

Improvement of DME HCCI Engine Performance by Fuel Injection Strategies and EGR

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ABSTRACT

The combustion and exhaust emission characteristics of a DME fueled HCCI engine were investigated. Different fuel injection strategies were tested under various injection quantities and timings with exhaust gas recirculation (EGR). The combustion phase in HCCI was changed by an in-cylinder direct injection and EGR, due to changes in the in-cylinder temperature and mixture homogeneity. The gross indicated mean effective pressure ($IMEP_{gross}$) increased and the hydrocarbon (HC) and carbon monoxide (CO) emissions decreased as the equivalence ratio was augmented. The $IMEP_{gross}$ with direct injection was greater than with the port injection due to retarded ignition timing resulting from latent heat of direct injected DME fuel. It was because that most of burn duration was completed before top dead center owing to higher ignitability for DME with high cetane number. However, HC and CO emissions were similar for both injection locations. In the case of direct injection, the $IMEP_{gross}$ increased until direct injection timing was 260 crank angle degrees (CAD) and decreased afterwards. In the case of port injection, the injection timing was not an influential parameter for $IMEP_{gross}$ change as the combustion characteristic changed from a relatively lean to locally-rich combustion due to a change in the mixture homogeneity with the delay of direct injection timing. Locally-rich combustion is supposed to lead to a higher combustion temperature, resulting in nitrogen oxide (NOx) formation at the retarded injection timing. In this study, optimal direct injection timing for HCCI combustion was approximately 260 CAD. NOx increased after direct injection timing was retarded more than 260 CAD. EGR could increase $IMEP_{gross}$ because combustion phase was shifted from before to after top dead center due to tardy combustion, while HC and CO emissions increased due to the lower combustion temperature. The combustion efficiency was improved by direct injection and a higher equivalence ratio, resulting in a higher combustion temperature. The thermal efficiency was increased due to late combustion,

as the EGR rate was augmented and the direct injection timing was retarded. Additionally, there was a drop of thermal efficiency due to an increased equivalence ratio resulting in early auto-ignition.

INTRODUCTION

The homogeneous charge compression ignition (HCCI) engine is a promising concept for future automobile engines and stationary power plants. In HCCI, a premixed air/fuel mixture is inhaled as in conventional spark ignition (SI) engines, it is auto-ignited as in conventional compression ignition (CI) engines. HCCI uses spontaneous ignition, which is governed primarily by chemical kinetics during the compression process with the premixed air/fuel mixture. Basic characteristics of the HCCI engine, such as very little cyclic variation and no flame propagation, have been studied by previous researchers [1,2]. HCCI engines emit less nitrogen oxide (NOx) than conventional engines due to the low combustion temperature resulting from the high air excess ratio [3]. Particulate matter (PM) emitted from the HCCI engine is also less than conventional CI engines due to the lean homogeneous charge mixture [3-5]. However, HCCI engine operation has some disadvantages, such as difficulty in controlling the combustion phase, a much lower maximum load than typical SI and CI engines, high pressure and noise at a low air excess ratio or high load condition, and higher carbon monoxide (CO) and hydrocarbon (HC) emissions than SI and CI engines due to the low combustion temperature [4,6].

Dimethyl ether (DME) is considered an alternative fuel for the CI engine [7]. DME is a high cetane numbered fuel similar to diesel that does not produce soot due to its molecular structure. DME has no double C-C bonds and contains oxygen molecules which oxidize intermediate soot product. DME is easily evaporated and mixed well with the air so that the rich regions avoided, leading to soot-free combustion. The ignition delay of

DME in a CI engine is shorter than that of diesel under the same operating conditions as the evaporation rate of DME is two or three times faster than diesel [8-10]. Because of the shorter ignition delay, various control schemes to change the auto-ignition timing have been proposed in DME HCCI such as internal and/or external exhaust gas recirculation (EGR), fuel mixing, and additives [11].

EGR is considered an effective method for controlling the combustion phase and reducing exhaust emissions. In SI engines, EGR can reduce NO_x emissions without the penalties of an increase in other emissions or excessive fuel consumption [12]. The effects of EGR are classified as dilution, thermal, or chemical effects [13,14]. The three EGR effects affect the combustion rate, burn duration, combustion temperature, and exhaust emission. In HCCI, thermal and dilution effects are more important than chemical effects of EGR due to the lower combustion temperature. The retarded onset of auto-ignition with EGR for a fixed induction air temperature is due to the thermal effect of the cooler compressed gas temperatures during the compression stroke [15-17]. Oxygen (O₂) dilution due to EGR retards the auto-ignition timing [17]. The heated intake charge by EGR advances the auto-ignition timing [18].

There is a potential for improved volumetric efficiency due to fuel evaporation in the cylinder resulting in increased air density with the direct injection during induction stroke. Fuel stratification by direct injection can also improve the HCCI engine with variation of the injection timing [19]. Therefore, Direct injection can be one of the ways to improve emissions and power in HCCI engine.

In this study, the effect of fuel injection strategy and EGR on an HCCI engine combustion fueled with DME was investigated. Output performance and emission were characterized with EGR rate and for the various injection locations in port and cylinder at different injection timings.

EXPERIMENTAL SYSTEM AND CONDITIONS

ENGINE – Figure 1 shows a schematic diagram of the experimental apparatus used in this study. The detailed specifications of the test engine are presented in Table 1. The engine was a four-stroke, water-cooled, and single-cylinder double overhead camshaft (DOHC) engine. The engine speed and load were controlled by an alternating current (AC) dynamometer (82 kW, Unico Co.). The DME fuel injector (Denso Co.) was placed upstream, 30 cm away from the intake port, to produce a homogeneous air/fuel mixture. DME was pressurized to 5 MPa by nitrogen gas. The DME injector was inclined at 30° to improve air/fuel mixing. A small amount (500 ppm) of lubrication enhancer (Infineum, R655) was added to the DME to protect the injector and fuel feeding system.

MEASUREMENT SYSTEM – The in-cylinder pressure was measured by a piezoelectric pressure transducer (Kistler, 6052b). The intake and exhaust pressures were

gauged by two piezoresistive pressure transducers (Kistler, 4045A5). The intake and exhaust gas temperatures were measured by two thermocouples (K-type) near the intake and exhaust ports, respectively. The exhaust gases were analyzed with a gas analyzer (HORIBA MEXA-1500d) to measure the HC, CO, CO₂, NO_x and O₂ in the exhaust and CO₂ in the intake. The experimental data were acquired by a data acquisition system (IO Tech, Wavebook 516 series with 12 bits of resolution, 1 s/channel sampling rate, and an accuracy of 0.025%).

EXPERIMENTAL CONDITIONS – To compare the effect of injection location on the DME HCCI combustion, the injection location was changed between the port and in-cylinder. The equivalence ratio (ϕ) was changed from 0.2 to 0.35. In-cylinder direct injection timing was varied from 20 to 350 CAD to investigate the effect of injection timing. The EGR rate was changed from 0 to 60% to improve the IMEP_{gross}.

In-cylinder and intake/exhaust pressure data for one hundred consecutive engine cycles were used for the analysis of combustion and performance [20]. To catch the rapid change of the auto-ignition timing, the temporal resolution of the acquisition data was approximately 0.175 CAD.

Figure 2 shows the injection timing of the port and in-cylinder direct injections. The injection timing of the port injection was maintained at 20 CAD and the direct injection was changed from 20 to 350 CAD.

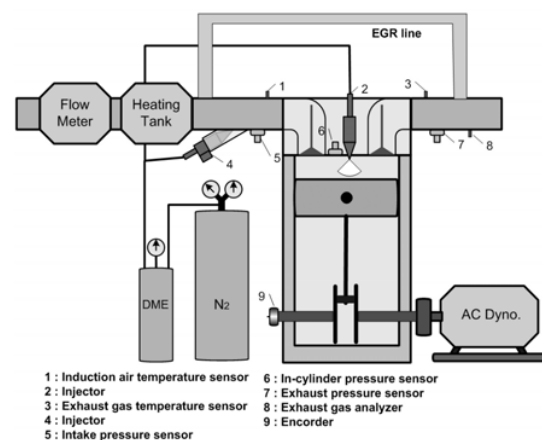


Figure 1 Experimental apparatus

Table 1 Engine specification

Bore (mm)		82
Stroke (mm)		93.5
Compression ratio		13
Intake/ Exhaust valve duration		228/228
Intake/ Exhaust lift duration (CAD)		8.5/8.4
Valve timing (CAD)	IVO (BTDC)	1
	IVC (ABDC)	39
	EVO (BBDC)	42
	EVC (ATDC)	6
DME injection pressure (MPa)		5

Table 2 Experimental conditions

Engine speed (rpm)	1,000
Equivalence ratio (ϕ)	0.21 ~ 0.35
Injection location	Port/In-cylinder
Injection timing (CAD)	20 ~ 350
EGR rate (%)	0 ~ 52

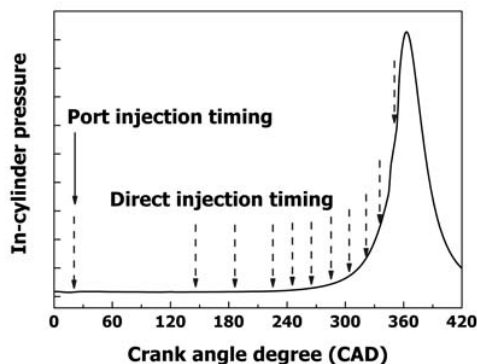


Figure 2 DME injection timing.

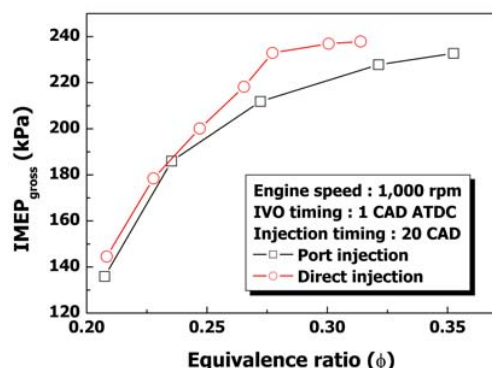


Figure 3 IMEP_{gross} comparison between port and direct injection as a function of injection location and equivalence ratio

RESULTS AND DISCUSSION

LOCATION OF DME INJECTION – The injection location of the DME fuel was at the port and center of the in-cylinder. The injection quantity of DME was varied and the injection timing was maintained at 20 crank angle degree (CAD) after top dead center (TDC) of intake. Figure 3 shows the gross indicated mean effective pressure (IMEP_{gross}) variation as a function of injection location and injected DME quantity. There was similar result between the port injection and in-cylinder direct injection with regards to the convergent tendency of IMEP_{gross} as injected DME quantity was increased. The ignition timing was advanced and the burn duration

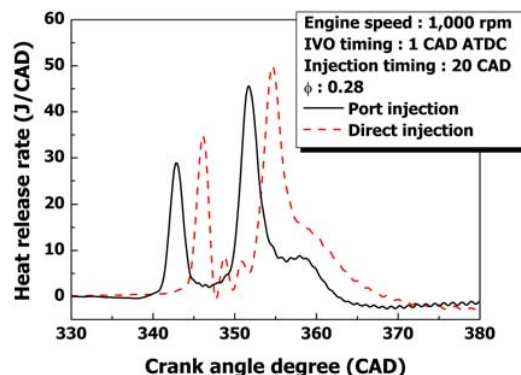


Figure 4 Heat release rate comparison between port and direct injection

was reduced during the compression stroke as DME quantity increased, resulting in increased negative work. To this end, the IMEP_{gross} did not increase further.

The IMEP_{gross} at the direct injection under same injection timing with port injection was higher than at the port injection due to the elongated ignition delay resulting from slow reaction rate, which was mainly affected by lower mixture temperature. The injected fuel at the port produced a homogeneous mixture during the induction stroke without additional temperature drop in the cylinder. However, there was a temperature drop due to the latent heat of liquid DME when it was injected into the cylinder. Auto-ignition timing was retarded and negative work of the direct injection decreased due to the temperature drop with the direct injection of DME fuel. Therefore, the IMEP_{gross} of the direct injection was bigger than the port injection. The heat release rate, as shown in Fig. 4, represented the late auto-ignition with direct injection. The start of the heat release with direct injection was more delayed than with the port injection while the burn duration was similar.

In HCCI combustion, nitrogen oxide (NO_x) emission is very low due to the lower combustion temperature. In this section, NO_x emission was not presented because the value was lower than 50 ppm. Hydrocarbon (HC) and carbon monoxide (CO) are presented in Fig. 5 as a function of injection location and equivalence ratio. HC and CO emission decreased as the equivalence ratio increased. The oxidation of HC and CO was more vigorous with increased combustion temperature due to a higher equivalence ratio in the combustion engine. HC emission with direct injection was lower than the port injection at a higher equivalence ratio, while CO emission with direct injection was similar to the port injection. The main source of HC emission was the air/fuel mixture of the crevice volume. These mixtures were incompletely oxidized up to the end of the expansion stroke due to the lower combustion temperature resulting in severe quenching. The amount of the air/fuel mixture in the crevice region was reduced by direct injection, because the premixed mixture with direct injection was less than the port injection.