ABSTRACT

In this study, a spectroscopic method was used to measure the combustion characteristics of a test diesel engine when operated on dimethyl ether (DME) under a homogenous charge compression ignition (HCCI) combustion process. A numerical analysis was made of the elementary reactions using Chemkin 4.0 to perform the calculations. The results of the analysis showed that compression ratio changes and the methane additive influenced the autoignition timing in the DME-HCCI combustion process. In the experiments, reducing the compression ratio delayed the time of the peak cylinder pressure until after top dead center, thereby increasing the crankshaft output and thermal efficiency. The addition of methane enabled the DME-HCCI engine to provide crankshaft output equivalent to that seen for diesel engine operation at a low equivalence ratio. This paper discusses these effects in reference to the experimental and calculated results.

INTRODUCTION

In recent years, a great deal of research has been done on homogeneous charge compression ignition (HCCI) combustion [1,2,3,4,5], which can simultaneously reduce emissions of nitrogen oxides (NOx) and particulate matter (PM) that usually require a trade-off with respect to the combustion gas temperature. The HCCI combustion process has the advantage of obtaining both high efficiency and low emissions because the engine can operate in a lean mixture region beyond the ordinary limits of flame propagation. However, HCCI combustion has the drawback that it is difficult to control the ignition timing and the combustion period.

The objective of this study was to gain further insights for expanding the operating region of HCCI engines and achieving better control of ignition timing. A spectroscopic method was used to investigate the behavior of combustion intermediate products, and heat release histories were measured to investigate the factors influencing the development of low-temperature flames that influence the occurrence of autoignition. Dimethyl ether (DME) was used as the main test fuel and methane was used as a fuel additive. One feature of DME is that it can be produced from a wide variety of materials. It is also characterized as being a clean energy source because its molecules contain oxygen but not sulfur. Furthermore, its cetane number is equal to or higher than that of diesel fuel. For these reasons, it has attracted attention in recent years as an alternative fuel for diesel engines.

This paper discusses the influence of compression ratio changes and the methane additive on the combustion characteristics of an HCCI test engine when operated on DME. A comparison is also made between the experimental data and the results of calculations performed with Chemkin 4.0 in a numerical analysis of the elementary reactions.

TEST FUELS

CHARACTERISTICS OF DME AND METHANE

The main test fuel used in the experiments was dimethyl ether (DME). In the blended fuel experiments, methane was added to the main fuel of DME to create the test fuel. The properties of each fuel are given in Table 1 [6].

DME has attracted interest as an alternative fuel for HCCI engines [7]. This fuel has a negative temperature coefficient region in which the ignition delay is not shortened even if the temperature is higher at the onset of compression. The combustion process is characterized by multiple-stage heat release attributed to low-temperature reactions (LTR) and to high-temperature reactions (HTR).

Because methane has a high autoignition temperature and a cetane number of 0, mixing it with DME is an
effective way to control the ignition timing. While DME shows a two-stage heat release pattern due to LTR and HTR, methane displays only a single-stage heat release pattern induced by HTR.

**OXIDATION PROCESSES OF DME AND DME/METHANE**

The oxidation reaction process of the DME/methane fuel blend is outlined in Fig. 1 [8,9]. DME reactions are divided into two processes: one that proceeds from the first addition of O₂ and follows a path involving a second addition of O₂ depending on the temperature region (denoted as (1) in the figure), and another process in which the reaction proceeds without the addition of O₂ (denoted as (2)). In the low-temperature region below 800 K, the reaction is accelerated by a chain-branching step (cool flame region). As the temperature rises further, the process changes to reaction (2), which is a chain propagation step, so the acceleration of the reaction ceases (negative temperature coefficient region, NTC [10,11,12,13] A subsequent temperature rise induces reaction (3), resulting in the production of excess OH radicals (blue flame region) and leading to autoignition. In conjunction with the temperature rise, two-stage ignition [14] occurs owing to the progression from a cool flame through the NTC region to autoignition, as the reactions proceed from (1) to (3).

In the case of the DME/methane fuel blend, reaction (1) induced by DME takes place in the low-temperature region. As the temperature subsequently increases, the process proceeds to reaction (3), at which point the OH radicals produced from H₂O₂ are supplied to methane. Methane actively undergoes its initial H-atom abstraction reaction, and its oxidation reaction proceeds to reaction (5). These reactions give rise to two- or even three-stage ignition. It is reported that the reaction process here involves the production of HCHO and CO by DME (LTR), the production of CO by DME (1st HTR), and the oxidation of CO and methane (2nd HTR).

**EXPERIMENTAL CONDITIONS**

**TEST EQUIPMENT AND CONDITIONS**

The test engine used in this study was an air-cooled single-cylinder diesel engine, having the specifications given in Table 2. The configuration of the test equipment used is shown in Fig. 2.

A crystal pressure transducer was installed in the top of the cylinder head to measure the cylinder pressure. In order to investigate the engine operating conditions, measurements were made of the combustion chamber inner wall temperature Tₖ, intake air temperature Tᵢ, and exhaust gas temperature Tₑ. All fuels were introduced into a premixer [15], installed approximately 1,000 mm upstream of the intake valve, for formation of the premixed mixture. The configuration of the fuel supply system is shown schematically in Fig. 3.

The heat release rate and radical light emission behavior were analyzed under each set of operating conditions. The LTR and HTR definitions used in analyzing the heat release rate are shown in Fig. 4. The onset time of LTR was defined as the point when the heat release rate showed a positive value (i.e., greater than 0) and the onset time of HTR was defined as the point of the lowest heat release rate in the interval between the peak heat release rate of LTR and the peak heat release rate of HTR (i.e., the point where the heat release rate began to rise again after decreasing). Flame light in the combustion chamber was sampled by means of a quartz detector.

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**Table 1 Properties of test fuels**

<table>
<thead>
<tr>
<th>Fuel</th>
<th>DME</th>
<th>Methane</th>
</tr>
</thead>
<tbody>
<tr>
<td>Molecular Formula</td>
<td>CH₃OCH₃</td>
<td>CH₄</td>
</tr>
<tr>
<td>Cetane Number</td>
<td>&gt;55</td>
<td>0</td>
</tr>
<tr>
<td>Auto-ignition Temp [K]</td>
<td>623</td>
<td>905</td>
</tr>
<tr>
<td>Low Heat Release Value [MJ/kg]</td>
<td>28.4</td>
<td>50.3</td>
</tr>
<tr>
<td>Stoichiometric A/F Ratio [kg/kg]</td>
<td>8.96</td>
<td>16.9</td>
</tr>
</tbody>
</table>

**Table 2 Specifications of test engine**

<table>
<thead>
<tr>
<th>4-Cycle Air-cooled Diesel Engine</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of Cylinders</td>
</tr>
<tr>
<td>Bore×Stroke</td>
</tr>
<tr>
<td>Displacement</td>
</tr>
<tr>
<td>Compression Ratio (Diesel)</td>
</tr>
<tr>
<td>Compression Ratio (DME/methane)</td>
</tr>
<tr>
<td>Valve Arrangement</td>
</tr>
<tr>
<td>Type of Piston</td>
</tr>
</tbody>
</table>
observation window holder attached to the top of the cylinder head. The sampled flame light passed through an optical fiber cable having a core diameter of 1 mm and was introduced into a polychromator [16,17], which separated the light into wavelengths of 395.2 nm, corresponding to the characteristic spectrum of HCHO, and 306.4 nm, corresponding to that of the OH radical [18,19,20]. The wavelength resolution of the polychromator was 4.0 nm. Following separation, the light at each wavelength was fed into a photomultiplier, and the respective output voltage was regarded as the light emission intensity.

A spacer was inserted between the cylinder and the crank case to vary the engine compression ratio $\varepsilon$ while running the engine at a constant speed of $N = 1400$ rpm. The compression ratio $\varepsilon$ was reduced in increments of one over a range from 19:1 to 10:1. In the experiments, the equivalence ratio $\phi$ of the fuels was varied in a range that allowed the test engine to be operated at these compression ratios.

In the experiments using the DME/methane fuel blend, the compression ratio was varied among three levels of $\varepsilon = 12$, 15 and 18:1, while keeping the engine speed constant at 1400 rpm. The equivalence ratio of DME in the fuel blend was kept constant, and that of methane was varied in a range that did not exceed the equivalence ratio of DME.

NUMERICAL ANALYSIS OF ELEMENTARY REACTIONS

The oxidation reactions of DME were analyzed numerically and the results were compared with the data measured experimentally with the test engine. The IC Engine software in Chemkin 4.0 was used to perform the calculations in the numerical analysis of the elementary reactions [21]. The scheme used was the elementary reaction model proposed by Curran et al. [22,23,24], involving 336 elementary reactions and 78 chemical species. The operating conditions used in the calculations were the same as those used in the experiments.

RESULTS AND DISCUSSION

INFLUENCE OF VARYING COMPRESSION RATIOS ON DME-HCCI COMBUSTION

Waveforms measured for various compression ratios

Figures 5 and 6 show typical waveforms measured for engine operation under HCCI combustion at a constant engine speed $N$ of 1400 rpm, a relatively low equivalence ratio $\phi$ of 0.32 and varying compression ratios $\varepsilon$ of 19, 17, 15, 13, 12, and 11:1. The crank angle $\theta$ (deg.) is shown along the horizontal axis. From the top, the vertical axes indicate the cylinder pressure $P$ (MPa), heat release rate (H.R.R., J/deg.) found geometrically from an indicator diagram, and the light emission intensity $E_{\text{OH}}$ (V) at 306.4 nm, corresponding to the OH radical [25,26], and the light emission intensity $E_{\text{HCHO}}$ (V) at 395.2 nm, corresponding to HCHO [25,26,27], at the observation point in the
combustion chamber. The charging efficiency $\eta_c (%)$, exhaust gas temperature $T_{ex}$ (K) and combustion chamber inner wall temperature $T_w$ (K) are shown below the crank angle as indicators of the engine operating state in this experiment.

The cylinder pressure waveforms in Figs. 5 and 6 show that the cylinder pressure began to rise sharply at an earlier crank angle with a higher compression ratio. In the case of a low compression ratio of $\varepsilon = 12:1$ or $11:1$, the sharp rise in the cylinder pressure waveform occurred in the vicinity of top dead center (TDC). These results indicate that lowering the compression ratio shifted the peak cylinder pressure to a point after TDC.

The H.R.R. waveforms in the figures mainly show a pattern of two-stage heat release. This pattern of multistage heat release is characteristic of compression ignition combustion of DME. The first and second stages are indicative of LTR and HTR, respectively. It is said that compression ignition reactions with DME theoretically show a pattern of three-stage heat release, consisting of the first-stage LTR that produce formaldehyde, second-stage HTR that produce the CO radical, and third-stage HTR that produce CO$_2$. However, as reported [28] elsewhere, it has been observed that the second- and third-stage reactions may be integrated into a single stage accompanying an increase in the equivalence ratio. The results obtained in this study showed that the second- and third-stage reactions were also integrated at a low equivalence ratio as a result of increasing the compression ratio.

Figure 7 shows typical heat release waveforms measured for HCCI combustion at an equivalence ratio of $\phi = 0.36$ and varying compression ratios of $\varepsilon = 19$, 17, 15, 13, 12 and 11:1. Line a is for the LTR onset time $\theta_{LS}$ and line b is for the HTR onset time $\theta_{HS}$. In Fig. 8, the LTR onset time $\theta_{LS}$ is shown along the x-axis, the equivalence ratio along the y-axis and the maximum H.R.R. $Q_{lmax}$ of LTR along the z-axis. It is seen in Figs. 7 and 8 that the LTR onset time shifted to a later crank angle as the compression ratio was reduced. As the compression ratio is reduced, the compression pressure at the same crank angle is lower compared with that for a higher compression ratio.
Accordingly, it takes a longer crank angle interval for the in-cylinder gas to reach the temperature level for the onset of LTR, which explains this shift to a later crank angle. The temperature rise originating from the LTR is also delayed to the extent that $\theta_{HS}$ shifts to a later crank angle, which means that the HTR onset time $\theta_{HS}$ also moves to a later crank angle as a result. The maximum H.R.R. $Q_{Lmax}$ of LTR increased with a lower compression ratio.

Figure 9 shows the LTR duration as a function of the equivalence ratio $\phi$. The LTR duration $L$ became longer as the compression ratio was reduced. That can be explained by the fact that the reactions slowed down and proceeded at a slow rate accompanying a reduction of the compression ratio. The heat release from LTR is shown in Fig. 10 in relation to the equivalence ratio $\phi$. It is seen that the heat release from LTR increased at a lower compression ratio. Three main factors are thought to be responsible for increasing the in-cylinder gas temperature in the progression to HTR. They are the temperature rise due to compression of the gas by piston motion, the temperature rise due to heat release from LTR, and the transfer of heat to the gas from the combustion chamber walls. In the crank angle interval from 90 deg. before top dead center (BTDC) to TDC, the piston speed becomes slower at later crank angles, with the result that the gas temperature rises more slowly due to compression. Accordingly, the progress of LTR during that interval presumably accounted for the increased heat release.

With a high equivalence ratio of $\phi = 0.40$ or more and a low compression ratio $\varepsilon$ of 11:1, the heat release from LTR decreased slightly. It is also seen in Fig. 10 that, at a compression ratio of 11:1, the LTR duration showed a larger rate of decrease with an increasing equivalence ratio compared with the other compression ratios. At a compression ratio of 11:1, it is assumed that heat transfer from the cylinder walls and the accumulated energy of the in-cylinder gas had a large effect on the gas temperature, with the result that the temperature level for the onset of high-temperature oxidation reactions was reached in a short period of time, which would account for the shorter LTR duration. Accordingly, even though the LTR maximum H.R.R. $Q_{Lmax}$ increased in the region of a high equivalence ratio at a compression ratio of 11:1, the shorter LTR duration presumably reduced the amount of heat released from LTR.

**Light emission behavior of intermediate products**

The light emission intensity waveforms of the intermediate combustion products in Figs. 5 and 6 indicate that the onset of a sharp rise in light emission intensity shifted to a later crank angle as the compression ratio was reduced. That shift to a later crank angle occurred incidental to the HTR onset time $\theta_{HS}$. Because the H.R.R. increased during the duration of LTR, it can be inferred that a low-temperature flame (cool flame) occurred in that interval. However, no corresponding behavior is observed in the light emission intensity waveform in the experimental results for $\phi = 0.32$ because light emission from a low-temperature flame is extremely faint.

Figure 11 shows typical waveforms measured for engine operation under DME-HCCI combustion at a compression ratio $\varepsilon$ of 12:1 and an equivalence ratio $\phi$ of 0.46. Figure 12 shows the waveforms measured under the same operating conditions when the resolution of the polychromator was set slightly higher than the normal level for HCHO alone. In the region denoted as A in Fig. 11, light emission behavior corresponding to HCHO can be observed in the LTR interval. The degeneracy of light emission behavior can also be seen in the LTR interval in the region denoted as B in Fig. 12.
J/deg.), in-cylinder gas temperature (K) and the mole fractions of the OH radical, HCHO, H2O2, CO radical and CO2. The vertical axis of the chemical species graph is given in logarithmic notation.

The H.R.R. waveform in Fig.13 shows a two-stage heat release pattern attributable to low- and high-temperature oxidation reactions. It is seen in the simulation results that the mole fraction of HCHO increased gradually at the point where heat release was manifested due to the low-temperature oxidation reactions. The numerical calculations of the elementary reactions were done at a zero dimension and under an adiabatic condition. Consequently, the simulation results do not accurately reproduce the experimental data, which were influenced by the in-cylinder flow of the combustion gas and heat loss. However, because both sets of results are for phenomena induced by low-temperature oxidation reactions during the heat release period, it is presumed that the light emission observed in the test engine at a wavelength corresponding to that of HCHO was due to the production of HCHO.

On the other hand, the OH radical concentration is low compared with that of HCHO. The production of the CO radical that characterizes high-temperature oxidation reactions rises together with the extinction of HCHO and eventually the reaction proceeds to the production of CO2. In other words, it is assumed that the sharp rise in light emission intensity observed in the waveforms measured experimentally with the test engine represents the detection of the continuous light emission spectrum of CO-O radiation that occurred in this process.

Expansion of operation region

Figure 14 shows the engine operating region for DME-HCCI combustion at various compression ratios and a constant engine speed of 1400 rpm. The knocking region indicated for each compression ratio is based on a judgment of noise resembling knocking during engine operation under that condition. It was judged that the engine misfired when the dynamometer reading was zero. As the compression ratio was lowered, it was possible to operate the test engine until a relatively high equivalence ratio. Stable operation was possible from an equivalence ratio $\phi$ of 0.23 at each compression ratio. In the case of a relatively high compression ratio, it was possible to operate the engine with torque output at an equivalence ratio below $\phi = 0.23$.

Improvement of crankshaft output and thermal efficiency

Figure 15 shows the crankshaft output Le measured in relation to the equivalence ratio for engine operation under DME-HCCI combustion at various compression ratios $\varepsilon$ from 19:1 to 11:1. The figure also shows the crankshaft output measured in relation to the equivalence ratio when the test engine was operated under its regular specifications (Table 1) at a compression ratio $\varepsilon$ of 21:1 using diesel fuel. It is seen that reducing the compression
ratio increased the crankshaft output. That is thought to be attributable to delayed ignition timing (i.e., combustion in the vicinity of TDC) and a higher equivalence ratio in the operable region of the engine. A lower compression ratio and a higher equivalence ratio made it possible to achieve crankshaft output from DME-HCCI combustion that approached the output seen for diesel operation at a lower equivalence ratio.

Reducing the compression ratio under DME-HCCI combustion retarded the time of the peak cylinder pressure to a point after TDC, thereby increasing crankshaft output and brake thermal efficiency. In the experiments, the highest brake thermal efficiency was obtained at a compression ratio $\varepsilon$ of 12:1 and the highest crankshaft output at $\varepsilon = 11:1$. The engine misfired at $\varepsilon =10:1$ and stable combustion could not be maintained.

**INFLUENCE OF ADDING METHANE**

**Operation region**

The stable operating region obtained at a compression ratio $\varepsilon$ of 12:1, 15:1 and 18:1 is shown in Figs. 17, 18 and 19, respectively. The operable region was defined as the region in which neither knocking nor misfiring occurred and the methane-based equivalence ratio $\phi_{\text{CH}_4}$ did not exceed the DME-based equivalence ratio $\phi_{\text{DME}}$. At compression ratios of $\varepsilon = 15:1$ and 18:1, measurements were made from $\phi_{\text{DME}} = 0.20$.

It is seen in Fig. 17 that the engine misfired when too much methane was added in the region of a low equivalence ratio at a compression ratio $\varepsilon$ of 12:1. In these experiments, misfire occurred at a combustion chamber inner wall temperature $T_W$ of 80°C when the amount of methane added exceeded a methane-based equivalence ratio $\phi_{\text{CH}_4}$ of 0.18 at a constant DME-based equivalence ratio $\phi_{\text{DME}}$ of 0.25. It was possible to steadily add more methane until the DME-based equivalence ratio $\phi_{\text{DME}}$ reached 0.30, but as $\phi_{\text{DME}}$ increased beyond
that level, the allowable region for the addition of methane narrowed.

It can also be seen in Figs. 17, 18 and 19 that the amount of methane which could be added relative to the DME-based equivalence ratio decreased with a higher compression ratio. For example, in the case of \( \phi_{\text{DME}} = 0.30 \), methane could be added to \( \phi_{\text{CH}_4} = 0.30 \) at a compression ratio \( \varepsilon \) of 12:1, but the allowable \( \phi_{\text{CH}_4} \) decreased to 0.16 at \( \varepsilon = 15:1 \) and to 0.10 at \( \varepsilon = 18:1 \). As the compression ratio was increased, the amount of methane that could be added in relation to the same quantity of DME was restricted by the occurrence of knocking.

Addition of methane at constant DME-based equivalence ratio \( \phi_{\text{DME}} \)

Figure 20-(a) shows typical examples of the waveforms measured when methane was added at various methane-based equivalence ratios of \( \phi_{\text{CH}_4} = 0, 0.10, 0.18 \) and 0.30 while keeping the DME-based equivalence ratio constant at \( \phi_{\text{DME}} = 0.30 \). The test engine was operated at an engine speed \( N \) of 1400 rpm and a compression ratio \( \varepsilon \) of 12:1. Figure 20-(b) shows an enlarged representation of the H.R.R. waveforms, and Fig. 20-(c) shows the crankshaft output as a function of \( \phi_{\text{CH}_4} \), which was increased in increments of 0.02 while keeping \( \phi_{\text{DME}} \) constant at 0.30. The numbers 1, 2, 3 and 4 in Fig. 20-(c) correspond to waveform nos. 1, 2, 3 and 4 in Fig. 20-(a).

It is seen in Fig. 20-(a) that the onset of a sharp rise in the cylinder pressure shifted to a later crank angle as the amount of methane was increased and that the peak cylinder pressure rose higher. The H.R.R. waveforms in Fig. 20-(b) indicate that both LTR (line a) and HTR (line b) began at a later crank angle with an increasing methane content. The HTR onset time \( \theta_{\text{HS}} \) (line b) in particular was substantially retarded. In addition, the maximum H.R.R. \( Q_{\text{Hmax}} \) of HTR rose as the amount of methane was increased. This rise in the H.R.R. of HTR is attributed to an increase in the injected heat value with a larger methane content.

Fig. 19 Operation region (DME/methane-HCCI, \( \varepsilon = 18:1 \))
The light emission intensity waveforms of the intermediate products in Fig. 20-(a) indicate that the onset of a sharp rise in light emission intensity shifted to a later crank angle as the amount of methane was increased. That is because the onset of the light emission behavior of the intermediate products occurred incidental to the delay of the HTR onset time $\phi_{\text{HTR}}$. The increase in the injected heat value with the addition of more methane caused the light emission intensity of the intermediate products to rise.

It is seen in Fig. 20-(c) that the crankshaft output rose as the methane-based equivalence ratio was increased. With a higher methane-based equivalence ratio, the time of the peak cylinder pressure was delayed and the injected heat value also increased, both of which presumably account for the increased crankshaft output. Brake thermal efficiency also rose with an increasing injected heat value also increased, both of which delay was further to 12º ATDC, but very little increase is seen in brake thermal efficiency. At point 3 ($\phi_{\text{CH}_4} = 0.18$), the time of the peak cylinder pressure was retarded further to 16º ATDC and no increase in brake thermal efficiency is observed. In the interval between points 1 and 2 in the waveform, brake thermal efficiency increased substantially because the appropriate time frame for the cylinder pressure to peak. On the other hand, brake thermal efficiency shows little increase in the interval between points 2 and 4 because the appropriate time frame for the peak cylinder pressure has already been reached.

At point 1 ($\phi_{\text{CH}_4} = 0$) in the brake thermal efficiency waveform, the peak cylinder pressure occurred at 7º ATDC, but the time of the peak cylinder pressure was retarded to the vicinity of 10º ATDC at point 2 ($\phi_{\text{CH}_4} = 0.10$), and the waveform shows a substantial increase in brake thermal efficiency. At point 3 ($\phi_{\text{CH}_4} = 0.18$), the time of the peak cylinder pressure was retarded further to 12º ATDC, but very little increase is seen in brake thermal efficiency. At point 4 ($\phi_{\text{CH}_4} = 0.30$), the time of the peak cylinder pressure was retarded to 16º ATDC and no increase in brake thermal efficiency is observed. In the interval between points 1 and 2 in the waveform, brake thermal efficiency increased substantially because the time of the peak cylinder pressure steadily approached a crank angle range of 10º~15º ATDC, which is thought to be an appropriate time frame for the cylinder pressure to peak. On the other hand, brake thermal efficiency shows little increase in the interval between points 2 and 4 because the appropriate time frame for the peak cylinder pressure has already been reached.

Figure 21-(a) shows typical examples of the waveforms measured for various methane-based equivalence ratios $\phi_{\text{CH}_4}$ of 0, 0.10, 0.16 and 0.20 while keeping the DME-based equivalence ratio constant at $\phi_{\text{DME}} = 0.22$. The test engine was operated at an engine speed $N$ of 1400 rpm and a compression ratio $\varepsilon$ of 15:1. Figure 22-(a) shows typical waveforms measured for varying equivalence ratios $\phi_{\text{CH}_4}$ of 0, 0.10, 0.16 and 0.20 while keeping $\phi_{\text{DME}}$ constant at 0.22. The test engine was operated at an engine speed $N$ of 1400 and a compression ratio $\varepsilon$ of 18:1. Figures 21-(b) and 22-(b) show the crankshaft output as a function of 0.02 increases in $\phi_{\text{CH}_4}$ while keeping $\phi_{\text{DME}}$ constant at 0.22 under each set of conditions, respectively.

Looking at the waveforms for engine operation under DME-HCCI combustion in Figs. 21-(a) and 22-(a), it is seen that LTR and HTR began at a later crank angle for $\varepsilon$.

\begin{table}[h]
\centering
\begin{tabular}{|c|c|c|c|}
\hline
$\varepsilon$ & $Q$ & $\phi_{\text{DME}}$ & $\phi_{\text{CH}_4}$ \\
& [J/cycle] & DME & CH$_4$ \\
\hline
1 & 15 & 207 & 0.22 & 0 \\
2 & 15 & 288 & 0.22 & 0.10 \\
3 & 15 & 339 & 0.22 & 0.16 \\
4 & 15 & 389 & 0.22 & 0.22 \\
\hline
\end{tabular}
\caption{DME / methane-HCCI measured waveforms (1400 rpm)}
\end{table}

\begin{table}[h]
\centering
\begin{tabular}{|c|c|c|c|}
\hline
$\varepsilon$ & $Q$ & $\phi_{\text{DME}}$ & $\phi_{\text{CH}_4}$ \\
& [J/cycle] & DME & CH$_4$ \\
\hline
1 & 18 & 210 & 0.22 & 0 \\
2 & 18 & 296 & 0.22 & 0.08 \\
3 & 18 & 348 & 0.22 & 0.16 \\
4 & 18 & 377 & 0.22 & 0.20 \\
\hline
\end{tabular}
\caption{DME / methane-HCCI measured waveforms (1400 rpm)}
\end{table}
= 15:1 than for $\varepsilon = 18:1$. That can be attributed to a retarding of the LTR and HTR onset times at the lower compression ratio. It is observed in the H.R.R. waveforms in each figure that the onset of LTR and HTR occurred at a later crank angle as the amount of methane added was increased. However, the effect of the addition of methane on retarding LTR and HTR was smaller at the higher compression ratio. In the case of a high compression ratio, the pressure fields are larger, so the reactions were presumably more dependent on pressure than on the methane content.

The waveforms in Figs. 21-(b) and 22-(b) indicate that output increased with the addition of a larger amount of methane. That is because the addition of more methane increased the injected heat value of the fuel. However, when roughly the same amount of methane was added, output was higher at $\varepsilon = 15:1$ than at $\varepsilon = 18:1$. The reason for that is because time of the peak cylinder pressure at $\varepsilon = 15:1$ was retarded to a crank angle after TDC compared with that before TDC for $\varepsilon = 18:1$.

**Light emission behavior in 1st HTR and 2nd HTR**

Figure 23 shows the H.R.R. waveforms measured for varying methane-based equivalence ratios of $\phi_{CH4}$ of 0, 0.02, 0.04, 0.06 and 0.10 under conditions of a compression ratio $\varepsilon = 12:1$ and a constant DME-based equivalence ratio $\phi_{DME} = 0.30$. The H.R.R. waveforms indicate the occurrence of two-stage HTR as the methane-based equivalence ratio $\phi_{CH4}$ was increased. The first stage signifies the 1st HTR and the second stage (region B) the 2nd HTR. The reaction process of DME/methane proceeds from DME-induced LTR reactions producing HCHO and CO, to CO and methane oxidation reactions representing the 2nd HTR. In other words, the manifestation of the 2nd HTR is due to the addition of methane. The results in the figure also indicate the integration of the 1st and 2nd HTR as a result of increasing the methane-based equivalence ratio $\phi_{CH4}$.

In Fig. 24, waveform-a was measured for engine operation under DME-HCCI combustion at a DME-based equivalence ratio $\phi_{DME}$ of 0.30 and waveform-b was measured for DME/methane-HCCI combustion when a methane-based equivalence ratio $\phi_{CH4}$ of 0.08 was added to $\phi_{DME} = 0.30$. H.R.R. waveform-a shows evidence of the occurrence of 1st and 2nd HTR, and the boundary between these reactions is denoted by the dashed line. The light emission intensity waveforms denoted as waveform-b at wavelengths of 306.4 (OH radical) and 395.2 (HCHO) both show faint light emission behavior in the 1st HTR interval (R1 region), but after passing the dashed line and entering the 2nd HTR interval (R2 region), a sharp rise in light emission intensity is observed. It is generally said that the 1st HTR represent DME oxidation reactions and that the 2nd HTR are reactions which consume methane. Therefore, it is presumed that the light emission behavior seen in the 1st HTR interval is attributable to DME-induced oxidation reactions and that seen in the 2nd HTR interval can be ascribed to the oxidation reactions of CO and methane.

**Various mixing ratios of DME and methane under a constant injected heat value**

Figure 25-(a) shows selected measured results in a range of ±5 J/cycle when the total heat value $Q_{in} (= Q_{DME} + Q_{CH4})$ of DME and methane injected per cycle was set at 411 J/cycle. Figure 25-(b) is an enlarged representation of the H.R.R. waveforms, and Fig. 25-(c) shows the crankshaft output obtained under each set of conditions. The letters a, b, c, d and e in Fig. 25-(c) correspond to the waveforms denoted by the same letters in Fig. 25-(a).
The H.R.R. waveforms in Fig. 25-(b) indicate that LTR (line 1) and HTR (line 2) began at a later crank angle as the proportion of the methane-based equivalence ratio was increased and that the HTR duration simultaneously lengthened. In addition, adding too much methane caused the test engine to misfire (plot e in Fig. 25-(c)). In the case of $Q_{in} (= Q_{DME} + Q_{CH4}) = 411 \pm 5$ J/cycle, LTR occurred under the conditions of (e), but the reactions did not proceed to the occurrence of HTR.

The light emission intensity waveforms of the intermediate products in Fig. 25-(a) show that the light emission intensity began to rise at a later crank angle as the proportion of the methane-based equivalence ratio was increased. That is presumably due to the effect of retarding the HTR onset time $\theta_{HS}$ (see Fig. 7) by the increased proportion of the methane-based equivalence ratio.

In Fig. 25-(c), the figure of each point shows the time of the peak cylinder pressure. The results in Fig. 25-(c) show that the crankshaft output rose and as the proportion of methane was increased. The reason for that is because the increased proportion of methane retarded the time of the peak cylinder pressure, bringing it closer to the optimum time frame.

**Results for addition of methane under a relatively high compression ratio**

Figure 26-(a) shows selected measured results in a range of $\pm 10$ J/cycle when the total heat value $Q_{in} (= Q_{DME} + Q_{CH4})$ of DME and methane injected per cycle during engine operation under DME/methane-HCCI combustion at a compression ratio $\epsilon$ of 15:1 was set at 360 J/cycle. Figure 26-(b) shows the crankshaft output measured for each set of conditions.

Figure 27-(a) shows selected measured results in a range of $\pm 5$ J/cycle when the total heat value $Q_{in} (= Q_{DME} + Q_{CH4})$ of DME and methane injected per cycle during engine operation under DME/methane-HCCI combustion at a compression ratio $\epsilon$ of 18:1 was set at 320 J/cycle. The crankshaft output obtained under each set of operating conditions is shown in Fig. 27-(b).

The H.R.R. waveforms in Figs. 26-(a) and 27-(a) show that LTR and HTR began at a later crank angle as the proportion of the methane-based equivalence ratio $\phi_{CH4}$ was increased. The HTR onset time $\theta_{HS}$ in particular was substantially retarded and the HTR duration was lengthened. It is seen in Figs. 26-(b) and 27-(b) that the crankshaft output increased as a result of retarding the onset time of LTR and HTR. That is attributed to the fact that retarding time of the peak cylinder pressure to the appropriate time reduced the amount of negative work.

These results indicate that the onset time of LTR and HTR can be controlled even at a relatively high compression ratio by adjusting the proportions of the DME and methane-based equivalence ratios. However,
in the case of a high compression ratio, there are severe restrictions owing to the occurrence of knocking, among other factors, as was made clear by the operation region diagrams in Figs. 18 and 19. Therefore, compared with a compression ratio $\varepsilon$ of 12:1, the onset time of LTR and HTR must be controlled more precisely.

Crankshaft output obtained with DME/methane fuel blend

Figures 28-(a) and (b) respectively show the crankshaft output and brake thermal efficiency obtained in relation to the injected heat value under engine operation on DME/methane-HCCI combustion at a compression ratio $\varepsilon$ of 12:1 when the proportions of the injected heat value $Q_{\text{DME}}$ and injected heat value $Q_{\text{CH}_4}$ were varied. For comparison, the figures also show the crankshaft output and brake thermal efficiency obtained with the ordinary diesel engine specifications (Table 2) and diesel fuel as the test fuel. Similarly, Figs. 29-(a) and (b) and Figs. 30-(a) and (b) show the corresponding results obtained at compression ratios of $\varepsilon = 15:1$ and $18:1$, respectively.

For DME/methane-HCCI combustion, the proportion of methane $Q_{\text{CH}_4}$ in the total injected heat value was increased in increments of 5%, and the figures show the crankshaft output obtained in relation to each proportion of methane. In Fig. 28-(a), it is seen that the addition of methane resulted in higher output for DME/methane-HCCI combustion than for DME-HCCI combustion. In addition, by adjusting the mixing ratios of the two types of fuel, the output obtained with DME/methane-HCCI combustion approached that of diesel engine operation. Figure 28-(b) indicates that high brake thermal efficiency was maintained.

In Figs. 29- and 30-(a) and (b), it is seen that the effect of methane on improving crankshaft output and brake thermal efficiency at compression ratios $\varepsilon$ of 15:1 and 18:1 was divided between a region of a large effect (region L) and one of a small effect (region S). These results were strongly influenced by the optimization of the ignition timing by the proportions of the DME and methane-based equivalence ratios. This suggests that
the attainment of high output under a high compression ratio requires precise adjustment of the mixing ratios of the two types of fuel. In this study, the highest output of 1.83 kW was obtained under conditions of a compression ratio $\varepsilon$ of 12:1, a DME-based equivalence ratio $\phi_{\text{DME}}$ of 0.30 and a methane-based equivalence ratio $\phi_{\text{CH}_4}$ of 0.30. The highest thermal efficiency of 30.5% was obtained under engine operating conditions of a compression ratio $\varepsilon$ of 15:1, a DME equivalence ratio $\phi$ of 0.20 and a methane equivalence ratio $\phi$ of 0.20.

CONCLUSIONS

The experimental results of this study made the following points clear with respect to compression ratio changes:

(1) The onset time of low-temperature reactions (LTR) and that of high-temperature reactions (HTR) were further delayed with a lower compression ratio and the LTR duration became longer.

(2) The amount of heat released by LTR increased as a result of lowering the compression ratio.

(3) Presumably, the onset time of HTR was substantially
influenced by the amount of heat released during the LTR duration.

(4) When the injected heat value of the fuel was increased at a lower compression ratio, faint light emission corresponding to HCHO was observed during LTR.

(5) Reducing the compression ratio had the effect of retarding the time of the peak cylinder pressure. The highest brake thermal efficiency was obtained at a compression ratio $\varepsilon$ of 12:1.

(6) Reducing the compression ratio enabled the test engine to operate at a relatively high equivalence ratio, and the crankshaft output approached the level obtained at a low equivalence ratio under ordinary diesel engine operation.

The experimental results also made the following points clear with respect to a two-fuel blend:

(7) A low compression ratio facilitated engine operation in the region of a high injected heat value $Q_{in}$ ($Q_{DME} + Q_{CH4}$). Additionally, the region allowed for the addition of methane increasingly narrowed with a higher compression ratio.

(8) When methane was added under a condition of a constant DME-based equivalence ratio, the higher methane-based equivalence ratio delayed the occurrence of LTR and HTR. Furthermore, the amount of heat released by HTR increased. However, increasing the compression ratio reduced the effect of the addition of methane on delaying LTR and HTR.

(9) When methane was added under a condition of a constant DME-based equivalence ratio, evidence of the 2nd HTR was seen in the heat release waveform as a result of the higher methane-based equivalence ratio. The light emission behavior of the intermediate products of the 1st and 2nd HTR was observed by means of emission spectroscopy.

(10) Under a condition of a constant injected heat value, the onset time of LTR and that of HTR were retarded as the methane ratio was increased.

(11) As the amount of methane added was increased under a condition of a constant injected heat value of DME, brake thermal efficiency rose together with the crankshaft output in the interval where the time of the peak cylinder pressure was suitably retarded. With a further increase in the methane additive, the crankshaft output rose because of the resulting higher injected heat value. Brake thermal efficiency, on the other hand, did not show any appreciable increase because the time of the peak cylinder pressure had already reached the vicinity of a suitable ignition timing.

(12) Adjusting the proportions of DME and methane to an optimum mixing ratio made it possible to control the time of the peak cylinder pressure, and higher output was obtained than with DME-HCCI combustion. This indicates that precise adjustment of the mixing ratio is necessary in the case of a high compression ratio.

(13) At a compression ratio of $\varepsilon = 12:1$, relatively high output was sustained for engine operation under HCCI combustion. The crankshaft output attained approached that of the regular diesel engine specifications when operated on diesel fuel. The HCCI engine was able to sustain high brake thermal efficiency.

REFERENCES


