TED Plaza

A Study of Low Compression Ratio Diesel Engines Operated with Neat Dimethyl Ether (DME)

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ABSTRACT

A new concept of using DME as an alternative fuel in direct injection compression ignition engines with low compression ratios was presented to seek a combustion regime with the highest thermal efficiency. The concept was experimentally evaluated by a comparison of performance and emissions between a DME fueled engine and the corresponding conventional diesel engine. The result demonstrated that the DME fueled engine is superior to the conventional diesel engine in terms of thermal efficiency, emissions and engine noise particularly at low compression ratios. However NOx emission is unacceptably high and needs to be reduced by EGR or after treatment systems.

Keywords : *DME, Compression ignition engine, Ignition delay, Performance, Emissions*

1. INTRODUCTION

The use of dimethyl ether (DME) as an alternative fuel (see Table 1) appears to be a promising approach in minimizing soot emission from conventional diesel engines. The low self-ignition temperature of 508K and the high oxygen content of 34.8 % (by mass) are two major factors that characterize low soot and unburned hydrocarbon (THC) emissions [1-3]. Since the first introduction of the concept by Sorenson et al. in 1995[1], a considerable number of studies [2-7] have suggested that DME may be widely used for heavy duty engines in tracks and buses in some parts of the world where soot emission is particularly a serious problem, such as in congested urban areas. Regarding the power output, the brake mean effective pressure of a compression ignition (CI) engine fueled with DME is comparable to that of a corresponding diesel engine. Moreover, DME readily mixes with gaseous fuels such as liquefied petroleum gas because of its similar nature to alkine fuels. IC engines operated with such blended fuels offer advantages in performance and emissions as described above. From the view point of the CO_2 emission, the use of DME for small size IC engines for passenger cars are also desirable. DME has its unique advantages in its application to CI engines as described above, but there are various problems that need to be solved.

The problems encountered in the previous studies are: 1) Fuel injection tends to occur at earlier timings because of low nozzle opening pressure setting; 2) Combustion is relatively insensitive to the injection timing; 3) Despite the extremely low emissions of soot and unburned hydrocarbons, NOx emission is relatively high because of the advanced injection timing; and 4) DME tends to cause leakage and wear in the injection system because of

Table 1 Characteristics of DME and Diesel fuel

	DME	Gas-oil (JIS No.1)
Molecular Structure	$(CH_3)_2O$	$C_nH_{1.87n}$
Molecular Weight, *g/mole*	46.069	170
Boiling Point, ^oC	–24.8	180/370
Self-ignition Temperature, ^oC *(Cetane Number)*	350 (>55)	250
Heat of Vaporization, *kJ/kg*	467	300
Stoichiometric A/F, *kg/kg*	9	14.6
Liquid density @ 20 ^oC, kg/m^3	668	829.3
Lower Heating Value, *kJ/kg*	28430	43200
C % *(wt)*	52.2	86
H	13	14
O	34.8	0
S	0	0.14
Aroma	0	–
T90, ^oC	–	351.0
Critical Pressure, *atm*	52.4	–
Critical Temperature, ^oC	126.95	–

its low viscosity. The problem can be overcome by adding additives or using blended fuels composed of DME and diesel fuel mixed at desired ratios [3]. Relating to the problems 1) and 2), the effects of fuel spray characteristics on combustion and emissions will be discussed in this paper. The measures to reduce NOx emission will also be shown.

During the course of studying DME application to CI engines, the fuel was found to have potential in a wide range of applications because of its easy ignitability. That is, the fuel may be used in low-compression-ratio direct-injection engines (hereafter referred to as LCR DI-CI), which are expected to offer all the above-mentioned advantages. Such advantages of LCR CI-DI engines operated over a wide range of engine loads are considered to be even more attractive in view of various difficulties in combustion control and the high cost involved in gasoline-direct-injection (GDI) spark-ignition (SI) engines. GDI-SI engines still have difficulties in meeting the stringent emission standards, especially the fine particulate matter standard.

After some exploratory studies on LCR-CI engines, the present study is directed towards an in-depth analysis of the new concept, LCR-CI engine. Engines operated with DME are expected to retain the advantages of high efficiency and low emissions if exhaust catalytic converters as well as EG are employed in the system.

Fig.1 presents the states of DME on the P-T diagram, and the trajectories of pressure and temperature during the compression stroke of engines with various compression ratios. The pressure and temperature trajectories were estimated based on the assumption of adiabatic compression. Kapus ran his engine with a compression ratio of 15.76, but didn't employ lower compression ratios because of the increased engine noise generated by the rapid premixed combustion [3]. The temperatures on the trajectory for a compression ratio of 12.36 acquired in our experiment are well higher than the DME's ignition temperature at an atmospheric pressure. The operation of a DME fueled CI engine at this compression ratio looks possible from this figure. In our previous work, a DI-CI engine fueled with DME was operated successfully at compression ratios of 10.19 and 11.16 .

The question of classifying the channels inevitably comes up, with many conflicting views on this topic. It should be remembered that the significance of any classification based on the channel dimension is subjective, and is invariably affected by the type of fluid (liquid or gas), operating conditions (temperature and pressure), the type of flow (single phase, boiling or condensation), and the length to diameter ratio. Nevertheless, a simple classification scheme serves as a preliminary guide to relate the channels with the physical world.

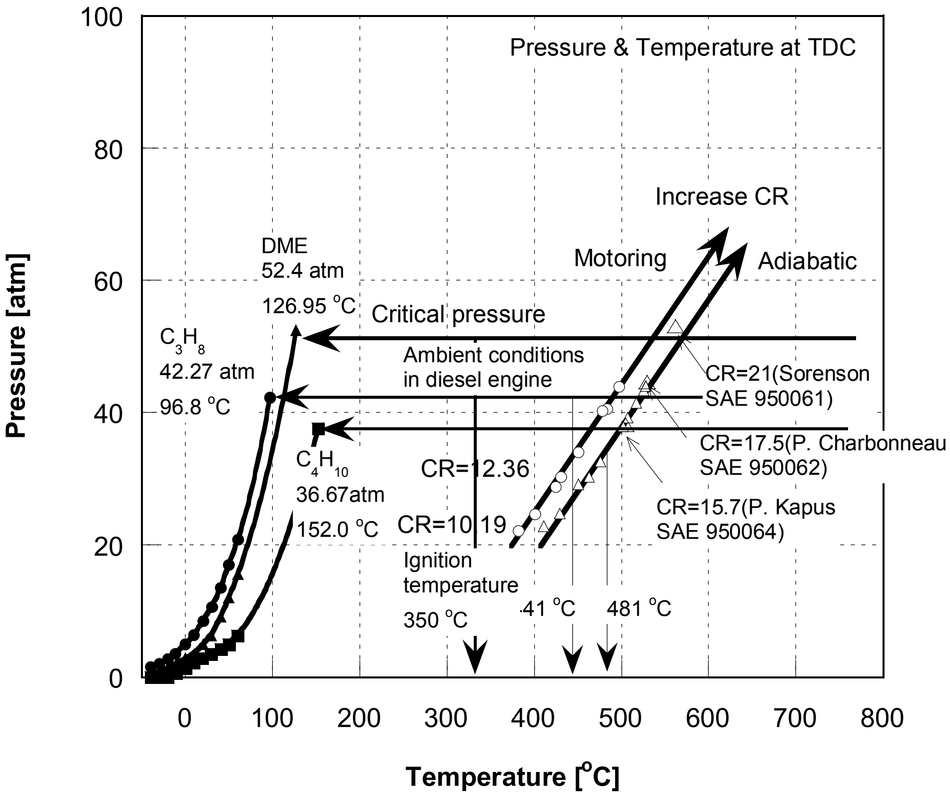

Fig. 1 PT diagram of DME.

2. EXPERIMENTAL

The present study of an LCR-CI engine operated with DME is mostly an experimental endeavor together with a thermodynamic analysis of combustion processes. The key issues in the design of an LCR-CI engine fueled with DME are reliable self-ignition and acceptable pollutants emission under stable engine operation. To investigate the factors affecting these issues, various parameters including compression ratio, start of fuel injection, injector opening pressure, and kind of fuel were changed in the experiments.

2.1 Engine and apparatus

The engine used in this study was basically the same engine as that employed in our earlier studies [5-7]. It is a typical direct-injection compression ignition engine manufactured by Yanmar Diesel Corporation with the specifications listed in Table 2. In order to vary the compression ratio of the engine, a thin copper spacer was inserted between the engine cylinder head and the cylinder block. The injection timing was altered at every 12 degrees crank angle by shifting the tooth of the injection pump gear. The injection timing given in the paper is not dynamic but nominal one. The difference between the nominal and dynamic timing is listed in Table 3. A pressure transducer to monitor the injection pressure, a needle-lift sensor, and an in-cylinder pressure transducer were installed in the engine. The engine apparatus was also interfaced with an emission measurement device (Horiba Co. MEXA-8220M) which includes measurement of total unburned hydrocarbons (THC). Since the main THC component in the exhaust was unburned DME, the response of the FID was calibrated to correct the relative molecular sensitivity [9]. A correction factor of 1.51 was used throughout the experiments.

After reviewing various methods employed for pressurizing DME in the previous studies, it was decided to use bottled nitrogen gas to maintain the fuel feed pressure at 3.43MPa. This measure was found to be effective to prevent the vapor lock within the fuel system. DME naturally absorbs the nitrogen which may increase the fuel NO formation, but there were few differences in NO emission between nitrogen and helium gas used for pressurizing DME during a short period of engine operation. With this method of fuel feeding, the engine was operated successfully with neat DME at a needle opening pressure of 8.82MPa. When DME is used as fuel, the needle opening pressure needs to be set at a lower value in comparison to that for diesel fuel, 20.1 MPa because of the lower constant of volume of elasticity with DME. One of the problems that a DME fueled DI CI engine faces on the injection system is a large amount of fuel leakage. The fuel leakage was measured under motoring condition using the same injection system as that for firing condition, and it was found that the leakage amounts to as high as 15% of the supplied fuel amount irrespective of the load. Such a leak volume was rather high, but, on the other hand, it is beneficial in the sense that the leaked fuel would help cool the injector and prevent the fuel from being prematurely vaporized in the injector. The amount of DME leakage and N_2 consumption in the container were both used in calculating the fuel consumption in the engine experiment.

Another problem in the injection system for DME is the needle lift behavior at low nozzle opening pressures. Fig.2 (a) shows the needle lift records at two nozzle opening pressures of 6.86 MPa and 8.82 MPa. As can be seen in the figure, when the nozzle opening pressure is 6.86 MPa, the needle starts to rise earlier in time and bounced with oscillation during the closing period, prolonging the injection duration. This oscillation was probably caused by the DME vapor in the pressure chamber of the injector. The nozzle opening pressure was the main factor governing the leaked fuel quantity at the injector, and the relationship between the leaked fuel quantity and the nozzle opening pressure was investigated as shown in Fig.2 (b). It is seen that the fraction of leaked fuel to the supplied fuel increases with nozzle opening pressure and that the fraction of leakage measured at firing condition doubles the leakage quantity at the injection experiment at motoring condition. The leakage quantity depends also on the design and degree of the abrasion of the injection system. It is difficult to determine the optimum nozzle opening pressure because a low nozzle opening pressure doesn't necessarily yield low injection pressures.

Table 2 Engine specification.

Engine Type	NFD-13K
Bore x Stroke	92 × 96 mm
Displacement	638cc
Original CR	17.7
Rated output	8.45kW/2600rpm
Injection pump	Jerk Type (Plunger 8mm dia)
Injector	0.26mm × 4

Table 3 The difference between nominal and actual injection timing.

Fuel	Nominal injection timing	Actual injection timing
DME	–17/–5 ^{o}CA ATDC	–12 ATDC/–1 ATDC
Diesel fuel	–17/–5 ^{o}CA ATDC	10/2 ^{o}CA ATDC

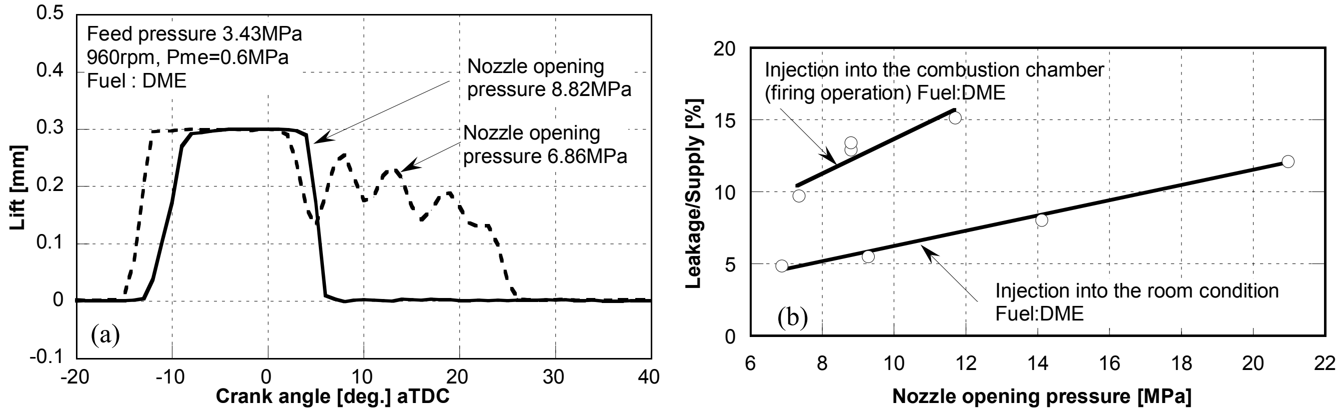

Fig.2 Effect of nozzle opening pressure on (a) the needle lift histories and on (b) the leakage of DME.

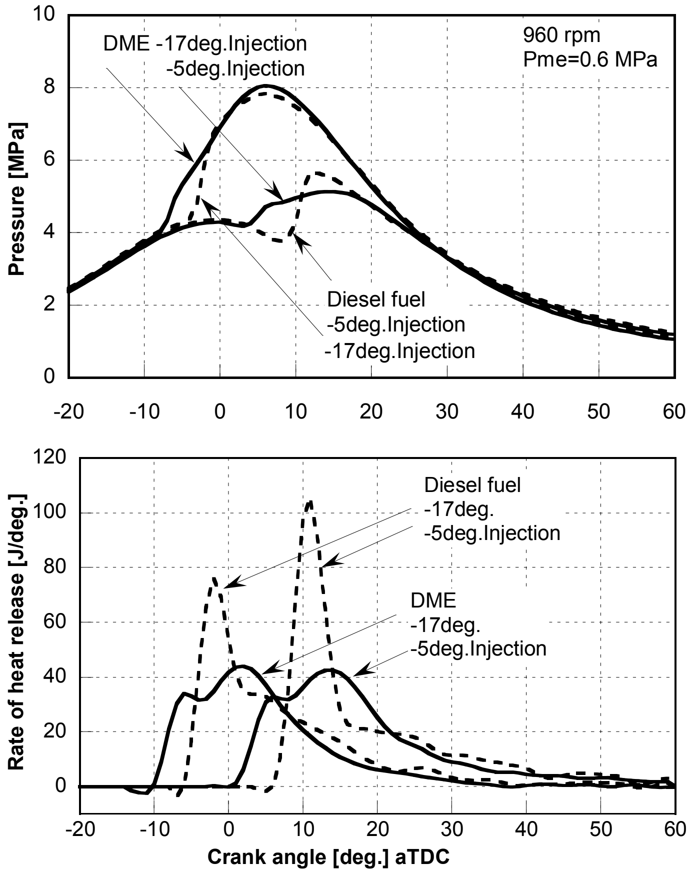

Fig.3 Pressure-time (p-t) and heat release histories of DME and diesel fuel for injection time at -17 and -5 ATDC.

2.2 Engine performance and emissions at original compression ratio

Engine performance and emissions at an original compression ratio of 17.7 are presented by comparing the results obtained with DME to those with diesel fuel. First, the pressure-time diagram and the rate of heat release at the same mean effective pressure of 0.6 MPa are compared to each other, as shown in Fig.3, for injection timings at 17 ^{o}CA and 5 ^{o}CA BTDC, respectively. For both injection timings, the start of pressure rise with DME occurred earlier in time than that with diesel fuel because of the DME's earlier start of needle lift by 4~5 ^{o}CA and short ignition delay.

The rate of heat release for DME presents a short "spike" in the initial premixed combustion stage and a moderate heat release during the diffusion combustion stage. The low peak of the rate of heat release for DME is due to the smaller amount of accumulated fuel during the ignition delay as compared with that for diesel fuel. Another reason for the weak spike and moderate heat release could be attributed to the lower mixing rate for DME. The lower initial momentum for DME spray jet must have produced a slower mixing rate for DME.

Fig.4 shows a comparison of the engine performance between DME and diesel fuel. The exhaust gas temperature and the brake specific energy consumption (BSEC) are plotted against Pme with the injection timing as a parameter. It should be noted that the exhaust gas temperature is lower for DME by around 50 ^{o}C and BSEC is lower for DME by well over 10%. The reason for the lower energy consumption with DME will be discussed later in details.

Fig.5 shows a comparison of emissions between DME and diesel fuel at various Pme at two different injection timings. As expected, soot emission was zero for DME at all operating conditions tested, while soot emission with diesel fuel increased with engine load. Similarly, THC emission was almost negligible for DME even under high loads. On the other hand, NOx emission was much higher with DME as compared to that with diesel fuel at an injection timing of 17 ^{o}CA BTDC, while it was somewhat lower with DME at an injection timing of -5 ^{o}CA BTDC.

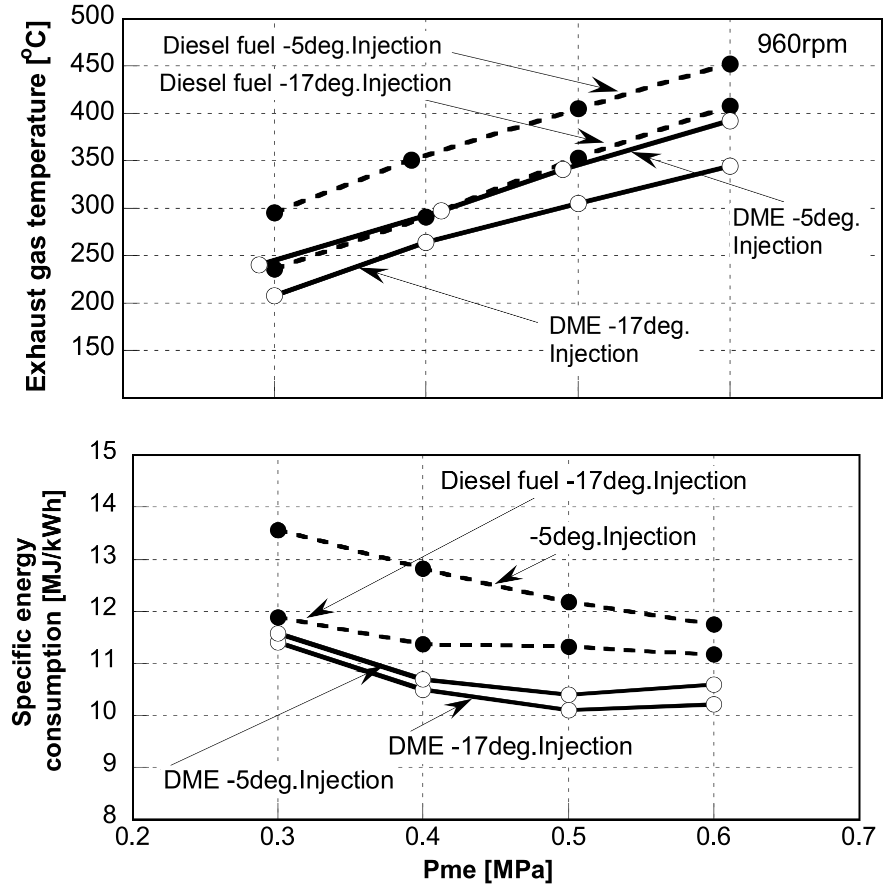

Fig.4 Exhaust-gas temperature and specific energy consumption for varied engine load.

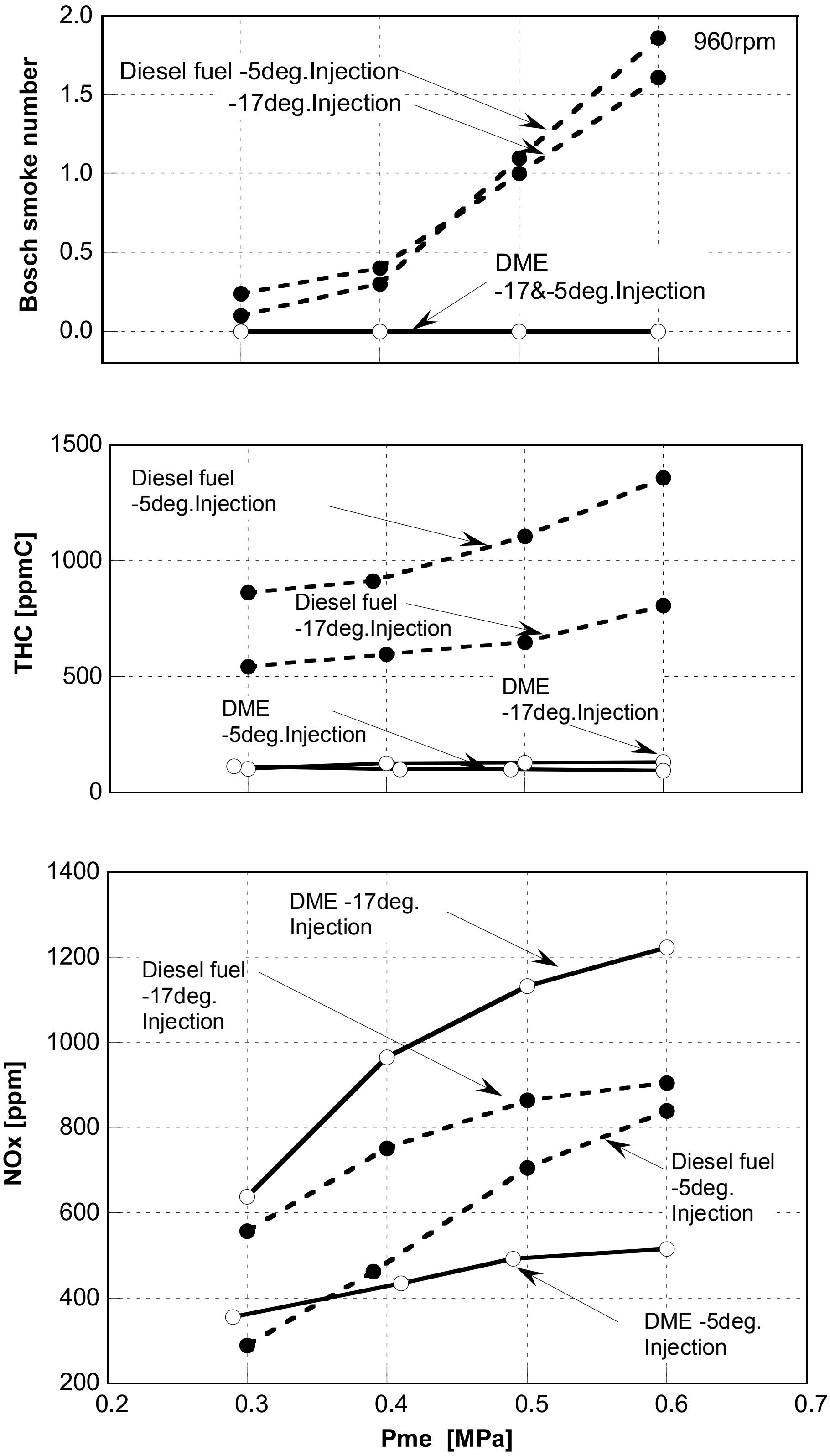

Fig.5 Smoke, THC and NOx emission when operated by DME and diesel fuel.

3. LCR-CI ENGINE AND DISCUSSION

The minimum brake specific fuel consumptions for various types of IC engine are plotted against compression ratio in Fig.6. The plots at left are the minimum BSFCs for SI engines, while the data for CI engines at right scatter at compression ratios ranging from 15 to 23. There are no data between two groups at compression ratios around 14. The facts that increased compression ratio generates knocking with SI engines and reduced compression ratio yields poor ignition with CI engines are the reasons for the lack of data in this range of compression ratio. However, the plots in the figure clearly suggest the possibility to achieve the lowest brake specific consumption at a compression ratio near 14. DME may have potential to achieve the lowest BSFC at compression ratios around 14. One of the latest models of SI engines is the direct injection type, and this type of engine could be easily modified to use DME. It might be possible to operate all types of internal combustion engines with DME. The effects of compression ratio on performance and emissions of a DME engine are discussed in this section to seek the lowest BSFC at a compression ratio around 14.

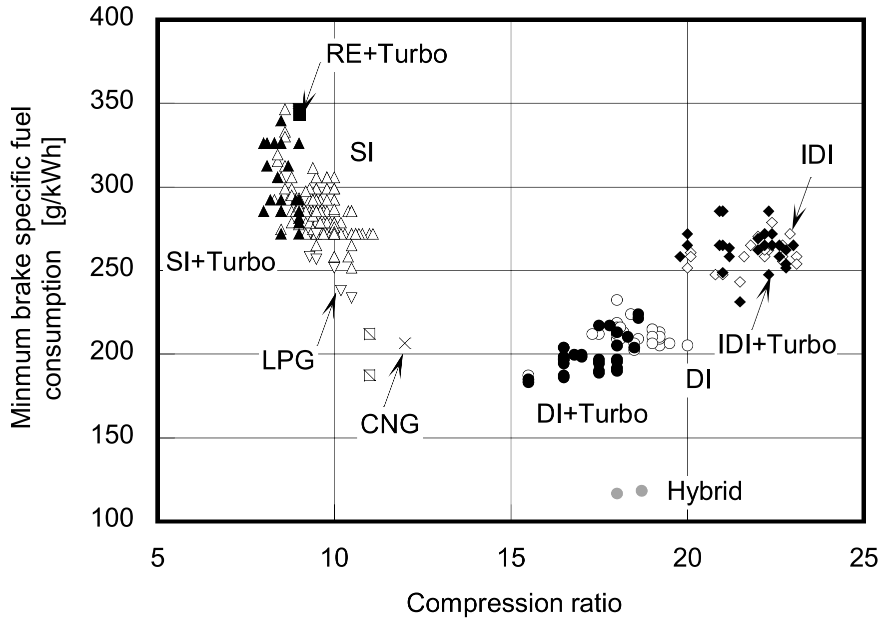

Fig.6 Relationship between the compression ratio and minimum brake specific fuel consumption.

3.1 Combustion behavior

As shown in Figs.4 and 5, the effect of injection timing on the specific energy consumption is marginal at a Pme of 0.6 MPa for both DME and diesel fuel even if the ignition timing was varied from –17 ^{o}CA to –5 ^{o}CA. Accordingly an experiment on the effects of compression ratio on performance and emissions was conducted at a fixed injection timing for both fuels. Fig.7 shows the pressure-time histories and the rates of heat release for both DME and diesel fuel at an injection timing of –12 ^{o}CA and a Pme of 0.6 Map. Unlike the steady and rapid rise in the pressure-time curves for diesel fuel, "bending points" can be observed for DME at pressures of approximately 5 MPa, which occurrs at -5 ^{o}CA, -3 ^{o}CA, and 3 ^{o}CA for compression ratios of 17.7, 13.89, and 12.36, respectively. Note that the bending points corresponded to crank angles where the peak of respective spikes in the rate of heat releases were observed. The moderate diffusion combustion occurring after the small initial pre-mixture combustion is the cause for the bending in the pressure-time records.

Fig.8 shows the ratio of the thermal efficiency, the ratio of degree of constant volume and ignition delay against compression ratio for both fuels at the same operating conditions as in Fig.7. Here the word “ratio” means a relative value to the reference one at a compression ratio of 17. Since the heat release occurs mainly near TDC and ends earlier with DME, the thermal efficiency for DME is considerably higher than that for diesel fuel at all compression ratios tested but particularly at low compression ratios. It is also seen that the thermal efficiency for DME remains almost constant regardless of the compression ratio, while it decreases for diesel fuel at low compression ratios. The higher thermal efficiency for DME is reflected from the higher ratio of degree of constant volume. When a rapid heat release occurs near TDC, it often results in a noisy operation. This, however, was not the case, as shown later. The ignition delay shown in Fig.8 is defined as the crank angle between the start of needle lift and a crank angle when the accumulated heat release reached 5 % of the total heat supplied. The ignition delay for DME is apparently shorter than diesel fuel because of the higher Cetane number of DME.

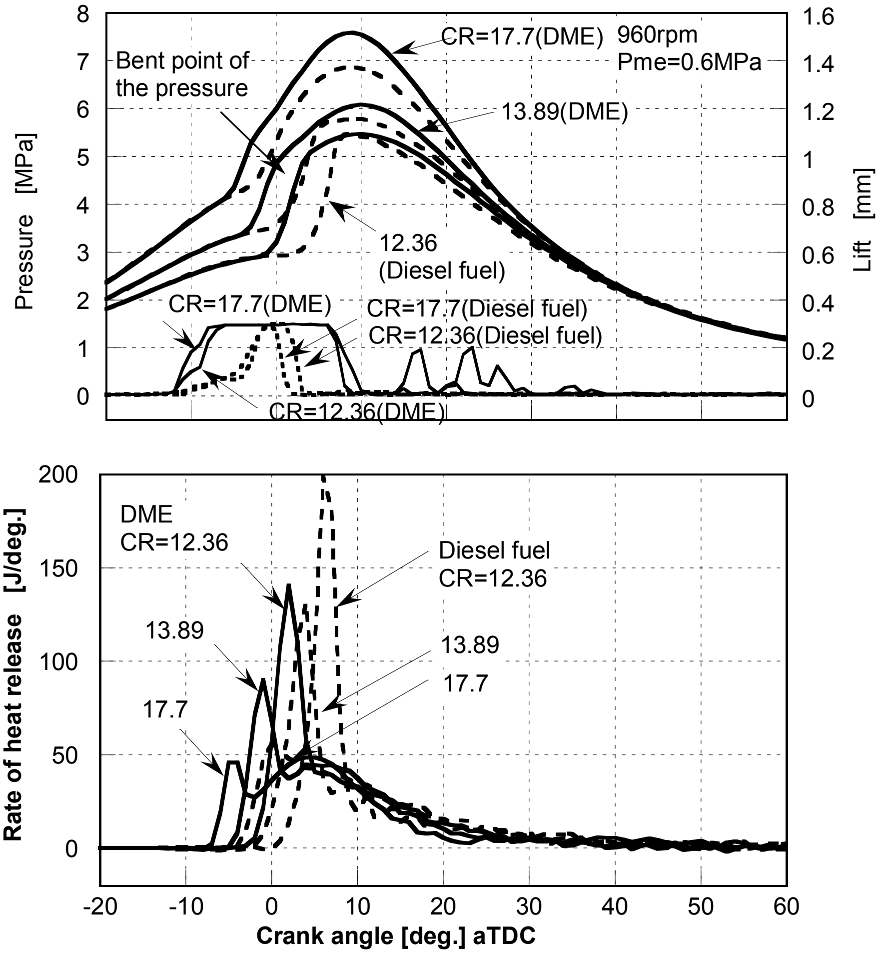

Fig.7 Pressure-time (p-t) and heat release rate histories for varied compression ratios.

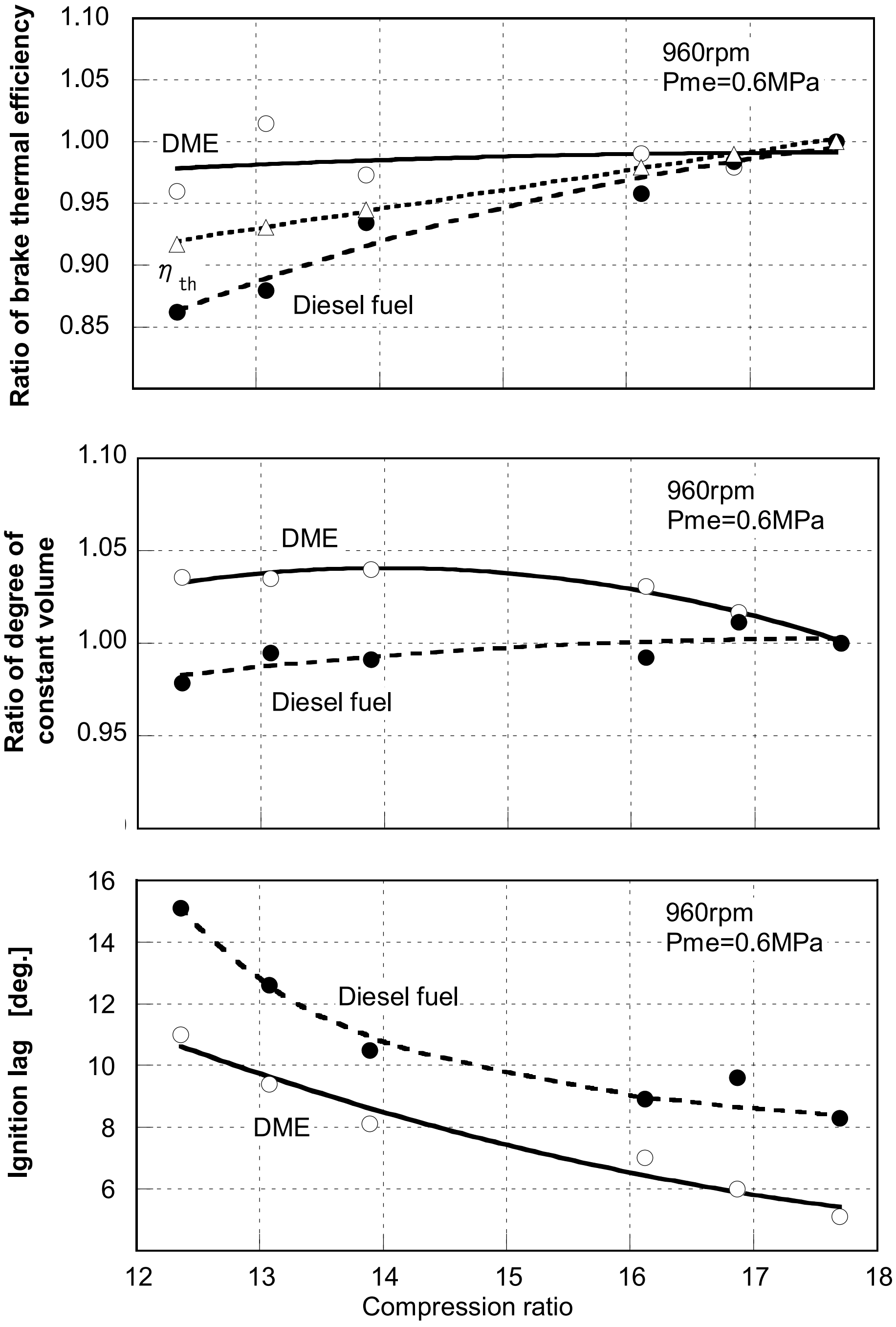

Fig.8 Relative brake thermal efficiency, relative degree of constant volume, and ignition lag for varied compression ratios

3.2 Exhaust emissions

Exhaust emissions were measured at a time when the above results were obtained. Fig.9 presents a comparison of exhaust emissions between two fuels at various compression ratios. CO and THC emissions are considerably low for DME and NOx is also lower for DME than diesel fuel. Shown in Fig.10 are the variations of exhaust-gas temperature, the fraction of the heat loss to the coolant against the input energy, and sound pressure level with compression ratio. The higher ratio of degree of constant volume shown in Fig.8, lower exhaust gas temperature and lower cooling heat loss shown in Fig.10, all support the superior thermal efficiency for DME.

The mean brake thermal efficiency is expressed as follows;

$$\eta_e = \eta_{th}\eta_{glh}\eta_b\eta_m(1-\varphi_w)$$

where

η_e *: Brake thermal efficiency*

η_{th} *: Theoretical thermal efficiency*

η_{glh} *: Degree of constant-volume combustion*

η_b *: Combustion efficiency*

η_m *: Mechanical efficiency*

φ_w *: Cooling loss ratio*

The factors affecting the brake thermal efficiency, η_e , are η_{glh} , η_b and η_w when the fuel is changed. The mechanical efficiency, η_m , is considered to be identical to both DME and diesel fuel at the same Pme condition. The degree of constant volume, η_{glh} , for DME increases with the decrease in compression ratio, as shown in Fig.8 because of the increased pre-mixed combustion. The significant low concentrations of CO and HC for DME imply a very high combustion efficiency for DME close to 99% at any compression ratios tested. Furthermore, the exhaust gas temperature for DME is lower than that for diesel fuel by 25~50 K, and the difference in the exhaust gas temperature between DME and diesel fuel becomes large at low compression ratios. The lower cooling loss to the combustion chamber walls for DME could be indebted partly to the reduced radiation heat transfer for the semi-luminous flame of DME.

As pointed out earlier, the engine operated with DME is quieter than with diesel fuel due to the shorter ignition delay for DME. Fig.10 shows that the sound pressure level for DME remains almost constant even at low compression ratios in contrast to the different trend with diesel fuel.

3.2 NO_x reduction by a catalyst

The performance of the present LCR DI-CI engine operated with DME was found to be superior to conventional diesel engines, but NOx emission was still unacceptably high. In order to demonstrate the possibility to reduce NOx emission from an engine fueled with DME, a Cobalt-aluminum base selective catalyst was tested. A small amount of DME was injected upstream of the catalyst as a reducing agent and the amount of the added DME was measured by a volume type flow meter. The concentration of THC, mainly DME, measured by FID at the tailpipe of the engine is about 1.21 times higher as compared with pure DME concentration because the relative mole sensitivity of DME is 0.62. Since the introduction of DME results in the increase in THC concentration due to the presence of un-reacted raw fuel after the catalyst, THC was measured together with NOx varying the catalyst temperature.

Fig.11 shows the result. As can be seen in the figure, a considerable reduction of NOx was achieved by the catalyst with DME injection. In particular, when DME was added in amounts ranging 4,000-8,000 ppmC, NOx was remarkably removed in a window of catalyst temperature between 300-350 ^{o}C. However, at the same time there arose another unfavorable problem, i.e. increased THC emission at low exhaust gas temperatures. The increased THC emission resulting from the DME addition needs to be reduced by an oxidation catalyst.

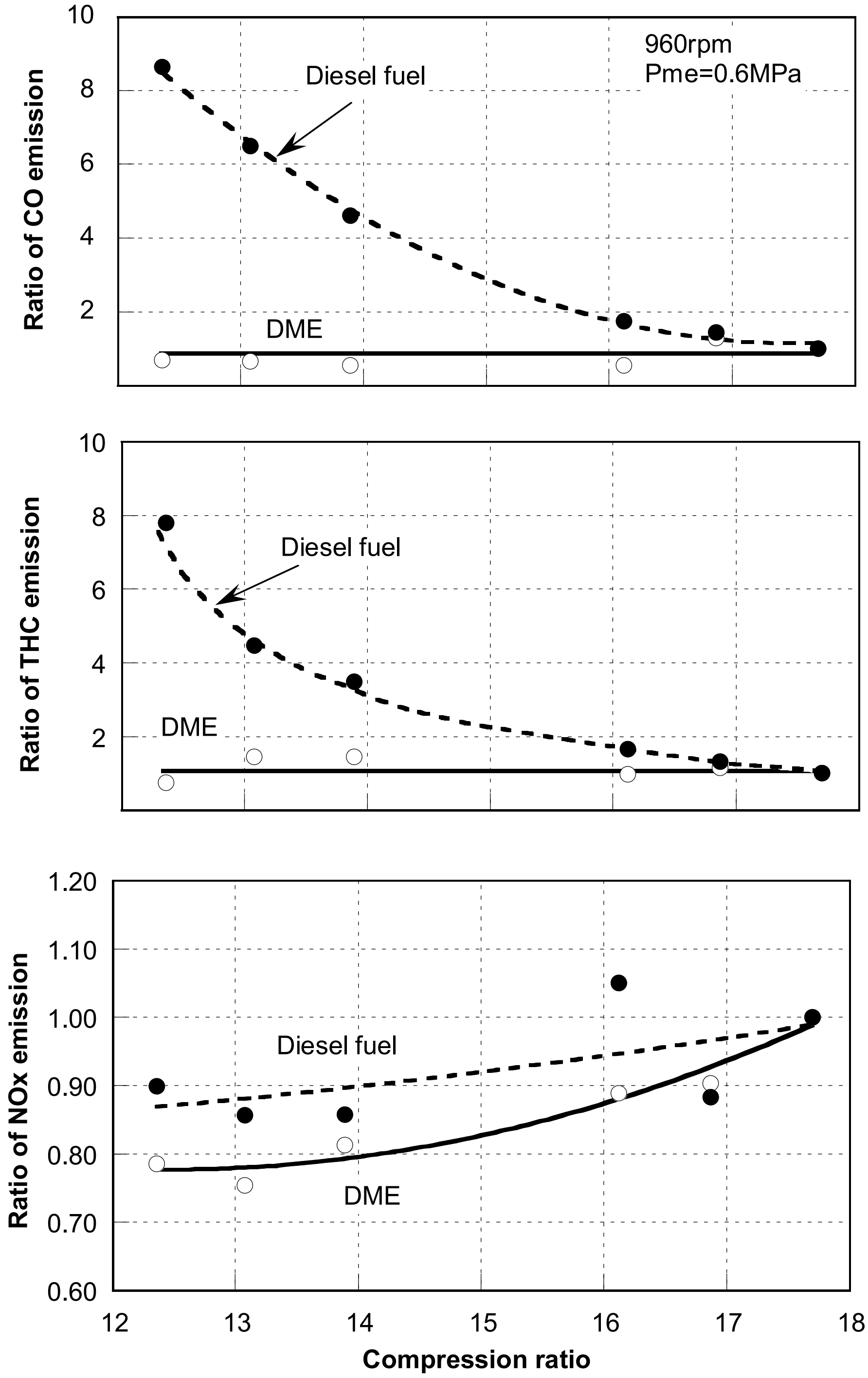

Fig.9 Carbon monoxide, THC and NOx emission for varied compression ratios.

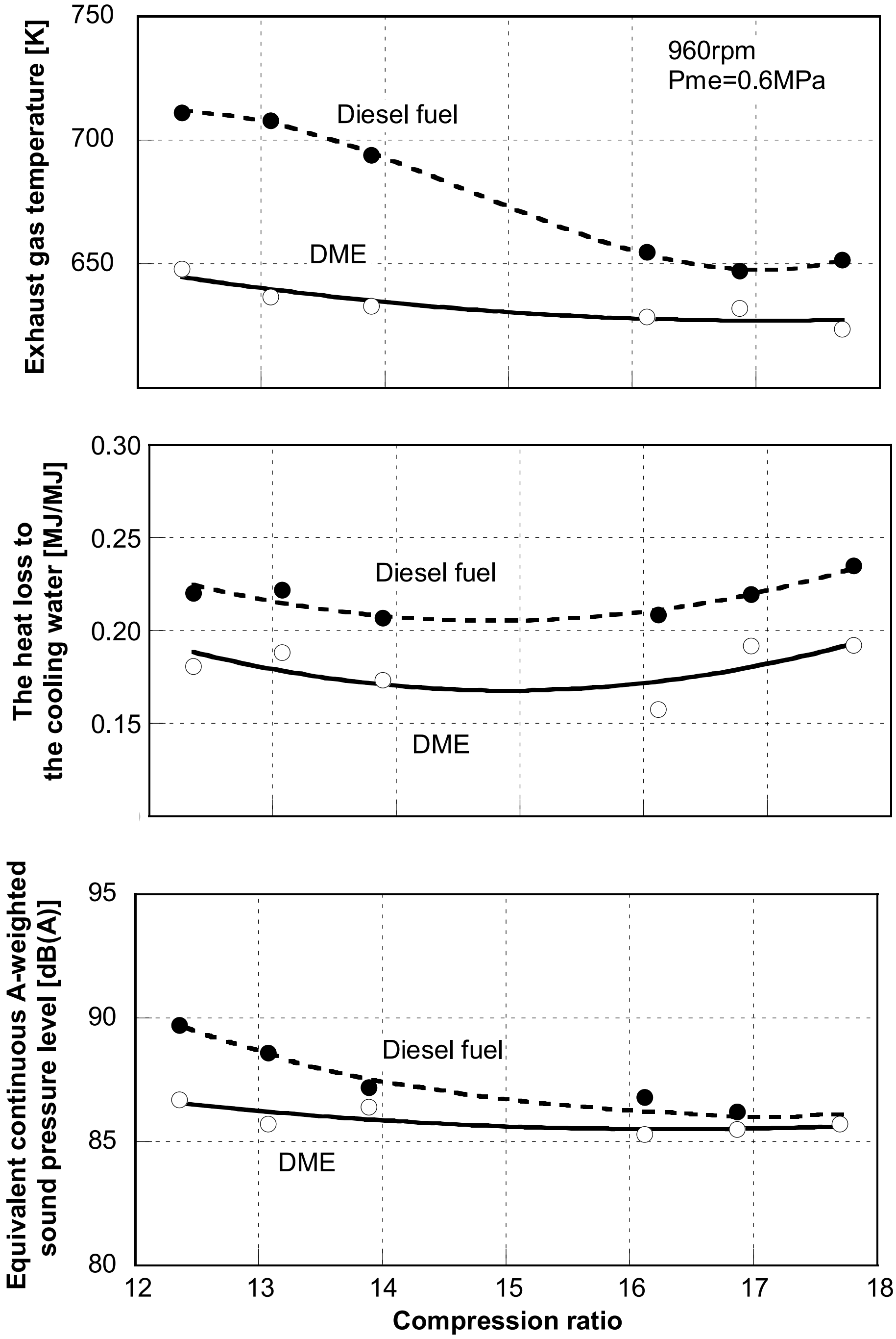

Fig.10 Exhaust gas temperature, heat loss to coolant and sound pressure measurements for varied compression ratios

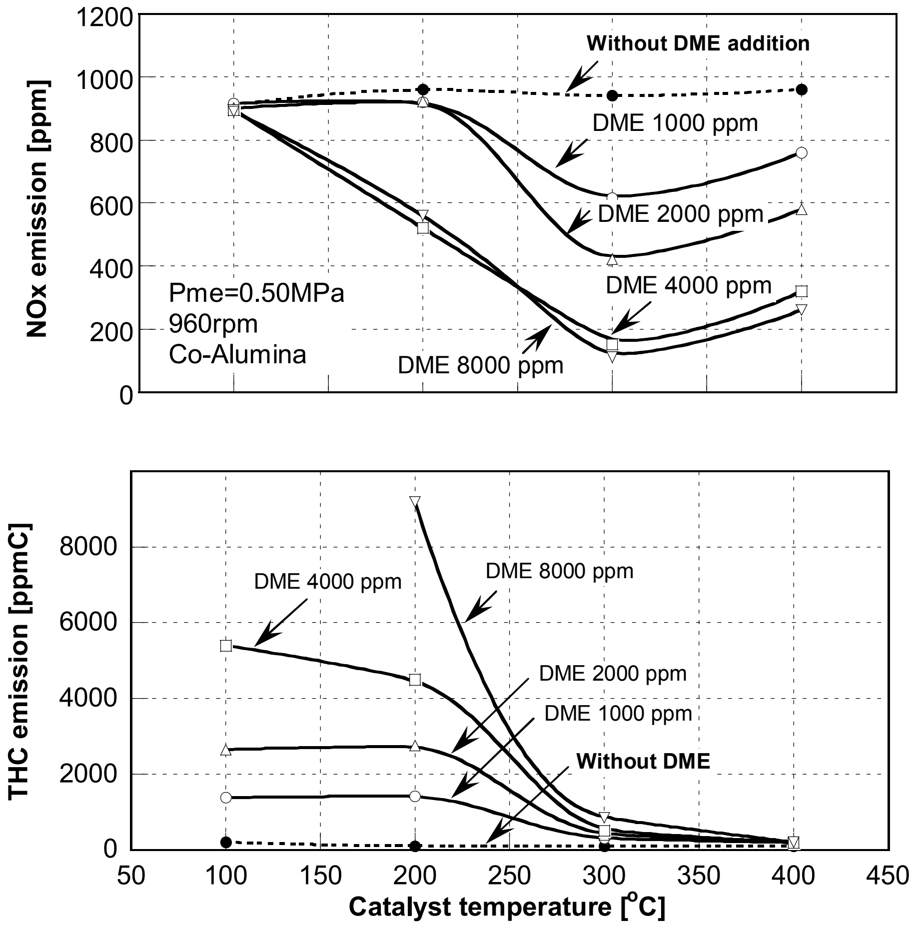

Fig.11 NOx and THC emission with variation of the catalyst temperature.

3. SUMMARY

DME may be widely used in compression-ignition (CI) engines as an alternative fuel in some parts of the world, mainly because of the low soot production. While this consideration is actively being evaluated in the field, a new methodology of using DME as a fuel was discussed in this paper, focusing on its use in low-compression-ratio (LCR) direct-injection (DI) CI engines.

As an alternative fuel to diesel fuel, it would be economical to use DME in LCR DI-CI engines because of the fuel's relatively low self-ignition temperature. The DME engine will not require complex and high-cost devices to achieve low emissions, such as an electronically controlled high-pressure injection system and exhaust traps or plasma after-treatment units. The same is true when compared with gasoline-direct-injection (GDI) spark-ignition (SI) engines, which are equipped with expensive and maintenance-demanding electric ignition system and combustion control devices.

Among the findings from the present study of DME operated LCR-CI engine are:

1. The lowest compression ratio for easy start and stable operation of the DME CI engine tested is around 12.
2. THC emission from the DME LCR-CI engine is low and remains almost constant irrespective of the compression ratio.
3. Soot emission from the DME LCR-CI engine is negligibly low at all compression ratios examined.
4. The brake thermal efficiency of the DME LCR-CI engine is almost constant over compression ratios ranging from 12.36 to 17.7.
5. NOx emission from the DME CI engine is unacceptably high and this needs to be reduced by EGR and after treatment devices.

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