A Study on Combustion Control and Operating Range Expansion of gasoline HCCI

Kyeonghyeon Lee[1], Seokwon Cho[2], Namho Kim[2], Kyoungdoug Min[2*]

Abstract

Because of the combustion principle of a gasoline homogeneous charge compression ignition (HCCI) engine, the operating range is limited within a narrow area. The objectives of this study were to extend the operating range of gasoline HCCI combustion and to develop control logic. To extend the high load operating range, several strategies including external exhaust gas recirculation (EGR), EGR stratification, fuel stratification and valve timing swing were adopted. Among these strategies, EGR stratification, asymmetric injection and open valve injection are novel techniques. The high load boundary of the low speed region was improved more than that of the high speed region. The improvement in the low load boundary was due to the direct injection during negative valve overlap. In terms of stabilizing the HCCI combustion phase, the peak pressure value and pressure rising rate of a cycle were important factors when considering the ringing intensity equation. Coefficient of

[1]Interdisciplinary Program in Automotive Engineering, Seoul National University, Seoul, Korea

[2]Department of Mechanical and Aerospace Engineering, Seoul National University, Seoul, Korea

Corresponding author:
School of Mechanical and Aerospace Engineering, Seoul National University, Gwanak-ro 1, Gwanak-gu, Seoul, Korea.
Email: kdmin@snu.ac.kr
Tel: +82-2-880-1661

variation of combustion was also used to judge the stabilization of the combustion. In this study, using these variables, the engine was controlled within the maps which were determined from the experiment. The indicated mean effective pressure calculated from real-time data followed the target load successfully without severe problems.

Keywords

Homogeneous charge compression ignition, ignition delay, exhaust gas recirculation, stratification, operating range, combustion control

1. Introduction

There have been increasing concerns about eco-friendly internal combustion engines because of air pollution, global warming and limited oil resources. Conventional engines, such as SI and CI engines, have been required to enhance their fuel economy and reduce pollutants. Thus, alternative combustion methods that could increase efficiency and reduce emissions have been actively studied during the past decades. The HCCI engine is an example of an alternative internal combustion engine.

Within the HCCI operating range, fuel consumption is lower at steady state conditions than for conventional throttled SI engine operation.[1] The NOx emission is typically two orders of magnitude lower than for SI engines, and the level of PM is too low to be detected by modern smoke meters. Partial combustion products in the form of unburned hydrocarbons and carbon oxides can be removed with simple oxidation catalyst used in conventional engines. In spite of the listed advantages, HCCI engines have not been commercialized yet. There are several issues, such as an unstable combustion phase, excessive dependence on the wall temperature, a limited operating range and the indirect control method, unlike in conventional engines.

The improvement in efficiency can be explained in detail. Because the HCCI engine is operated in the wide open throttle (WOT) condition, the air/fuel ratio varies with the load, just as it does in a diesel engine. A conventional SI engine requires a stoichiometric air/fuel mixture for catalyst operation. This is because unburned hydrocarbons are more likely to react with molecular oxygen than with NOx. Consequently, a catalyst in SI engines operating under an excess air condition will fail to reduce NOx emissions. In contrast, as because HCCI operation results in low emissions of NOx, the engine can be operated successfully on non-stoichiometric mixtures. Because HCCI engines can be run under a wide open throttle (even for a part load operation), the pumping losses are significantly lower than in an SI operation at low and intermediate loads. This contributes to the relatively high efficiency of HCCI combustion at the part load operation. Other advantages over SI operation are faster combustion, which means that the process is more like theoretical constant volume combustion, and the reduction in heat loss due to the low combustion temperature. Under the condition with same baseline at 1500 r/min and 4.5 bar of net IMEP, the HCCI mode indicated 34.5% thermal efficiency, which is about 7% higher than that of SI mode (32.2%) in this study.

To commercialize HCCI engines, the operating range should be extended to the practical domain. In addition, it is vital to achieve a stable controlling mechanism in the extended range. For stable combustion, the operating range of an HCCI engine is significantly narrower than that of conventional engines. Because of the nature of combustion, a mixed charge ignites across the entire combustion chamber above the high load limit, thereby increasing the pressure rising rate, noise and vibration. Below the low load limit, the ignition delay is prolonged so that misfire can occur because of the low equivalence ratio and the low exhaust gas temperature. This narrow operating range of HCCI is one of the reasons that it cannot be adapted to real cars and even if it is adapted, the engine has to operate with both HCCI and SI modes, so its fuel economy effect is significantly small.

As HCCI engines operate within short combustion duration, deterioration of noise, vibration and durability of the engine occurs due to simultaneous ignition of the homogeneous mixture in the cylinder. To alleviate this stated problem, mixture composition should be formed to have different ignition timings depending on its position in the cylinder. Machrafi et al. described the effect on ignition timing and heat release rate of NO, CO, CH_2O in EGR of the previous cycle.[2] Shi et al. reported a phenomenon that the peak pressure of a cycle in the cylinder decreases as EGR portion increases.[3] In short, it is expected that the ignition timing and pressure rising rate will be different at each position in the cylinder when the mixture composition are different. This is called stratification which induces non-uniform mixture composition. Cho et al. evaluated the effect of internal EGR conditions, such as homogeneity and air-fuel ratio even temperature and rates using a rapid intake compression expansion machine.[4] Stratification is adapted usefully to expand the high load boundary of the operating range. In this study, external EGR stratification is used to form the stratified condition of the mixture in the cylinder during operation.

Furthermore, air-fuel ratio is one of the main factors which determines the start of combustion because if affect the auto-ignition delay significantly.[5] The difference in air-fuel ratio between certain positions in the cylinder can lead to different ignition timings; thus, fuel stratification strategy is adopted for this application. Brands et al. had investigated the effects of mixture stratification on ignition and combustion in a single cylinder engine with optical diagnostics.[6] They found the combustion apparently depends also on the small-scale inhomogeneity in the study. In this study, asymmetric injection strategy was used to form a stratified charge by injecting the fuel into just one of the two intake ports so that the fuel stratified condition could be achieved. Direct injection has stratification effect due to gasoline engine's distinctive characteristic which exhibits a strong tumble effect but cannot form fully homogeneous mixture in the cylinder.

Many researchers have tried to overcome the issues of HCCI engines. A homogeneous charge helps successful HCCI combustion, but a premixed charge has no potential for changing the combustion phasing. Moreover, nearly every local point ignites at the same time, causing a rapid pressure increase in the combustion chamber. Direct injection could offer solutions to these problems. Marriott et al. reported that combustion phasing could be changed by adopting direct injection.[7] Guohong et al. suggested two-stage direct injection to control combustion to manipulate combustion phasing.[8]

Urushihara et al. demonstrated early NVO injection to secure stability of engine operation at extremely diluted combustion.[9] In this research, it was found injecting whole fuel during exhaust compression advanced combustion phasing but also deteriorated thermal efficiency. To overcome, strategy of split fuel injection and reduction of reforming fuel amount which was injected in NVO period was adopted. Song et al. extended the low-load limit by injecting fuel during negative valve overlap (NVO). Direct injection and NVO injection strategy were also simultaneously used and it was possible to expand low-load limit drastically.[10] And also intensive numerical study was also performed in order to understand the chemical effect during recompression process.[11] Likewise, many researchers contributed their efforts in order to obtain higher efficiency and lower harmful emissions simultaneously with expansion of operating range when NVO strategy is used in gasoline HCCI engine. [12, 13, 14, 15, 16, 17, 18]

The development of the HCCI engine controller was studied with a stability analysis by Kang et al. In this study, operating set points, such as the EGR rate, the air-fuel ratio etc., were determined by a breathing process model and a combustion model.[19, 20] The engine controller was then designed to follow those set-points in the HCCI combustion mode. Matthews et al. developed a closed-loop controller, including an integrator, to control the load of an HCCI engine using an electromagnetic variable valve timing system with pre-heated intake air.[21] Recently, controlling HCCI combustion has become the main part of the HCCI research field. Lahti et al. developed a physics-based individual cylinder real-time model to predict the engine state and control the engine with the state information.[22] Jung et al. developed a closed-loop control logic for DME using external and rebreathed EGR.[23] As the result, stable stoichiometric HCCI operation was achieved with combustion-phasing retard with acceptable pressure rise rate in a higher load region. Likewise, recently, tremendous amount of control studies, mainly model based, have been conducted in gasoline HCCI field [24, 25, 26], not only to achieve stable combustion state but also to control combustion process during operation. [27]

However, previous research studies were only focused on the extension of the operating range or just on combustion control, meaning that the combined effect of the two parts was not studied thoroughly. Practically, primary reference fuels or unpractical fuels were widely used [28], and the ability to maintain constant combustion timing was emphasized. In this study, the stratification concept was applied to further enlarge the operating range and to operate the HCCI engine in a stable state using commercial gasoline fuel. In the control part, transition between different operating strategies was attempted.

2. Experimental set-up

2.1 Single cylinder research engine

A single cylinder gasoline engine was used for gasoline HCCI combustion. This engine was equipped with a continuously variable valve timing (CVVT) system, which was operated by hydraulic pressure from the engine oil pump at both the intake and exhaust cams. The adjustable valve timing range was 40 crank angle (CA) degrees for both the intake and exhaust valves. A low lift camshaft (2 mm) was installed instead of a normal camshaft (6 mm) for HCCI combustion. This low lift camshaft had a short duration compared to the normal camshaft.

Information about the cam position from the hall sensors at both camshafts was transferred to the ECU and CVVT unit actuators, which were operated by the ECU command signal. Detailed specifications and the condition of the test engine are shown in Table 1.

The detailed valve profiles and an example of the cylinder pressure history are shown in Figure 1. Unlike conventional SI engines, which use valve overlap, the NVO strategy was used to perform HCCI combustion. Thus, the cylinder pressure in HCCI combustion had two peaks because of the second compression during the exhaust process. The characteristics of combustion were analysed with AVL Indimodule System basically deriving from in-cylinder pressure pegged with the intake pressure.

A special intake manifold was constructed to adopt dual port fuel injection (PFI) injectors, which enabled following of the stratified charge experiment. The direct injector injected fuel directly into the cylinder when it was needed. The moisture of EGR gas was firstly eliminated through moisture filter. A compressor forced the EGR gas into the position in front of the intake valve. While forcing the EGR gas to move to the intake manifold, the temperature of the gas decreased because of heat transfer. Thus, the EGR gas was pre-heated to approximately 60°C by an EGR heater whose operating principle is based on a heat exchange process between the EGR gas and the hot engine coolant.

2.2 Equipment for engine control

The LabVIEW program and devices from National Instruments were used for real-time analysis and control. A USB X-6341 DAQ board was used, and the calculation was conducted with PXI-8196 based on the PXI-1042Q platform. The PXI-7833R FPGA board produced signals that were needed for high rate real-time control, such as fuel injection and spark ignition timing. Furthermore, by interlocking these devices with MOTEC PC-ECU M800, information was transferred between the devices.

3. Strategies for extending the operating range

In this study, several strategies were applied to extend the operating range of a target gasoline HCCI engine. The strategies were valve timing control, stratification of external EGR, fuel stratification using PFI and DI injectors including asymmetric injection, open valve injection using a PFI injector, and NVO injection. Among these strategies, EGR stratification, asymmetric injection and open valve injection were novel techniques.

It was hard to provide all information to the reader because the data was enormous. Every test points in the operating range have own parameters including air-fuel ratio, fuel injection parameters, residual gas fraction and EGR rate which are different with other points. However, lambda (λ) was maintained at least more than one for overall operating points in this study to maximize fuel economical advantage of HCCI. The amount of EGR used in all EGR strategies were less than 6% of the total volume of the mixture in order to secure sufficiency combustion stabilities in this study.

3.1 EGR stratification

External EGR was used to lower the ringing intensity and to extend the high-load boundary of the gasoline HCCI engine. In addition, stratification effect was added to the external EGR method in this study. Many previous studies have considered external EGR as a strategy for extending the operating range, but all previous studies assumed that the external EGR gas is uniformly mixed evenly across the combustion chamber. As a result, ignition delay was increased causing another simultaneous prolonged combustion.

K. Lee, one of the authors of this study, and other authors proved stratified charge through the asymmetric EGR induction and port fuel injection in same engine geometry with this experiment by using 3-dimensional computational method. And they showed that each gas mixture at other location in combustion chamber had different composition and auto-ignition characteristic. [29]

Extensive computational calculations to elicit local gas fraction and temperature were carried out with chemical mechanism within each unit volume. Consequently, it proves that the ignition in the chamber does not occur simultaneously because ignition delay would be differentiated up to four times according to the position in the combustion chamber. Furthermore, stratification can make as much as twice the differences in maximum heat release rate (HRR) and prolong combustion duration. Maximum HRR and pressure rising rate (PRR) are lowered by stratification and it is appeared as the prevention of simultaneous combustion and prolongation of combustion. As mentioned before, stratification and prolonged combustion phenomenon cannot be considered independently. Hence, stratification of the external EGR gas in the cylinder was proposed for higher load boundary.

In this study, the supply line of the external EGR gas was divided into two lines, which were connected to the intake port. The external EGR supply lines were long enough to make sure the tip was located in front of each intake valve. By closing or opening the control valve, the direction of the supplied EGR gas could be selected. External EGR i.e. e-EGR strategy means supplying the EGR gas uniformly to both intake ports. On the contrary, stratified EGR i.e. s-EGR strategy was realized by providing the EGR gas to just one of both valves. Temperature of the external EGR gas was an important factor influencing the characteristics of HCCI combustion. In this study, as it was interpreted above, engine coolant was used to prevent excessive cooling of the external EGR gas.

Figures 2(a) to (c) show results of the application of the external EGR and stratified EGR conditions at 2000 r/min and 3.4 bar of IMEP. In this study, all the load values are indicated with net IMEP. The figures compare the in-cylinder pressure, rate of heat release (ROHR), mass fraction burned 50 (MFB50), and maximum IMEP of three cases which are no external EGR case, uniform external EGR (e-EGR) and stratified external EGR (s-EGR) cases. The combustion duration in the stratified EGR case was prolonged more than 50% compared to the uniform external EGR case within the criterion of ROHR. And Figure 2(c) shows the timing of the MFB50 of the two cases with the base case. The MFB50 of the stratified EGR case had 14.2 aTDC CA which was the mean value of a hundred cycles. This was retarded relative to that of the uniform external EGR case whose mean value of a hundred cycles was 12.5 aTDC CA.

Figure 2(d) shows the result of a high-load boundary at 1500 r/min and 4.02 bar of IMEP. The maximum load was shifted to a higher value, 4.42 bar of IMEP, by adopting the EGR stratification strategy. In the case of 2000 r/min, the effect was larger than in the case of 1500 r/min. The base value was 2.87 bar of IMEP, so the high load boundary improved by 19.7% when external EGR was used and improved by 25.8% when stratified EGR was used.

As interpreted in the introduction, prolonging ignition delay was derived from the chemical effect of EGR itself. Spontaneous ignition in the cylinder could be prevented by forming different EGR concentration so that the pressure rising rate can be decreased per cycle. As ringing intensity has to be under 5MW/m^2 during the operation, EGR was introduced to decrease RI values; thereby, in-cylinder condition could afford to ignite more fuel than before. As a result, the high load boundary can be improved and stratification strategy has an additional effect as aforementioned.

3.2 Application of fuel stratification by asymmetric injection using a PFI injector

Because the air-fuel ratio is a significant factor of HCCI characteristics, the stratification of fuel could affect the start of combustion. A PFI injector is generally used to make a premixed charge, but it was paradoxically used for the stratification of the mixture in this study.

Two PFI injectors were installed to the intake manifold, which was specially designed for EGR and fuel stratification. The fuel spray was aimed at the intake valve stem, and the injector had a relatively small Sauter Mean Diameter (SMD), below 50 μm. If only one injector was used for fuel injection, fuel stratification was produced because of the tumble motion of the fresh air. The effect of asymmetric injection was explained in the introduction. Figures 3(a) to (c) show the results of asymmetric injection. The experimental conditions were 1500 r/min of engine speed and 4.0 bar of IMEP. Figure 3(a) shows a clear difference between the two cases. The maximum pressure of the stratified PFI case was lowered by 3.5 bar than the dual PFI case which had an almost like premixed charge. This tendency can be observed in Figure 3(b). The combustion phase was significantly prolonged and the peak of heat release rate (HRR) was decreased. The MFB50 of the stratified PFI case was delayed by 5 CA degrees.

K. Lee et al. performed a numerical study about the effect of stratified mixture on combustion to examine influences on HRR, PRR and etc. from combustion during expansion stroke due to delayed ignition.[29] In this past study, stratified chamber was formed by asymmetric fuel injection and five representative points were selected to calculate ignition delay at each point using temperature, pressure and gas composition of corresponding point.

The difference in maximum HRR between representative points was up to two times and combustion duration up to two and a half times which indicated asynchronous combustion. The result from past paper indicated simultaneous combustion which is the one of technical barriers in commercialization of HCCI, could be mitigated with stratification and concurred with experimental result, Figure 2 and 3 of this paper study. This phenomenon leads to prolongation effect of the combustion duration which is similar with the EGR stratification cases. Consequently, higher load boundary can be improved.

3.3 Fuel stratification by open valve injection using a PFI injector

Generally, PFI is conducted before the intake stroke to provide sufficient time for the premixed charge because the droplets injected from a PFI injector are not fully vaporized because of their size. In this study, a specially manufactured PFI injector was used. The number of injection holes was 18, whereas the normal injector had 4 holes. The fuel was pressurized up to 6 bar, which is twice that of normal PFI injectors. Thus, the SMD of the specially manufactured PFI injector was less than half of that of conventional PFI injectors.

To achieve a stratified charge, fuel was injected at 300 bTDC CA which was followed after the intake valve opening (IVO) at 312 bTDC CA. The fuel injected from the PFI injector was then charged into the cylinder directly. This strategy is called open valve injection (OVI) in this study. The temperature of the charge decreased because of the latent heat of the fuel, and more fresh air could be supplied to the cylinder compared to a conventional engine equipped with PFI. Under a low engine speed, DI could be inadequate because of its long penetration length, small mass of air and weak tumble motion. It was found that unstable combustion occurred when DI was performed on the target engine at below 2000 r/min.

3.4 Simultaneous application of stratified external EGR and stratified fuel charge

The EGR and fuel injection strategies both had two directional selections in the intake manifold. EGR and fuel could be induced into the same port or through different ports. The former is called the in-line strategy, and the latter is called the cross strategy in this study. The high load boundary of each case is plotted in

Figure 4. The IMEP of the OVI case was higher than that of closed valve injection (CVI) because of the charge cooling effect. The cross case enabled a higher IMEP to be achieved than the inline case. In the inline case, fuel was delivered into the cylinder with external EGR, so less fuel evaporated during the intake stroke. Un-vaporized fuel diffused to the other side, so the stratification effect decreased.

3.5 NVO injection and split injection strategy using a DI injector

The advantage of direct injection was the flexibility of the injection timing. Injection during NVO was considered an effective method to extend the low load limit of the gasoline HCCI engine. [16-18] Recompressed fuel during NVO was reformed into intermediate products, and it chemically enhanced the reaction rate.

The DI injection timing was 400° BTDC, and in the NVO strategy, combustion was very stable because of the advanced combustion timing. Thus, the equivalence ratio could be lowered below 0.7 without abnormal combustion. In the region between the NVO strategy and the base operating zone, split injection was attempted. In this strategy, a part of the fuel was injected at NVO by a DI injector to enhance the reactivity of combustion, and the rest of the fuel was injected by a DI injector or PFI injector according to the operating point.

3.6 Operating range and emissions of a gasoline HCCI engine

A gasoline HCCI engine has a narrow operating range because the start of combustion should happen at a reasonable place near the firing TDC. Various strategies have been adopted to control the ignition delay. Proper ignition delay means that stable combustion is possible for a broader operating range without an excessive ringing intensity (RI) or a large coefficient of variation (COV) of the IMEP values. These RI and COV values will be interpreted in section 4.

The broadened operating range is plotted in Figure 5. The shape of the point in the figure represents the strategy that was used. The range of the base operation is represented in blue. Only 0.3 ~ 0.5 bar ranges were covered at each speed condition for the target engine. A load range lower than the low load boundary of the base operation is marked as hatched green. In this region, both the NVO injection and the split injection strategies were employed; hence, the chemical reaction was enhanced by fuel injection during the recompression stroke. A load range higher than the high-load boundary of the base operation is highlighted as hatched red. The strategies used include open valve injection, fuel stratification by direct injection and asymmetric PFI, external EGR, and EGR stratification. By adopting the aforementioned strategies, the operating range was greatly extended compared to the base operation. The possible load range at the 1500 r/min condition was extended from 3.6~4.0 bar to 1.9~4.5 bar of IMEP, and the same tendency could be observed in the higher speed ranges.

The emission characteristics of NOx depend on load. As the load increases, the NOx emission also increases. However, it is under 200 ppm, which is considerably low compared to the conventional SI engine at steady state operating conditions. In the low load region, under three bar of IMEP, the engine-out NOx emission was measured under 30 ppm. Load limits under the increase of the load have two limitations, ringing intensity and tremendous NOx emission. If the latter limitation is defined as the point that NOx emission of HCCI engine exceeds conventional SI engine emission level, since the former reveals prior to the latter limitation while increasing the engine load, it is concluded to focus on combustion stabilization and knock mitigation as a factor of limitation for this study. Thus, COV and RI were used to implement those limitations.

4. Combustion control

There are two real-time based indexes to judge the HCCI combustion phase. The values were mainly derived from the fuel injection mass and the CVVT cam phase, which are the main variables for control.

First, the ringing intensity, described in equation (1), was used to determine the engine noise and knock behavior of combustion. Ringing intensity is firstly introduced by Eng, J. [30] as the left form of equation (1) using various constants and Yun, H. [31] had used the index in a single cylinder engine as the right simplified form. This index is related to the peak pressure (PP), engine speed (unit of r/min) and maximum pressure rising rate (MPRR) per angle in a cycle. In this study, the objective target value was set to be under 5 MW/m^2 for 100 cycles' mean value of the RI, and an experiment was conducted to achieve this goal.

$$\mathrm{RI} = \frac{1}{2\gamma}\frac{(\beta\frac{dP}{dt_{max}})^2}{P_{max}}\sqrt{\gamma R T_{max}} \approx 2.88\ \times 10^{-8}\ \times \frac{(\mathrm{MPRR}\ \times \mathrm{Speed})^2}{\mathrm{PP}}\ \ (\mathrm{MW/m^2}) \tag{1}$$

Second, the coefficient of variation of IMEP (i.e., COV) was used. As described in equation (2), this value is related to the standard deviation (σ) of the IMEP values and a hundred cycle mean IMEP value. A large COV value means that HCCI combustion is unstable. In this study, the COV was calculated from the 100 cycle IMEP data and controlled such that it did not exceed 5%.

$$\mathrm{COV} = \frac{\sigma}{\mathrm{Mean\ IMEP}} \times 100\ \ (\%) \tag{2}$$

Through these two calculated values and the IMEP value, the control system could analyze the present state of the combustion phase in real-time. The operating region with the EGR supply was not considered in the control part of this study. Because of the lag of the cycle-by-cycle variation in the EGR rate, it was not sufficient to follow the transient control speed, which varies over 50 r/min in a second.

From the following experiments, the injection duration and injection timing of the PFI and DI injectors were determined, so the map data were organized for each load and speed. Furthermore, the spark plug duration and timing were determined to prevent the discontinuity of the HCCI combustion when misfire caused by a low residual gas temperature occurred. These map data were put in the control logic, and the logic was driven with the LabVIEW program from National Instruments.

The controller controlled three parts, fuel injection, the spark plug and the cam phase. The fuel injection part consisted of six variables including the injection duration and timing of PFI and the variables of the first and second DI. The spark plug part consists of two variables, the duration and timing of the plug signal. These eight variables, which are shown in Figure 6, were controlled by the LabVIEW program. Likewise, the CVVT cam phase was controlled by LabVIEW logic with an analogue voltage output.

4.1 Control logic

The purpose of control logic was to achieve stabilized HCCI combustion with a simultaneous chasing target load. $\vec{T}$, represented in equation (3) and Figure 7, is a continuously compensated engine input matrix value that was derived from the maps from the input target load value to follow the target load value.

In $\vec{T}$, the first six values on the right hand of equation (3) determined the fuel control in the fuel injection map, such as the duration and timing at a certain speed. Furthermore, the cam phase value was also compensated for from the input target load, and the analogue voltage value was transferred from the logic to the ECU to shift the CVVT cam phase using map data that was already uploaded in the ECU. A smaller cam phase value means that the exhaust cam phase is retarded and that the intake cam phase is advanced; a larger cam phase means the opposite. All values of $\vec{T}$ are shown in Figure 6 and all nine values can be seen in Figure 6.

$$\vec{T} = \begin{bmatrix} PFI\ Timing \\ PFI\ Duration \\ DI1\ Timing \\ DI1\ Duration \\ DI2\ Timing \\ DI2\ Duration \\ Spark\ Timing \\ Spark\ Duration \\ Cam\ phase \end{bmatrix} \tag{3}$$

The core logic of the control loop is interpreted below. The engine was controlled based on this control loop in real-time, and the value k in Figure 7 is the real-time proportional gain value, which helped the target load chase the input value. Furthermore, the compensator of the controller, which is shown in Figure 7, functioned as an arbiter, so it judged the present combustion phase and produced a compensated value for each different combustion phase.

All experimental cases had different combustion phases. As it was interpreted in the previous section, the controller controlled the engine with two variables, the COV and the RI. By checking the COV and RI values, the compensator detected the present combustion phase and judged whether to shift the combustion phase to a richer mixture area or a leaner area. Furthermore, when misfire occurs, the logic detects the misfire and outputs the spark duration value for the next five cycles to ignite the mixture.

The case of COV > 5% and RI > 5% can hardly exist due to its own counter characteristics which represent the combustion instability and knocking. Nevertheless, as it can be derived from the definition of a COV value, a COV value can soar when there is change in the engine load. Thus, this case can be interpreted as the knock occurring phase right after the target load changes. The logic targets the cam phase to decrease the knocking phenomena. No variable except the cam phase variable is changed by the logic.

In the case of COV > 5% and RI ≤ 5 MW/m2, the logic determines that the phase is now in low knock occurring situation but also in low combustion stability. Thus, the logic targets the cam phase to obtain more fresh air and increase the amount of fuel injection. The logic varies the cam phase variable and the six variables which are directly related with the fuel input amount.

COV ≤ 5% and RI > 5 MW/m2 shows that the combustion phase is stable but knocking occurs due to a large amount of fuel injection. Thus, the logic targets to lower the amount of fresh air by altering the cam phase and decrease the amount of fuel. Likewise, the logic varies the cam phase variable and the six variables which are directly related with the fuel input amount.

In the case of COV ≤ 5% and RI ≤ 5 MW/m2, HCCI combustion is stabilized. However, the actual controlled load of the combustion could be different from the target load which was inputted at the first time due to decrease in the wall temperature and many other time dependent parameters. Thus, in this case, the logic controlled the engine to follow the initial input load. However, the actual controlled load of the combustion could be different from the target load, which was first input because of a decrease in the wall temperature and many other time dependent parameters. Thus, in this case, the logic controlled the engine to follow the initial input load.

4.2 Control results

Based on the developed control logic and map data, HCCI combustion was successfully controlled in the operating range. Except for in the region with the EGR addition, control while moving between operating points was possible. If two points belong to regions operated by a different strategy, the control logic varied the control factors to shift the first operating point to the other domain.

Figure 8(a) shows the load control result. At a constant speed, 1500 r/min, the target load was changed from 3.8 to 2.9 bar. The split injection strategy using DI and PFI was sequentially employed at 3.8 bar of IMEP and a 1500 r/min point. However, direct fuel injection was adopted alone at 2.9 bar of IMEP and a 1500 r/min point. According to the control logic, PFI injector stopped fuel injection, and the DI injector increased the amount of fuel after the load-shift command. As shown in the figure, the load was changed instantaneously such that it could be considered real-time control. The controller is always switched on and keeps the load controlled even before changing the target load. When the target load is changed, the controller manipulates the engine control variables including cam phasing and fuelling quantity to follow and maintain the changed input load. During this time, the controller detects COV and RI, and continues to control the both values to be under five.

Figure 8(b) also presents load control at a constant speed. The load should increase from 2.2 to 3.8 bar of

IMEP. The transition to a higher load was operated by a different strategy, and the stability of combustion decreased in many cases compared to the decremental load change case. Therefore, the logic complemented the unstable phenomena of the incremental load transition by operating the spark plug during the first 5 cycles after the load-shift command to minimize misfired cycles and achieve stability. However, it was hard to prevent misfire during a large incremental load change, and it seemed that the energy of the residual gas from the previous cycle was insufficient to initiate combustion.

Continuous load changes appear in Figure 8(c). The target load varied from 3.8 to 2.0 bar of IMEP and then to 2.9 bar of IMEP with a short interval of approximately 40 cycles. The operating strategies for each target load were different from each other, such as in the cases discussed above, and the control logic successfully altered the operating strategy twice in a short period.

Figure 8(d) shows the ability to maintain a constant load against a decrease in the speed from 1750 to 1500 r/min. As the engine speed varied, the operating parameters, such as the amount of air and the start of combustion, were changed, so the control logic made it possible to maintain a constant load. The controller controls the load but the engine speed is controlled manually by spinning the knob of the engine speed controller. In this case, the target load was fixed and controlled with 2 bar of IMEP by the controller continuously, and the engine speed controller which is driven by spinning the control knob controlled the engine speed from 1500 r/min to 1750 r/min. Figure 8(e) shows a constant load against an increase in the speed from 1500 to 1750 r/min. Regardless of an increase or decrease in the engine speed, the control logic could effectively maintain a constant load. Figure 8(f) shows the case in which both the load and speed change.

5. Conclusion

A gasoline HCCI engine has many advantages compared to conventional engines. The emissions and efficiency are especially outstanding. However, because gasoline HCCI combustion is controlled by chemical kinetics, controlling combustion is very difficult, unlike conventional gasoline or diesel engines, which have a direct control tool. Furthermore, because of its combustion principle, an HCCI engine should operate within a limited range.

The objectives of this study were twofold. First, the operating range of gasoline HCCI combustion was extended. To extend the operating range, several strategies were tested. Among these strategies, EGR stratification, asymmetric injection and open valve injection were novel techniques. Second, control logic for an HCCI engine was developed. The developed logic allows controller to choose proper strategies to operate engine without sever problem of knocking-like or misfire conditions. Besides, this logic can make to the current load to chase the target well, even if it is in transient condition, which engine load and speed are changing drastically between different values.

1. Based on a naturally aspirated gasoline HCCI engine, the operating range was extended by a compound strategy. The compound strategy involved valve timing swing, fuel stratification using direct injection, asymmetric PFI and open valve PFI, EGR stratification by asymmetric induction and NVO injection. The stratification strategy of fuel and EGR is a unique system for extending the high load limit.
2. Map based control was conducted using determined values from the compound strategy. The aim of this control was transient performance that could operate in an enlarged operating range. Although the control could not include the operating region with EGR because of its characteristic, it showed good performance. In addition, combustion stability was secured and knocking characteristics was suppressed by utilizing COV and RI such that load control was successfully achieved.

Even if it is necessary to study further to meet the conventional operating range requirements for gasoline HCCI combustion, nowadays new concepts of high efficiency engines are being developed using state-of-the-art technologies such as new intake cooling systems or higher tumble inducing intake ports. Thus, further operating range expansion would be possible. However, as it is revealed in this study, there are combustion instability problems while increasing the load within HCCI operating range. Even though it was minimized with the ignition of the spark plug 5 cycles prior to the load change, it should be required to study further due to drivers need the load to be changed immediately in actual vehicles.

Acknowledgements

The single cylinder engine and port fuel injectors used in this study were donated by Hyundai Motor Company and Hyundai-KEFICO. The authors would like to express gratitude to them.

References

[1] Stanglmaier, H. and Roberts, E., Homogeneous Charge Compression Ignition (HCCI): Benefits, Compromises, and Future Engine Applications, SAE (1999), doi:10.4271/1999-01-3682.

[2] Marchrafi, H., Cavadias, S. and Guibert, P., An experimental and numerical investigation on the influence of external gas recirculation on the HCCI autoignition process in an engine: Thermal, diluting, and chemical effects, Combustion and Flame (2008), doi:10.1016/j.combustflame.2008.05.001.

[3] Shi, L., Cui. Y, Deng, K. et al., Study of low emission homogeneous charge compression ignition (HCCI) engine using combined internal and external exhaust gas recirculation (EGR), Energy (2006), doi:10.1016/j.energy.2005.12.005.

[4] Cho, G, Jeong, D, Moon, G et al., Controlled auto-ignition characteristics of methane–air mixture in a rapid intake compression and expansion machine, Energy (2010), doi:10.4271/2006-01-0415.

[5] Bradley, D. and Kalghatgi, G., Influence of autoignition delay time charactertistics of different fuels on pressure waves and knock in reciprocating engines, Combustion and Flame (2009), doi:10.1016/j.combustflame.2009.08.003.

[6] Brands, T., Hottenback, P., Koss, H. et al., Effects of Mixture Stratification on Ignition and Combustion in a GCAI Engine, SAE (2014), doi:10.4271/2014-01-1270.

[7] Marriott, C. and Reitz, R. Experimental Investigation of Direct Injection-Gasoline for Premixed

Compression Ignited Combustion Phasing Control. SAE (2002), doi:10.4271/2002-01-0418.

[8] Guohong, T., Zhi, W., Jianxin, W. et al., HCCI Combustion Control by Injection Strategy with Negative Valve Overlap in a GDI Engine, SAE (2006), doi:10.4271/2006-01-0415.

[9] Urushihara, T., Hiraya, K., Kakuhou, A. et al., Expansion of HCCI Operating Region by the Combination of Direct Fuel Injection, Negative Valve Overlap and Internal Fuel Reformation, SAE (2003), doi:10.4271/2003-01-0749.

[10] Song, H, Padmanabhan, A, Kaahaaina, N et al., Experimental study of recompression reaction for low-load operation in direct-injection homogeneous charge compression ignition engines with n-heptane and i-octane fuels, International Journal of Engine Research (2009), doi:10.1243/14680874 JER03309.

[11] Song, H. and Edwards, C. Understanding chemical effects in low-load-limit extension of homogeneous charge compression ignition engines via recompression reaction, International Journal of Engine Research (2009), doi:10.1243/14680874JER03409.

[12] Canakci, M. and Reitz, R., Experimental optimization of a direct injection homogeneous charge compression ignition gasoline engine using split injections with fully automated microgenetic algorithms, International Journal of Engine Research (2003), doi:10.1243/146808703762826642.

[13] Shi Y. and Reitz, R. Optimization of a heavy-duty compression-ignition engine fueled with diesel and gasoline-like fuels. Fuel (2010), doi:10.1016/j.fuel.2010.02.023.

[14] Lee, C. and Lee, K., An experimental study of the combustion characteristics in SCCI and CAI based on direct-injection gasoline engine, Experimental Thermal and Fluid Science (2007), doi:10.1016/j.expthermflusci.2006.12.003.

[15] Standing, R., Kalian, M., Ma, T. et al., Effects of Injection Timing and Valve Timings on CAI Operation in a Multi-Cylinder DI Gasoline Engine, SAE (2005), doi:10.4271/2005-01-0132.

[16] Hunicz, J., An experimental study of negative valve overlap injection effects and their impact on combustion in a gasoline HCCI engine, Fuel (2014), doi:10.1016/j.fuel.2013.09.079.

[17] Hunivz, J., Kordos, P. and Ignaciuk, P., Experimental Investigation into Thermal and Chemical Effects of Negative Valve Overlap Injection in a Gasoline HCCI Engine, SAE (2014), doi: 10.4271/2014-01-2660.

[18] Lee, J. and Song, H., Experimental and Computational Study on Recompression Reaction of Pilot-Injected Fuel During Negative Valve Overlap in a Gasoline-fueled Homogeneous Charge Compression Ignition Engine, International Journal of Automotive Technology (2014), doi:10.1007/s12239-014-0111-x.

[19] Kang, J., Chang, C., Chen, J. et al., Concept and Implementation of a Robust HCCI Engine Controller. SAE (2009), doi:10.4271/2009-01-1131.

[20] Kang, J.. Sensitivity Analysis of Auto-ignited Combustion in HCCI Engines. SAE (2010), doi: 10.4271/2010-01-0573.

[21] Matthews, J., Santoso, H. and Cheng, W., Load Control for an HCCI Engine. SAE (2005), doi: 10.4271/2005-01-0150.

[22] Lahti, J. and Moskwa, J., Dynamic Engine Control for HCCI Combustion. SAE (2012), doi:10.4271/2012-01-1133.

[23] Jung, D. and Iida, N., Closed-loop control of HCCI combustion for DME using external EGR and rebreathed EGR to reduce pressure-rise rate with combustion-phasing retard, Applied Energy (2015), doi:10.1016/j.apenergy.2014.10.085.

[24] Bidarvartan, M., Thakkar, V. and Shahbakhti, M., Grey-box Modeling and Control of HCCI Engine Emissions, American Control Conference (2014), doi:10.1109/ACC.2014.6859420.

[25] Ebrahhimi, K., Koch, C. and Schramm, A., A Control Oriented Model with Variable Valve Timing for HCCI Combustion Timing Control, SAE (2013), doi:10.4271/2013-01-0588.

[26] Ebrahhimi, K., Koch, C., Model Predictive Control for Combustion Timing and Load Control in HCCI Engines, SAE (2015), doi:10.4271/2015-01-0822.

[27] Das, P., Subbaroa, P. and Subrahmanyam, J., Control of combustion process in an HCCI-DI combustion engine using dual injection strategy with EGR, Fuel (2015), doi:10.1016/j.fuel.2015.07.009.

[28] Kalghatgi, G., Auto-Ignition Quality of Practical Fuels and Implications for Fuel Requirements of Future SI and HCCI Engines, SAE (2005), doi:10.4271/2005-01-0239.

[29] Lee, K., Min, K.. Study of a Stratification Effect on Engine Performance in Gasoline HCCI Combustion by Using the Multi-zone Method and Reduced Kinetic Mechanism, SAE (2009), doi:10.4271/2009-01-1784.

[30] Eng, J., Characteriziation of Pressure Waves in HCCI Combustion, SAE (2002), doi:10.4271/2002-01-2859.

[31] Yun, H., Wermuth, N. and Najt, P., Extending the High Load Operating Limit of a Naturally-Aspirated Gasoline HCCI Combustion Engine, SAE (2010), doi:10.4271/2010-01-0847.

Bore × Stroke	86 × 86 mm
Compression ratio	10.5 : 1
Displaced volume	498 cm^3
Port fuel injection	6 bar / 18 holes
Direct injection	70 bar / 4 holes
PFI timing	CVI: 540 bTDC CA OVI: 300 bTDC CA
DI timing	bTDC 400 CA
Throttle position	WOT
Intake temperature	Room temperature
Standard Valve Timing	IVO: 312 bTDC CA IVC: 140 bTDC CA EVO: 160 aTDC CA EVC: 296 aTDC CA

Table 1. Salient features of the single cylinder test engine.

WOT: Wide Open Throttle
IVO: Intake Valve Opening
IVC: Intake Valve Closing
EVO: Exhaust Valve Opening
EVC: Exhaust Valve Closing

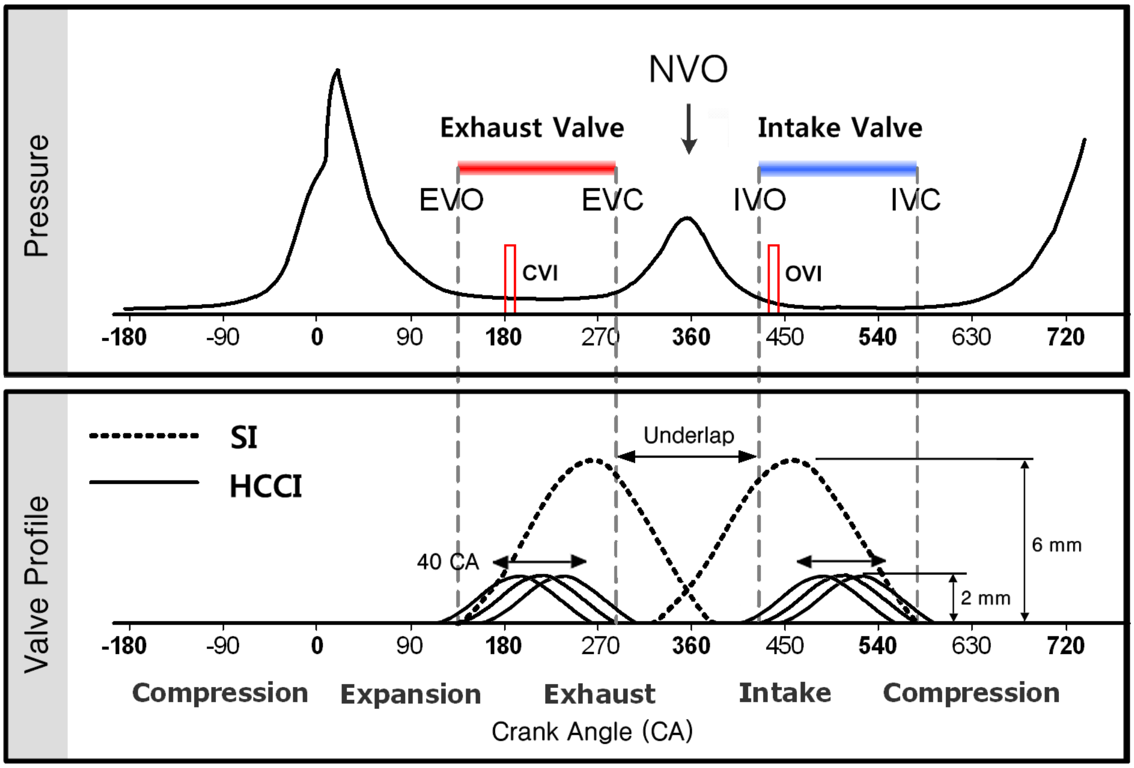


Figure 1. Valve operating strategy for HCCI combustion.

NVO: Negative Valve Overlap
EVO: Exhaust Valve Opening
EVC: Exhaust Valve Closing
IVO: Intake Valve Opening
IVC: Intake Valve Closing
CVI: Closed Valve Injection
OVI: Opened Valve Injection

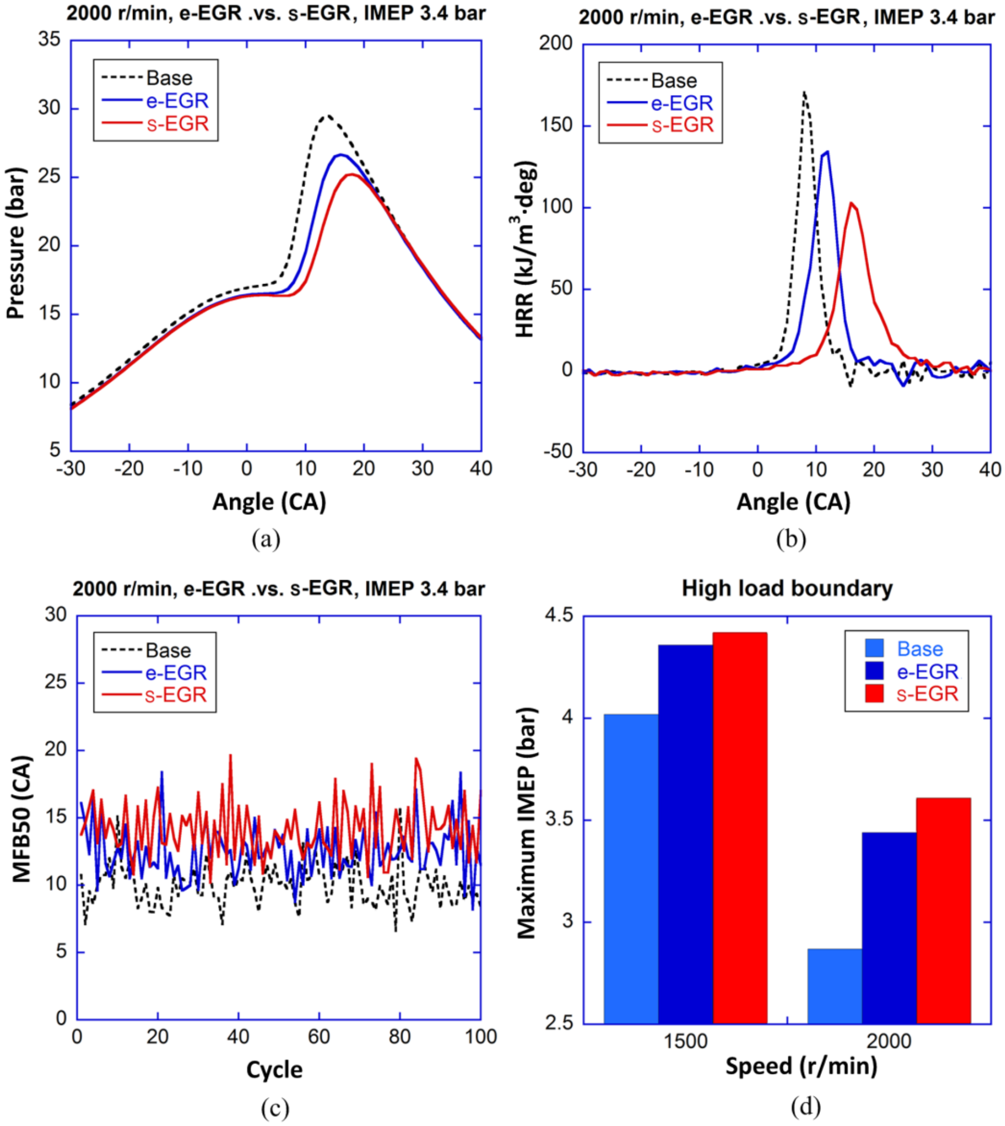
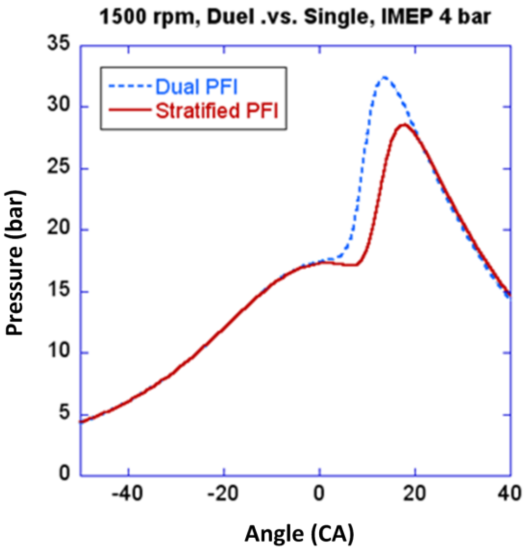
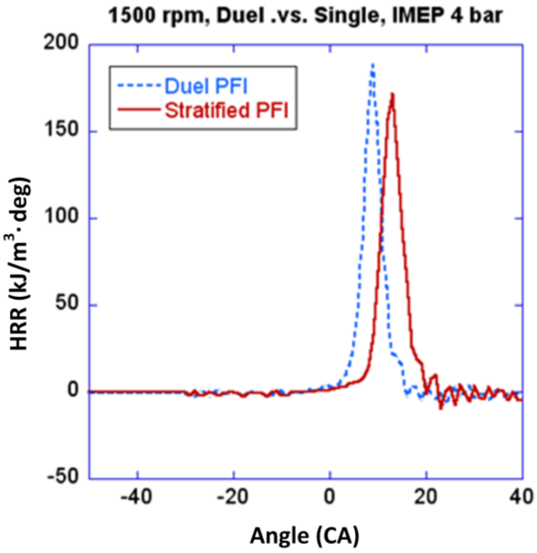


Figure 2. Results of the experiment using stratified EGR; (a) effect of stratified EGR on the pressure curve; (b) effect of stratified EGR on the heat release rate; (c) effect of stratified EGR on the MFB50; (d) high load boundary according to EGR strategies.

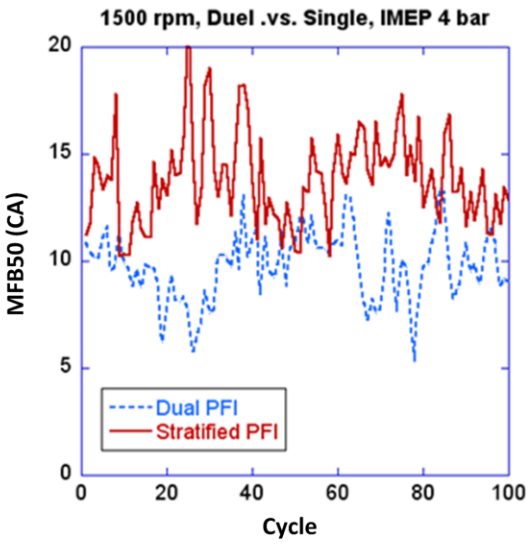
eEGR: external Exhaust Gas Recirculation
s-EGR: stratified Exhaust Gas Recirculation
IMEP: Indicated Mean Effective Pressure
HRR: Heat Release Rate



(a)



(b)



(c)

Figure 3. Results of the experiment using fuel stratification; (a) effect of asymmetric PFI on the pressure curve; (b) effect of asymmetric PFI on the heat release rate; (c) effect of asymmetric PFI on the MFB50

IMEP: Indicated Mean Effective Pressure
PFI: Port Fuel Injection
HRR: Heat Release Rate
MFB50: Mass Fraction Burned 50%

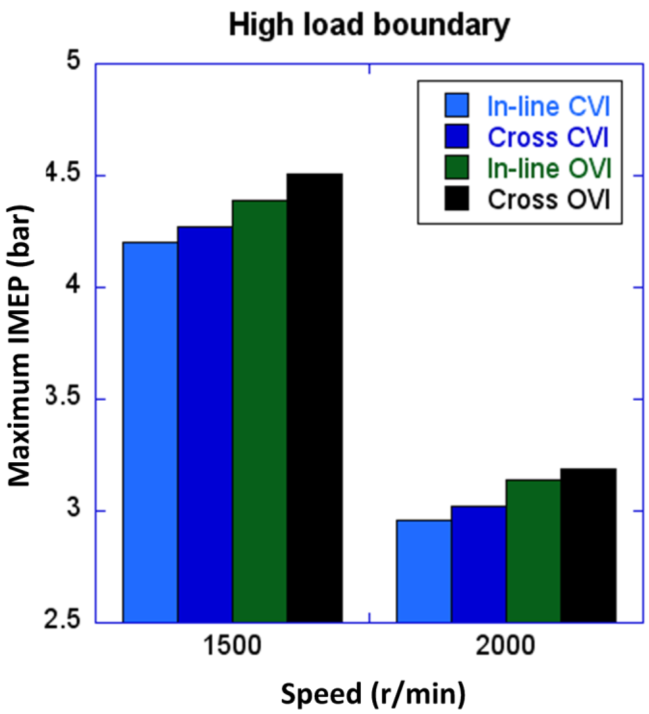


Figure 4. Result of the high load boundary in the simultaneous application of stratified external EGR and stratified fuel charge.

CVI: Closed Valve Injection
OVI: Open Valve Injection
IMEP: Indicated Mean Effective Pressure

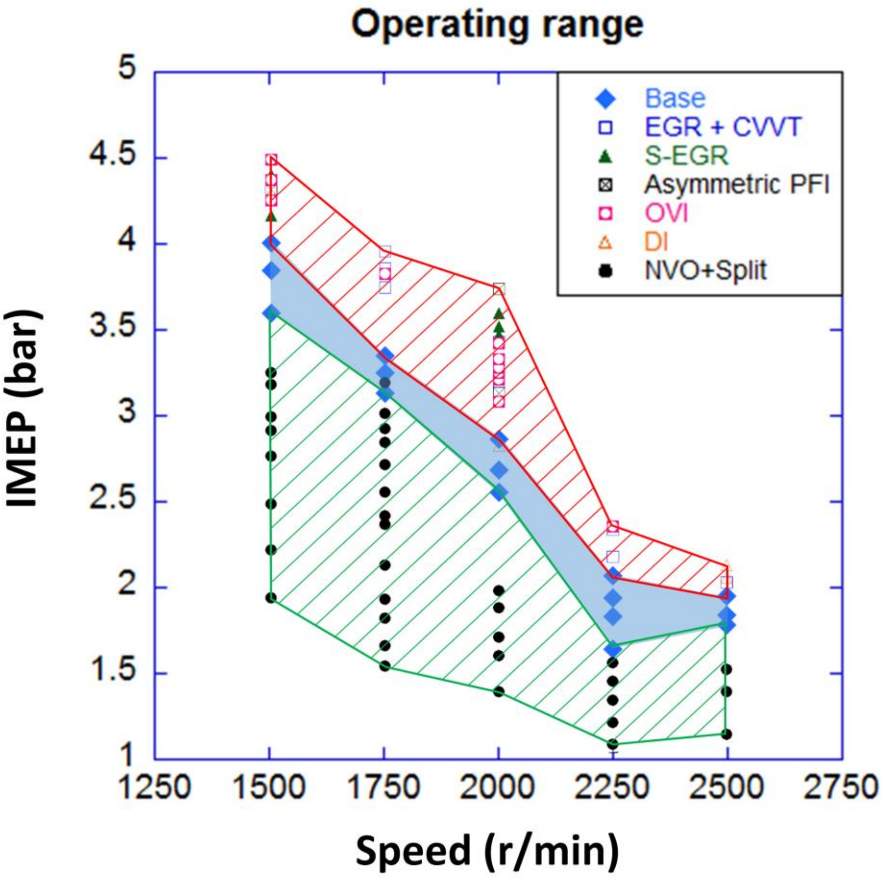


Figure 5. Extended operating range

IMEP: Indicated Mean Effective Pressure
EGR: Exhaust Gas Recirculation
CVVT: Continuously Variable Valve Timing
S-EGR: Stratified Exhaust Gas Recirculation
PFI: Port Fuel Injection
OVI: Open Valve Injection
NVO: Negative Valve Overlap

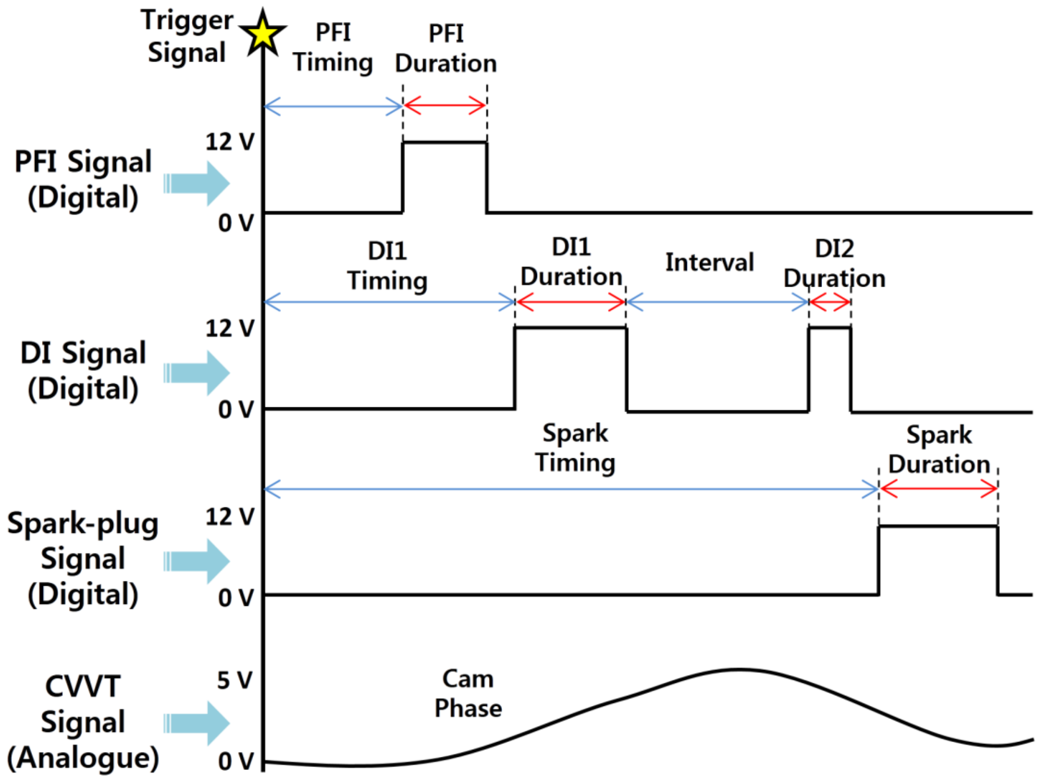


Figure 6. Signal schematic diagram of the control variables.

PFI: Port Fuel Injection
CVVT: Continuous Variable Valve Timing

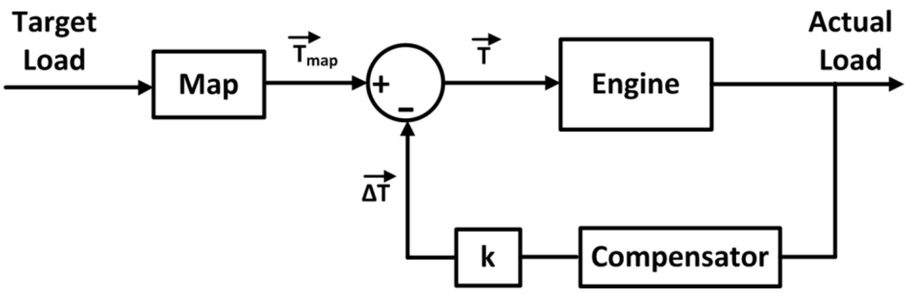


Figure 7. HCCI controller.

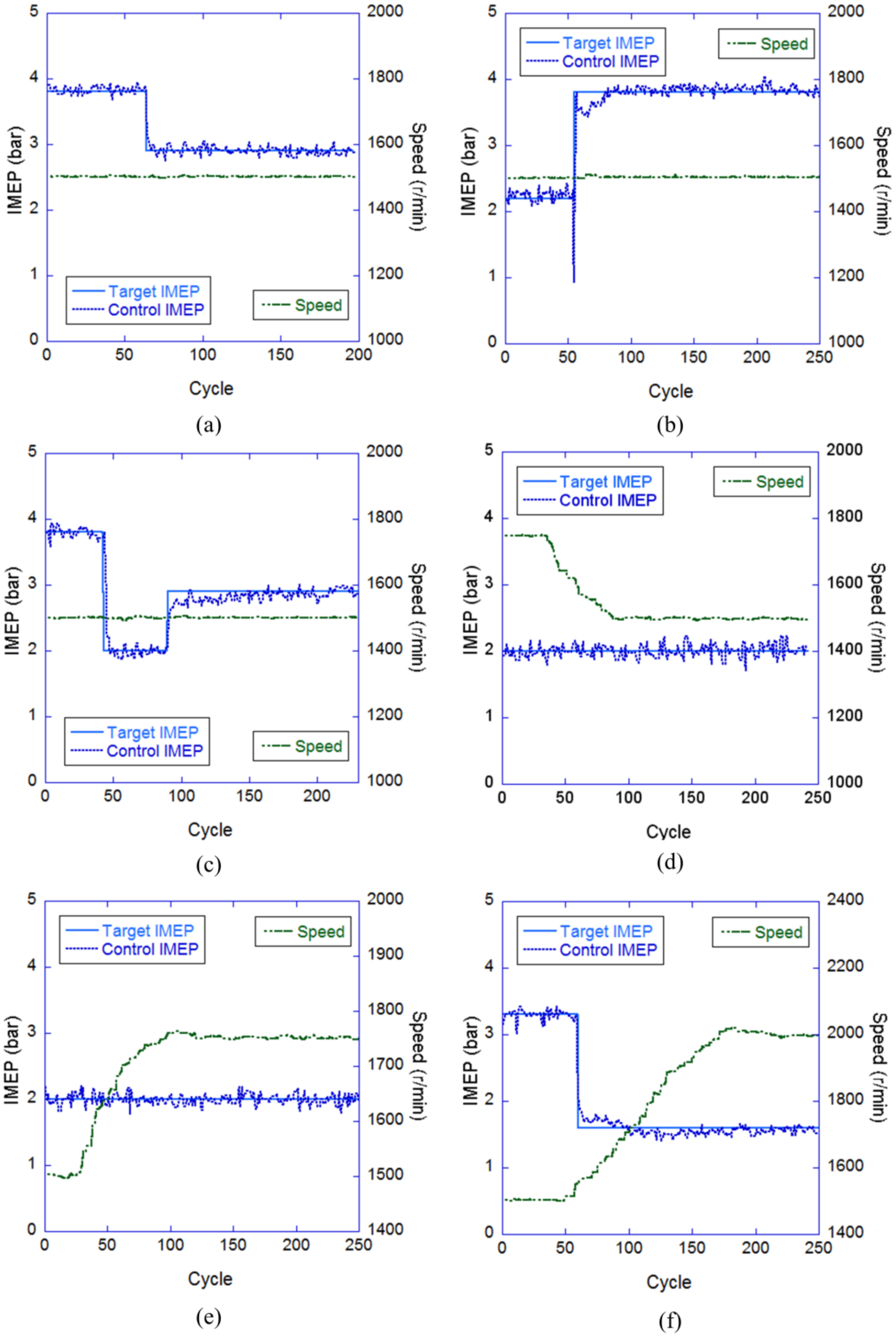

Target IMEP
Control IMEP
Speed
IMEP (bar)
Speed (r/min)
Cycle
(a)
(b)
(c)
(d)
(e)
(f)

Figure 8. Control results; (a) 3.8 to 2.9 bar of IMEP, decremental load control at a constant speed; (b) 2.2 to 3.8 bar of IMEP, incremental load control at a constant speed; (c) 3.8, 2.0 to 2.9 bar of IMEP, compound load control at a constant speed; (d) 1750 to 1500 r/min, decremental speed control at a constant load; (e) 1500 to 1750 r/min, incremental speed control at a constant load; (f) 3.3 to 1.6 bar of IMEP load control at 1500 to 2000 r/min.

IMEP: Indicated Mean Effective Pressure