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Homogeneous charge compression ignition of LPG and gasoline using variable valve timing in an engine

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Abstract

The combustion characteristics and exhaust emissions in an engine were investigated under homogeneous charge compression ignition (HCCI) operation fueled with liquefied petroleum gas (LPG) and gasoline with regard to variable valve timing (VVT) and the addition of di-methyl ether (DME). LPG is a low carbon, high octane number fuel. These two features lead to lower carbon dioxide (CO_2) emission and later combustion in an LPG HCCI engine as compared to a gasoline HCCI engine. To investigate the advantages and disadvantages of the LPG HCCI engine, experimental results for the LPG HCCI engine are compared with those for the gasoline HCCI engine. LPG was injected at an intake port as the main fuel in a liquid phase using a liquefied injection system, while a small amount of DME was also injected directly into the cylinder during the intake stroke as an ignition promoter. Different intake valve timings and fuel injection amount were tested in order to identify their effects on exhaust emissions and combustion characteristics. Combustion pressure, heat release rate, and indicated mean effective pressure (IMEP) were investigated to characterize the combustion performance. The optimal intake valve open (IVO) timing for the maximum IMEP was retarded as the λ_{TOTAL} was decreased. The start of combustion was affected by the IVO timing and the mixture strength (λ_{TOTAL}) due to the volumetric efficiency and latent heat of vaporization. At rich operating conditions, the θ_{90-20} of the LPG HCCI engine was longer than that of the gasoline HCCI engine. Hydrocarbon (HC) and carbon monoxide (CO) emissions were increased as the IVO timing was retarded. However, CO_2 was decreased as the IVO timing was retarded. CO_2 emission of the LPG HCCI engine was lower than that of the gasoline HCCI engine. However, CO and HC emissions of the LPG HCCI engine were higher than those of the gasoline HCCI engine.

Keywords: HCCI (homogeneous charge compression ignition); LPG (liquefied petroleum gas); DME (di-methyl ether)

1. Introduction

The homogeneous charge compression ignition (HCCI) engine is a promising concept for future automobile engines and stationary powerplants. The main concepts of HCCI are breathing premixed air/fuel mixture, as in conventional spark ignition (SI) engines, and ignition without a spark plug, as in conventional compression ignition (CI) engines [1]. Ultra lean burn combustion is achieved through homogeneous mixture formation and compression ignition [1], enabling combustion temperature much lower than that of conventional SI and CI engines. Owing to this lean mixture low temperature combustion, nitrogen oxides (NO_x) emissions are reduced dramatically and fuel economy is improved. However, greater amounts of hydrocarbon (HC) and carbon monoxide (CO) emissions are released relative to conventional SI and CI engines [1,2]. The oxidation reactions of HC and CO emissions during the expansion stroke are reduced due to the lower combustion temperature.

Most previous studies in this field have been carried out in conjunction with conventional gasoline and diesel fuels. However, the early combustion of gasoline HCCI limits its operating range [1]. The early combustion leads to high combustion pressure and knock [1]. Delayed combustion can solve these two major problems. Delayed combustion

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Nomenclature

λ	relative air fuel ratio
θ	crank angle degree
κ	specific heat ratio
P	cylinder combustion pressure
Q	heat release
V	cylinder volume
Q_{HR}	integrated value of total heat release
m_{fuel}	injection quantity of fuel
Q_{HV}	low heating value of using fuel

Abbreviation

AC	alternating current
ABDC	after bottom dead center
ATDC	after top dead center
BBDC	before bottom dead center
BTDC	before top dead center
CAD	crank angle degree
CI	compression ignition
CO	carbon oxide
CO_2	carbon dioxide
DI	direct injection
DOHC	double overhead camshaft
DME	di-methyl ether
ECU	engine control unit
HC	hydrocarbon
HCCI	homogeneous charge compression ignition
IMEP	indicated mean effective pressure
IVO	intake valve open
LHV	low heating value
LPG	liquefied petroleum gas
MFB	mass fraction burned
NO_x	nitrogen oxides
SI	spark ignition
TDC	top dead center
VVT	variable valve timing

is obtained by the use of LPG and lowered compression pressure and temperature. LPG has a higher octane number than gasoline and it has a high latent heat of vaporization, which can lower the compression pressure and temperature [3]. Moreover, LPG contains fewer carbon molecules than gasoline or diesel, and thus carbon dioxide (CO_2) can be reduced by using LPG fuel in vehicle engines [4]. The properties of DME are very similar to those of diesel. The cetane number of DME is higher than that of diesel. Because of its high cetane number, DME is suitable as an ignition promoter in the present study [5].

Another important issue in relation to the HCCI engine is combustion phase control. Hot residual gas supplies heat to the combustion chamber and promotes HCCI combustion [6–9]. This hot residual gas can be controlled by a variable valve timing (VVT) device [10]. Moreover, the VVT device can improve volumetric efficiency by varying the intake valve's open and close timing [11].

In this research, LPG HCCI combustion and exhaust emission characteristics were investigated. The characteristics of the LPG HCCI engine are compared with those of the gasoline HCCI engine. VVT was used to control the residual gas and volumetric efficiency. DME was also used as an ignition promoter by direct injection. The effects of VVT and fuel quantity on the LPG and gasoline fueled HCCI combustion were investigated in order to assess the feasibility of LPG as a future alternative fuel.

2. Experimental apparatus

2.1. Engine

The specifications of the engine are given in Table 1. The base engine was a four-cylinder spark ignition (SI) engine and has a double overhead camshaft (DOHC) equipped with a VVT. A cylinder was modified for HCCI combustion and a DME direct injection system was installed in the cylinder head. An engine control unit (ECU) (Motec Co., M4) was employed to precisely control DME quantity and injection timing. A second ECU (ETAS Co.) was used to control port fuel injection quantity, timing, and intake valve timing.

Table 1
Engine specifications

Bore (mm)		82
Stroke (mm)		93.5
Compression ratio		13
Displacement (cc)		494
Intake/exhaust valve opening duration (CAD)		228/228
Intake/exhaust valve lift (mm)		8.5/8.4
Valve timing (CAD)	Intake valve open (ATDC)	−29 to 11
	Intake valve close (ABDC)	59–19
	Exhaust valve open (BBDC)	42
	Exhaust valve close (ATDC)	6
DME injection pressure (bar)		50
DME injector		Slit injector

Fig. 1 shows a schematic diagram of the experimental setup. The engine speed and load were controlled by an alternating current (AC) dynamometer. A liquid phase injection system was used for LPG injection. This system can improve the volumetric efficiency via liquid phase injection and lower intake air temperature due to LPG vaporization at the intake port. Additionally, reduced engine out emissions can be achieved with a precise mixture control, as in gasoline SI engines [12]. A slit injector (Denso Co.) was used to inject DME at a constant supply pressure of 50 bar using pressurized nitrogen gas. The DME injector was

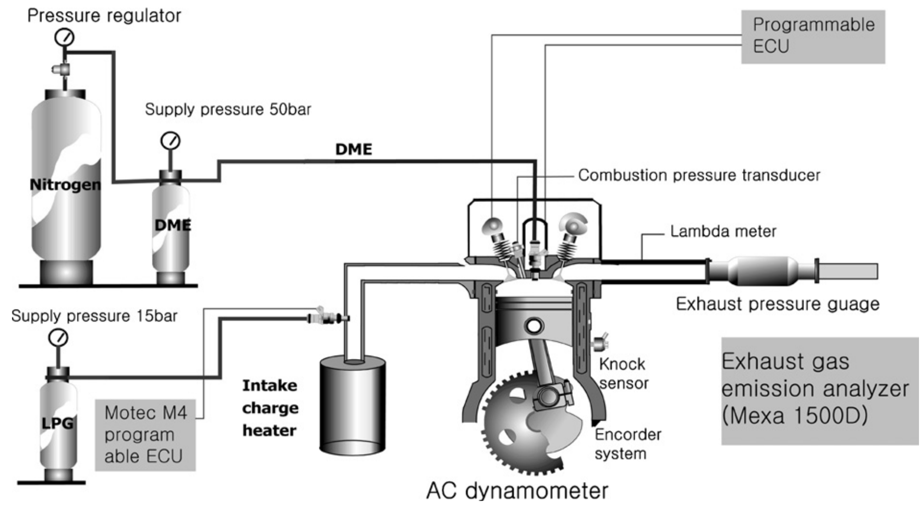

Fig. 1. Experimental apparatus.

located at the spark plug hole. Lubricity enhancer (Infineum, R655) of 500 ppm was added to the neat DME to avoid damage to the fuel injection system. The in-cylinder pressure was measured using a piezoelectric pressure transducer (Kistler, 6052b). The intake and exhaust manifold pressures were measured by two piezo-resistive pressure transducers (Kistler, 4045A5). The intake and exhaust temperature were measured by two K-type thermocouples, which were fitted on the intake and exhaust manifold. A wide band lambda meter (ETAS, LA4) was installed for the measurement of relative air/fuel ratio. Exhaust gases were analyzed with a gas analyzer (Horiba, Mexa 1500d) to measure the HC, NO_x, CO, and CO_2 emissions. The air flow rate was measured to obtain volumetric efficiency by a laminar flow meter (Meriam Co., 50MC2-2S). A data acquisition system (IOtech, Wavebook 512H) was employed to acquire all engine combustion and exhaust gas data.

The VVT system can freely vary the intake valve open and close timing. The intake valve open (IVO) timing was varied in a range of 29 crank angle degrees (CAD) before top dead center (BTDC) to 11 CAD after top dead center (ATDC), while the valve duration was fixed at 228 CAD. Generally, IVO timing is advanced as volumetric efficiency and residual gas are increased at low engine speed. Increased volumetric efficiency and residual gas promote combustion [13,14].

2.2. Experimental conditions

Table 2 shows the main experimental conditions used in this study. λ (relative air/fuel ratio) is defined as the ratio $(A/F)_{actual}/(A/F)_{stoichiometric}$. λ_{TOTAL} is defined as shown in [15]

$$\lambda_{TOTAL} = \frac{\lambda_{LPG}\lambda_{DME}}{\lambda_{LPG} + \lambda_{DME}} \quad (1)$$

Table 2
Experimental conditions

Engine speed (rpm)	1000
Intake valve open timing (CAD)	−29, −19, −9, 1, 11
DME injection timing (CAD)	110
λ_{TOTAL}	2.12, 2.41, 2.57, 2.77, 2.91
λ_{DME}	3.7
Intake charge temperature (°C)	30
Coolant/oil temperature (°C)	80/80

λ_{DME} was fixed at 3.7. and λ_{LPG} and $\lambda_{Gasoline}$ were varied from 2.12 to 2.91. The engine was run at 1000 rpm for various intake valve timings and air excess ratios. In the case of the HCCI engine, the gap of the lean operating limit and rich operating limit is increased as the engine speed is decreased [1]. Owing to this, the engine speed was limited to 1000 rpm. Fig. 2 shows the intake valve timing and the DME injection timing. The intake valve open (IVO) timing was varied from −29 CAD ATDC to 11 CAD ATDC. At

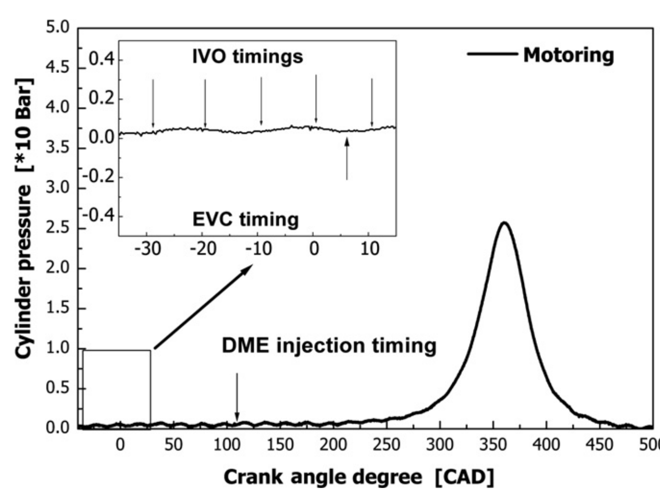

Fig. 2. Intake valve open, exhaust valve close and DME injection timing at 1000 rpm.

IVO timing of −29 CAD ATDC, the volumetric efficiency was 80%. At IVO timing of 11 CAD, it was 66.2%. The DME injection timing was fixed at 110 CAD ATDC for the formation of a homogeneous mixture. Gasoline and LPG were injected at the intake manifold at 470 CAD ATDC. Indicated mean effective pressure (IMEP) and heat release rate were calculated from the cylinder pressure values [15] using

$$\frac{dQ}{d\theta} = \frac{\kappa}{\kappa - 1} P \frac{dV}{d\theta} + \frac{1}{\kappa - 1} V \frac{dP}{d\theta} + \Delta Q_{\text{heat transfer}} \quad (2)$$

where

- θ crank angle degree,
- κ specific heat ratio,
- P cylinder combustion pressure,
- Q heat release,
- V cylinder volume.

The combustion efficiency was calculated from the integration of apparent heat release rate and the low heating value (LHV) of injected fuel using

$$\eta_{\text{comb}} = \frac{Q_{\text{HR}}}{m_{\text{fuel}} Q_{\text{HV}}} \quad (3)$$

where

- Q_{HR} integrated value of total heat release,
- m_{fuel} injection quantity of fuel,
- Q_{HV} low heating value of using fuel.

The DME oxidation reaction has a two-stage auto-ignition process; heat release with low temperature reaction (LTR) and high temperature reaction (HTR) [16]. The heat release of LTR was approximately 8–12% of the total heat release. To eliminate the effects of LTR, the burn duration (θ_{90-20}) is defined as the period between the duration for 20% mass fraction burned (MFB) and the duration for 90% MFB.

The LHV difference of LPG and gasoline used in this study was less than 0.3%. The proportions of propane and butane in the LPG used in this study were 60% and 40%, respectively.

3. Results and discussion

The main benefit of employing liquefied petroleum gas (LPG) in the engine is the reduction of greenhouse gases [12]. LPG is one of the most promising low carbon alternative fuels for spark ignition (SI) and homogeneous charge compression ignition (HCCI) engines. LPG HCCI engine combustion and emission characteristics were investigated to verify the feasibility of LPG for alternative fuel applications.

3.1. Combustion characteristics of LPG HCCI engine

Fig. 3 shows the combustion pressure and the heat release rates with respect to λ_{TOTAL} and the IVO timing.

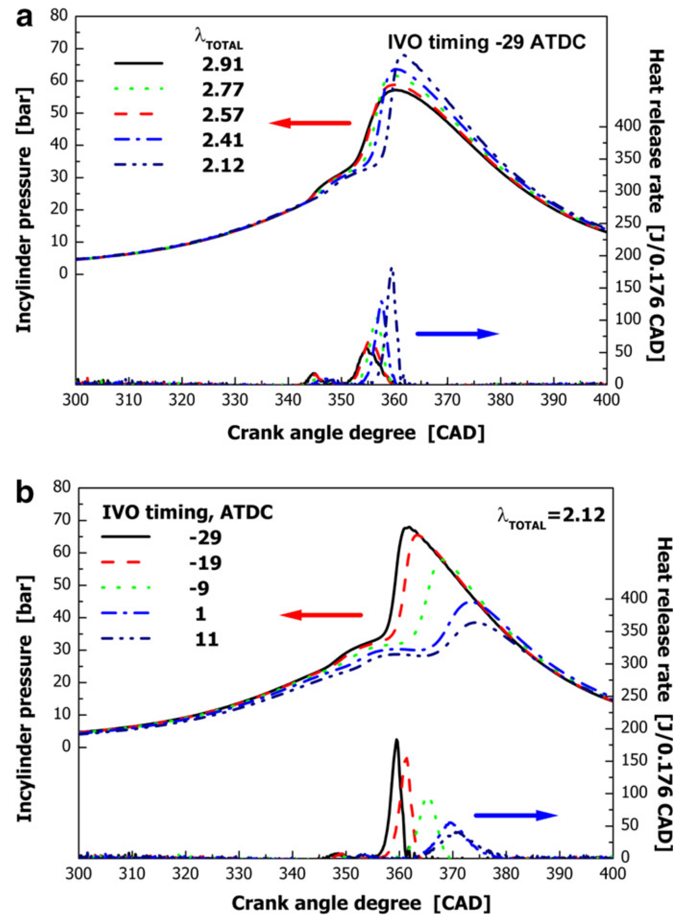

Fig. 3. Cylinder combustion pressure and heat release rate of LPG HCCI engine at 1000 rpm: (a) effects of λ_{TOTAL} at a fixed IVO timing (−29 CAD ATDC), (b) effects of IVO timing at a fixed λ_{TOTAL} (2.12).

The peak combustion pressure and the rate of pressure rise were increased due to the more combustible charge as λ_{TOTAL} was decreased, as shown in Fig. 3a. However, the start of the combustion point was retarded as λ_{TOTAL} was decreased. The retarded combustion at the lower λ_{TOTAL} condition could be explained by the difference of the octane number of fuel and the difference in the latent heat of LPG vaporization. The HCCI engine combustion is influenced by the chemical formation, temperature, and pressure of the air/fuel mixture [1]. LPG is the one of the high-octane number fuels [1,12]. The injection quantity of high-octane number fuel was increased as the cetane number of total fuel was decreased. The addition of high-cetane number fuels brings an advanced ignition timing [17]. The cetane number of fuel was decreased as λ_{TOTAL} was decreased due to decreased $\lambda_{\text{Gasoline}}$ or λ_{LPG} at fixed λ_{DME}. A liquid phase LPG injection system was used in this study. Liquid LPG was injected to the intake port and immediately vaporized [12]. The latent heat of vaporization of LPG used in this study is 370 kJ/kg and that of gasoline is 306 kJ/kg. Due to the latent heat of vaporization, the temperature of the LPG/air mixture before start of combustion was 2.04 K lower than that of the gasoline/air mixture. In the case of the LPG HCCI engine, the temperature of the air/fuel mixture is inversely proportional to the injection quantity of LPG. The octane number and temperature difference accounts for the difference in the start of the combustion.

The peak combustion pressure was decreased and the start of the combustion was retarded as the IVO timing was retarded (Fig. 3b). This is attributed to reduced volumetric efficiency and decreased hot residual gas.

Fig. 4 shows the rate of heat release from the LTR of the LPG HCCI engine compared to the gasoline fueled HCCI engine. In the case of LPG, the mixture pressure and the temperature showed a greater decrease, relative to gasoline, due to the latent heat of vaporization. The starting point of the LPG HCCI combustion LTR was more retarded as the IVO timing was retarded relative to gasoline HCCI combustion due to the higher octane number and the higher latent heat of vaporization.

The IMEP of the LPG HCCI engine with respect to λ_{TOTAL} and the IVO timing is shown in Fig. 5. At λ_{TOTAL} of 2.91, the maximum IMEP was acquired when the IVO timing was −9 CAD. However, at λ_{TOTAL} of 2.12, the maximum IMEP was acquired when the IVO timing was 1 CAD. On the left side of this map, the IMEP drop is due to negative work during compression resulting from early combustion. When the IVO timing was later than 1 CAD ATDC, an IMEP drop was observed, due to increased incomplete combustion. The CO emission, an index of incomplete combustion, was increased at later IVO conditions [18]. At earlier IVO timing, IMEP was suppressed by negative work. However at the latest IVO timing, IMEP was decreased due to incomplete combustion. This is verified by the HC emission results presented in the following section. The optimal IVO timing for the maximum IMEP was retarded as λ_{TOTAL} was decreased. The indicated mean effective pressure (IMEP) can be varied by the engine operating conditions, particularly the intake valve timing. For this reason, comparisons between LPG and gasoline were performed under fixed λ_{TOTAL}, not fixed IMEP. The LHV of LPG and gasoline at λ_{TOTAL} of 2.12 was 671.90 J and 673.80 J, respectively.

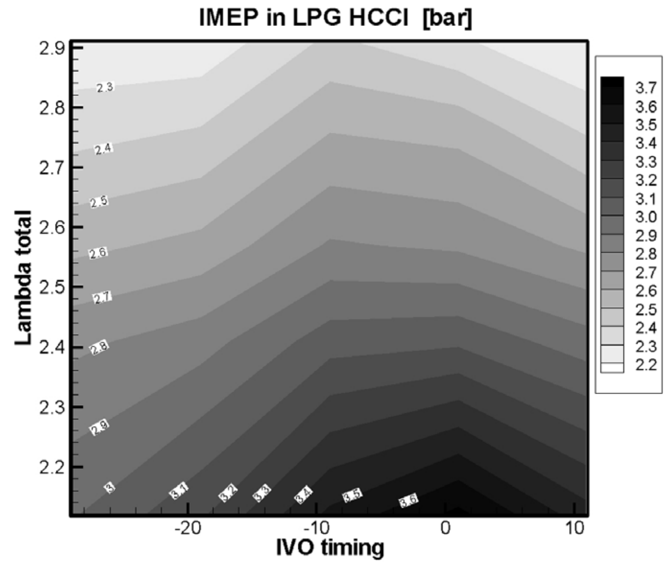

Fig. 5. IMEP of LPG HCCI engine with respect to λ_{TOTAL} and IVO timing at 1000 rpm.

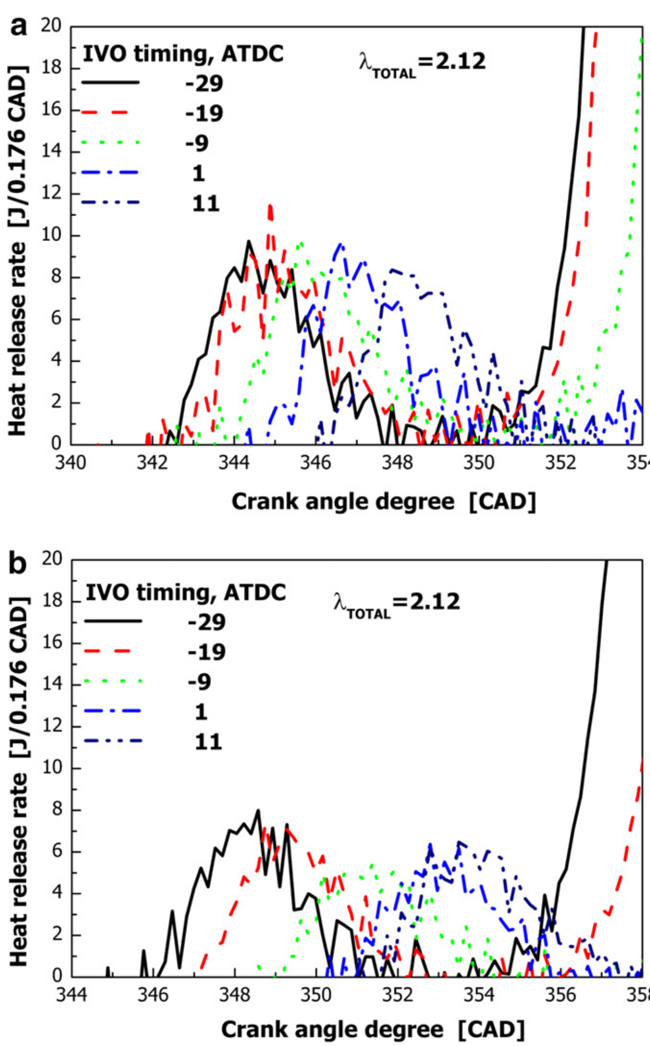

Fig. 4. Heat release rate of LTR at 1000 rpm, $\lambda_{TOTAL} = 2.12$: (a) gasoline HCCI engine and (b) LPG HCCI engine.

The IMEP of the gasoline HCCI engine is shown in Fig. 6. A decrease in IMEP for the gasoline HCCI was observed at richer mixtures. This is attributed to early combustion arising from the higher volumetric efficiency and the lower λ_{TOTAL}.

Fig. 7 shows the θ_{90-20} of the LPG HCCI engine with respect to λ_{TOTAL} and the IVO timing. θ_{90-20} was defined in the previous section. The shortest θ_{90-20} was observed at earlier IVO timing with richer mixtures. This is attributed to the higher volumetric efficiency and the larger injection quantity of LPG. The volumetric efficiency was 80%, 79.5%, and 77.2% at −29, −19, and −9 CAD ATDC, respectively. Meanwhile, the volumetric efficiency dropped rapidly at 1 CAD ATDC and 11 CAD ATDC, i.e., 70.8%

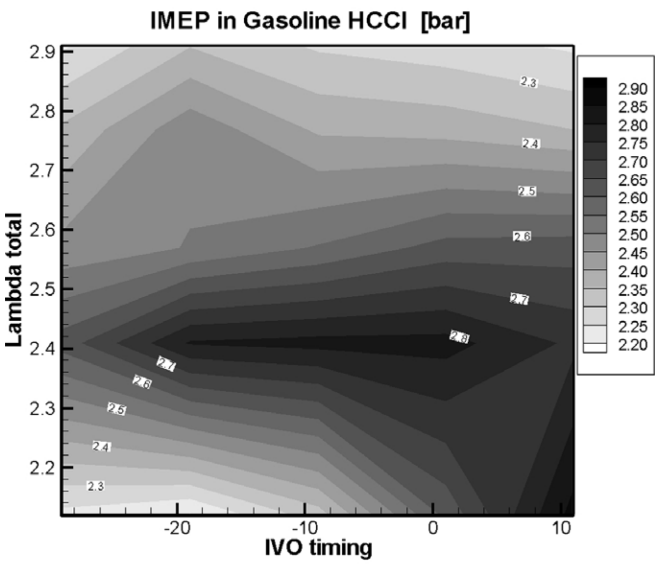

Fig. 6. IMEP of gasoline HCCI engine with respect to λ_{TOTAL} and IVO timing at 1000 rpm.

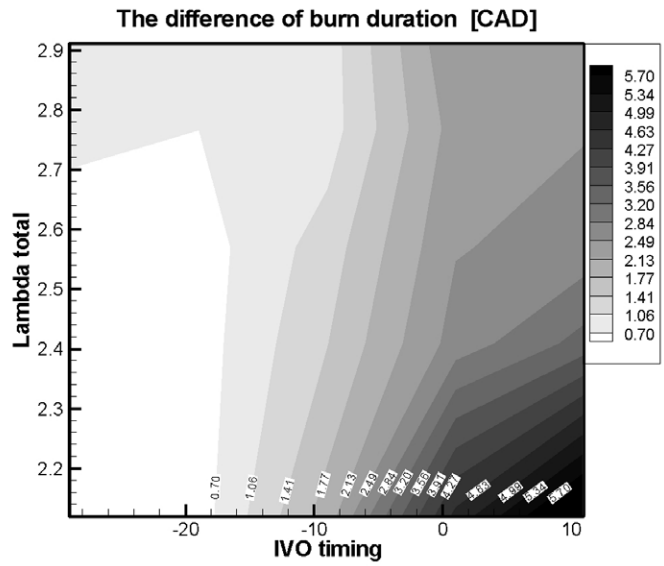

Fig. 8. Burn duration difference between LPG HCCI engine and gasoline HCCI engine with respect to λ_{TOTAL} and IVO timing.

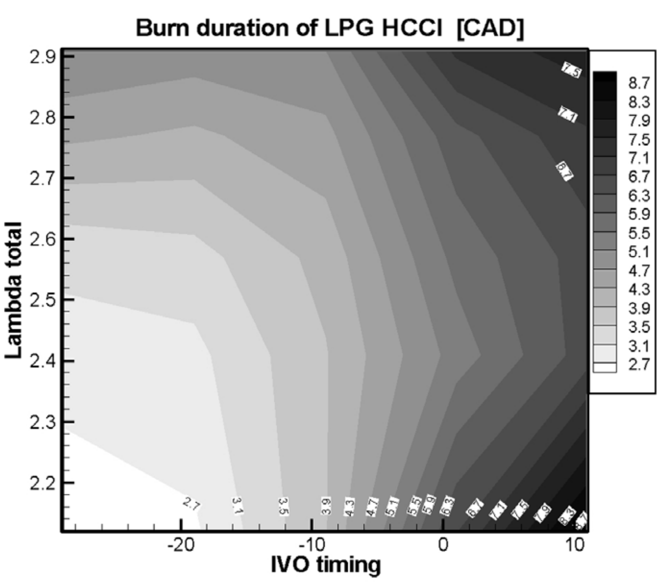

Fig. 7. Burn duration of LPG HCCI engine with respect to λ_{TOTAL} and IVO timing at 1000 rpm.

and 66.2%, respectively. Due to the long θ_{90-20}, the knock limit of the LPG HCCI engine was expanded. Longer θ_{90-20} was observed at rich conditions rather than at lean conditions. At later IVO timing, θ_{90-20} was increased for both richer and leaner conditions. At leaner conditions, long θ_{90-20} originates from the lean mixture. However, at richer conditions, long θ_{90-20} results from late combustion, which occurs in the expansion stroke due to the large latent heat of vaporization of LPG and lower volumetric efficiency. The effect of latent heat of vaporization in LPG is more clearly observed in Fig. 8, which shows the θ_{90-20} difference between the LPG and the gasoline HCCI engines. θ_{90-20} of LPG was generally longer than that of gasoline. Moreover, the θ_{90-20} difference is increased as the IVO timing was retarded and λ_{TOTAL} was lowered. This is attributed to a longer LPG θ_{90-20} resulting from the higher latent heat of vaporization and higher octane number of LPG.

Fig. 9 shows the durations for 90% MFB with respect to λ_{TOTAL} and the IVO timing. The duration for 90% of the MFB was increased as λ_{TOTAL} was decreased and the IVO timing was retarded. This is because of the lower volumetric efficiency and higher latent heat of vaporization, as explained above. Fig. 10 also shows a CAD difference of 90% MFB between the LPG and the gasoline HCCI engines. Fig. 10 indicates that the difference in θ_{90-20} originates from retarded LPG combustion. The end of combustion was more retarded at later IVO timing and richer

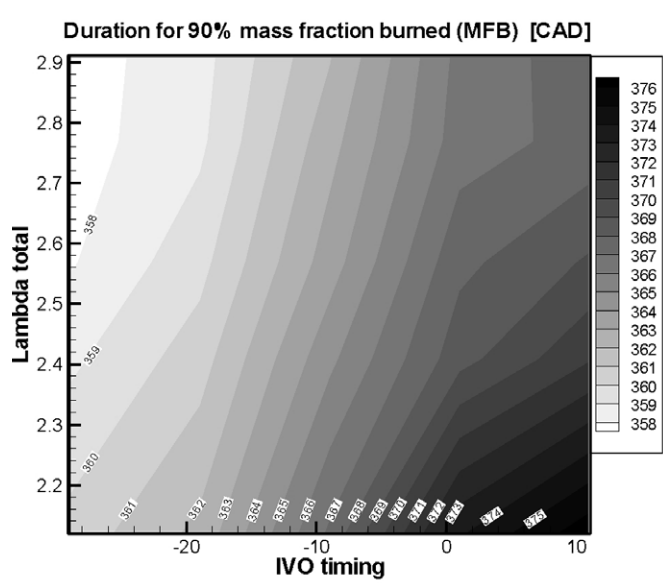

Fig. 9. Duration for 90% MFB in LPG HCCI engine with respect to λ_{TOTAL} and IVO timing.

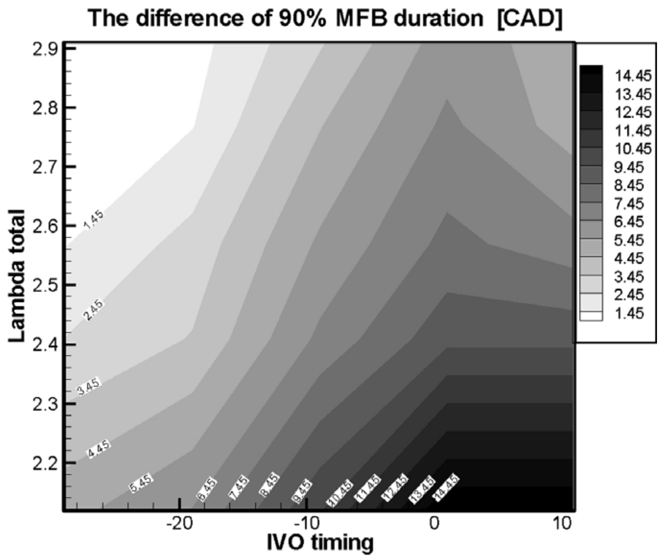

Fig. 10. The difference in the burn duration for 90% MFB between LPG HCCI engine and gasoline HCCI engine with respect to λ_{TOTAL} and IVO timing.

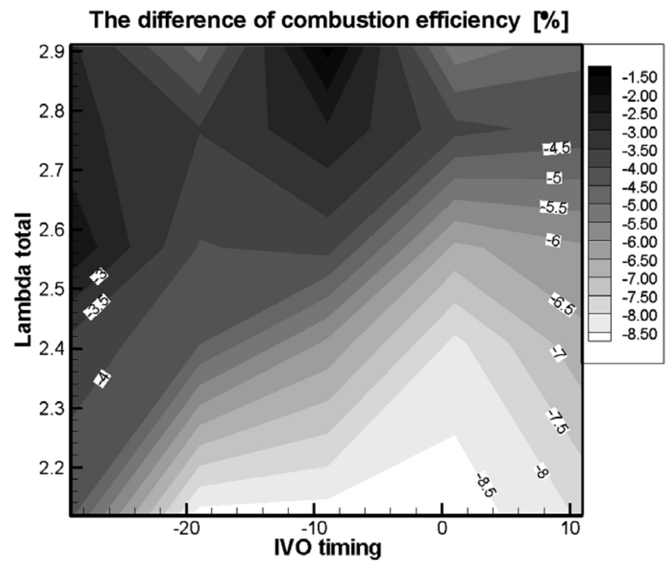

Fig. 12. The difference of combustion efficiency between LPG HCCI engine and gasoline HCCI engine with respect to λ_{TOTAL} and IVO timing at 1000 rpm.

conditions than at earlier IVO timing and leaner conditions, because the air/fuel mixture temperature at the end of compression was decreased at later IVO timing.

Figs. 11 and 12 show the combustion efficiency of LPG HCCI engine and the difference of combustion efficiency between LPG and gasoline HCCI engines with respect to λ_{TOTAL} and the IVO timing. The λ_{TOTAL} was a dominant factor of the combustion efficiency. The combustion efficiency was increased as the λ_{TOTAL} was decreased. However, the combustion efficiency of LPG HCCI engine was lower than that of gasoline HCCI engine at every experimental condition. This efficiency drop was due to incomplete combustion. The increased CO and HC emissions of LPG HCCI engine also show the evidence of incomplete combustion. In case of gasoline HCCI engine, knocking and early combustion were observed at most of experimental conditions. The combustion efficiency calculation considered the heat transfer from combustion chamber to coolant and oil. This heat transfer is approximately 15% of total LHV of fuel [19].

3.2. Exhaust emissions of LPG HCCI engine

Fig. 13 shows the CO_2 emission from an LPG HCCI engine with respect to the IVO timing and λ_{TOTAL}. The

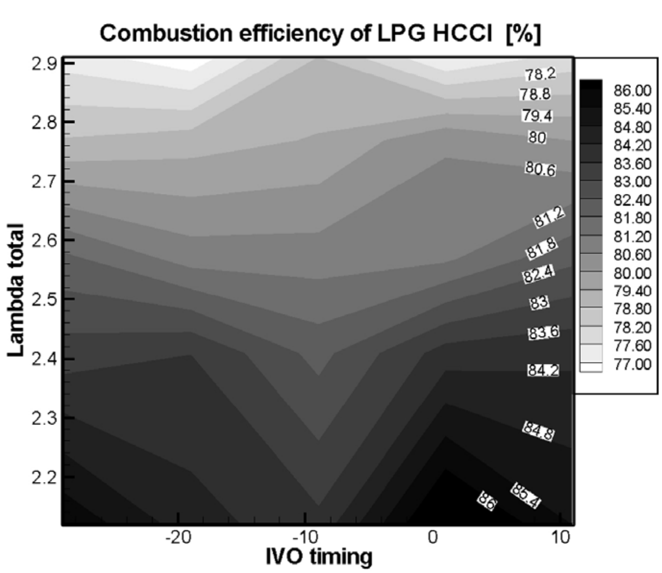

Fig. 11. Combustion efficiency of LPG HCCI engine with respect to λ_{TOTAL} and IVO timing at 1000 rpm.

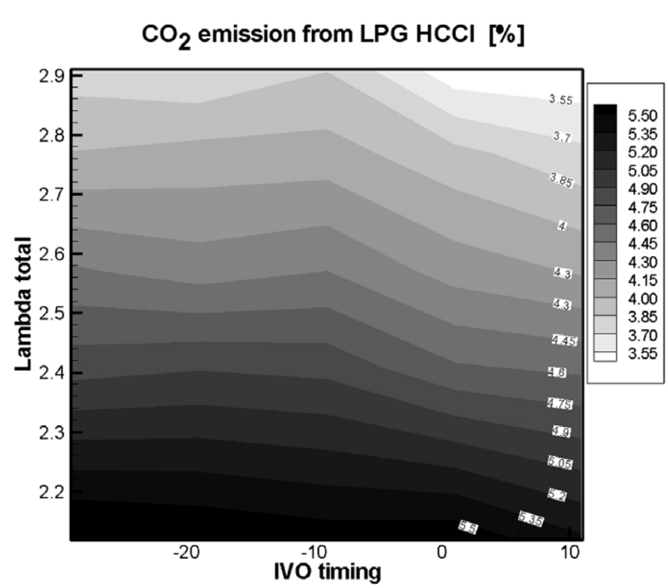

Fig. 13. CO_2 emission from an LPG HCCI engine with respect to λ_{TOTAL} and IVO timing.

CO_2 emission was naturally reduced at the leaner condition as λ_{TOTAL} increased. Moreover, the CO_2 emission was reduced as the IVO timing was retarded. When the IVO timing was retarded, the amount of hot residual gas, which can promote combustion, as well as the volumetric efficiency, were reduced because of the reduced valve overlap and late intake valve closing. In the case of low engine speed, late intake valve opening permits entry of less fresh air than early IVO timing due to the inertia of the air. Because of the reduced volumetric efficiency, the combustion starting point at later IVO timing was retarded. The amount of latent heat of vaporization did not vary when the same amount of LPG was injected. However, the volumetric efficiency was decreased and the temperature drop at the end of the compression stroke was increased. Due to the temperature drop, the combustion start point was retarded as the IVO timing was retarded. The late combustion resulted in increased unburned fuel and decreased combustion temperature.

To confirm this trend, HC emissions with respect to IVO timing and λ_{TOTAL} are shown in Fig. 14. The HC emission was increased due to the late combustion at the retarded IVO timing and the increased λ_{TOTAL}. More HC emission was exhausted due to the stronger quenching effect under the lower combustion temperature together with the weaker HC oxidation reaction during the expansion stroke.

Fig. 15 shows the CO emission from the LPG HCCI engine as a function of the IVO timing and λ_{TOTAL}. The CO emission was increased as the IVO timing was retarded due to late combustion occurring after top dead center (ATDC). Lowered combustion temperature and pressure due to late combustion results in a weaker oxidation reaction of CO during the expansion stroke [1].

Fig. 16 shows the CO_2 emission difference between the LPG and gasoline HCCI engine. LPG is one of the low-carbon fuels that can be used in SI engines. Therefore, the CO_2 emission from the LPG HCCI engine was lower than that from the gasoline HCCI engine. LPG has a higher octane number and higher temperature of ignition than gasoline. The LPG properties lead to a lower combustion rate relative to gasoline. The lowered combustion rate is also described in Fig. 17, which shows the HC emission difference between the LPG and the gasoline HCCI engine. Late IVO timing conditions of the LPG HCCI engine result in greater HC emission relative to the gasoline HCCI engine. The increased HC emission results from the lack of an oxidation reaction due to late combustion [1]. Late combustion according to IVO timing was also observed in the

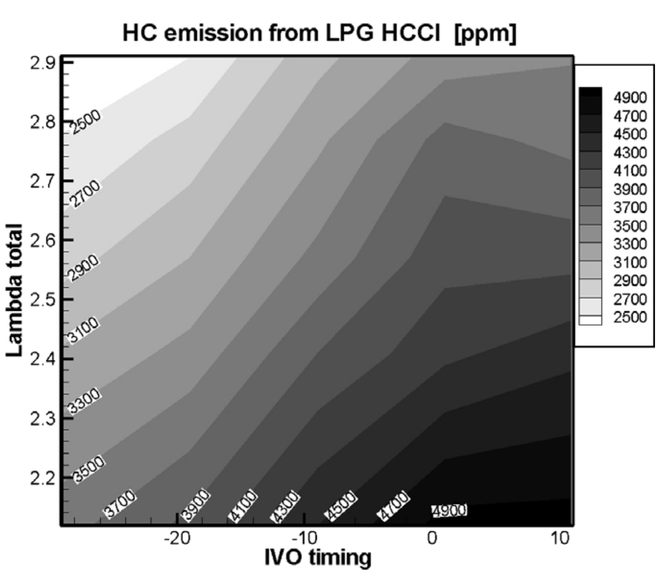

Fig. 14. HC emission from an LPG HCCI engine with respect to λ_{TOTAL} and IVO timing.

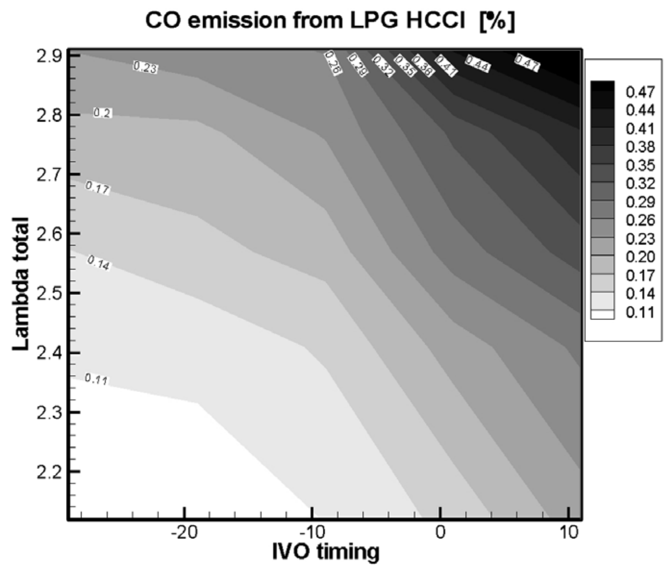

Fig. 15. CO emission from an LPG HCCI engine with respect to λ_{TOTAL} and IVO timing.

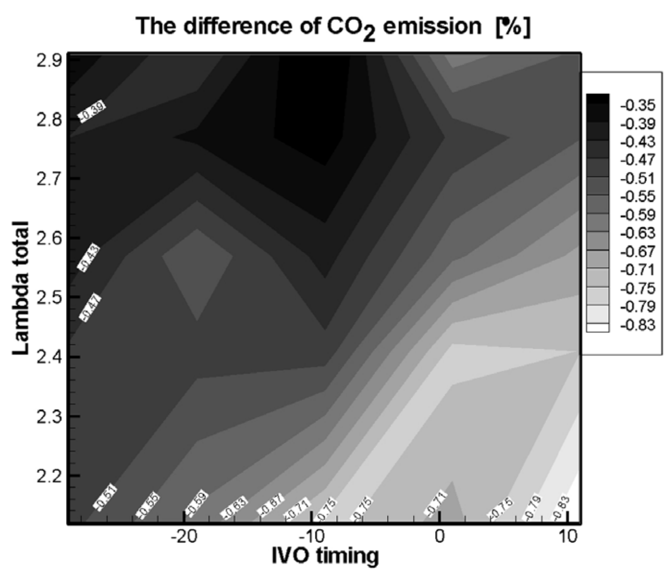

Fig. 16. CO_2 emission difference between LPG and gasoline HCCI engine with respect to λ_{TOTAL} and IVO timing.

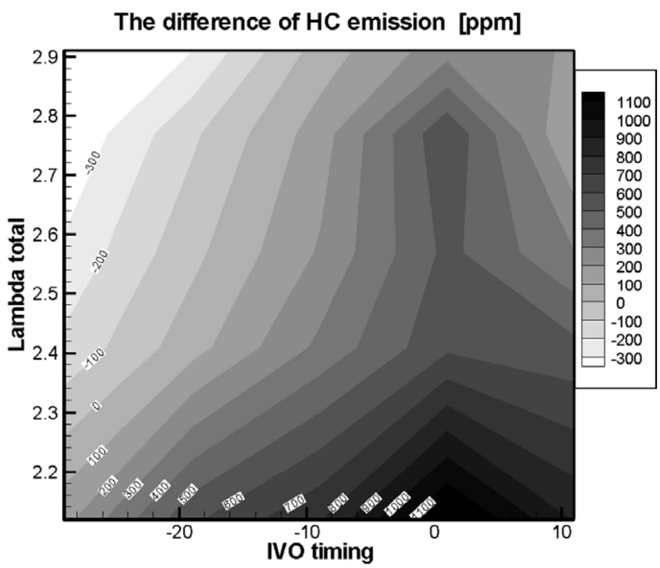

Fig. 17. HC emission difference between LPG and gasoline HCCI engine with respect to λ_{TOTAL} and IVO timing.

gasoline HCCI engine. However, the start of combustion in the gasoline HCCI engine was earlier than in the LPG HCCI engine due to the lower octane number and lower heat of vaporization of gasoline. The faster combustion leads to higher combustion temperature and pressure, which can oxidize the HC emission. Meanwhile, the start of combustion in the gasoline HCCI engine was retarded dramatically at 11 CAD ATDC. The dramatic retardation was similar to that observed for the LPG HCCI engine at 1 CAD ATDC. The HC emission difference was reduced at 11 CAD ATDC. The HC emission increase was caused by the start of ignition retardation. The retardation of the start of combustion accounts for the increased HC emission due to a lack of oxidation and a slower combustion rate.

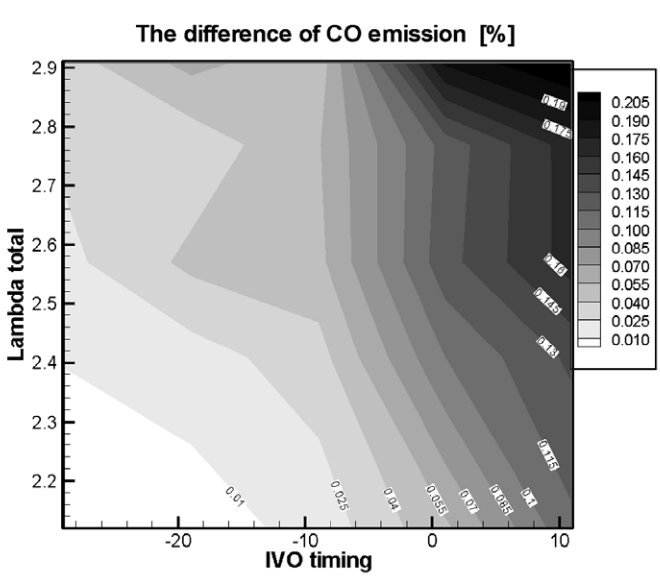

Fig. 18. CO emission difference between LPG and gasoline HCCI engine with respect to λ_{TOTAL} and IVO timing.

Fig. 18 shows the CO emission difference between the LPG and gasoline HCCI engines. The CO emission from the LPG HCCI engine is increased slightly compared to that from the gasoline HCCI engine. The CO emission is increased dramatically with retarded IVO timing and increased λ_{TOTAL}. This is because of incomplete combustion due to low combustion temperature [18].

4. Conclusions

The effects of IVO timing and fuel quantity on exhaust emissions and combustion characteristics in LPG and gasoline fueled HCCI engines controlled by a VVT were investigated. A heat release analysis was performed in order to verify the HCCI combustion characteristics. The exhaust emissions and combustion characteristics from the LPG HCCI engine were compared with those of a gasoline HCCI engine. The following conclusions were drawn from the experimental results:

(1) The peak combustion pressure and the rate of pressure rise were increased due to the increase in combustible charge as λ_{TOTAL} was decreased. However, the peak combustion pressure was decreased and the start of the combustion crank angle was retarded as the IVO timing was retarded due to reduced volumetric efficiency and an increase in the residual gas.
(2) At earlier IVO timing, IMEP was suppressed by negative work. However, at the latest IVO timing, IMEP was decreased due to incomplete combustion. The optimal IVO timing for the maximum IMEP was retarded as λ_{TOTAL} was decreased. This is because the negative work during compression increased as λ_{TOTAL} was decreased due to early combustion.
(3) The CO_2 emission from the LPG HCCI engine was naturally reduced at the leaner condition as λ_{TOTAL} increased. Moreover, the CO_2 emission was reduced as the IVO timing was retarded.
(4) The HC emission was increased due to late combustion at the retarded IVO timing and increased λ_{TOTAL}. More HC emissions were exhausted due to the stronger quenching effect under lower combustion temperature together with weaker HC oxidation reaction during the expansion stroke. The LPG HCCI engine yielded more HC emission than the gasoline HCCI engine due to late combustion originating from the latent heat of vaporization.
(5) The CO emission from the LPG HCCI engine was increased as the IVO timing was retarded due to late combustion occurring after top dead center (ATDC). The CO emission from the LPG HCCI engine increased slightly compared to the gasoline HCCI engine. Lowered combustion temperature and pressure due to late combustion results in weaker oxidation of CO during the expansion stroke.

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