Technology Enablers for Advanced Gasoline Compression Ignition Engines

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Abstract. To meet current emission regulations and increasingly demanding global fleet CO_2 standards on fuel economy and future trends towards life cycle GHG emissions, advanced combustion engines remain significant in the passenger vehicle sector to achieve high efficiency and low emissions over the full operating range. Lean burn gasoline compression ignition (GCI) technology has shown to have the most potential in reaching these goals, although it faces challenges in the operating range. In this study, an advanced GCI engine is considered with the capability to operate under two combustion modes, namely low temperature combustion (LTC) and GCI. They are enabled with the use of two Hyundai in-house developed technologies; an advanced valve control mechanism known as continuously variable valve duration (CVVD) and a high-pressure gasoline injection system. At low load, the engine utilizes dual CVVD and dual CVVT (continuously variable valve timing) mechanisms for both intake and exhaust valvetrains to enable NVO (negative valve overlap) to trap hot residuals. The hot residuals enable low load auto-ignition, and the phasing control of that auto-ignition is achieved by varying the degrees of NVO, and in turn, the amount of hot residuals. This is done in real time with respect to the changes in engine loads or operating conditions, significantly improving cylinder-phasing control in LTC mode, and subsequently is able to realize a fuel economy gain and reduced emission benefits. Early direct injection during NVO or intake stroke is used to form homogenous or quasi-homogenous air-fuel mixture for LTC. At mid to high loads, late direct injection of gasoline or GCI mode is employed to take advantage of the high volatility and high resistance to auto-ignition of gasoline fuel, so that diesel-engine like high combustion efficiency can be achieved with lower soot emissions for the same NOx level as diesel combustion. In this paper, the key control and fueling technologies that enable this multiple mode combustion are introduced, and the resultant analysis on engine test and CFD simulation at LTC and GCI operating conditions are presented in detail.

Keywords: Gasoline Compression Ignition, Low Temperature Combustion, Negative Valve Overlap, Continuously Variable Valve Duration, Continuously Variable Valve Timing, High Pressure Gasoline Direct Injection.

1. Introduction

Ever increasing regulatory demand to reduce CO_2 emissions and conserve global resources has led to an increased focus on advanced internal combustion engine development to improve overall engine efficiency. Meanwhile, more and more stringent emissions standards for NOx, hydrocarbon, and particulate matter requires significant reductions in engine-out emissions. Spark ignition (SI) gasoline engines have lower emissions but also lower combustion efficiency as compared to compression ignition (CI) diesel engines, primarily due to throttling at low load and knocking at mid to high loads. Compression ignition diesel engines have higher efficiency, but suffer from a high NOx vs. soot trade-off. Gasoline compression ignition (GCI) engines were then proposed to combine the features of SI low emissions and CI high efficiency [1], and have been widely studied numerically and experimentally [2-5]. Hyundai's initial GCI test has demonstrated that GCI combustion mode can result in better thermal efficiency and lower soot emissions for the same engine-out NO_x level than those of a comparable diesel engine, and the maximum brake thermal efficiency achieved with pump-grade AKI 87 E10 fuel was 43.3% [6]. This result was determined to come from extended ignition delay of gasoline AKI 87 E10 fuel versus #2 ULSD diesel fuel and the decreased fuel pressure requirement, which reduced pump work and parasitic losses.

However, GCI combustion faces challenges in actual engine applications at low load and high load operations. More specifically, the low cetane number of gasoline fuel, estimated to be around 15 maximum compared to a cetane number greater than 40 for a typical #2 ultra-low-sulfur diesel fuel, leads to long ignition delays especially under low boost and ultra-lean condition as in low load operation, resulting in large cycle-to-cycle variation and even misfire. To improve combustion stability at low load, either a fuel with higher reactivity is needed, such as a gasoline-like fuel with RON around 70 [7]; or auxiliary devices such as a heater to elevate charge air temperature and/or an exhaust rebreathing method need to be applied [5].

At high load and high speed, too rapid combustion and pressure rise rates can damage the engine and/or result in excessive noise, and as such, should be avoided. Methods to reduce the fuel reactivity rates at these conditions are inducing exhaust residuals into the combustion chamber, and using mixing controlled combustion strategy as in diesel engines. Alternately, multiple fuel injections can be used to create a mixture stratification with and without spark assist [8, 9]. Nonetheless, a high-pressure gasoline fuel injection system is needed to improve fuel atomization, vaporization, and air/fuel mixing for soot reduction. Sellnau et al. [5] showed that higher fuel injection pressures over 1000bar are needed to enable GCI combustion while minimizing PM emissions. However, production-ready gasoline direct injection (GDI) systems are not currently available at pressures above 1000 bar. Common rail fuel injection systems designed for diesel fuel typically have fuel injection pressures on the order of 2000 bar. There are challenges to implementing diesel common rail systems in gasoline compression ignition engines due to insufficient durability of the components with the reduced fuel lubricity of gasoline compared to diesel fuel as well as controlling the fuel properties during compression/expansion [10].

In the current work, two technologies Hyundai has developed in-house are introduced to GCI engines: an advanced valve control mechanism known as continuously variable valve duration (CVVD) and a high-pressure gasoline fuel system consisting of a high-pressure fuel pump and high rate of injection injectors. The feasibility of using these two technologies to enable GCI combustion during both low and high load operations has been experimentally and numerically demonstrated. At low load, the engine utilizes the dual CVVD and CVVT (continuously variable valve timing) mechanisms for both intake and exhaust valvetrain to enable NVO (negative valve overlap) in order to trap hot residuals and promote auto-ignition for LTC. At mid to higher loads, the high pressure gasoline fuel system capable of 1800 bar enables late direct injection of gasoline or GCI mode, so that diesel-engine like high combustion efficiency can be achieved with lower soot emissions for the same NOx level as in diesel mode.

2. Two Key Technologies for GCI

2.1 CVVT& CVVD Valvetrain

Hyundai Motor Group has launched engines with CVVD on the market beginning in 2019 [11-13]. CVVD maintains the valve lift and changes the valve's duration. By combining with CVVT, these two technologies independently control the valve's opening and closing timing. To realize LTC, a new valvetrain system with dual CVVD and dual CVVT was designed uniquely. With that, the engine can move from conventional timings to relatively short durations with enough phasing authority seamlessly, which enables NVO to trap hot residuals and promote low load auto-ignition for LTC. In LTC, the air and fuel forms a homogenous mixture and burns rapidly at certain conditions, but it is known to be not easy to control. The combustion phasing control in this paper, however, is achieved flexibly by varying the extent of NVO, and in turn, the amount of hot residuals. This can be done in real time, significantly improving cylinder phasing control in LTC mode, and subsequently is able to realize increased fuel economy and reduced emissions. The success of using CVVD & CVVT to enable LTC has been well demonstrated numerically and experimentally in our six-stroke engine development project [14, 15].

CVVD uses a fixed shape cam, but adjusts the valve duration by changing the instantaneous rotational speed of the cam [11-13, 16].

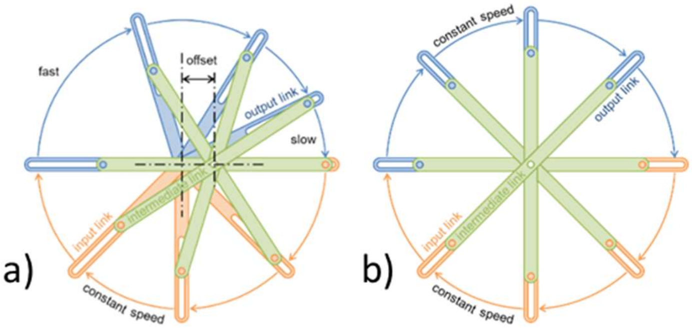

Fig. 1. CVVD Mechanism

Fig. 1 shows the mechanism for altering the instantaneous angular velocity used in CVVD. In Fig. 1-a), the rotation centers of the input and output link are the same, but the center of the intermediate link is offset. In this case, even when the input link rotates at a constant speed, the rotational speed of the output link varies depending on the position. In Fig. 1 –b), the center of the intermediate link coincides with the center of the input and output link, so the output link rotates at the same speed as the input link. CVVD controls the valve opening time by changing the center position of the intermediate link.

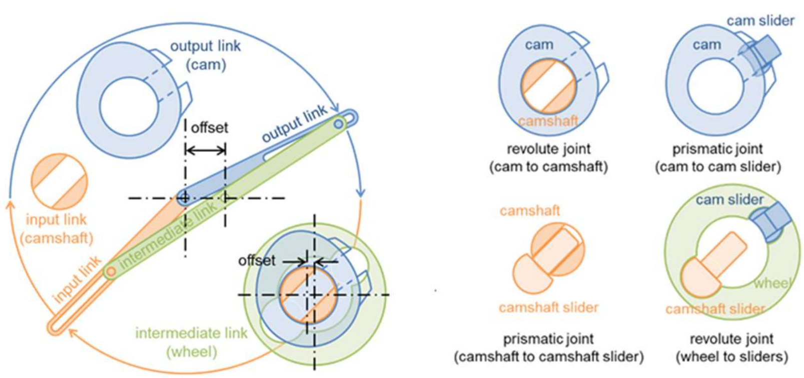

Fig. 2. Links and joints structure of the CVVD system

In Fig. 2, the link structure of Fig. 1 is illustrated with actual engine parts. The input link corresponds to the camshaft, and the output link corresponds to the cam lobe. In a conventional engine, the cam is integrated with the camshaft, but the CVVD separates the cam lobe from the camshaft to change the rotational speed of the cam to realize variable valve duration even with the same cam shape. In addition, the wheel corresponding to the intermediate link and sliders connecting these links are shown. Fig. 2 shows the combination of these five links. The wheel position controller that changes the position of the wheel is not shown in the figure.

Fig. 3 compares the CVVD model currently in mass production (Gen-I) with the latest version developed for the advanced combustion engine (Gen-II). The mass production model applies the CVVD system only to the intake cam, but in the GCI engine, CVVD is applied to both intake and exhaust. In addition, a Gen-II CVVD mechanism was developed to reduce the number of parts and improve valve dynamics. The Gen-I system is sufficient for general engine operation, but more aggressive valve operation is required for effective GCI operation, so an improved CVVD system was used.

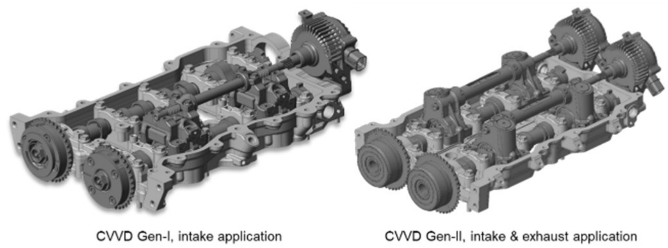

Fig. 3. Comparison of the CVVD Gen-I and Gen-II

Fig. 4 compares the wheel position controllers of CVVD Gen-I and Gen-II. The camshaft assembly is the same for Gen-I and Gen-II. Unlike Gen-I, for Gen-II, the screw shaft is integrated with the wheel housing. Both operate with a worm and worm gear mechanism. Gen-I's wheel housing moves horizontally, while Gen-II's moves vertically. Gen-II has a structure that supports the wheel housing from both the upper and lower parts, so the assembly is more mechanically rigid.

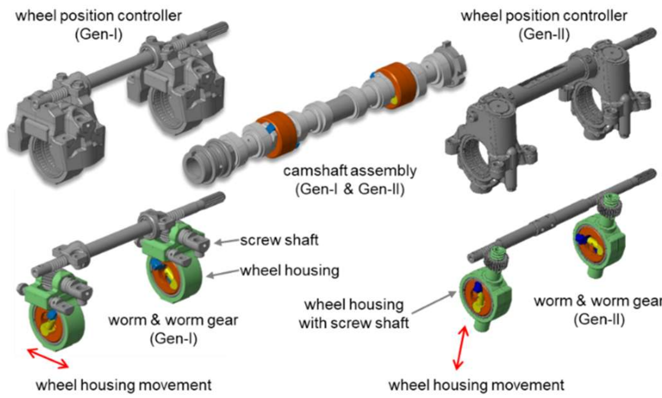

Fig. 4. Comparison of wheel position controller of the CVVD Gen-I and Gen-II

Fig. 5 shows the cam carrier assembly of the GCI engine. In addition to using CVVDs for both intake and exhaust, electric CVVTs were also used. The e-CVVTs have a maximum range of authority of 100 degrees.

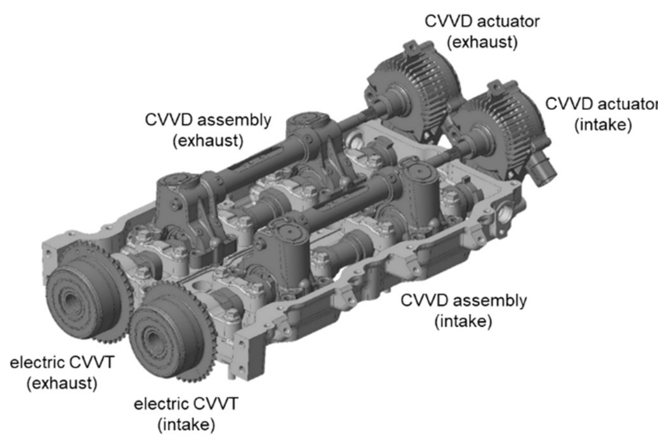

Fig. 5. Cam carrier assembly of GCI engine with CVVD applied to intake and exhaust cams

The valve profiles of the CVVD system with the CVVTs at the reference position are shown in Fig. 6. In order to avoid interference with the piston, the intake valve has a fixed opening point while the exhaust has a fixed closing point. The intake CVVT moves in the retard direction and the exhaust CVVT moves in the advance direction to implement a NVO strategy.

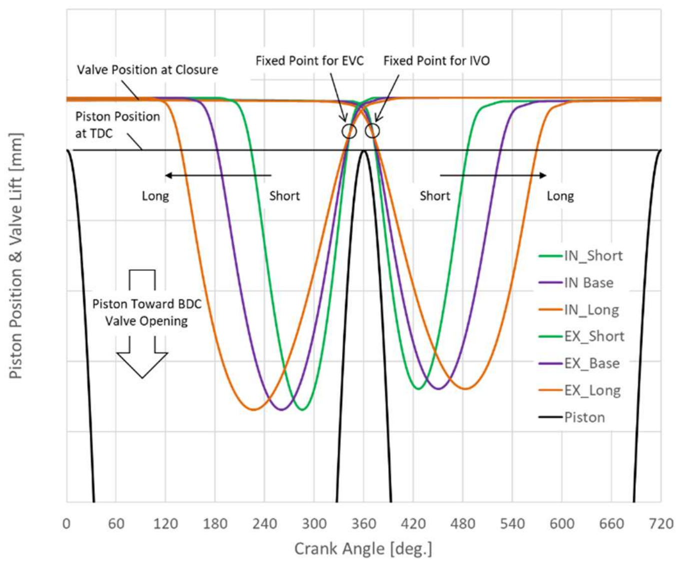

Fig. 6. Valve profiles of the CVVD system when the CVVTs are in reference position

2.2 High pressure gasoline fuel system

At low load, LTC is an effective way to get a fuel benefit, but the pressure rise rate at mid to high loads may not be acceptable. To overcome this phenomenon, the overly rapid combustion needs to be controlled using different fueling strategy, which is referred to as GCI combustion. With that, a late injection is efficient in controlling combustion phasing and pressure rise rate. Consequently, this showed the need for the development of a gasoline high fuel pressure system to inject the fuel and initiate the combustion around TDC timing.

By combining a cost-effective common rail approach with GDI technology, an applicable 1800bar GDI system has been designed.

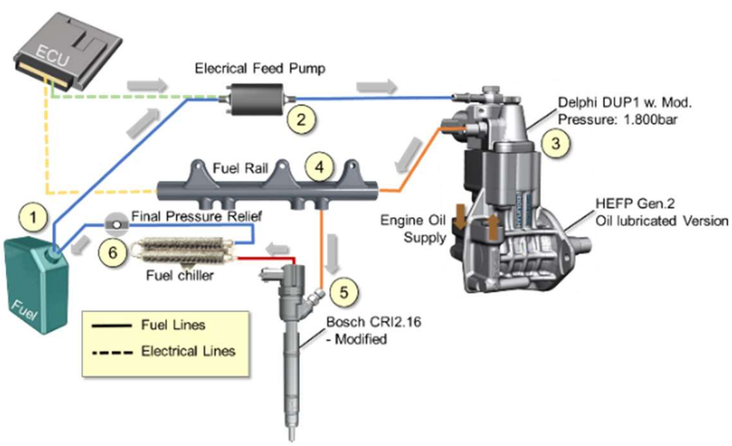

Fig. 7. Setup of 1800bar GDI system

The major effort incurred for such an injection system is stabilizing the fuel during pressurization and expansion [17-19]. As shown in Fig. 7, some system approaches need to change in order to process a high evaporation point fuel, starting from a standard fuel tank ①, to the electrical feed pump ②. Here the first change is by using the ECU as power supply for the feed pump. This allows controlling the rail pressure via supply flow and keeps the fuel above ambient pressure level to avoid fuel cavitation. As a result, even during suction stage in the high-pressure pump ③, the fuel cannot reach evaporation condition. Here the fuel will be pressurized up to 1,800bar and discharged into the rail ④. The rail itself is a single governor concept as is common in GDI applications. The connection to the injector ⑤, is via steel pipes, as is typical for diesel common rail and central mounted injectors, as conventional O-Ring fittings cannot survive the high pressure levels. Since a servo-actuated electrically driven injector has leakage, extra precautions have to be taken within the return line system ⑥ to the fuel tank. The fuel pressure in the return line has to be increased to 5 bar in order to avoid boiling gasoline and/or auto-ignition. Since the return fuel expands due to pressure drop from the 1,800 bar contained in the rail, the fuel will heat up to a potential maximum of ~125°C above the fuel tank temperature due to energy conservation. Therefore, the fuel needs to be cooled via a fuel chiller to a reduced temperature level of ~50°C prior to expansion to ambient pressure, as it returns into the fuel tank.

Among the entire injection system setup, two components necessitate additional engineering effort for proper functionality. One of them is the high pressure pump ③. The pump needs to be not only robust, but also cost efficient. This design is based on our internally developed HEFP Gen.2 roller bearing pump, which can easily be applied with engine oil lubrication, to compensate for poor gasoline lubricity. To protect the plunger and the high-pressure chamber of the high-pressure pump, following measures are necessary.

One is to avoid cavitation by never having the fuel inlet lower than 1 bar absolute. The other is to improve material surface strength by utilizing DLC coating on the plunger. Using this new pump design, the pump itself is designed to be capable of above 3,000bar. A Delphi DUP1 pump element is utilized for pressurizing the fuel. Here, a robust friction reducing coating has been applied in order to ensure durability.

The other component with a higher engineering effort is the injector ⑤. The selected Bosch CRI2.16 common rail diesel injector is a robust injector with high technical maturity for diesel emission standards. As an optimization for this application the spray pattern was retargeted according to piston bowl shape, fuel atomization and combustion characteristics. As a result, the servo-hydraulic circuit and command piston pre-loads were changed. A parametric study on this configuration was performed by 1D-simulation. Table 1 lists the major performance target of the prototype 1800bar GDI fuel injection equipment.

Table 1. Major performance targets of 1800bar GDI fuel injection equipment

Item	Design Value	Design Accuracy
Max Pressure	1800 bar	-
Min. Injection	~1.5 mm^3/stroke	10%
Shot to Shot Variation	-	< 2.5%
FIE Efficiency	> 50%	-
Fuel Type	RON 91	-
Durability (proto)	500 h	

3. Engine Setup

In this study of technology demonstration, two engine setups were used sequentially to confirm the feasibility of using dual CVVD and high pressure fuel injection system to enable LTC at low load and GCI at mid to high load conditions. The engine specification is listed in Table 2. The gasoline fuel used was AKI 87 E10. One of the test engines was a Hyundai 2.0L engine with the addition of CVVD & CVVT mechanism on both intake and exhaust valvetrains (LTC-01). The engine also utilized low pressure EGR, 350bar fuel injection, and increased 14:1 compression ratio to assist auto-ignition.

Table 2. Engine specification

Engine	LTC-01	GCI-01
Displacement [L]	2.0 (4-cylinder)	2.2 (4-cylinder)

Bore × Stroke [mm]	81x97	85.4x96.0
Direct Injection	Side Injection with 6 Holes	Central Injection with 8 Holes
Max Fuel Pressure of DI [bar]	350	1000
Max Fuel Pressure of PFI [bar]	-	5
Piston	Shallow Cavity	Bowl
Comp. Ratio	14	16
Valve System	CVVD & CVVT	16 Valves HLA, Chain Drive
Turbocharger	-	Variable Geometry Turbine
EGR	Low Pressure Cooled EGR	High/Low Pressure Cooled EGR Loops

The other engine used in this study was Hyundai's 1st generation of GCI engine (GCI-01). It is modified from a four-cylinder 2.2L turbocharged, compression ignition engine. It is outfitted with both port and direct fuel injection systems. With the spark plug on the engine, it is capable of SI and GCI operations. 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 direct injection (DI) injectors are centrally mounted in the flat cylinder head with bowl in piston design combustion chamber. Details on experimental instrumentation and measurement can be found in [6].

With the demonstrated feasibility of using those two technologies to enable LTC and GCI, Hyundai designed its 2nd generation of GCI engine (GCI-02) by adding CVVD and upgrading the high-pressure gasoline fuel system to the GCI-01 engine. The GCI-02 engine is capable of running multimode combustion – SI for cold start, LTC at low load and GCI at mid to high load conditions. GCI-02 engine testing is currently ongoing, and results will be reported in our DOE funded project "Co-Optimized Mixed-Mode Engine & Fuel Demonstrator for Improved Fuel Economy while Meeting Emissions Requirements"[20].

4. Results and Discussions

4.1 GCI Mid-to-High Load Operation

With the GCI-01 engine, GCI operational range was explored for mid to high load conditions, BMEP range from 5 to 20bar. Gasoline compression ignition could be attained without any assist from spark, and the fuel benefit from lean combustion and no knocking were noticeable compared with a conventional gasoline engine. In this test, double injections (pilot+main) with fuel pressures of 250-800bar and late SOIs around TDC were used. All test data points were taken using the following criteria:

1) NOx emissions ≤ 5g/kWh,
2) Maximum pressure rise rate ≤ 8bar/deg., and

3) Soot emission FSN (Filter Smoke Number) <1

No EGR was used in these tests. Fig. 8 shows the BSFC, burn duration and soot contours for different engine speed and load in GCI combustion mode [6]. The maximum brake thermal efficiency obtained was about 43.3%. It can be seen that the burn duration was very short, soot emissions and fuel pressure were low at mid load. As speed and load increase, burn duration increases, leading to high exhaust gas temperature and smoke. The rate of fuel injection with the existing fuel system was determined to be too low. In this engine test, the stock diesel fuel equipment with common rail system was used directly for gasoline fuel for a limited total test time. The maximum fuel pressure attainable with gasoline fuel was 800bar without causing fuel bubbles in the fuel return line. A higher rate of injection and fuel pressure are clearly needed to improve fuel atomization and vaporization for reducing burn duration and soot emissions, and to increase load at higher engine speeds.

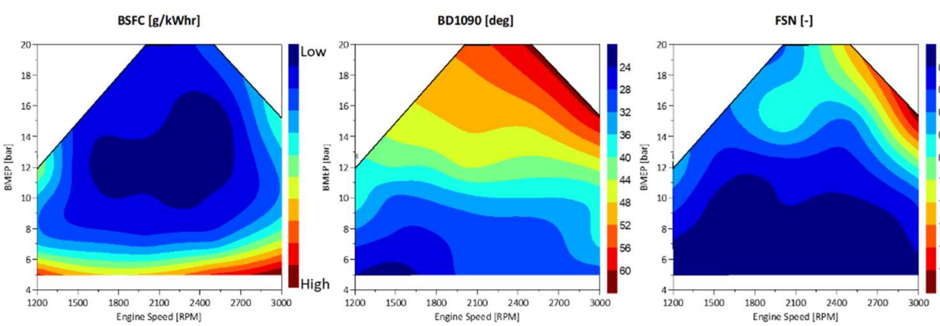

Fig. 8. BSFC, burn duration (BD1090), and soot (FSN) contours of GCI mode. (BD1090 and FSN reproduced from [6]. Copyright © SAE International)

The benefits of using high injection pressure for GCI were studied in combustion CFD simulations. The commercial CFD code Converge [21] was used in this study, and the RANS-based Re-Normalization Group (RNG) κ-ε turbulence model and primary reference fuel chemistry model were applied. The CFD model was first validated with GCI engine data at 2000rpm and 20bar BMEP condition, and then used to predict the combustion performance at rated condition (4500rpm and 18bar BMEP) with different fuel injection pressure.

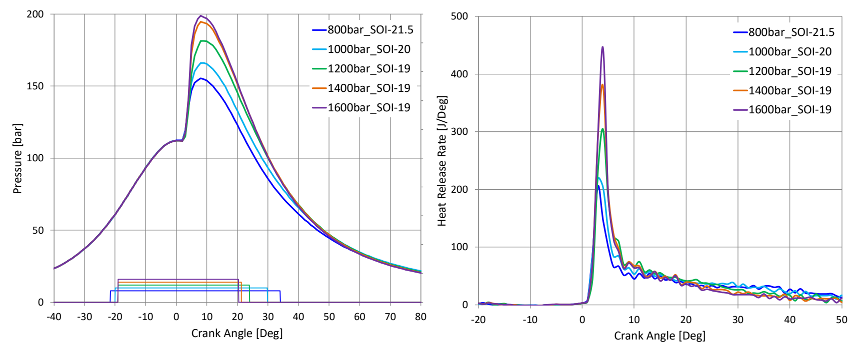

Fig. 9. Fuel injection pressure sweep at rated condition (4500rpm/18.1bar) - combustion pressure and heat release rate

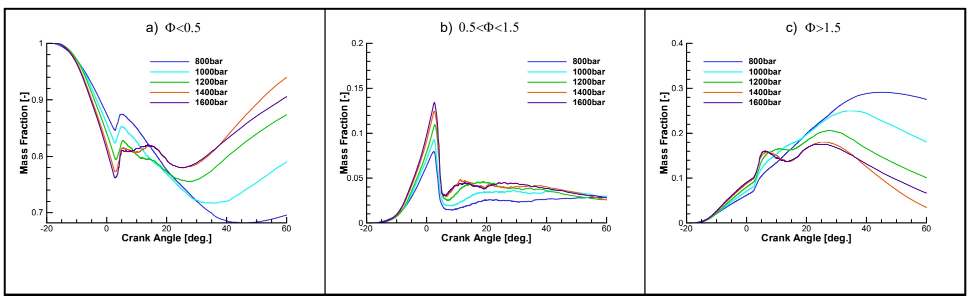

Fig.10. Mass fraction of equivalence ratio ϕ bins: a) ϕ<0.5, b) 0.5<ϕ<1.5, c) ϕ>1.5

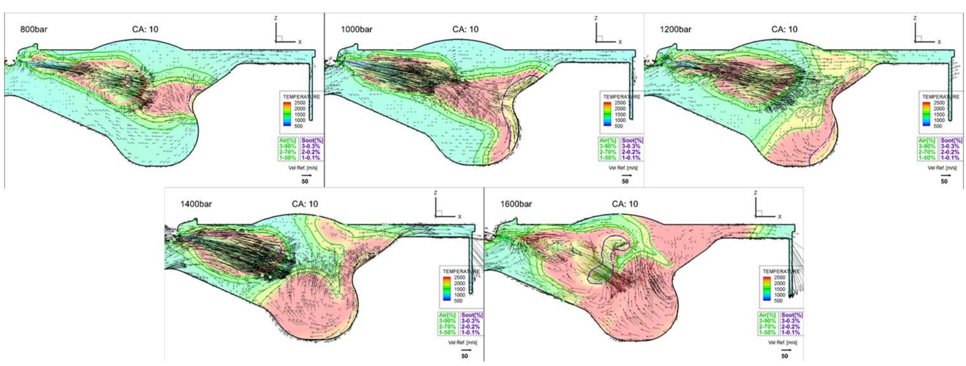

Fig.11. Fuel injection pressure sweep at rated condition (4500rpm/18.1Bar) - temperature contour superimposed with velocity vector, air and soot line contours

The fuel injection pressure from 800 bar to 1600 bar was investigated at rated conditions. No EGR was used. The fuel quantity was kept the same, as higher injection pressure shortened the fuel injection pulse-width and hydraulic duration. Fig. 9 shows the combustion pressure and heat release rate for different injection pressures. The SOI timing was adjusted for each case to have similar start of combustion timing. Clearly, with higher injection pressures, fuel atomization and mixing have been improved as shown in Fig.10 by the reduced mass fraction of an overly rich mixture (equivalence ratio >1.5), and it results in a stronger premixed combustion as indicated by the increased heat release rate peak. Fig.11 shows the temperature color contour superimposed with velocity vector, air and soot line contours for each case. Higher injection pressures increased the combustion burn rate, as indicated by the larger area of high temperature contour and the higher maximum cylinder pressure as shown in Fig. 9. Fig.12 compares the combustion performance for each case. Higher injection pressure significantly decreased the soot (~90%) and HC (~99%) emissions, and increased thermal efficiency, while also increasing NOx emissions. 1400bar injection pressure resulted in the best ITE improvement of 14.4% over the baseline 800bar injection pressure case. It should be noted that the pressure rise rates in these simulated cases were too high, >15bar/deg. compared to our specified 8bar/deg., but this can be reduced by using and optimizing the multiple fuel injections.

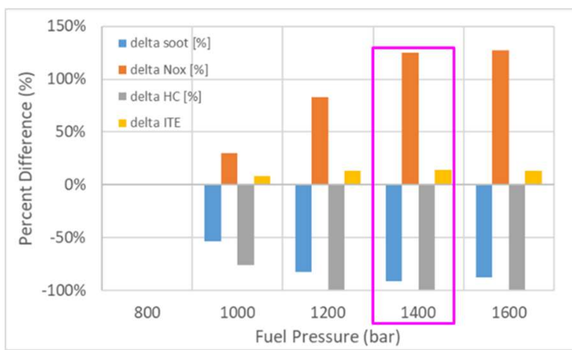

Fig.12. Fuel injection pressure sweep with BSL EU6 piston at rated condition (4500rpm/18.1bar) - performance summary

4.2 GCI Low Load Operation

To improve GCI low load combustion stability and extend low load operation range, one of the approaches that was effective was cylinder pressure feedback control. It reads the combustion status of each cylinder in real time and adjusts the fuel injection timing (CA50 control) and the fuel quantity (IMEP balancing control) across cylinders if any deviation is observed.

Fig. 13 and Fig. 14 show the effects of CA50 and IMEP balancing controls for the GCI-01 engine. The CA50 control adjusted the main fuel injection timing and start of injection, dynamically for the target combustion phasing for each cylinder independently on a cycle by cycle basis. IMEP balancing control was also used to adjust the fuel quantity dynamically by trimming fuel pulse widths to target the torque request for each cylinder. With CA50 control on, each cylinder had the same combustion phasing. With IMEP balancing control on, the IMEP was balanced out across the cylinders. With combustion stability defined as COV of IMEP less than 3%, the GCI low-load limit was extended down to about 2.5bar BMEP from 5.5bar at 1500rpm.

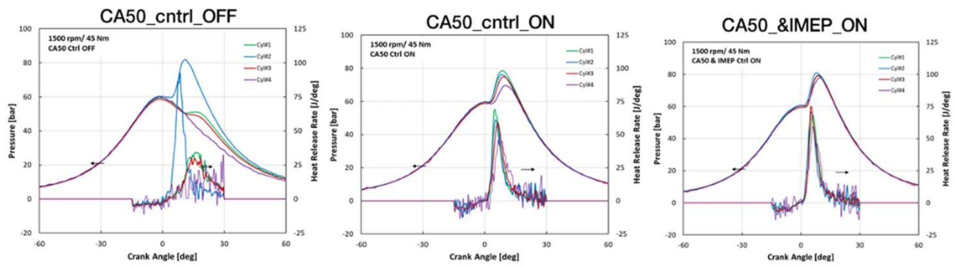

Fig. 13. Combustion Improvement with CA50 and IMEP control

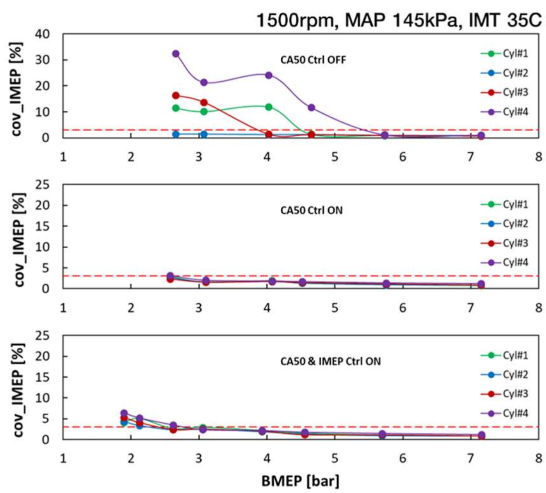

Fig. 14. Effect of CA50 and IMEP control on extending GCI limit

To further extend the low load limit, different parameter sweeps were performed, including boost level, intake air temperature, fuel injection strategy, fuel injection pressure, spark assist, coolant temperature, EGR, etc. The GCI low load limit was then further extended down to about 2 bar BMEP without changing the fuel specification/reactivity. The fuel economy at this load, however, has some penalty due to high pumping loss and unburnt hydrocarbons. In other words, if the gasoline fuel's reactivity is not changed, the engine needs to have some additional levers to boost the reactivity and eventually attain stable combustion and a fuel economy gain over SI.

From that perspective, low temperature combustion was adopted by using negative valve overlap in this study. This combustion has been investigated for a long time, but is not easy to realize or manipulate on an engine. With the combination of Dual CVVD and Dual CVVT, this novel combustion system was well controlled with our LTC-01 engine. As both duration and timing can be controlled for intake and exhaust valves, trapped hot residuals and fresh air can be altered to adjust in-cylinder thermodynamic condition for promoting auto-ignition, especially at low load condition. This is one of the key technology enablers for a novel combustion at low load, or LTC.

Fig. 15 shows an example of negative valve overlap (NVO) used for LTC strategy development. By closing exhaust valve earlier and opening intake valve later, more hot residuals and less cold fresh air are trapped. Depending on the timing and duration used, the in-cylinder fuel air mixture reaction can be then adjusted and controlled.

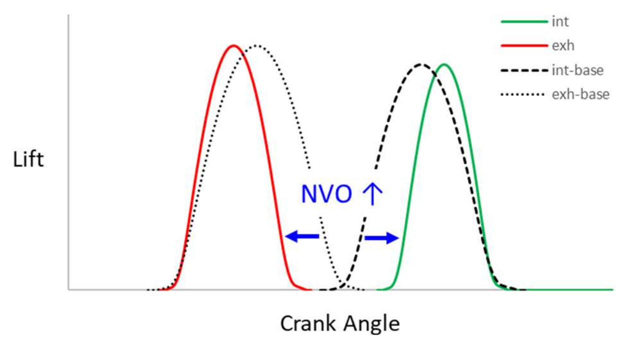

Fig. 15. Negative valve overlap

Fig. 16 compares the typical combustion pressures between LTC and SI modes. Compared with SI, LTC has higher combustion rate and pressure rise rate. The NVO recompression can be also seen clearly for the LTC mode.

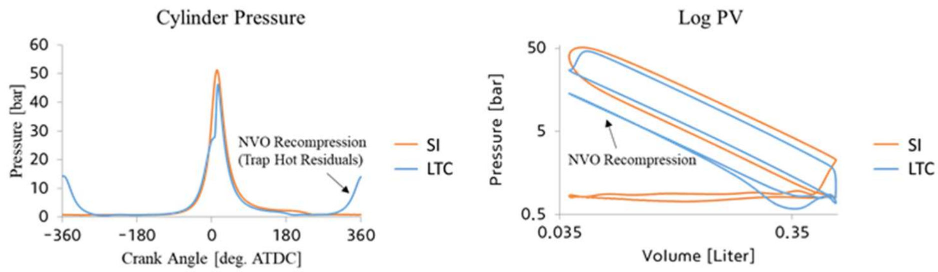

Fig. 16. Combustion pressures of LTC and SI

With intake/exhaust valve duration of approximately 100 CAD and NVO of about 200 CAD, LTC was achieved for 0.5 - 3.5 bar BMEP and 1500 - 3000 rpm engine speed. Fig. 17 shows the corresponding NVO, COV of IMEP, max pressure rise rate, and NOx emissions for LTC. In this study, single direct injection with fuel pressure of 250bar was used, and the SOI timing was 160 to 400 CAD BTDC, depending on NVO used and engine load. NVO duration controls the in-cylinder hot residuals ratio, thus the gas temperature at intake valve closure (IVC) and start of combustion. When NVO duration is small and the gas temperature at IVC is relatively low, then an early SOI during NVO may be needed. For example, injecting fuel in the exhaust re-compression process to have the injected fuel undergo chemical reactions with the hot recompressed exhaust gases produces species with different reactivity, that is, fuel reforming, thereby promoting the auto-ignition of the air-fuel mixture in the main compression stroke. It should be pointed out that to have such fuel reformation during NVO, overall fuel-lean condition is required to have oxygen available in the burnt gases/residuals. In this engine test, the lambda was approximately 1.6. The combustion phasing was controlled by varying the start of injection timing and by adjusting the amount of valve overlap through the CVVD mechanism. Low load range was limited by combustion stability, while high load was restricted by the maximum pressure rise rate for LTC. However, compared to GCI combustion, LTC, enabled by CVVT& CVVD now can operate at lower load down to 0.5bar BMEP with demonstrated low emissions and high efficiency; the fuel economy gain by LTC over SI is approximately 10-15%.

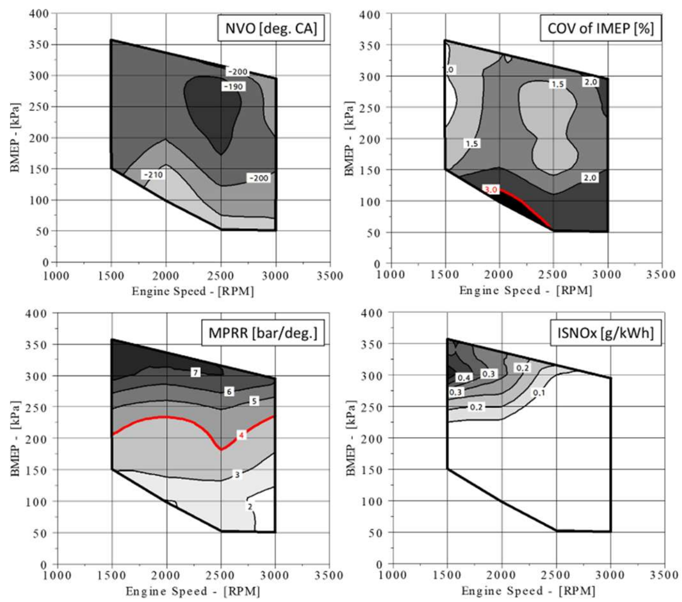

Fig. 17. NVO, COV of IMEP, max pressure rise rate, and BSFC improvement for LTC

With the low load results of GCI and LTC, it is clear that dual CVVT & CVVD for both intake and exhaust valvetrain will help to address GCI low load challenges, by enabling LTC mode at low load operation and realizing a fuel economy gain and emissions reduction.

The fuel economy gain of LTC over SI came from mainly three parts: reduced pumping losses, faster combustion rate, and fuel-lean combustion. Regarding the NOx emissions, it is very low as shown in Fig. 17 due to the low temperature combustion initiated by significant amount of trapped internal EGR.

5. Future work

Mode switching is critical to developing multimode compression ignition engines. The mode switch typically has been very difficult due to the widely differing operating conditions of the different modes. During the mode switch, thermal and gas exchange processes need to vary quickly to prevent rapid changes in torque output which may be unpleasant to the end customer. This remains a very active area of development and exploration which will be crucial to solve.

In LTC mode, in-cylinder combustion temperatures are typically very low, as a result NOx seen in tailpipe emissions is very low as well, which may not need a complicated aftertreatment. However, GCI combustion mode is operated outside typical stoichiometric operation and will require a dedicated aftertreatment, which needs to be explored further.

6. Conclusions

Gasoline compression ignition has long been explored to improve fuel efficiency. To realize this novel combustion, two technical enablers were addressed in this study: dual CVVD & dual CVVT mechanism and high-pressure gasoline fuel system. Both systems were developed and tested in-house.

To gain GCI combustion with high efficiency, it is required to have a late injection near TDC for gasoline fuel. Increased fuel pressure enables reducing fuel injection times significantly, allows multiple injections near TDC, and improves fuel-air mixing and air utilization. As the engine operation moves to higher loads at higher engine speeds, securing high fuel injection rates become critical because insufficient rate of fuel injection causes soot and high exhaust temperatures.

At low loads, it is not easy to control LTC with respect to the changes in engine loads or operating conditions. Negative valve overlap is used in this study to trap hot residual gas and increase in-cylinder temperatures sufficiently to allow and control LTC operation. As engine load increases, NVO is decreased accordingly to avoid high pressure rise rates.

These systems are capable of allowing combustion operation within the desired parameters. Important future development opportunities to explore are mode transition and range expansion, as well as aftertreatment.

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