Demonstration of Better than Diesel Efficiency and Soot Emissions using Gasoline Compression Ignition in a Light Duty Engine with a Fuel Pressure Limitation

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Abstract

Increasing regulatory demand to reduce CO_2 emissions has led to a focus on advanced combustion strategy development to improve overall engine efficiency. Gasoline compression ignition (GCI) has been demonstrated by others to have the potential to meet future CO_2 regulations and emissions while achieving comparable to better efficiency than conventional diesel compression ignition (DCI). Soot and NO_x emissions are also reduced significantly by using gasoline instead of diesel in compression ignition engines due to differences in composition, fuel properties, and reactivity. In comparison with diesel fuel, gasoline has a higher volatility and more resistance to autoignition, therefore, its longer ignition delay time will allow for better mixing of the air-fuel charge before combustion. In this study, a GCI combustion system has been tested in a Hyundai 2.2L engine as part of a US Department of Energy funded project. A double-injection strategy was tested from mid-to-high loads (5-20 bar BMEP) and for engine speeds in the range of 1200-3000 rpm. Up to 43.4% brake thermal efficiency was achieved using the GCI mode versus 41% using DCI mode. The GCI mode has demonstrated two distinct strategies that work at different load ranges, partially premixed compression ignition (PPCI) and mixing-controlled compression ignition (MCCI). Overall, this study shows that for similar engine-out NO_x levels, GCI mode had higher brake thermal efficiency than DCI with lower fuel pressure and EGR required.

Introduction

Regulatory demand around the world for a cleaner environment has led automakers and research institutions to focus on developing new concepts to reduce engine-out emissions. This has the potential to extend the existence of the combustion engine as a reliable, cost-effective, and low-emitting machine for the foreseeable future. Some of the advanced engine combustion concepts under active development are homogeneous charge compression ignition (HCCI), and partially premixed compression ignition (PPCI). These types of combustion modes combine lean operation with copious levels of exhaust gas recirculation (EGR) and intake boost with high compression ratios (CR>13.5) to enable improved brake thermal efficiency (BTE), NO_x, and soot emissions.

In HCCI combustion mode, fuel is injected very early in the compression stroke, in which the air-fuel charge undergoes a homogeneous mixing prior to autoignition. This combustion mode can guarantee a significant reduction in NO_x and lower soot emissions. However, the combustion is controlled by the chemical kinetics of the mixture. Therefore, controlling the HCCI combustion phasing proves to be difficult [1]. On the other hand, in PPCI combustion mode, the fuel is injected later during the compression stroke with multiple injections to avoid fully premixing, this gives the fuel sufficient time to partially premix with air, known as “premixed enough” [2-3]. For the PPCI mode, the in-cylinder fuel stratification determines the reactivity. Subsequent autoignition is controlled in-part by the timing of the injection events, i.e., large injection at TDC [3]. Therefore, PPCI can still achieve some of the benefits of HCCI, like reduced NO_x and smoke but with reductions in pressure rise rates and improved combustion phasing control.

When gasoline fuel is used in a PPCI combustion mode, the combustion mode can also be referred to as a gasoline compression ignition (GCI) combustion mode. Gasoline fuels have different chemical compositions. Therefore, they differ in reactivity and their behaviors under different temperature and pressure conditions. As a result, the ignition delay time before the autoignition varies [4]. With compression ignition (CI) engines, this is defined as time between end of injection (EOI) and start of combustion (SOC). The fuel is considered gasoline when its cetane number (CN) < 30 or has a research octane number (RON) of > 60 and is considered diesel fuel when its CN > 30 (typical diesel fuel has CN of 40-60) [2]. Gasoline fuel has a higher volatility and is more resistant to autoignition when compared to diesel fuel. Therefore, its longer ignition delay time will allow for better mixing of the fuel and air before combustion compared to diesel [2], as a result, soot can be reduced. Generally, the heat release with GCI combustion starts after the end of injection, which prevents injecting fuel during the main combustion heat release, i.e., diffusion combustion [2]. NO_x production is lower when combustion temperature is lower which can be done by adding EGR to the mixture, also leaner mixtures result in lower combustion temperatures. This can be achieved by delaying the combustion after the injection event to give more time for the air and fuel to mix together or by acceleration the mixing event.

In the literature, researchers showed that a GCI engine can run successfully on gasoline fuel at mid-to-high loads and achieve low NO_x and soot emissions. Kalghatgi et al. [2] performed their test at 1200 rpm using single-injection and found that injecting gasoline after -30 degATDC will guarantee a stable PPCI combustion, otherwise, the air-fuel mixture will be over-mixed, i.e., too mixed as opposed to mixed enough, and the combustion will transition to HCCI. Their results showed that, at the same load, lower boost pressure and EGR level can be used with gasoline compared to diesel while lowering NO_x

and smoke. Later on, Kalghatgi et al. [3] applied double-injection strategy to shape and reduce the heat release rate (HRR) and its peak, respectively. Double-injection strategy got also higher loads at the same conditions.

Manente et al. [5] performed a pilot sweep in a double-injection GCI strategy with 0% EGR and found that by retarding the pilot's start of injection (SOI_Pilot), the stratification increases, causing the pressure rise rate (PRR) and indicated specific fuel consumption (ISFC) to decrease. Also, NO_x production decreases due to decrease in combustion temperature. On the other hand, soot increases due to forming less homogeneous air-fuel mixture. In their work, they achieved a peak indicated thermal efficiency (ITE) of 47.8% in a 2.0L single cylinder.

Delphi [6] conducted their experiment in a single cylinder engine using a gasoline injection system in a GCI mode. The GCI mode produced less NO_x when compared to using diesel injector in a GCI combustion mode. The authors attributed the variance to the fuel spray characteristics injecting gasoline at low pressure in a diesel injector. They later employed an exhaust rebreathing strategy in a four-cylinder engine to attain low load operation [7,8], and achieved a peak BTE of 40% and 42%, respectively.

Cung et al. [9] investigated the influence of injection timing and pressure, boost pressure, and lambda on PPCI combustion. They showed that boost and lambda have the most effect on combustion and emissions. Cung et al. performed their test at 1000 rpm using single-injection and found that the combustion mode is HCCI-like when the SOI is between -140 to -60 degATDC and then switches to GCI after -30 degATDC, passing through a transition region between -60 to -30 degATDC. Similar results were also confirmed by Kalghatgi et al. [2] for the mode transition between HCCI and PPCI.

Cung and Ciatti [10] investigated optimizing GCI triple-injection strategy at 2000 rpm and load of 8-14 bar BMEP in a diesel engine and achieved best BTE of 38.3% at 8 bar BMEP. They indicated that mixing-controlled compression ignition combustion (MCCI) started at 8 bar BMEP, but it is more obvious at higher loads especially with higher amount of stratifications due to late injection. As a result, FSN increased by load increase. The authors suggested using higher injection pressure or boost pressure to tackle this issue. They identified the challenges to run GCI mode at high loads as high combustion noise resulted from the early premixed combustion.

Recently, Zhang et al. [11,12] used a single cylinder optical engine to show the effect of injection strategies on efficiency, their effect while transitioning from HCCI to PCCI combustion mode. The images showed that the mixture transitions from homogeneous to stratified occurs by increasing the number of injections. Also, closer injection to TDC will create diffusion combustion mode or MCCI. While the pilot injection shifts the combustion phasing in a double-injection strategy, an additional poet injection can improve the indicated efficiency by optimizing the balance between mixing and local temperature.

In summary, GCI combustion performance and efficiency is affected by many variables, such as, number of injections, injection timing, quantity split, fuel pressure, boost pressure, and EGR. Previously, their effect on GCI was mainly investigated in a PPCI combustion mode. In this work the authors went further to define the MCCI combustion mode as a better strategy for high loads. To the best knowledge of the authors, current engine achieved better BTE compared to previous presented small displacement 4-Cyl engines and has the capability to be a production intent rather than a research engine. The objective of this work is to demonstrate the benefits of GCI mode versus DCI mode with respect to efficiency and emissions in a light duty multi-cylinder engine without the use of EGR. A double-injection strategy was used throughout the load sweep from 5-20 bar BMEP at engine speeds in the range of 1200-3000 rpm. A detailed cylinder pressure analysis is performed at representative points of interest to show the differences in both injection strategies and heat release of GCI versus DCI. Operational maps of the GCI mode strategy are presented to show the performance and emissions across the speed and load range and giving insight into potential challenges and areas for improvement. This project is funded in part by the US Department of Energy under the collaborative project entitled Co-optimized multi-mode SI-GCI engine (DE-EE0008478).

Experimental Setup and Methods

Engine Setup

A Hyundai four-cylinder 2.2L turbocharged, compression ignition engine was used for this study. The engine has a high compression ratio (CR) of 16:1 and is equipped with conventional valvetrain system for both intake and exhaust camshafts. The base direct injection (DI) injectors are centrally mounted in the flat cylinder head with bowl in piston design combustion chamber was used for all experimental studies.

The base turbocharger for the engine was already sized appropriately for high dilution conditions. Compressed intake air was cooled using a liquid-to-air heat exchanger. The engine also has a high pressure loop EGR cooler and cold side EGR valve. The base diesel fuel pump was limited to a maximum of 1000 bar for use with gasoline to provide for a realistic product development target. Although the base fuel pump on current engine is designed for diesel fuel, gasoline fuel was used in this test without any lubricity additive since total test time was limited to less than 40 hours. As a result, the durability of the fuel pump was not affected. The test engine specifications are shown in Table 1. The engine schematic layout with main instrumentation is shown in Figure 1.

Table 1. Engine specifications

Engine	4-Cylinder CI Engine
Displacement [L]	2.199
Bore [mm]	85.4
Compression Ratio [-]	16:1
Camshaft arrangement	DOHC
Valve System	16 valves HLA, chain drive
Piston	Bowl
Turbocharger	E-VGT
EGR	high pressure cooled EGR loop
Fuel sys. Max Pressure [bar]	2000

Engine Controller

A rapid-prototyping PiInnovo M670 engine control unit (ECU) was utilized as the development platform. The ECU is capable to communicate with all the various engine actuators and sensors consisting of pedal, DI injectors, EGR valve, and variable geometry turbo (VGT).

The ECU was interfaced using ETAS INCA software to manipulate the calibration parameters. The M670 is highly configurable and based on the Simulink platform for control logic development, thus allowing for complete customization of an engine control strategy.

The engine control strategy relies on an engine speed set point and pedal request, both of which are controlled using the test cell console. These inputs initiate the calculation of mass fuel set-point, EGR set-point, lambda set-point, and boost pressure set-point. The output of these set-point calculations is in the form of injector pulse-width and timing commands, EGR, and VGT duty cycles. The controller can operate the engine actuators in both manual and open loop modes via calibration tables.

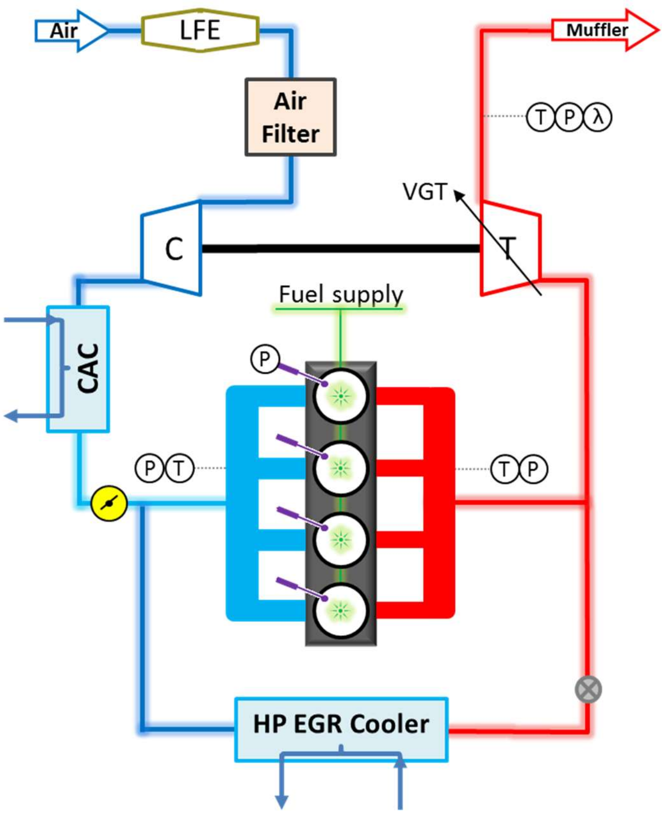

Figure 1. Engine schematic and instrumentation layout

Experimental Instrumentation

The engine was tested using a 400 kW AC dynamometer with AVL PUMA 2.0 for dynamometer controls and data acquisition. The engine was instrumented for temperature and pressure with sampling of 10 Hz. A HBM T40 torque flange was used for dyno torque measurement. Fuel mass flowrate was measured using the AVL 7351 CST Coriolis meter. Fuel temperature was controlled with an AVL 753CH fuel temperature control unit. Air flow was measured with a Meriam laminar flow element (LFE). Both LFE and fuel flow meter measurements were used to calculate the air-fuel ratio (AFR). Engine emissions were measured using a Horiba MEXA 7500HEGR emissions bench. Total hydrocarbons (THC), carbon monoxide (CO), carbon dioxide (CO_2), oxygen (O_2), and nitrogen oxides (NO_x) were measured before and after the DOC/DPF. Additionally, carbon dioxide (CO_2) is measured in the intake manifold to enable EGR volume calculation. Soot concentration and filter smoke number (FSN) is measured using an AVL 415S smoke meter. Engine coolant and oil temperature were controlled to 85°C during testing. High speed cylinder pressure was measured with Kistler 6056A sensors paired with an AVL X-Ion analog to digital converter in conjunction with AVL 4P4G MICRO IFEM charge amplifiers. AVL Indi-Com was used for high speed data acquisition, recording 200 consecutive cycles from -90 CAD to 90 CAD TDC firing, utilizing 0.2 CAD resolution. Outside the firing window, 1 CAD resolution was used. A 720 tooth crank angle encoder was installed, which enables for 0.5 CAD resolution. The -90 CAD to 90 CAD TDC firing window derives CAD due to the higher resolution requirement than which is provided by the crank angle encoder (AVL 365C01).

Tested Fuel

Conventional E10 gasoline fuel with an anti-knock index (AKI) of 87 was used in the GCI tests. This is a market-representative research octane number and has approximately 10% ethanol content and initial boiling point (IBP) of 26.6°C.

The diesel fuel used is a #2 ultra-low-sulfur diesel (ULSD) which is less volatile than gasoline with an IBP of 172.1°C. Since the test cell was not equipped with a diesel fuel tank, diesel was delivered using a separate fuel cart which also outputs the fuel mass flow via the Coriolis meter in the cell. Key fuel properties of both fuels are reported below in Table 2.

Table 2. Tested fuel properties

Fuel	87 AKI E10	#2 ULSD
IBP [°C]	26.6	172.1
T10 [°C]	37.7	206.9
T50 [°C]	65.3	257.9
T90 [°C]	158.6	322.4
FBP [°C]	210.8	352.0
Specific Gravity at 15.56 °C	0.7224	0.8518
RON	91.5	-
AKI	87	-
Lower Heating Value [MJ/kg]	42.041	42.424
Gross Heating Value [MJ/kg]	45.077	45.200
Carbon (Wt%) [m/m]	81.91	86.92
Hydrogen (Wt%) [m/m]	14.31	13.08
Oxygen (Wt%) [m/m]	3.78	0.16
Ethanol (Wt%) [m/m]	10.89	0
Density (at 15.56 °C) [g/ml]	0.7217	0.8509
Stoichiometric AFR [air mass/fuel mass]	14.70	14.48

Test Conditions

The engine tests were conducted at engine speeds within the range of 1200-3000 rpm using a double-injection strategy and fuel pressure limit of 1000 bar. Fuel pressure was set for each speed and load point and held constant while SOI for both injections were adjusted manually so that the individual cylinders' combustion phasing was similar. Number of injections used for both GCI and DCI were exclusive to double-injection strategy only to present a simplified scientific study for a production application rather than presenting calibration results. The profile of each injection and their corresponding SOI range are depicted in Figure 2. SOI and fuel split percentage on mass basis (Qsplit) for the pilot injection (PI) and main injection (MI) were varied during the test. Mainly, PI was adjusted to keep the pressure rise rate (PRR) < 8 bar/deg and MI was adjusted to keep the CA50 around 10 degATDC. Boost pressure was initially set according to the load by controlling the VGT actuator on the turbo. The loads for DCI mode tests were swept from 5 to 20 bar BMEP for a constant NO_x emissions target of < 5 g/kWh. The EGR valve position was varied to adjust the level of EGR at each speed and load point to attain desired engine-out NO_x targets. However, it was quite difficult to control the EGR to a precise level using current hardware and calibration, therefore, there are some areas with inconsistencies. Smoke was targeted to a filter smoke number (FSN) < 1.

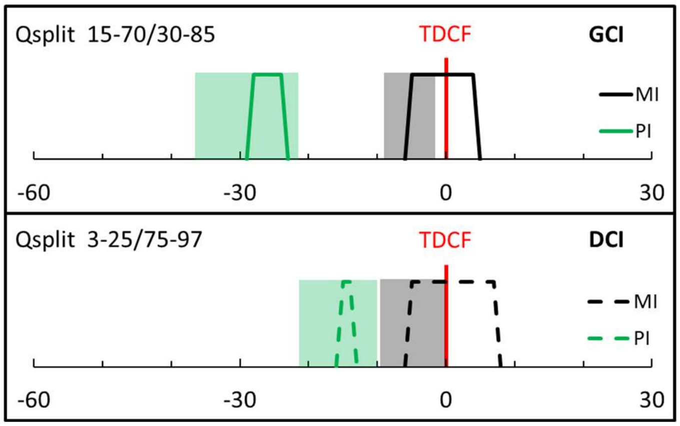

Figure 2. Double-injection strategy schematic for GCI and DCI. Range of SOI variations for pilot (PI) and main (MI) injections are depicted with an example of one injection profile scenario.

For GCI mode tests, constant NO_x load sweeps from 5 to 20 bar BMEP were varied at engine speeds up to 3000 rpm. Due to the higher volatility of gasoline, lower fuel rail pressures were used for GCI mode compared to DCI mode without sacrificing the engine-out smoke due to better mixing. The higher anti-knock properties of gasoline allowed earlier pilot injection for stronger premixed combustion as shown in Figure 2. To maintain the engine-out NO_x target, no EGR was used for GCI testing. The experimental test conditions compared to DCI are summarized in Table 3.

Table 3. Experimental test conditions

Combustion Mode	**GCI**	**DCI**
Fuel	87 AKI E10	#2 ULSD
Engine Speed [rpm]	1200-3000	
BMEP [bar]	5-20	
EGR [%]	0	Swept for a const. NO_x target
Injection Pressure [bar]	Up to 775	Up to 1000
Injection Strategy	Double-injection: PI + MI	

Results and Discussion

Load Sweep Tests

Engine testing was conducted at specific speed and load points and parametric studies were conducted at each point. For each data point, the appropriate strategy was applied by varying parameters, such as, injections timing and quantity, AFR, and EGR (only for DCI) to get the best steady state BTE optimized for NO_x < 5 g/kWh. The constant NO_x load sweep is a compilation of all the best optimized runs which are summarized in the load sweep plots as shown in Figures 3 and 4. Only data at speeds 1500 and 2000 rpm are presented in this paper for brevity.

Figure 3 presents the results of the 1500 rpm constant NO_x load sweep for both GCI and DCI modes. In general, GCI achieved better fuel efficiency compared to DCI except at the lowest load (5 bar BMEP). The improved fuel efficiency is due to higher AFR used with the GCI at these conditions allowed by the better fuel mixing of gasoline.

Comparing GCI to the DCI mode at the same speed and load points showed that GCI mode could be operated with a fuel pressure of around 400 bar across the load sweep, which is 60% lower than with the DCI at higher loads while maintaining NO_x within the target. Despite operating with lower fuel pressure, GCI mode has lower engine-out smoke number (FSN) than the DCI points for the same NO_x level, same load, and similar efficiency. To operate DCI with NO_x target of 5 g/kWh, EGR was added to increase charge dilution which resulted in lower combustion temperature and reduced NO_x production. For DCI mode at 15 bar BMEP, more EGR was needed to reduce NO_x further, but due to control issue, it was not possible to increase any EGR further. For the same reason, EGR couldn't be added at all at 20 bar BMEP.

The NO_x target was observed for GCI mode without adding EGR for all loads. The longer ignition delay of gasoline allows for enough time for the air-fuel mixture to reach globally lean condition which lowers combustion temperature for less NO_x production. Also slight decrease in NO_x level can be noticed by decreasing the load from 15 to 5 bar due to reduced combustion temperature.

As shown in Figure 3, smoke number (FSN) for GCI is much lower compared to DCI due to the gasoline fuel properties which results in better mixing of gasoline fuel. This allowed for much lower fuel pressure with GCI. The smoke number at 12 bar BMEP is not available for DCI. Finally, slight higher PRR can be noticed when running GCI compared to DCI.

Constant NO_x load sweep test results of GCI mode versus DCI at 2000 rpm are presented in Figure 4. For similar NO_x target, GCI combustion attained similar or better fuel consumption compared to DCI across the load range. At 12 bar BMEP, GCI achieved best brake specific fuel consumption (BSFC) of 199 g/kWh (BTE ~43%), which was 5.7% improvement over DCI. Similar NO_x was achieved without the use of EGR. Adding EGR at higher load was required for DCI to help maintaining engine out NO_x emissions below the target. Additionally, it was observed that, as engine speed increased above 2500 rpm, EGR was not needed for DCI mode to meet the NO_x target, due to the shorter residence time for nitrogen reactions which reduced the NO_x production.

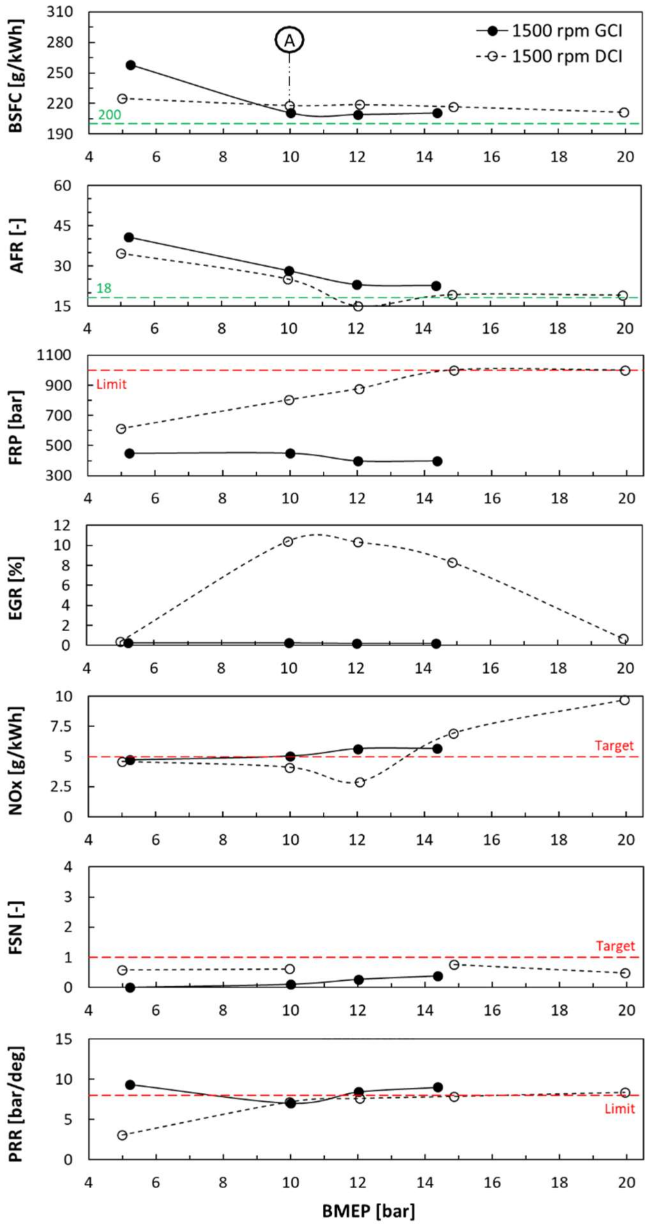

Figure 3. Constant NO_x load sweep for GCI combustion mode versus DCI combustion mode at 1500 rpm

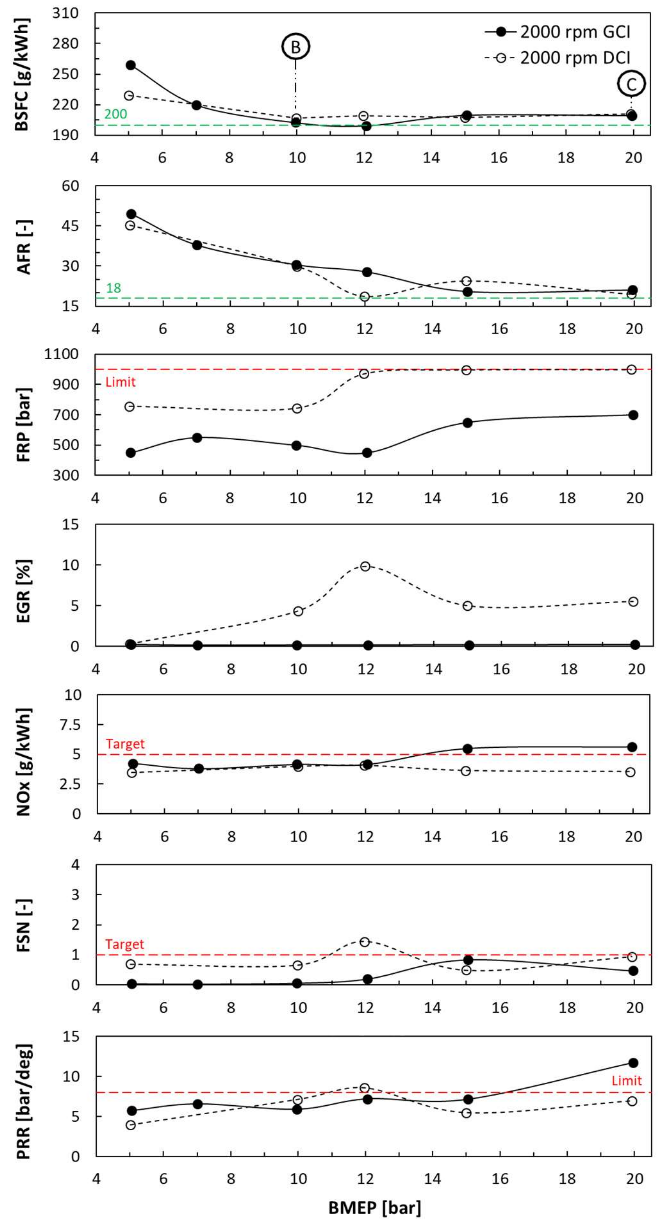

Figure 4. Constant NO_x load sweep for GCI combustion mode versus DCI combustion mode at 2000 rpm

The fuel pressure for GCI was approximately 300 bar lower comparing to the fuel pressure used for DCI, nevertheless, the smoke number was below 1 FSN except at loads of 15 and 20 bar BMEP. GCI combustion has an improved NO_x-Soot trade-off behavior compared to DCI and is able to attain similar NO_x and better soot level with lower BSFC. Running at lower fuel pressure results in reduced parasitic losses of the fuel pump shaft leading to improved BSFC.

One of the challenges to run the GCI mode is the higher PRR as load is increased. This is shown in Figure 4 at 2000 rpm/ 20 bar BMEP. This can be mitigated by increasing number of injections for instance, which helps to shape the heat release rate and enable fuel to find available oxygen in-cylinder. All tests were within the combustion stability limit of covIMEP < 3%.

Summary – Optimized Points Comparisons

Two optimal points with regard to fuel efficiency and emissions are each selected from Figures 3 and 4 and compared to the best fuel efficiency point for the set of tests, namely, points 1500 rpm/ 10 bar BMEP, 2000 rpm/ 10 bar BMEP, and 2500 rpm/ 15 bar BMEP, as shown in Figure 5. GCI mode demonstrated improved fuel efficiency at all speeds. Best GCI fuel consumption is 6% lower than DCI at 2500 rpm/ 15 bar BMEP. Due to the high volatility of gasoline, the required fuel pressures during testing were much lower when using GCI over the DCI with differences as much as 44% at 1500 rpm and 10 BMEP. The improved mixing of gasoline and oxygenation due to ethanol content caused close to zero levels of soot. Ethanol has lower hydrogen to carbon ratio and lower boiling temperature than gasoline and these two properties have positive effects on diffusion burn [13]. Similar NO_x levels to DCI were achieved using gasoline fuel but with no EGR requirement. However, engine-out CO and HC emission were noticeably higher. The CO emission production is a result of a globally lean homogeneous mixture resulting in lower combustion temperature [2-3,5] and the HC emissions production is due to the piston and wall wetting [5].

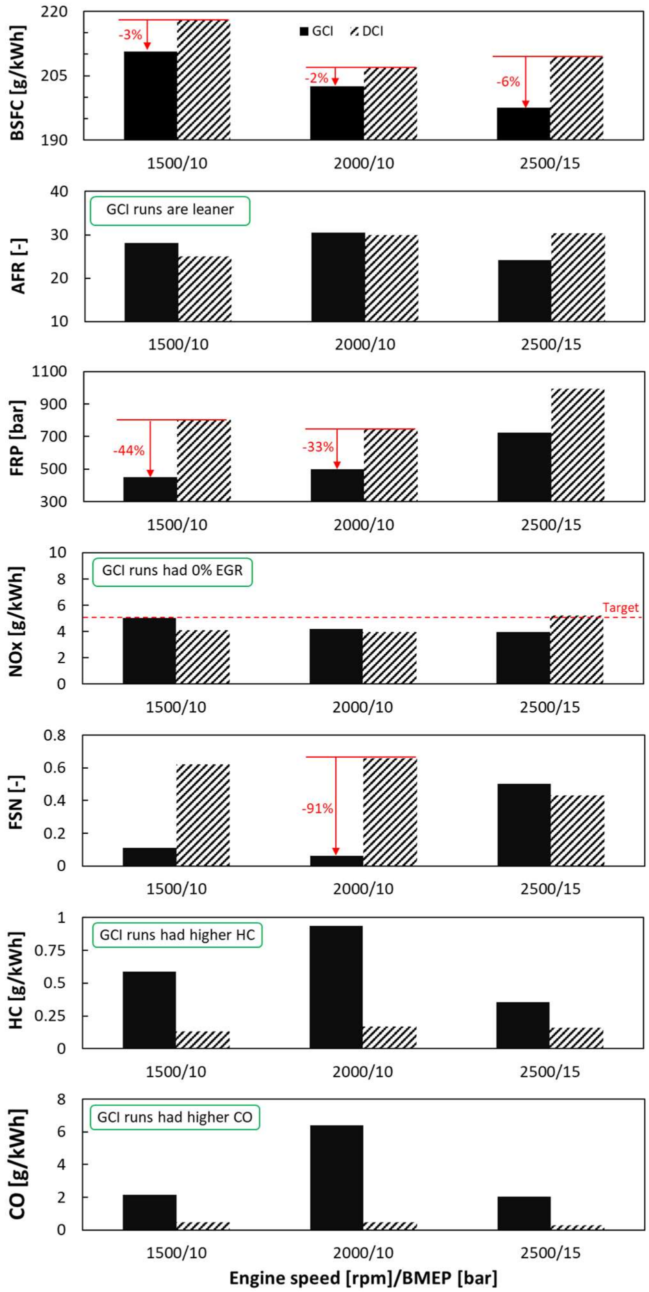

Figure 5. Best points comparisons at each tested engine speed

The experimental results were then compared to a benchmark 2.0 L TGDI SI engine. The GCI mode was able to show a 17.6% improvement over the baseline SI engine at 1500 rpm/ 10 bar BMEP, 18.5% improvement at 2000 rpm/ 10 bar BMEP, and 19.8% improvement at 2500 rpm/ 15 bar BMEP as shown in Figure 6. In addition, the GCI mode showed a maximum of 6.5% thermal efficiency improvement at 2500 rpm/ 15 bar BMEP, over the DCI mode Therefore, GCI mode is better than diesel for a similar NO_x and soot level for the same engine hardware

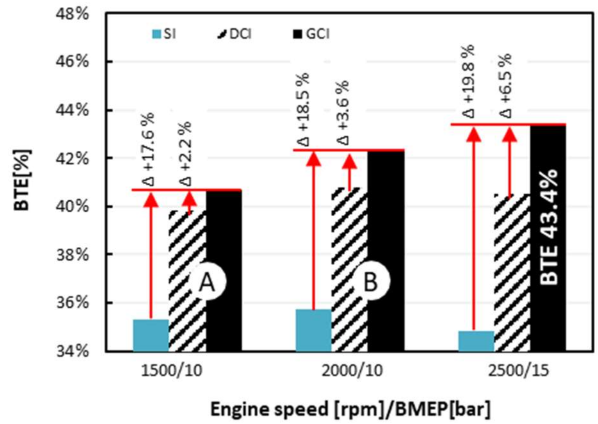

Figure 6. Brake thermal efficiency percent improvements for GCI over SI and DCI combustion modes

Pressure and Heat Release Analysis

To provide further insight into the GCI combustion modes, a high speed in-cylinder pressure and heat release rate were analyzed and compared to those of DCI. Two distinct combustion modes were observed during the GCI testing, namely PPCI and MCCI. Only three cases are presented here for brevity at different operating conditions, 1500 rpm/ 10 bar BMEP, 2000 rpm/ 10 bar BMEP, and 2000 rpm/ 20 bar BMEP.

As shown in Figure 7, the corresponding combustion pressure and heat release rate of GCI combustion are compared with DCI combustion. The pulse widths of the pilot and main injections events are plotted to highlight the differences. Figures 7A, 7B, and 7C are referring to points on Figures 3 and 4 for full condition data. In general, for GCI mode to achieve similar NO_x versus PM trade-off compared to DCI, the fuel pressure could be set lower by approximately 300 bar at all presented points. Considering that GCI and DCI combustion modes at different load points resulted in similar NO_x and Soot levels, it is observed that GCI combustion has an improved NO_x-Soot trade-off behavior than DCI and is more fuel efficient.

As shown in Figure 7A, at 1500 rpm/ 10 bar BMEP, the longer ignition delay of gasoline allowed for a noticeable earlier pilot injection for GCI compared to DCI, although corresponding main injection start at the same time. Combined with larger quantity split for GCI, the available time for mixing before the combustion resulted in a stronger initial of HRR. It is an evidence of a premixed combustion depicted by the sudden elevated peak of HRR and the early rise of the combustion pressure before the start of main injection. The main injection then quenches slightly the premixed combustion initiated by the pilot injection, and the mixing-controlled combustion followed. In this case, the combustion is referred to as an MCCI combustion which is similar to that of DCI by exhibiting a double-hump HRR shape, but the GCI differs by the strong start of premixed combustion.

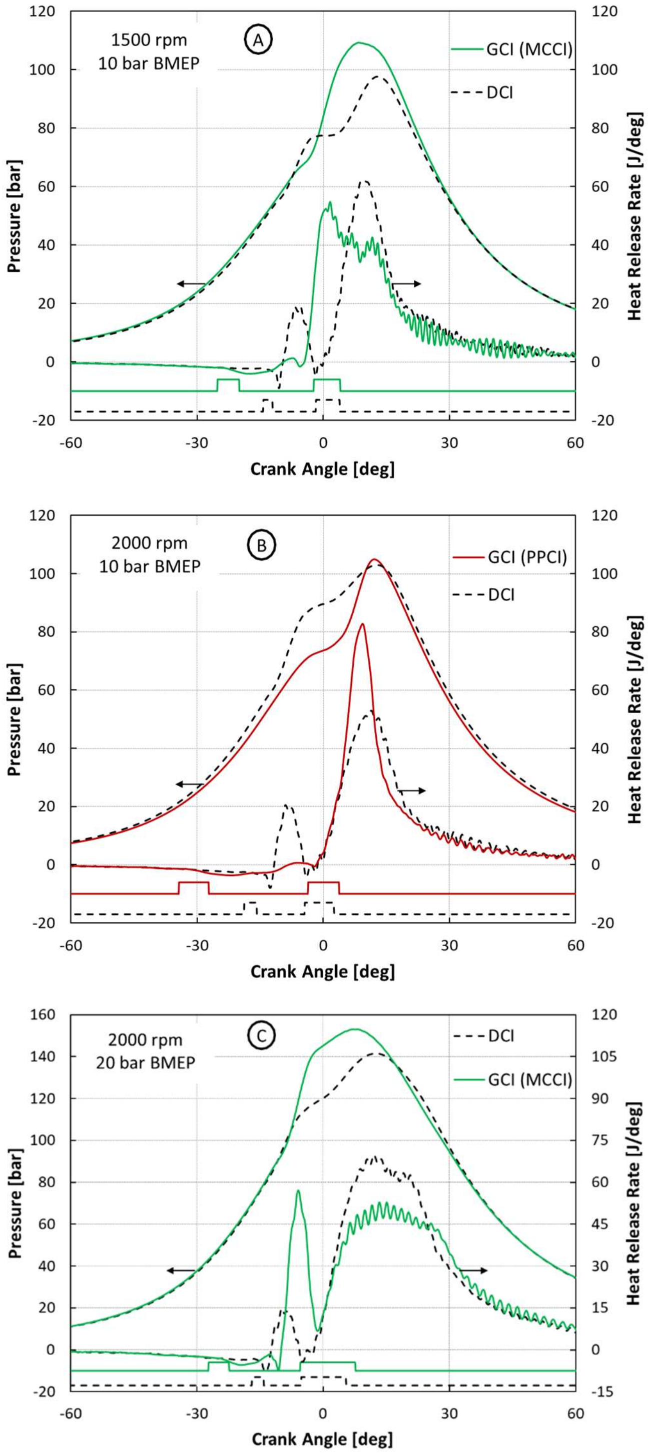

Figure 7. GCI versus DCI strategy- Cylinder pressure, heat release rate, and injections pulse width (represented in square signal). Figures A, B, and C are referenced to points on Figures 3 and 4.

The combustion at 2000 rpm/ 10 bar BMEP operating condition is shown in Figure 7B. The pilot injection for GCI is further advanced compared to that of the GCI in Figure 7A. At these conditions, the mixture had even more time for mixing, leading to a dominant partial premixed combustion as indicated by a single HRR peak and sharp pressure rise rate (PRR). This combustion is defined to be a PPCI combustion. On the other hand, DCI combustion exhibits the characteristics of MCCI, with a typical double-hump HRR shape.

By increasing the load to 20 bar BMEP at 2000 rpm, the optimum GCI combustion mode exhibits MCCI characteristics as shown in Figure 7C. At this condition, the pilot injection started later, as in Figure 7A, and with less Qsplit amount comparing to that in Figure 7B. Compared to DCI, the GCI has stronger initial premixed combustion and is characterized by a larger first hump of HRR. This is followed by a mixing-controlled combustion after the start of the main injection similar to DCI combustion.

Optimized Operational Map for GCI engine

The optimum GCI strategy depends on the operating conditions, mainly: load, speed, boost pressure, SOI, Qsplit, and injection pressure. To further explore on the effect of such parameters, both strategies were run at the same speed and load to highlight the differences. Two condition are presented in Figures 8 and 9 and they include: Qsplit, injection pressure, boost pressure, BSFC, BTE, and PRR for comparison. Corresponding combustion pressure and heat release rate are plotted along with the pulse widths of the pilot and main injections events to highlight the differences. Full operating condition for cycles A and B are referenced in Figure 3 and 4.

As shown in Figure 8, MCCI combustion is compared with PPCI combustion at 1500 rpm/ 10 bar BMEP. Both strategies share the same Qsplit and boost pressure, however, a lower injection pressure combined with longer dwell time between the pilot and the main injection for the PPCI resulted in more homogenized mixture. The PPCI combustion is characterized by an elevated single HRR peak. In contrast shorter dwell time between pilot and main injection implies a less homogeneous mixture with a subsequently slower heat release and pressure rise rates. In addition, the MCCI is distinct here by injecting the main injection during the heat release resulting in mixing-controlled combustion. Having the PRR within the limit (< 8 bar/deg), MCCI strategy was picked as the optimum at this operating point since both strategies have the same BSFC.

Another scenario to highlight the differences between the MCCI and PPCI strategies are shown in Figure 9 at 2000 rpm/ 10 bar BMEP. Both strategies share the same injection pressure. The earlier larger pilot combined with less boost pressure characterize the PPCI strategy compared to MCCI. At this conditions, the PPCI strategy delivered better fuel efficiency with PRR within the limit, and hence, the PPCI strategy was picked as the optimum at this operating point.

In summary, the following are the authors' observations for running both GCI combustion modes, PPCI and MCCI. Although, both GCI strategies can run at the same operating points most of the time, there is only one optimum strategy that gets better thermal efficiency or stays within the operating limits. In PPCI strategy, the pilot injection is more advanced and have larger amount of the fuel split with the main (Qsplit), in addition the boost and fuel pressure are lower compared to MCCI. Whereas in MCCI strategy, the pilot injection is less advanced and lower in Qsplit amount, in addition, the boost and fuel pressure are higher. Some of these mentioned variables could be the same for both strategies except the pilot injection timing, which plays the key role in determining the strategy (PPCI or MCCI) in combination with the injected amount (as a function of fuel pressure and amount split with the main injection).

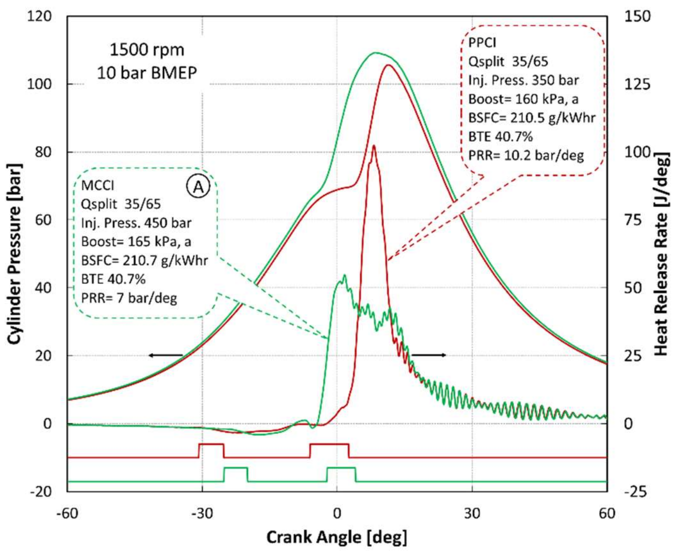

Figure 8. Example of running both GCI strategies at 1500 rpm/ 10 bar BMEP: different injection pressure and timing

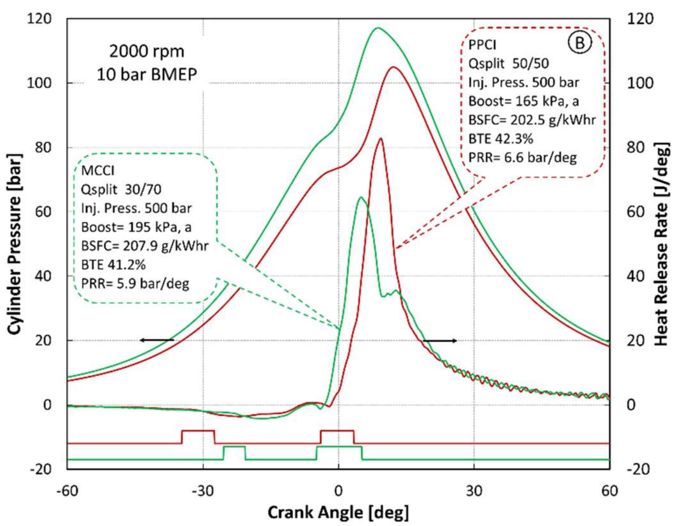

Figure 9. Example of running both GCI strategies at 2000 rpm/ 10 bar BMEP: different boost pressure, Qsplit, and injection timing

In the current study, an investigation of all engine speeds and loads was conducted to find the optimum GCI strategy at each point. The recommended operating map is demonstrated as shown in Figure 10. The proposed map suggested a GCI engine running through the entire engine speeds at medium to high load range. For low load, the engine could be started by using spark ignition mode since it is difficult to autoignite gasoline at lower temperature conditions. The shaded area at speeds higher than 3000 rpm was not tested currently and is to be explored in future work since a dedicated gasoline fuel pump is needed for this purpose. Points A, B, and C are referenced on Figures 3 and 4. Also the injection strategies at different speeds and loads are illustrated along with the fuel rail pressure (FRP). Full GCI maps are shown in Figure 11 for the tested region in this study.

As shown in Figure 10, the optimum operating regions for both GCI strategies, PPCI and MCCI, are depicted. It was determined that PPCI combustion shows improvement at lower loads as a result of the attempted testing. For achieving higher loads with GCI operating condition, the optimized injection strategy lean towards MCCI combustion where high level of fuel stratification and late injection reduces the PRR which enables running GCI at this conditions with better control over the combustion phasing. Nonetheless, the PRR is still a challenge at higher loads as shown in the upper right corner of Figure 11a. The engine-out emission results shown in Figure 11b, c, d, e give an evidence of the selected GCI combustion mode in operation. With PPCI mode, the NO_x and soot (FSN) are reduced, However, CO and HC emissions are higher [2-3]. With MCCI mode, it is the opposite by having more NO_x and soot and less CO and HC [10]. Increased smoke as speed is increased can be fixed by increasing the injection pressure.

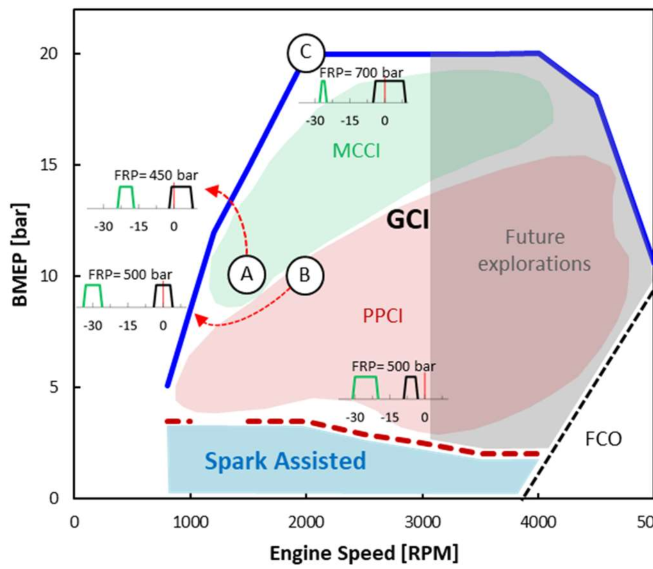

Figure 10. Proposed full map operation of the GCI engine.

It was observed that for PPCI mode, the start of injection of the pilot is always before -30 degATDC. For MCCI mode, the injection events are retarded compared to PPCI and the main is injected during the start of heat release generating a mixing-controlled mode similar to DCI. This explains the need for higher injection pressure at higher loads where the MCCI mode is applied as shown in Figure 11f. The split amount for the pilot of both GCI modes increases by increasing engine speed and decreases by increasing the load. The PPCI Qsplit range is between 40/60 to 70/30, whereas for MCCI it is between 15/85 to 30/70.

As shown in Figure 11g, the brake thermal efficiency is lower at low-load due to fuel reactivity issue at lower temperature conditions, especially at higher speeds where cycle time are shorter and the longer ignition time for gasoline becomes an issue. Finally, Figure 11h shows as load and engine speed are increased the BD1090 increases and this results in higher exhaust gas temperature (EGT) as the mixing rate of fuel and air is unable to keep up due to fuel pressure limitations. Longer burn durations result in wasted heat release, which is observed with higher EGT temperatures.

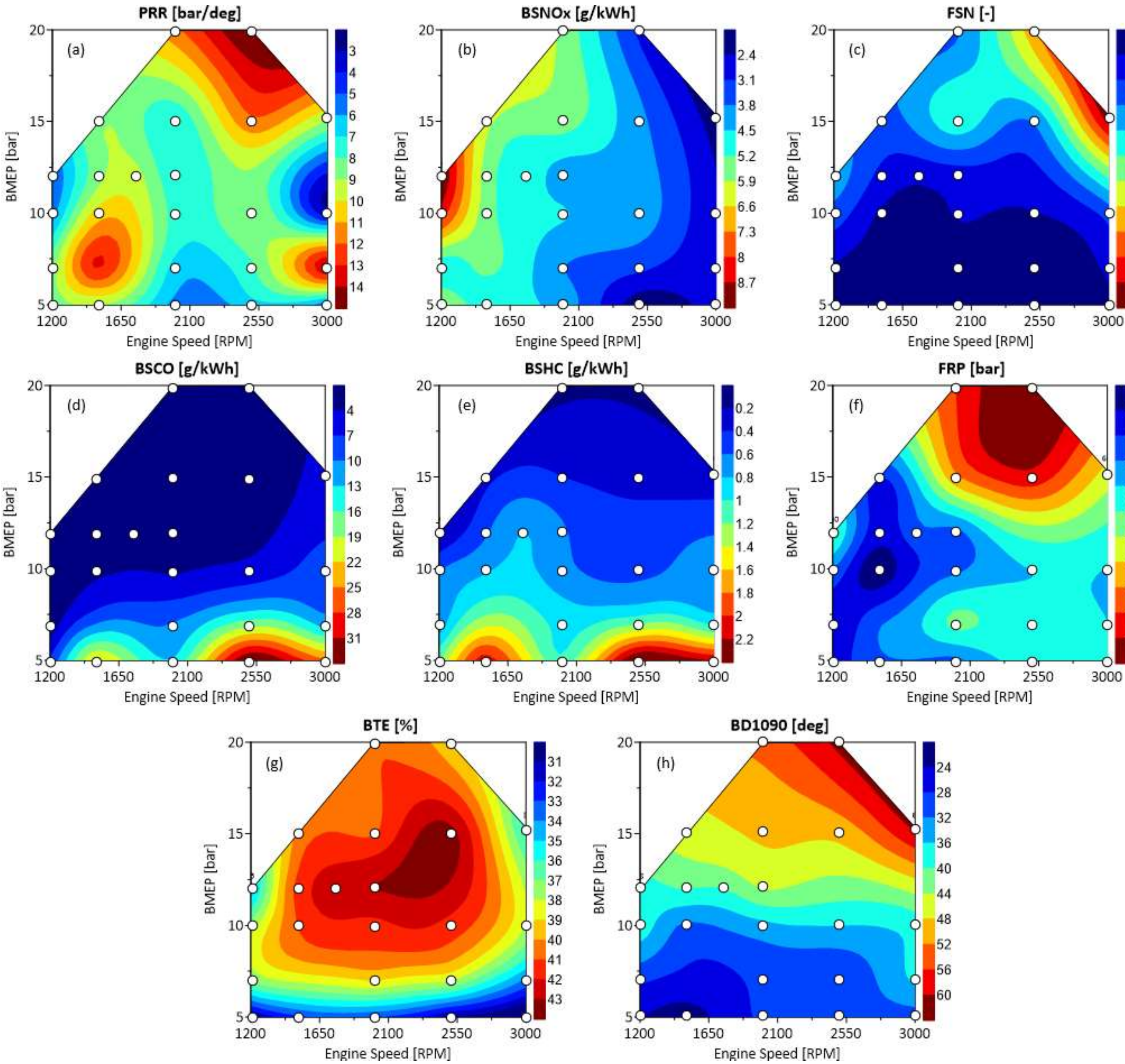

Figure 11. GCI Contour maps. (a) pressure rise rate, (b) $BSNO_x$, (c) FSN, (d) BSCO, (e) BSHC, (f) fuel rail pressure, (g) break thermal efficiency, (h) BD1090

Summary and Conclusions

Experiments have been conducted in a Hyundai 2.2L engine to demonstrate the benefits of GCI mode versus DCI mode at engine speed in the range of 1200-3000 rpm and loads in the range of 5-20 bar BMEP. Parameters varied include SOI_Pilot, Qsplit, injection pressure, and boost pressure of the double-injection strategy were varied to demonstrate both PPCI and MMCI strategies of the GCI combustion mode. A deep dive into the steady state operational data as well as detailed cylinder pressure analysis was performed to highlight the findings of this study which are summarized as follows:

- GCI mode is highly efficient compared to DCI with upwards of 43.4% brake thermal efficiency at 2500 rpm and 15 bar BMEP without optimization. This corresponded to a smoke FSN of 0.5 and $BSNO_x$ of 3.97 g/kWh. This was achieved with an inlet pressure of 2 bar, abs. and no EGR.
- GCI mode was able to show an improvement over a baseline SI engine with best point at 2500 rpm/ 15 bar BMEP by 19.8%.
- In PPCI strategy, larger pilot Qsplit and lower injection and boost pressures were applied, whereas in MCCI strategy, a later injection timing and smaller Qsplit for the pilot were applied in conjunction with higher boost pressures.

- CFD analysis is recommended to further optimize the combustion recipe. By looking into the effect of engine speed and load on the in-cylinder fuel-air mixture distribution in the cylinder in addition to its interaction with the piston, we can better understand and develop the combustion strategy for MCCI and PPCI modes and optimize the combustion hardware to best harness the benefits of both.
- In general, the performance of GCI mode can be improved by optimizing of numerous relevant factors such as the boosting system, swirl, fuel injectors, and increasing the number of injections.

Challenges to apply GCI technologies include:

- Ensuring low-load combustion stability is difficult with low reactivity fuels like gasoline. Traditionally charge air heating and increased boost pressure has been used to address this.
- Maintaining high load maximum pressure rise rates to < 8 bar/deg, with heat release rate shaping while avoiding excessive EGT temperatures. More injection event helps to shape heat release rate but increased fuel pressure >775 bar is needed to improve the trade-off.
- Higher fuel pressure than 775 bar were not possible using the stock diesel fuel pump. This was due to fuel cavitation in the fuel pump caused by internal heat generation because gasoline has less lubricity and higher volatility compared to diesel fuel. A fuel pump designed for the use with gasoline fuel pump capable of higher injection pressures >1000 bar is needed.

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Definitions/Abbreviations

HCCI	Homogenous Charge Compression Ignition
PPCI	Partially Premixed Compression Ignition
EGR	Exhaust Gas Recirculation
CR	Compression Ratio
BTE	Brake Thermal Efficiency
TDC	Top Dead Center
GCI	Gasoline Compression Ignition

CI	Compression Ignition	**DI**	Direct Injection
RON	Research Octane Number	**AFR**	Air-Fuel Ratio
ATDC	After Top Dead Center	**FSN**	Filter Smoke Number
HRR	Heat Release Rate	**CAD**	Crank Angle Degree
SOI	Start of Injection	**Qsplit**	fuel split percentage on mass basis
PRR	Pressure Rise Rate	**PI**	Pilot Injection
ITE	Indicated thermal efficiency	**MI**	Main Injection
BMEP	Brake Mean Effective Pressure	**CA50**	Crank-angle location of 50% fuel mass burned
MCCI	Mixing-Controlled Compression Ignition	**BSFC**	Brake Specific Fuel Consumption
DCI	Diesel Compression Ignition	**FRP**	Fuel Rail Pressure