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# The Effect of Ethanol Injection Strategy on Knock Suppression of the Gasoline/Ethanol Dual Fuel Combustion in a Spark-Ignited Engine

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## Abstract

Ethanol is becoming more popular as a fuel component for spark-ignited engines. Ethanol can be used either as an octane enhancer of low RON gasoline or splash-blended with gasoline if a single injector is used for fuel injection. If two separate injectors are used, it is possible to inject gasoline and ethanol separately and the addition of ethanol can be varied on demand. In this study, the effect of the ethanol injection strategy on knock suppression was observed using a single cylinder engine equipped with two port fuel injectors dedicated to each side of the intake port and one direct injector. If the fuel is injected to only one side of the intake port, it is possible to form a stratified charge. The experiment was conducted under a compression ratio of 12.2 for various injection strategies. From the experimental results, it was found that injecting ethanol to the left side only of the intake port while both intake valves were open required approximately 52.1% (at 1500rpm) or 60.6% (at 2000rpm) less ethanol compared to the case in which ethanol is injected to both sides of intake port while intake valves are closed under a similar level of knock frequency. Furthermore, the engine load was maintained to the same level.

## Introduction

Ethanol is becoming more widely used for spark-ignited engines and has the potential to replace gasoline as a fuel due to energy security issues. Ethanol fuel use in the U.S. increased dramatically from approximately 1.7 billion gallons in 2001 to approximately 13.2 billion in 2013 [1]. There are many advantages to using ethanol as a fuel in spark-ignited engines. Ethanol has an intrinsically high octane number, a higher latent heat of vaporization and faster laminar flame speed, which can suppress knock behavior during combustion [2, 3, 4]. Additionally, the presence of oxygen molecules in ethanol enhances the oxidation rate of fuel, and thus CO and THC emissions can be decreased [5, 6]. However, due to ethanol having a higher latent heat of vaporization and boiling point it is unfavorable for the cold start condition in a spark-ignited engine [7-10]. Furthermore, ethanol has a lower energy density, so when it is used in real cars, the cruising distance becomes shorter due to restricted fuel tank size [11, 12]. Therefore, ethanol is widely used currently in blended form with gasoline rather than its pure form. However, fueling ethanol in a gasoline-blended form has a limit of impossibility for real-time variation of the ethanol-gasoline ratio according to various engine operating conditions. Thus, to educe the potential merits and overcome the demerits of ethanol as a fuel, on-demand blending ratio change of gasoline and ethanol is needed.

The concept was first introduced by Cohn et el. [13], In this study, demonstrating the direct injection of ethanol in a conventional PFI type engine, the authors proposed the addition of ethanol during knock occurrences during engine operation. Stein et al. [11], using a 3.5 L turbocharged engine, observed that an E85 DI and a gasoline PFI system facilitate the maintenance of spark timing at MBT until the engine load is raised up to 18 bar of BMEP. Zhuang et al. [14] investigated the effect of ethanol and gasoline on engine load, BSFC, volumetric efficiency and emission characteristics with an ethanol direct injection (EDI) and gasoline port injection (GPI) system. As the amount of ethanol was increased, the charge cooling effect led to an increase in volumetric efficiency. The authors also studied the effects of injection pressure and timing [15], and the results showed that further knock mitigation effect is observed when ethanol is injected after the intake valves are closed. Kim et al. [16], used a dual fuel system of ethanol port injection (EPI) and gasoline direct injection (GDI) on a single cylinder gasoline engine to exploit the charge cooling effect of gasoline, investigated the knock mitigation effect with the emission characteristics of EPI at two compression ratios, 9.5 and 13.3. As a result, the charge cooling effect of both ethanol and gasoline was verified while switching the EPI timing from conventional timing to the timing that opens the intake valves. The dual fuel system increased efficiency and decreased harmful emissions simultaneously. To enhance anti-knock characteristics in the combustion process, charge stratification of the air-fuel mixture was introduced by Li, et al. [17]. The authors observed the knock mitigation effect when different RON fuels were injected into each of two intake ports. The result demonstrated the potential of charge stratification to improve knock behaviors, which can lead to a better performance at high loads.

Many researchers have investigated the knock mitigation effect of ethanol using both the blended form and the additional injection form using the dual injection system. However, the effect of charge stratification and the swing of EPI system timing on knock suppression has not been studied before. Therefore, in this study, first, the effect of ethanol addition using EPI to enhance the anti-knock characteristic will be verified. Second, the effect of ethanol injection in various injection timings, which can lead to different volumetric efficiency and different enhanced charge cooling effects, will be investigated. Last, the effect of the charge stratification of ethanol port injection utilized with one of two EPI injectors will be observed.

# Experimental Setup

## *Experimental apparatus*

The detailed engine specifications for the experiment are shown in Table 1. A single cylinder gasoline engine modified from a Hyundai 2 L Theta-II Turbocharged GDI engine was used. Based on the original engine, which has CR of 9.5, a newly designed piston was used to achieve a higher CR of 12.2. The in-cylinder pressure was measured with a Kistler 6041A piezoelectric pressure transducer and the intake manifold pressure was measured with a Kistler 4045A5 absolute pressure sensor. Cylinder pressure pegging was conducted by setting the mean pressure at intake BDC (± 2°CA) equal to the mean pressure of the intake port, assuming the air flow stops [18]. Data from 300 cycles were logged to calculate the average, and the pressure data were sampled at 0.1°CA for high resolution, which made it possible to analyze the knock phenomena precisely.

Table 1. Engine specifications

| | |
|---|---|
| Displacement | 500 cc |
| Engine | Naturally aspirated single cylinder |
| Bore | 86 mm |
| Stroke | 86 mm |
| Compression ratio | 12.2 |
| Intake valve open / close | 1.9° bTDC / 58.1° aBDC |
| Exhaust valve open / close | 12.1° bBDC / 31.9° aTDC |

The characteristics of combustion were analyzed with an AVL Indimodule system, and exhaust gas emissions were analyzed using a HORIBA MEXA-7100 DEGR gas analyzer. A MoTeC M800 ECU generated the injection signal for the GDI injector at a specified timing, and a DG-535 pulse generator generated the injection signal for the EPI injectors and was synchronized with the GDI signal. With the pulse generator, EPI timing and duration were manually controlled. The desired injection pressure for GDI was achieved with a high pressure pump operated using the driving force of the exhaust camshaft. The ethanol for EPI was pressurized using nitrogen gas.

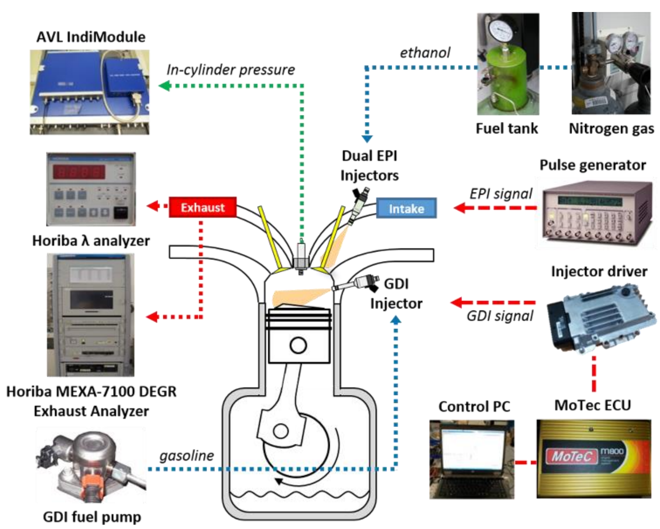


Figure 1. Schematic diagram of the experimental apparatus

## *Injection system*

In this study, two PFI injectors and one DI injector were used to supply ethanol and gasoline, respectively. Ethanol was injected with dual PFI injectors (EPI; ethanol port injection), and gasoline was injected with a single side-mounted DI injector (GDI; gasoline direct injection). The PFI injector was specially designed so that the SMD of the fuel droplets were approximately 50 μm under 3 bar of fuel injection pressure. The dual PFI system enabled the stratification of fuel charge by cutting off one of those injectors and also reduced the injection duration, which led to the injection of fuel while the intake valves were open. The schematic diagram of the injection system is shown in Fig. 2.

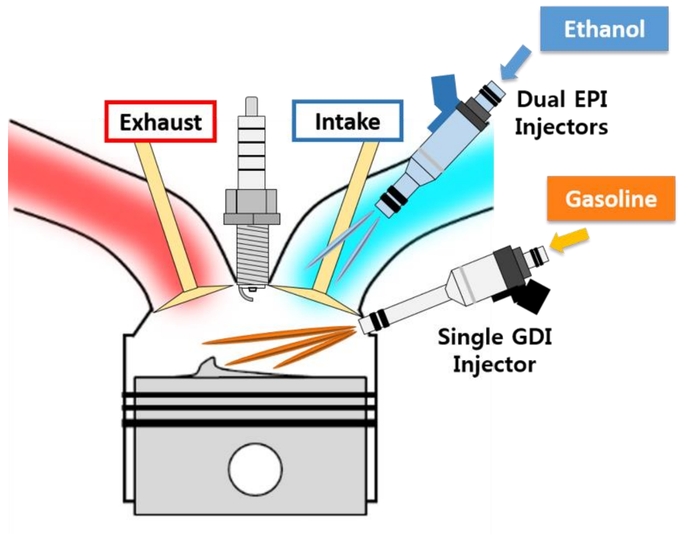


Figure 2. Schematic diagram of the injection system

## *Stratified injection strategy of ethanol*

To obtain the stratification of the air-fuel mixture in the combustion chamber, a stratified injection strategy was used. The stratified injection strategy meant that the ethanol was injected to the left side only of the two intake ports; consequently, different concentrations of right and left air-fuel mixture in the cylinder would be achieved while the valve operation strategy did not changed. Whether a lower pressure rising rate and prolonged combustion duration were achieved or not, the same air-fuel ratio was maintained during the experiments so that the injection duration at the stratified condition had to be longer than the normal condition. The schematic diagram of each strategy is shown in Fig. 3.

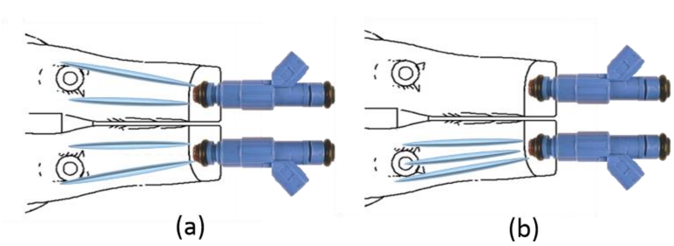


Figure 3. Top view of the ethanol port injection system: (a) normal injection, (b) stratified injection

Figure 4. Specially designed intake manifold for fuel charge stratification

Furthermore, a specially designed intake manifold with a thin wall inside of it was used as shown in Fig. 4. To prevent the mixing of the left and right sides of intake mixture during the intake process; the shape of the thin wall was perfectly matched with that of intake port in the head unit. Therefore, the charge stratification in the intake process could be achieved while injecting to one side of the intake ports. It is not certain that the fuel charge stratification had produced the stratified condition in the combustion chamber so further study like optical research would be needed. However, the following results show there are different phenomena when the fuel charge stratification was implemented.

## *Test fuels*

For GDI, regular RON 91 gasoline was used. Even if the gasoline has a lower latent heat of vaporization than ethanol, it is not negligible because gasoline was fueled directly into the cylinder in liquid form using a direct injection system. For EPI, 99% anhydrous ethyl alcohol was used. The properties of fuels are shown in Table 2.

Table 2. Test fuel properties

| | Ethanol | Gasoline |
|---|---|---|
| H/C ratio | 3 | 1.8 ~ 1.87 |
| O/C ratio | 0.5 | 0 |
| Gravimetric oxygen content (%) | 34.78 | 0 |
| Density [kg/m$^3$] | 785 | 735 ~ 750 |
| Research Octane Number | 107 | 91 |
| LHV [MJ/kg] | 26.9 | 43 ~ 44 |
| Latent heat of vaporization [kJ/kg] | 840 | 305 ~ 380 |

## *Experimental procedures*

The experimental conditions are shown in Table 3. Gasoline for DI and ethanol for PFI were pressurized to 100 bar and 6 bar, respectively. Intake air pressure was maintained at 0.92 bar and ambient temperature was maintained at 30 ± 1 ℃ for all experimental cases. Engine oil temperature and coolant temperature were maintained at 85 ℃ during operation.

Ethanol has a higher combustion speed than gasoline. Accordingly, within the same air-fuel ratio, MBT timing in the dual fuel combustion case is retarded compared to gasoline combustion due to the increased laminar speed of ethanol. In this study, for all cases the spark timing was fixed to 23 bTDC CA, which enabled CA50 of the combustion phasing to be 8° ± 0.5° aTDC in the CVI case. In the CVI case, it was MBT timing.

Table 3. Experimental conditions

| | |
|---|---|
| Engine speed | 1500, 2000 RPM |
| Lambda | 1 |
| Intake pressure | 0.92 bar |
| Gasoline injection timing | 305° bTDC |
| Gasoline injection pressure | 100 bar |
| Ethanol injection timing | CVI: 540° bTDC<br>OVI: 305°, 270°, 240°, 210° bTDC |
| Ethanol injection pressure | 6 bar |
| Intake air temperature | 30 ± 1 ℃ |
| Engine oil temperature | 85 ℃ |
| Coolant temperature | 85 ℃ |

Knock frequency is defined as the proportion of knock-occurring cycles to total experimental cycles. As the knock frequency builds up continuously due to thermal accumulation in the cylinder during the operating cycle, the testing procedure needed to be specified. Therefore, fixing other parameters including throttle angle and durations of EPI and GDI, only the advance of spark timing was used to induce knock-prone conditions, as shown in Fig. 5. Seven hundred fifty cycles, equivalent to one minute at 1500 rpm, were run as a purge interval for retardation of spark timing at 15 bTDC CA to achieve a steady state condition and avoid knock behavior. After that, the spark timing was advanced to 23 bTDC CA to generate knock behavior. The average of data from three hundred cycles was used for the analysis of combustion phase and knock behavior as mentioned in the previous section. Comparison of the pressure curves of knock-occurring cycles and averaged cycles is shown in Fig. 6. Fluctuation of the in-cylinder pressure, which is also a characteristic of knock, appears in the figure as well.

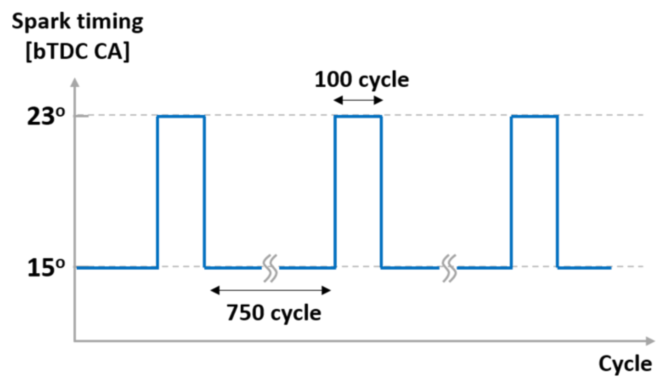


Figure 5. Specified engine operating procedure

For the ethanol additive experimental conditions, an open valve injection (OVI) strategy was adopted in addition to the conventional closed valve injection (CVI) strategy. An enhanced charge cooling effect using the higher latent heat of vaporization of ethanol can be achieved with the OVI strategy due to the injection timing and opening of the intake valves. [16] The conventional injection timing for the CVI strategy was 540° bTDC CA and timings of 305°, 270°, 240°, 210° bTDC CA were used for the OVI strategy. These cases were referred to as OVI305, OVI270, OVI240, and OVI210 in this study, respectively. The various timings of the OVI strategy show the difference in the charge cooling effect. Fig. 7 shows the specific injection timings of ethanol for both CVI and OVI.

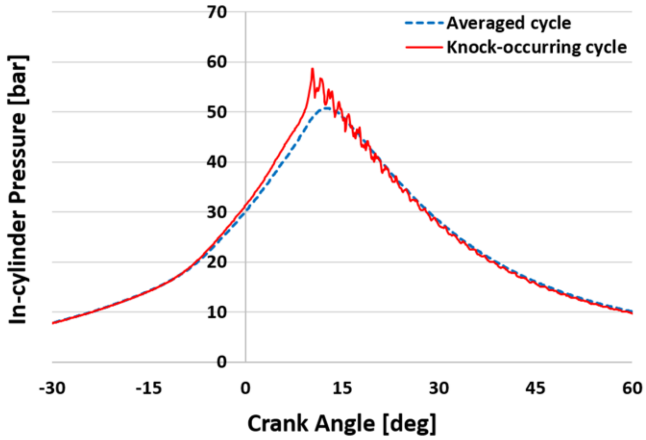


Figure 6. Comparison of in-cylinder pressure curves between averaged cycle and knock-occurring cycle

To achieve the stoichiometric condition in experimental cases, investigation of the injection amount according to the duration of injection signal was conducted before the engine experiments, and a calibration was conducted. With the calibration, H/C and O/C ratios could be calculated with various ethanol-gasoline ratios in the mixture using equations as shown in Eq. (1). In the equations, $N_e$ is the mole number of injected ethanol and $N_g$ is that of gasoline. These ratios were inputted to the lambda analyzer which is shown in Fig. 1 and manually corrected each time to measure the air-fuel ratio precisely. Additionally, the lambda value from the analyzer was validated using cross-over theory. Cross-over theory states that the stoichiometric condition can be achieved when the emission concentrations of oxygen ($O_2$) and carbon monoxide (CO) are equal. This is described in the literature [19], and Wu et al., verified the theory with oxygen-containing fuels such as ethanol [20, 21]. As per that study, cross-over theory can be used to control the lambda of different fuels with high accuracy.

$$O/C\ ratio\ = \left[\frac{N_e}{2 \cdot N_e + 8 \cdot N_g}\right]$$

$$H/C\ ratio\ = \left[\frac{6 \cdot N_e + 8 \cdot 1.87 \cdot N_g}{2 \cdot N_e + 8 \cdot N_g}\right] \tag{1}$$

The experiment consists of three parts as described in the introduction. First, with a higher compression ratio (12.2) than that of a conventional engine, the maximum IMEP was determined to achieve MBT timing without knock behavior. Then, ethanol was injected, and the knock mitigation effect which can lead to a higher operating load, was identified. Second, to investigate the further charge cooling effect of ethanol with OVI timing, the ethanol injection timing swing was conducted. As mentioned above, the excess air ratio and intake pressure was fixed while the EPI timing was changed. With varying OVI timing of EPI, knock frequency was also maintained to 10% ± 3% due to the protean characteristics of knock phenomena. Thus, the proportion of gasoline and ethanol in the fuel was changed for each condition. Lastly, with varied injection timing, the effect of stratification of EPI was investigated.

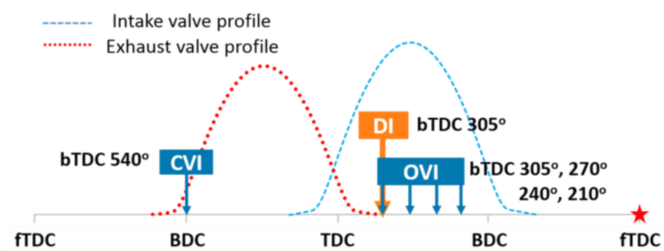


Figure 7. Injection timing for gasoline and ethanol

### *Knock detection method*

Based on cyclic analysis, "Knock transient" logic in AVL IndiCom software was used to detect the knock occurrence in this study. [22] First, the logic detected the timing of the maximum pressure in the cycle in units of CA. And then reference point was set where -2° shifted from the timing. After setting the reference point, the reference window and knock window were specified to the calculating range from the zero to 30° before the reference point and the zero to 30° after the reference point, respectively. High-pass filtered signal in each window was integrated in domain of crank angle, and the values are defined as $K_{int\_knock}$ and $K_{int\_ref}$ in this study. After that, knock ratio was calculated with the ratio of each $K_{int}$ value as shown in Eq. (2). The baseline constant abbreviated to BL in the equation, was 1.5 in unit of bar·deg when $K_{int\_ref}$ was zero. Thus, it prevented the denominator not to be zero. If the knock ratio value was over two, the cycle was judged as a knock-occurring cycle.

$$Knock\ Ratio = \left[\frac{BL +\ K_{int\_knock}}{BL +\ K_{int\_ref}}\right] \tag{2}$$

## Result and Discussion

### *Knock mitigation effect of ethanol*

#### The maximum load of MBT timing without ethanol addition

The experimental engine has a higher compression ratio than that of a conventional engine. Therefore as the load increased, even in medium operating range, the knock frequency started to rise. To identify the knock mitigation effect of ethanol addition in this study, first, the load that enabled the spark timing to be MBT timing was found without ethanol addition. This is called the g-MBT (gasoline MBT) case in this study. In the g-MBT case, the achieved maximum load was 5.5 bar of IMEP while the intake pressure was 0.82 bar and it had 29 bTDC CA of spark timing. If the load increased following the increase of intake pressure to maintain stoichiometric condition, the spark timing could not be advanced to MBT timing because of the knock behavior.

### Knock mitigation effect of ethanol addition using EPI system

Even if it is in a higher load condition than that of the g-MBT case above, ethanol addition using EPI enabled operation of the engine without knock behavior. At 0.92 bar of intake manifold pressure, without ethanol addition, as was expected, spark timing could not be advanced towards MBT timing and could only be advanced to 17 bTDC CA. This case is called g-knock (gasoline knock) in this study and the engine load was 6.97 bar of IMEP. In this condition, it was thought that the higher load could be achieved with further knock mitigation and simultaneous spark timing advance.

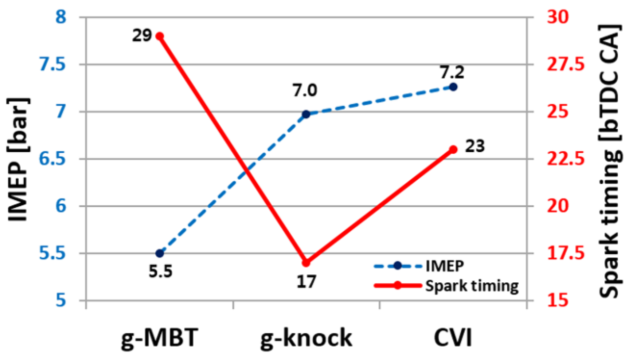


Figure 8. IMEP and spark timing of g-MBT, g-knock and CVI cases

Therefore, at the same intake manifold pressure as the g-knock case, ethanol was injected using the dual EPI system with CVI timing while advancing the spark timing towards the MBT timing. The 23 bTDC CA of spark timing for MBT was achieved with ethanol addition at 18.8% of total energy input. The load was 7.2 bar of IMEP. This case is called the CVI (dual, closed valve injection) case in this study. These results are shown in Fig. 8. The solid red line indicates the spark timing in units of bTDC CA and dark blue line indicates the load in units of bars of IMEP. There is a slight increase in IMEP in the CVI case while the total energy input between the g-knock case and the CVI case was almost identical (+ 0.1%). It is thought that the increase of combustion efficiency by ethanol addition [4] and the small advance of spark timing towards MBT timing were simultaneously responsible for the increase of the load.

## *Effects of ethanol injection timing*

The experimental cases were conducted to identify the variations in the knock mitigation effect as the ethanol injection timing swing was performed. As stated in the previous sections, each case used a different gasoline/ethanol energy ratio because the knock mitigation effects differed. All cases in this section were performed with the condition of normal injection, that is, both dual injectors were used and the engine speed was 1500 rpm.

### Comparison of fuel energy balance

Figure 9 shows the energy balances of input gasoline/ethanol for the experimental cases at 1500 rpm. In all cases, maintaining stoichiometric conditions, the ratio of ethanol/gasoline were varied to find the exact point where the knock frequency was 10%± 3%. The minimum ratio of ethanol, which was 8.9% of total input energy, was achieved with the OVI 305 case. This value of ethanol ratio is 19.5% lower than that of the CVI case. Other experimental cases also required less ethanol ratio than the CVI cases. This result shows that the enhanced charge cooling effect [16] reduced the overall in-cylinder mixture temperature and raised anti-knock characteristics. Therefore, less ethanol was required. The effect was maximized in the OVI305 case, where the ethanol was injected when the intake valves were being opened. Thus, more gasoline was induced into the cylinder than in the other late injection cases. Due to the degradation of the additional charge cooling effect, the demanded ethanol ratio was increased as the EPI timing was retarded.

In spite of the decrease of ethanol ratio in the OVI cases, the results show that the same level of the load could be achieved within 1% difference. Due to the late injection timings of OVI cases, the in-cylinder mixing condition deteriorated and efficiency may have decreased. However, because the volumetric proportion of port-injected fuel could be eliminated, increased volumetric efficiency was obtained using the OVI strategy [16]. It was already verified by detecting the increased lambda value while varying the injection timing and maintaining duration of fuel injections.

And there was no remarkable differences of combustion phase or in-cylinder pressure trace in all cases. Commonly, if the injection timing is retarded, the combustion phase which is usually represented by CA50, is also retarded. However, in this study, the fuel input amount and ratio were varied for each case to maintain the same level of knock frequency so it can be thought that the variation could compensate the combustion phasing. For example, when the injection timing was retarded in OVI cases, gasoline amount was decreased and ethanol amount was increased as shown in Fig. 9. Thus, faster laminar flame speed of ethanol could have influenced the combustion phase to be advanced. And between CVI and OVI cases, it is thought that higher volumetric efficiency of OVI cases influenced to have similar pressure curve with that of CVI case.

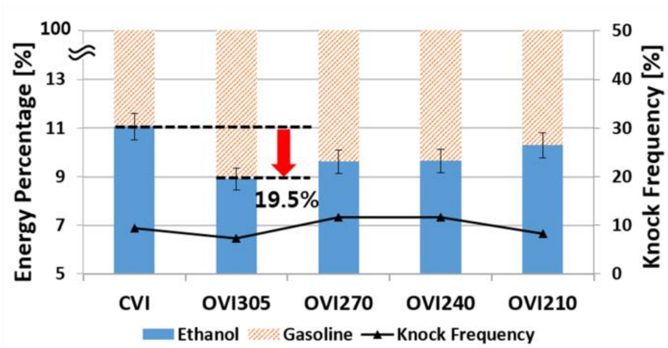


Figure 9. Energy percentages of demanded ethanol and gasoline at 1500 rpm

Observing less ethanol demand and the achievement of the same level of load in the experimental cases, the OVI strategy can be an effective choice to increase the anti-knock characteristics of dual fuel combustion when using an EPI and GDI system.

## *Effect of fuel charge stratification of additive ethanol*

Not only was the effect of ethanol injection timing monitored, but also the effect of fuel charge stratification was observed at 1500 rpm. For each case of OVI timings, instead of the normal injection strategy of using both EPI injectors, a stratified injection strategy using solely the left one of those injectors was conducted. The method of stratification is described in the previous section "Stratified injection strategy of ethanol" and in Fig. 3.

## Comparison of load and fuel energy balance

The demanded amounts of ethanol energy over total input energy of the fuels for each case are indicated in Fig. 10. "Stratified" in the figure indicates the stratified condition. As shown in the figure, when the stratification strategy was adopted, an average of 28% less ethanol energy percentage was demanded in implementation while maintaining the knock frequency. It is thought that further effect on the knock mitigation facilitated the reduction in the amount of ethanol demand, and gasoline percentages could be increased, simultaneously.

The most outstanding effect of stratification took place in the case of OVI305. 40.5% reduction of ethanol energy contribution was achieved when only the left injector was used for fuel charge stratification, compared to the normal injection condition. This case is called OVI305-L (left) in this study. For OVI305-L, 52.1% less ethanol was used, compared to the CVI case. The mass-based amount of demanded ethanol was also under half of that required for the CVI case. As the injection timing was advanced in OVI cases, the effect of charge stratification was improved, as shown in the figure as the rising slope due to retardation of the injection timing. It is thought that due to attenuated inlet air flow speed, the potential of injected fuel by EPI cannot be fully realized when the injection timing is retarded.

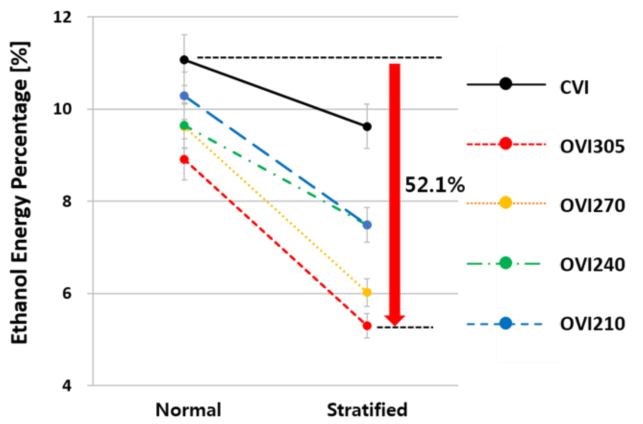


Figure 10. Energy percentages of demanded ethanol with stratification

Other right-side injected experimental cases were implemented as well, but in all cases, further effects were educed with only left-side injection. Regardless of the symmetric shape of the combustion chamber, this phenomenon is thought to be caused by the location of knock occurrence in the chamber. Thermal gradation of cylinder or other unknown factor might cause the asymmetry of knock occurrence locations in the combustion chamber and the knock may often be slightly on the left side of the cylinder. Consequently, injecting ethanol on the left side of the chamber led to a better effect on knock mitigation. To prove this theory, further study will be needed using knock detection devices to specify the location of knock occurrence.

## Load and emission characteristics

There was little decrease of IMEP (approximately 1.7% in stratified condition cases) over that of each CVI case when load changes were not observed in OVI normal injection cases. This is attributed to a less homogeneous condition of the mixture due to charge stratification.

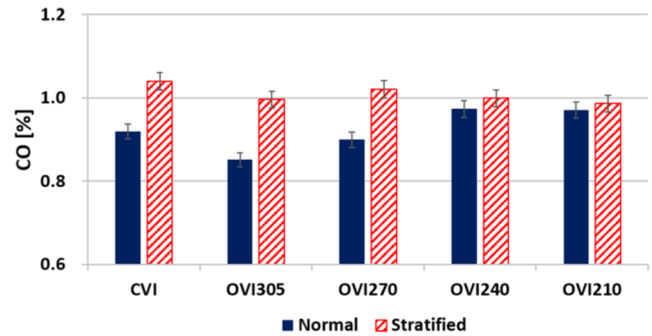


Figure 11. CO emission characteristics at the stratified condition

Figure 11 shows the changes in CO emissions by charge stratification in each ethanol injection timing. Increase of CO emissions was observed when the stratification was adopted for each injection timing, and the result corresponds to the decrease in IMEP. In OVI conditions, as the injection timing was advanced compared to stratified charge condition, less CO emissions were observed in the normal injection condition. It means that combustion phenomena in the normal injection condition are more likely to be complete, thus it is thought that more mixing time could be secured as the injection timing was advanced. However, further study would be needed because this study did not fully investigate it the reason why the CO emission of the OVI305 case is less than that of CVI case in condition of normal injection.

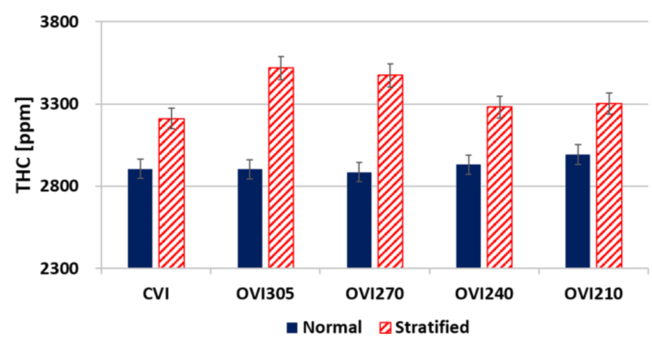


Figure 12. THC emission characteristics at the stratified condition

Figure 12 shows the unburned hydrocarbon emission changes between the conditions of normal injection and stratified injection. The level of unburned hydrocarbons in the exhaust gas is specified in terms of the THC in unit of ppm. In conditions of normal injection, the same levels of THC emission were observed regardless of the ethanol percentage changes. However, an increase of THC concentration was observed when the stratification was adopted. It is thought that more ethanol went into the crevice volume due to stratification. As injection timing was advanced, the mixture could permeate into the crevice in more time. Thus, the further increase was observed.

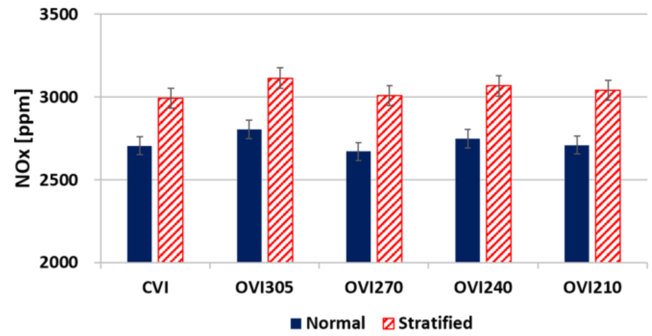


Figure 13. NOx emission characteristics at the stratified condition

NOx emission characteristics are shown in Fig. 13. In all cases, NOx emissions increased over 10% when the stratification was adopted. As it is mentioned above, the same level of knock frequency and stoichiometric conditions were maintained in all cases. Thus, despsite the more incomplete combustion observed, the ethanol amount inputted was reduced by further knock suppression effects in the stratified condition as shown in Fig. 10. As the ethanol's latent heat of vaporization is greater than that of gasoline, the charge cooling effect could be decreased in the cylinder. Therefore, NOx emissions could be increased by the increase in overall combustion temperature.

### *Effect verification of stratified open valve injection strategy at high speed*

Experimental implementation was conducted not only at 1500 rpm but also at 2000 rpm to verify the effect of stratification and open valve injection. Intake pressure was fixed to 0.92 bar again, and spark timing was advanced to 25 bTDC CA, unlike the 23 bTDC CA at 1500 rpm, to achieve MBT timing at the CVI condition.

#### Comparison of fuel energy balance

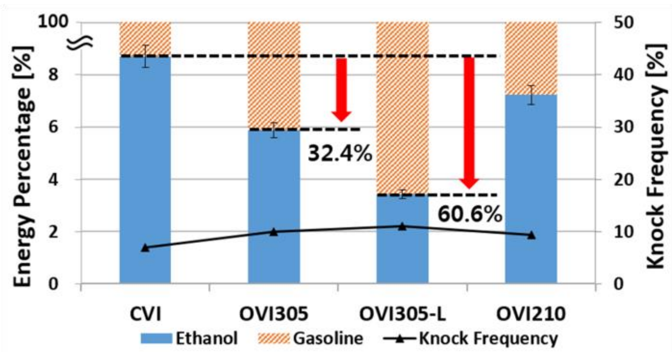


Figure 14. Energy percentages of demanded ethanol and gasoline at 2000 rpm

Figure 14 shows the energy balances of input gasoline/ethanol for the experimental cases at 2000 rpm. As explained in the previous sections, the aforementioned OVI305-L, indicates injection of ethanol to only the left side of the intake port while the EPI timing was bTDC 305 CA. As a result, the OVI strategy facilitates further knock suppression, which is shown in the case of OVI305 and even with late injection in the OVI210 case. In the OVI305 case, 32.4% less ethanol energy contribution was needed to maintain the same level of knock frequency. The effect of stratification was also observed. Similar to the result at 1500 rpm, stratification promoted the knock suppression characteristics, thus the demanded amount of ethanol was decreased as shown in Fig. 14, with a 60.6% decrement of ethanol energy proportion to total energy input achieved.

## Conclusions

In this research, the effects of ethanol injection strategies on knock suppression in a spark-ignited engine using the ethanol port injection and gasoline direct injection system were studied. CVI and OVI strategies for additive ethanol were tested and the effect of OVI timing swing was investigated. During the experiments for each OVI timing, the effect of stratification was also observed. The analysis of experimental results can be summarized as the following:

1. In this research, the experimental constraints were the 10% knock frequency, intake pressure and spark timing. Depending on the constraints, the demanded gasoline/ethanol ratio was varied. In all cases, the excess air ratio was maintained at one using verified crossover theory.

2. Using only gasoline direct injection at 1500 rpm, 5.5 bar of IMEP was achieved at 0.82 bar of intake pressure and 29 bTDC CA spark timing, which was MBT timing. While maintaining the same level of knock frequency, along with ethanol port injection, 7.21 bar of IMEP was achieved at 23 bTDC CA spark timing, which was MBT timing.

3. An ethanol injection timing swing was conducted from CVI timing to various OVI timings. In case of OVI305, a 19.5% lower ratio of ethanol energy input was demanded compared with the case of CVI, due to the effect of increased charge cooling on knock mitigation. In this case, no decrease in the load was observed.

4. At each OVI timing, the effect of the charge stratification was investigated. All cases showed the increased knock suppression effect. The case of OVI305-L, where ethanol was injected only to the left of the intake ports with the timing of OVI305, demonstrated the most drastic effect, in which a 52.1% lower ratio of ethanol energy input was demanded compared to the case of CVI. When the stratification strategy was adopted, due a decrease in homogeneity, CO emission was increased. Slight increases of NOx and THC emissions were also observed. Moreover, the stratification with the right side injection showed less effect than that of the left side injection.

5. Additionally, in this study, the effects of open valve injection and fuel charge stratification were verified not only at 1500 rpm but also at 2000 rpm. Similar effects were observed to the results at 1500 rpm. In the case of OVI305, 32.4% less energy input of ethanol was demanded while maintaining the same level of knock frequency. OVI305-L, the case that had the most remarkable effect, showed 60.6% less demand than that of the CVI case.

6. From the results above, it can be concluded that the open valve injection strategy and combined fuel charge stratification can suppress the knock behavior very efficiently. By minimizing ethanol consumption, the defect of ethanol (its low energy density) can be overcome during engine operation. However, there were slight increases of harmful emissions in stratified condition cases. In addition, it is not certain that the fuel charge stratification could show magnificent performance at very higher engine speeds. Thus, further studies are needed to investigate and exploit the benefits of ethanol.

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## Definitions/Abbreviations

| | |
|---|---|
| **EDI** | Ethanol direct injection |
| **GPI** | Gasoline port injection |
| **EPI** | Ethanol port injection |
| **GDI** | Gasoline direct injection |
| **RON** | Research octane number |
| **CR** | Compression ratio |
| **aTDC** | After top dead center |
| **bTDC** | Before top dead center |
| **aBDC** | After bottom dead center |

| | |
|---|---|
| **bBDC** | Before bottom dead center |
| **PFI** | Port fuel injection |
| **DI** | Direct injection |
| **LHV** | Lower heating value |
| **MBT** | Maximum brake torque |
| **RPM** | Revolution per minute |
| **CA** | Crank angle |
| **CA50** | Crank angle for 50% mass fraction burned |
| **IMEP** | Indicated mean effective pressure |
| **THC** | Total hydrocarbon |
| **NOx** | Nitrogen oxides |
| **CVI** | Closed valve injection |
| **SMD** | Sauter mean diameter |

## Appendix

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