat release rate is much lower for SI combustion.

Figure 5 and Figure 6 show similar comparisons between the two modes of combustion at progressively higher engine loads, nominally 4.5 and 6.5 bar net IMEP respectively. At higher engine loads less throttling is required. The start of compression pressure for SI combustion is higher than in Figure 4: 60 kPa for Figure 5 and 70 kPa for Figure 6. As a result the peak pressure differences between the two combustion modes is less pronounced than in Figure 4, but