# Di-methyl Ether Homogeneous Charge Compression Ignition Engine with Gasoline Port Injection

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## ABSTRACT

The combustion and emission characteristics in an engine were investigated under homogeneous charge compression ignition (HCCI) operation with regard to variable valve timing (VVT) and di-methyl ether (DME) direct injection. Two operation modes were tested to analyze the effects of gasoline adding. One is the DME direct injection HCCI mode and the other is Gasoline port injection with DME direct injection HCCI mode. Gasoline was injected at the intake port, while DME was also injected directly into the cylinder. Different intake valve timings, fuel injection timings and fuel injection quantities were tested for combustion phase control and exhaust emission reduction.

The indicated mean effective pressure (IMEP) was increased by adding of gasoline. Moreover, the CO emission was reduced. However, the HC and $CO_2$ emission was increased. The HC and CO emissions, which are the index of incomplete combustion, were decreased as the IVO timing was advanced and the injection timing was retarded. Moreover, the IMEP was increased as the IVO timing was advanced and the injection timing was retarded. The NOx emission was lower than 500 ppm at every experimental condition.

## 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 homogeneous 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 (NOx) 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.

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.

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[3- 6)]. This hot residual gas can be controlled by a variable valve timing (VVT) device[7)]. Moreover, the VVT device can improve volumetric efficiency by varying the intake valve's open and close timing[8)].

In this research, gasoline HCCI combustion and exhaust emission characteristics were investigated. 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 gasoline fueled HCCI combustion were investigated.

## EXPERIMENTAL APPARATUS

ENGINE - The specifications of the engine are given in Table 1. The base engine was a 4 cylinder spark ignition (SI) engine and has a double over head 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.

Figure 1 shows a schematic diagram of the experimental setup. The engine speed and load were controlled by an alternating current (AC) dynamometer. Additionally, reduced engine out emissions can be achieved with a precise mixture control, as in gasoline SI engines[9)]. A slit injector (Denso Co.) was used to inject DME at a constant supply pressure of 5 MPa using pressurized nitrogen gas. The DME injector was 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, NOx, 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[10,11].

EXPERIMENTAL CONDITION - Table 2 shows the main experimental conditions used in this study. (Relative air/fuel ratio) $\lambda$ is defined as the ratio $(A/F)_{actual}$ / $(A/F)_{stoichiometric}$. $\lambda_{TOTAL}$ is defined as shown in equation (1)[12].

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

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[12]. Owing to this, the engine speed was limited to 1000 rpm. The intake valve open (IVO) timing was varied from -29 CAD ATDC to 11 CAD ATDC. At 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 was 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[12] using equation (2).

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

The DME oxidation reaction has a two-stage auto-ignition process; heat release with low temperature reaction (LTR) and high temperature reaction (HTR)[13]. The heat release of LTR was approximately 8~12% of the total heat release.

To eliminate the effects of LTR, the burn duration ($\theta_{20\text{-}90}$) is defined as the period between the duration for 20% mass fraction burned (MFB) and the duration for 90% MFB.

Figure 2 shows the criterion of knocking used in this study. The knock intensity is obtained from cylinder pressure data using various methods[14]. The present study focuses on the knock probability. In order to determine the knock, the criterion of pressure oscillation due to knock is the pressure rise over 0.05 MPa during the expansion stroke[14]. The combustion pressures were analyzed for 100 cycles at each experimental condition to obtain IMEP and knock probability.

Table 1 Engine specifications

| | | |
|---|---|---|
| Bore (mm) | | 82 |
| Stroke (mm) | | 93.5 |
| Compression ratio | | 12 |
| Displacement (cc) | | 494 |
| Intake / Exhaust valve duration | | 228 / 228 |
| Intake / Exhaust valve lift | | 8.5 / 8.4 |
| Valve timing (CAD) | Intake Valve Open (ATDC) | -29 ~ 11 |
| | Intake Valve Close (ABDC) | 59 ~ 19 |
| | Exhaust Valve Open (BBDC) | 42 |
| | Exhaust Valve Close (ATDC) | 6 |
| DME injection pressure (MPa) | | 5 |
| DME injector | | Slit injector |

Table 2 Experimental conditions

| | |
|---|---|
| Engine speed (rpm) | 1000 |
| Intake Valve Open timing (CAD) | -29, -19, -9, 1, 11 |
| DME injection timing (CAD) | 110 |
| $\lambda_{TOTAL}$ | 2.12, 2.41, 2.57, 2.77, 2.91 |
| $\lambda_{DME}$ | 3.7 |
| Intake charge temperature ($^{O}C$) | 30 |
| Coolant / Oil temperature ($^{O}C$) | 80 / 80 |

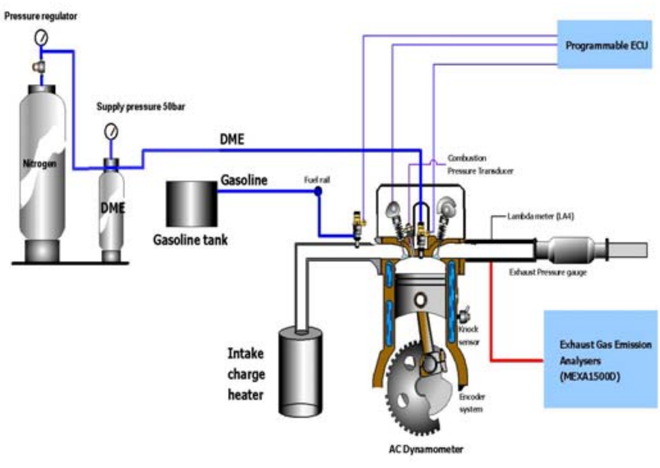



Figure 1 Experimental apparatus

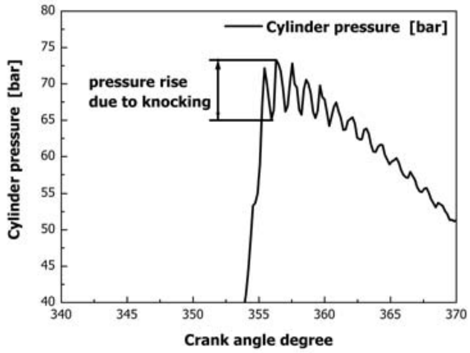



Figure 2 The criterion of knocking detection in this study

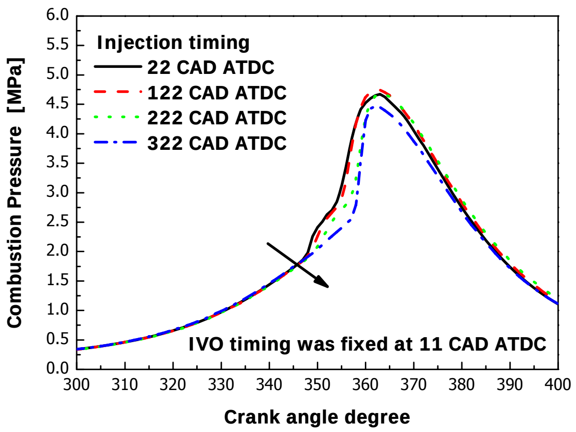



Figure 3 Cylinder combustion pressure of DME HCCI engine at 1000rpm as a function of injection timing

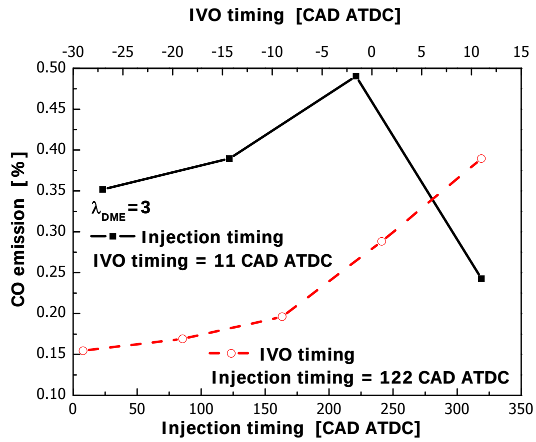
## RESULTS AND DISCUSSION

DME HCCI CHARACTERISTICS – Figure 3 shows the combustion pressure with respect to $\lambda_{TOTAL}$ of DME HCCI combustion. There was no cool flame at 322 CAD ATDC injection timing. This fact means that the combustion phenomenon was mixing controlled combustion like CI engine at the condition of 322 CAD ATDC. However, the other cases of injection timing show the evidence of HCCI combustion. This fact is supported by HC and CO emissions. These two emissions are regarded as the index of incomplete combustion. Figure 4 shows the HC and CO emissions of DME HCCI engine. In the case of injection timing at 322 CAD ATDC, HC and CO emission show the lowest level compare to other cases. This low level of HC and CO emissions means high temperature combustion.

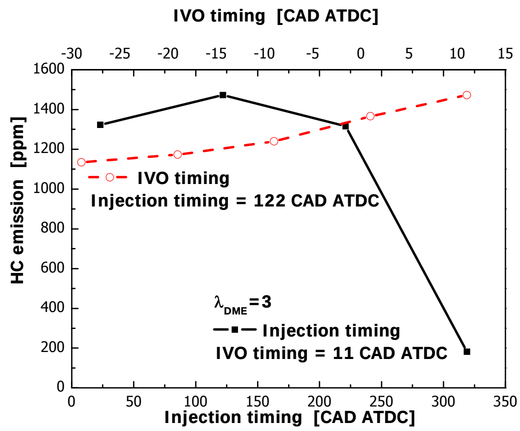
DME-GASOLINE COMBUSTION CHARACTERISTICS – Figure 5 shows the combustion pressure and the heat release rates with respect to $\lambda_{TOTAL}$ of DME-gasoline HCCI. The peak combustion pressure and the rate of pressure rise were increased due to the more combustible charge as the $\lambda_{TOTAL}$ was decreased as shown in Fig. 5. The start of the combustion point was retarded as the $\lambda_{TOTAL}$ was decreased. It is because that gasoline has anti-auto-ignition characteristics, which was originated from higher octane number. Combustion is influenced by the chemical formation, temperature and pressure of the air/fuel mixture[1).]

Figure 6 shows the combustion pressure and the heat release rates with respect to the IVO timing. The peak combustion pressure was decreased and the start of the combustion was retarded as the IVO timing was getting late, as shown in Fig. 6. This reduced volumetric efficiency and residual gas lead to late combustion.

The IMEP of the gasoline HCCI engine with respect to the $\lambda_{TOTAL}$ and the IVO timing is shown in Fig. 7. The horizontal axis represents the IVO timing and the vertical axis represents $\lambda_{TOTAL}$. This figure shows that $\lambda_{TOTAL}$ is the major parameter affecting the IMEP. There is an optimum amount of injected gasoline for attaining maximum IMEP, corresponding to an approximate value of 2.4 in the $\lambda_{TOTAL}$. When $\lambda_{TOTAL}$ was lower than 2.4, early combustion was observed and IMEP dropped rapidly. This is because most of the heat was released before TDC and the negative work during compression was increased as $\lambda_{TOTAL}$ was decreased. On the lower part of this map, IMEP is increased as the IVO timing is retarded. The volumetric




(a) CO emission




(b) HC emission

Figure 4 Exhaust emissions of DME HCCI engine at 1000rpm as a function of injection timing

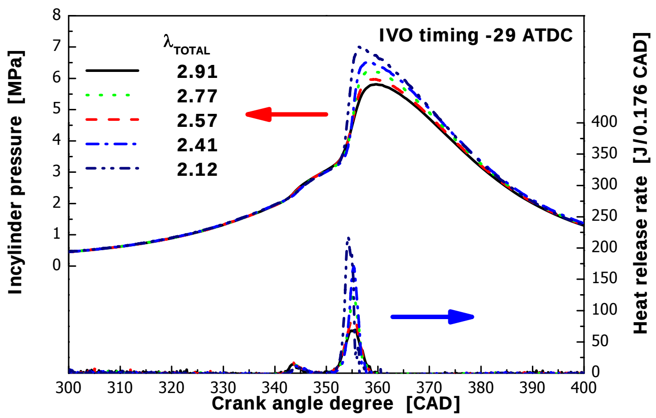



Figure 5 Cylinder combustion pressure and heat release rate of gasoline HCCI engine at 1000 rpm as a function of $\lambda_{TOTAL}$

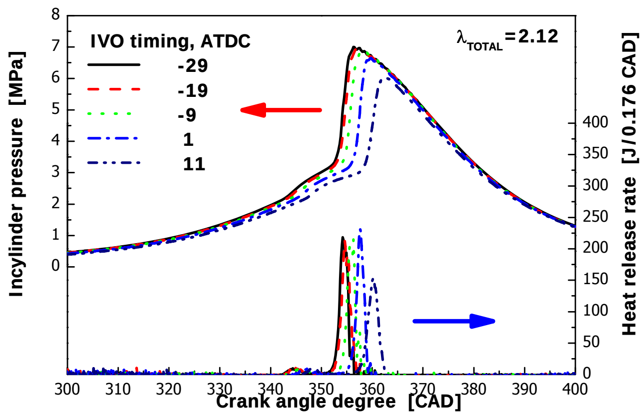



Figure 6 Cylinder combustion pressure and heat release rate of gasoline HCCI engine at 1000 rpm as a function of IVO timing

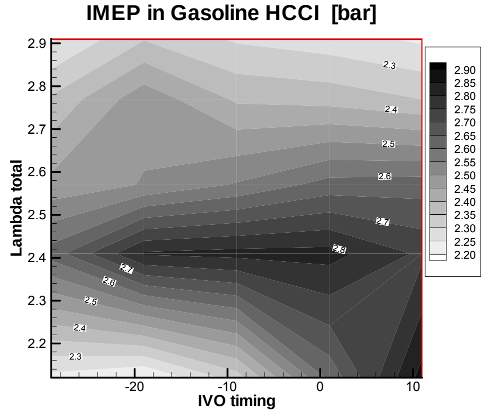



Figure 7 IMEP of gasoline HCCI engine with respect to $\lambda_{TOTAL}$ and IVO timing

efficiency and the amount of residual gas decrease due to late intake valve opening and closing. This reduced volumetric efficiency and residual gas lead to late combustion, which reduces the negative work.

DME-GASOLINE EXHAUST EMISSIONS - Figure 8 shows the HC emission from a gasoline HCCI engine with respect to $\lambda_{TOTAL}$ and the IVO timing. The HC emission increased as the IVO timing was retarded. The increased HC emission was due to delayed combustion. Late combustion leads to lower combustion pressure and temperature. The fuel that was not burned during the combustion process becomes the source of the HC emission. The HC emission could be oxidized by heat from the burned gas during the expansion stroke. However, the HCCI engine combustion temperature is lower than that of SI and CI engines. Thus, the main causes of increased HC emission from the HCCI engine are the strong quenching effect and the lack of an oxidation reaction during the expansion stroke. In addition, in the case of late IVO timing, the quenching effect was stronger than that for early IVO timing due to the low combustion temperature. This implies that lower combustion temperature creates more HC emission through reduced oxidation and increased quenching in the case of late IVO timing relative to early IVO timing.

The CO emission map is shown in Fig. 9. The upper right region of the CO emission map shows higher CO concentration than the other regions of the map. The combustion temperature was lower than the early IVO timing and low $\lambda_{TOTAL}$ conditions.

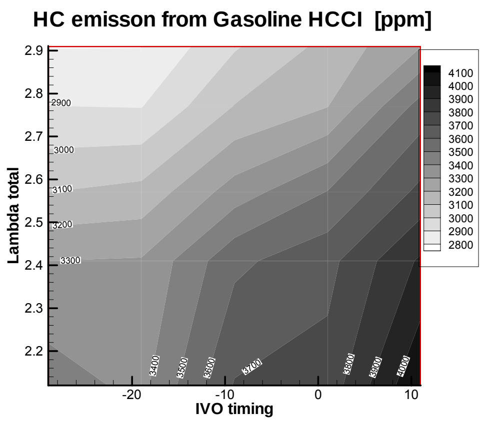



Figure 8 HC emission from a gasoline HCCI engine with respect to $\lambda_{TOTAL}$ and IVO timing

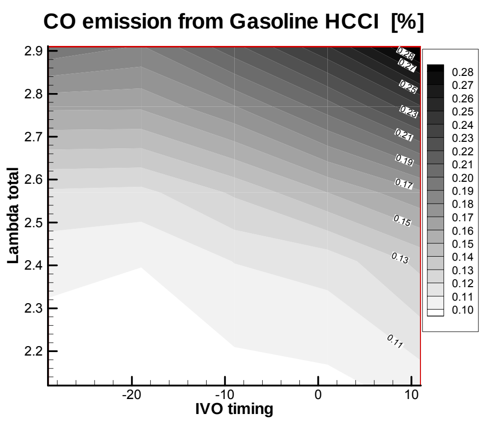



Figure 9 CO emission from gasoline HCCI engine with respect to $\lambda_{TOTAL}$ and IVO timing

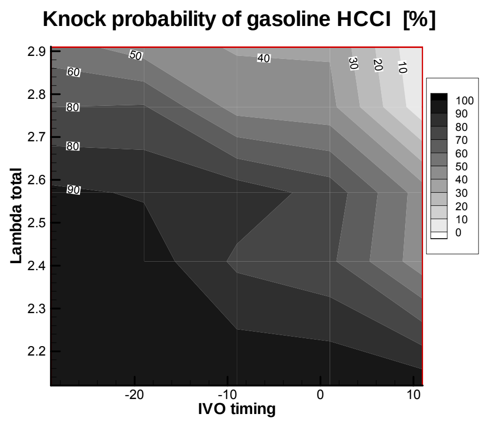



Figure 10 Knock probability of gasoline HCCI engine with respect to $\lambda_{TOTAL}$ and IVO timing at 1000 rpm

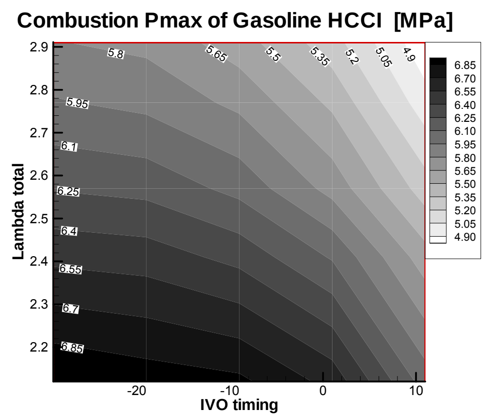



Figure 11 Maximum combustion pressure of gasoline HCCI engine with respect to $\lambda_{TOTAL}$ and IVO timing at 1000 rpm

Also, the oxidation reaction during the expansion stroke was weaker than the other conditions. However, in the case of HC emission, HC emission increased as $\lambda_{TOTAL}$ was decreased. This is because the increasing rate of produced HC emission during the combustion process was larger than the increasing rate of oxidized HC emission during the expansion stroke.

DME-GASOLINE KNOCKING CHARACTERISTICS - Figure 10 shows the knock probability of the gasoline HCCI. Every 100 cycle's cylinder pressure data for each experimental condition was used for the knock probability analysis. The knock probability was increased as $\lambda_{TOTAL}$ was decreased and the IVO timing was advanced. The reduced $\lambda_{TOTAL}$ and advanced IVO timing lead to higher combustion pressure and an increased rate of combustion pressure rise.

Figure 11 presents the maximum pressure rise of combustion with respect to the $\lambda_{TOTAL}$ and IVO timing. The maximum pressure rise of combustion increased as the $\lambda_{TOTAL}$ was decreased and IVO timing was advanced. Also, the knock probability of HCCI combustion increased as the maximum combustion pressure rise was increased, because the rich mixture results in vigorous combustion. Moreover, early intake valve closing allows more charge into the cylinder and traps more hot residual gas in the cylinder. Hence $P_{max}$ and maximum combustion pressure rise are key parameters of the knocking probability of HCCI engine combustion. The knock probability with respect to the $P_{max}$ is shown in Fig. 12. From every 25 engine operating conditions, which were tested in this study, the relation of the knock probability and the $P_{max}$ can be derived via a linear equation (3).

$$y = 47.2x - 215.15 \quad (3)$$

The equation shows the relations of cylinder pressure and knock probability. The knock probability can be estimated by monitoring the combustion pressure.

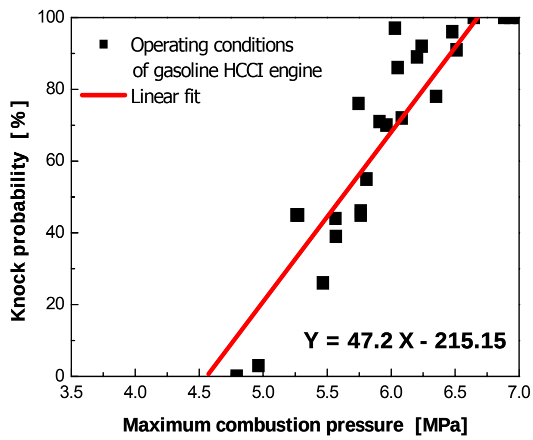



Figure 12 Knock probability of gasoline HCCI engine with respect to maximum combustion pressure at 1000 rpm

## CONCLUSION

The effects of IVO timing and fuel quantity on exhaust emissions and combustion characteristics in 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 following conclusions were drawn from the experimental results.

1. Combustion retardation was due to lower volumetric efficiency and internal residual gas.
2. The start of combustion was independent of the gasoline injection quantity.
3. The IMEP was determined by the gasoline injection quantity.
4. On the basis of IMEP drop, the high load operating range of the test engine was limited to under 0.5 $MW/m^2$.
5. High combustion pressure and pressure rise were main causes of knocking

6. The knock probability can be predicted by two derived equations.
7. The HC emission was increased as the IVO timing was retarded and $\lambda_{TOTAL}$ was decreased.
8. The CO emission was increased as the IVO timing was retarded and $\lambda_{TOTAL}$ was increased.

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## DEFINITIONS, ACRONYMS, ABBREVIATIONS

Nomenclature
λ : relative air fuel ratio
θ : crank angle degree
κ : specific heat ratio
P : cylinder combustion pressure
Q : heat release
V : cylinder volume

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 over head camshaft
DME : di-methyl ether
ECU : engine control unit
HC : hydrocarbon
HCCI : Homogeneous charge compression ignition
HTR : high temperature reaction
IMEP : indicated mean effective pressure
IVO : intake valve open
LHV : low heating value
LTR : low temperature reaction
MFB : mass fraction burned
NOx : nitrogen oxides
SI : spark ignition
VVT : variable valve timing