OPTIMIZATION OF INJECTION PRESSURE FOR FUEL CONSUMPTION AND EXHAUST EMISSIONS IN A DIMETHYL ETHER (DME) ENGINE WITH A COMMON RAIL TYPE INJECTION SYSTEM

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Abstract

Dimethyl Ether (DME) has been attracting attention as an alternative fuel for diesel engines because it can be produced from various feed stocks and does not create smoke. On the other hand, a DME truck require a large volume fuel tank in order to obtain the running distance equivalent to diesel trucks due to the lower energy content per unit volume of DME, and un-burned gas and fuel consumption of a DME engine tend to increase when large volume EGR are used. Therefore, DME engine research for the further improvement of fuel consumption and exhaust emissions still remains. It is important to optimize the injection pressure of the DME engine in order to improve fuel consumption and exhaust emissions. Compared to diesel fuel, DME has much lower viscosity, lower bulk modulus and higher vapour pressure, so the DME injection system commonly used a relatively low injection pressure up to 60 MPa. The effect of high injection pressure on DME engine performance has not yet been understood. In this study, the influence of injection pressure on fuel consumption and exhaust emissions were examined using a common rail type DME injection system with injection pressure up to 100 MPa, and it was demonstrated that the optimum injection pressures for fuel consumption and exhaust emissions of the DME engine in wide operating ranges.

Keywords: *diesel engines, alternative fuels, DME, common rail, injection pressure, fuel consumption, emissions*

1. Introduction

DME has been attracting attention as an alternative fuel for diesel engines because it can be produced from various feed stocks and does not create smoke. Research studies about the DME engine have been carried out since 1995. S.C. Sorenson et al. and P. Kapus had reported about the performance and emissions of the DME engine in 1995 [1, 2]. Many fundamental studies about

DME engines have been conducted in Japan after 1995 [3-6]. The DME city bus had been developed by K. F. Hansen et al. on 2000 [7], some DME trucks also had been developed by National institute of Advanced Industrial Science and Technology (AIST) and National Traffic Safety and Environment Laboratory (NTSEL) in Japan [8, 9]. Those demonstrated that DME trucks have much lower Particulate matter (PM) and NOx, without the use of diesel particulate filter (DPF) and NOx reduction catalyst. From these results, it has been understood that DME trucks require a large volume fuel tank in order to obtain the running distance equivalent to diesel trucks due to the lower energy content per unit volume of DME. Un-burned gas and fuel consumption tend to increase when large volumes EGR are used with reduction NOx. DME engine research for the further improvement of fuel consumption and exhaust emissions still remains.

On the other hand, it is necessary to optimize injection pressure due to its strong influence on engine performance. The influence of injection pressure on fuel consumption and exhaust emissions need to be investigated in order to optimize injection pressure. Compared to diesel fuel, DME has much lower viscosity, lower bulk modulus and higher vapour pressure, so that the DME injection system commonly used relatively low injection pressure. The influence of injection pressure on DME engine performance has been reported in some papers [10-12]. However, the influence of high injection pressure as over 60 MPa on engine performance has not yet been understood. The engine operating ranges in those experiments were narrow.

The purpose of this study is to improve the performance of a DME engine with a common rail type injection system. In this study, the influence of injection pressure on fuel consumption, NOx, CO and NMHC was examined to find the optimum injection pressure using an injection pressure of up to 100 MPa, and then the effective injection pressure for fuel consumption and exhaust emissions at a wide operating range were discussed.

2. Experimental setup

2.1. DME properties

Table 1 shows the main physical properties of DME. DME is a liquified gas and it has oxygen in its molecular. DME have to be pressed to maintain its liquid phase because DME has high vapour pressure. The low viscosity of DME makes it necessary to increase the amount of lubricant for it. In the case of the experiment, lubricant improver (LZ539ST) added 500 ppm wt. to the DME. The energy content of DME is low, approximately 1.8 times the injection volume quantity is required compared with diesel fuel to obtain the engine power equivalent to diesel engines.

Tab. 1. Properties of DME

Fuel		DME	Diesel fuel
Molecular formula		CH_3–O–CH_3	$C_{16}H_{34}$
Oxygen content	[wt. %]	34.8	0
Cetane number		>>55	55
Stoic, A/F ratio		9.0	14.6
Liquid density	[g/cm^3]	0.67	0.8–0.84
Kinetic viscosity	[cSt@303K]	0.197	3.256
Boiling point	[K]	248	410–650
Lower heating value	[MJ/kg]	28.9	42.7
Vapor pressure	[MPa@293K]	0.53	–
Bulk modulus	[N/m^2]	6.37E+08	1.49E+09

2.2. DME engine

Table 2 shows the specifications of the DME engine. A four cylinder, 4.6L diesel engine with a turbocharger was used. The fuel injection system changed from the base system to a common rail type DME injection system. The geometric compression ratio was 19.0 which was the same as the base diesel engine since the Cetane number of DME is close to that of diesel fuel. EGR gas was introduced to the inlet side of the compressor to obtain a large volume of EGR due to the fact that the DME does not create smoke. A diesel oxidant catalyst (DOC) was installed in the outlet side of the turbocharger to reduce un-burned gas because increasing amounts of un-burned gas will be expected by the large volume of EGR.

Tab. 2. Specification of the DME engine

	Base diesel engine	DME engine
Model	ISUZU 4HG1T	←
Cylinder	Inline 4	←
Bore × Stroke	115mm × 110mm	←
Total displacement	4570cc	←
Combustion chanmber	Re-entrant type	←
Compression ratio	19.0	←
Maximum Power	89kW/3200rpm	102kW/3000rpm
Maximum Torpue	325Nm/1800rpm	325Nm/1800rpm
Aspiration	Turbo charge	Turbo charge with intercooler
After-treatment	without	Oxdant catalyst (1.5L,Pt,4g/L)
Injection system	Inline-jerk	Common rail (Modified for DME)
Injection pressure	–	20～100MPa

2.3. Fuel injection system

Figure 1 shows the configuration of the DME fuel system. Table 3 shows the main specifications of the DME common rail injection system. DME was pressurized by the electric feed pump to approximately 1.5 MPa in order to keep it in the liquid phase. The fuel tank and the fuel pipe were changed pressurised structure due to the fact that the fuel pressure was high. DME was circulated through the fuel tank to the high pressure pump at 4.0 L/min and the fuel cooler was installed to avoid vaporization of DME. DME leaked into the high pressure pump due to the high fuel pressure, the leaked DME returned to the fuel tank by the re-liquefaction compressor. The re-liquefaction compressor was driven by the cam inside the high pressure pump. The rubber material in this fuel system was modified for DME to protect it from swelling. The injection nozzle hole increased to 0.32 mm to obtain a large quantity of fuel injection. Figure 2 shows the injection rate of the injection system when the injection quantity was 150 mm^3 / st at a high pressure pump speed of 1500 rpm. This condition is close to the maximum out put condition. The injection duration at an injection pressure of 100MPa was 13.0 degrees and was the same as the injection duration of the base diesel engine when the equivalent heating value was injected at the same condition.

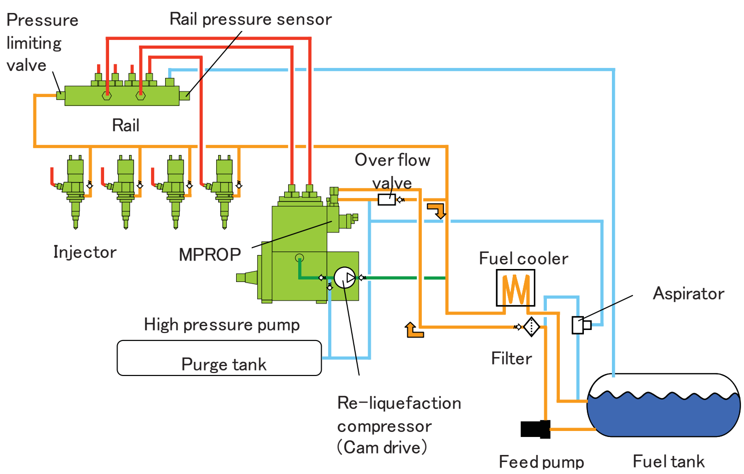

Fig. 1. Configuration of the DME fuel system using common rail

Tab. 3. Specification of the DME common rail injection system

		DME Common Rail Injection System
Fuel Metering Type		Suction control
High Pressure Pump Plunger	Diameter	8 mm
	Stroke	15 mm
	Surface Treatment	Carbon coating
Injector Nozzle	Orifice Number	5
	Orifice Dia.	0.32 mm
	Seat Dia.	1.9 mm
	Surface Treatment	Carbon coating
Common Rail	Volume	18.9 cm3
Injection Pipe	Inner Dia.	3.5 mm
	Length	650 mm

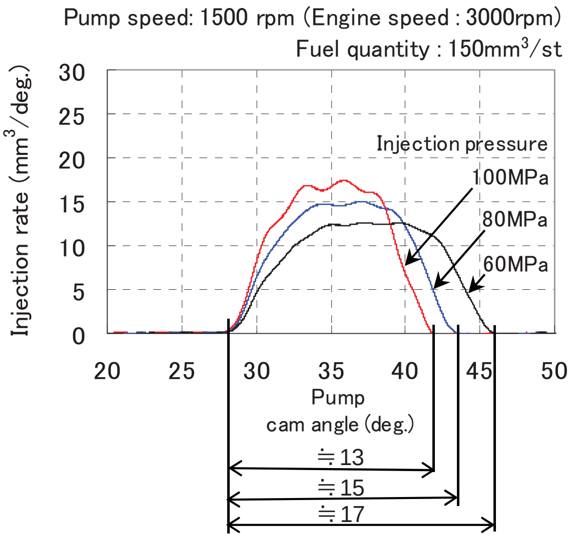

Fig. 2. Injection rate

2.4. Measurement system

Figure 3 shows the experimental engine system. The engine was coupled with a dynamometer for regulating the engine speed. The cylinder pressure was measured with a piezoelectric pressure transducer and logged in intervals of 0.5 crank angle degrees for combustion analysis. Exhaust emissions were measured with an exhaust analyzer (HORIBA MEXA7500DEGR). The exhaust gas was collected from two sample lines before DOC and after DOC to investigate the effects of un-burned gas reduction. CO and CO_2 were measured by the NDIR technique, and NOx was measured by the CLD technique, and THC was measured by the FID technique, and CH_4 was measured by the NMC (non-methane cutter)-FID technique. NMHC (non-methane hydrocarbons) was calculated by subtracting CH_4 from THC. The sensitivity of the FID technique for DME is lower than hydrocarbons without oxygen elements [8] due to the fact that DME has oxygen elements. In a current study, the correction factor of DME [8] was used when the combustion efficiency was calculated. The NMHC values in this paper did not use the correction factor.

2.5. Experimental condition

Table 4 shows the operating points in the experiment. The experiments were performed at four operating points to consider optimum injection pressures in a wide operating range. The parameters of the experiment were injection pressures and injection timing. EGR was used only at a low load as BMEP of 0.2 MPa to reduce NOx. The EGR ratio was 50 % at 1000 rpm and was 30 % at 3000 rpm.

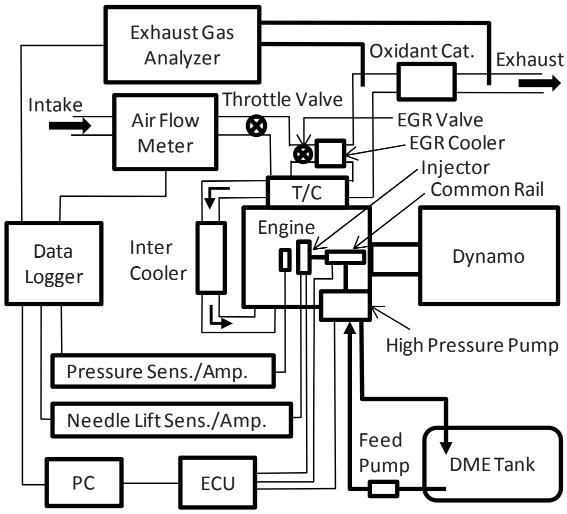

Fig. 3. Schematic diagram of engine experiment

Tab. 4. Operating points and experimental parameters

Operating points

	Engine speed (rpm)	BMEP (MPa)	EGR (%)
①	1000	0.2	50
②	1000	0.89	0
③	3000	0.2	30
④	3000	0.89	0

Parameters

Common rail pressure (MPa)	20 ～ 100
Injection timing (deg. ATDC)	−35 ～ 10

2.6. Analysis of fuel consumption

The influencing factors of fuel consumption was analysed to consider the influence of injection pressure. The methods for calculating the influencing factors will be covered in this section. Brake thermal efficiency η_e, indicated thermal efficiency η_i and mechanical efficiency η_m can be represented by equation (1) and equation (2) respectively [13].

$$\eta_e = \eta_i \cdot \eta_m , \tag{1}$$

$$\eta_i = \eta_{th} \cdot \eta_{gl} \cdot \eta_{comb} \cdot (1-\phi_w), \tag{2}$$

where:

η_e - brake thermal efficiency,
η_i - indicated thermal efficiency,
η_m - mechanical efficiency,
η_{th} - theoretical thermal efficiency of Otto cycle,
κ - specific heat ratio,
η_{gl} - degree of constant volume,
η_{comb} - combustion efficiency,
ϕ_w - cooling loss fraction.

The degree of constant volume η_{gl} was calculated by equation (3).

$$\eta_{gl} = \frac{1}{Q}\int \frac{dQ}{d\theta} \cdot \eta_{th\theta} d\theta , \tag{3}$$

where:
$\eta_{th\theta}$ - infinitesimal theoretical thermal efficiency of Otto cycle when combustion is in constant volume at crank angle,
Q - cumulative apparent heat-release,
θ - crank angle.

The combustion efficiency η_{comb} [14] was calculated by equation (4).

$$\eta_{comb} = 1 - \frac{\left[(CO \cdot H_{co}) + (THC \cdot K_{FID} \cdot H_{uDME})\right]}{BSFC \cdot H_{uDME}}, \tag{4}$$

where:
H_{co} - heating value of CO,
H_{uDME} - lower heating value of DME,
CO - CO in exhaust gas,
THC - THC in exhaust gas,
$BSFC$ - brake specific fuel consumption,
K_{FID} - correction factor of DME for FID.

The cooling loss fraction ϕ_w was calculated by equation (5).

$$\phi_w = 1 - \frac{Q}{Q_{inj\_mass\_cycle} \cdot H_{uDME} \cdot \eta_{comb}}, \tag{5}$$

where:
Q - cumulative apparent heat release,
$Q_{inj\_mass\_cycle}$ - fuel quantity per cycle,
H_{uDME} - lower heating value of DME,
η_{comb} - combustion efficiency.

3. Results and discussion

3.1. The influence of injection pressure and injection timing on the trade-off between fuel consumption and NOx

This section considers optimum injection pressures for the trade-off between fuel consumption and NOx.

Figure 4 shows the effect of various injection pressures and injection timing on the trade-off between fuel consumption (BSFC) and NOx. The trade-off at 1000 rpm tends to improve with decreasing injection pressure, the best results at both BMEP are achieved when an injection pressure of 25 MPa is used. The BSFC at the 25 MPa in 1000 rpm and the BMEP of 0.89 MPa tends to be worse when injection timing retards with reducing NOx. These BSFC and NOx in 1000 rpm and BMEP of 0.2 MPa simultaneously can be reduced by retarding the injection timing because BSFC will not be so worse. The trade-off at 3000 rpm and BMEP of 0.89 MPa on retarded injection timing side improves when the injection pressure increases from 60 MPa to 80 MPa. The trade-off almost does not change when injection pressure increases over 80 MPa. The best result at 3000 rpm and BMEP of 0.2 MPa is achieved when an injection pressure of 60 MPa is used. Both of the 40 MPa and 60 MPa results at this condition are almost same when the injection timing has been retarded.

It can be concluded from the results that the optimum injection pressure for the trade-off between BSFC and NOx is around 25 MPa at 1000 rpm at both BMEP. It is around 60 MPa at 3000 rpm with a BMEP of 0.2 MPa. It is around 90 MPa at 3000 rpm with a BMEP of 0.89 MPa.

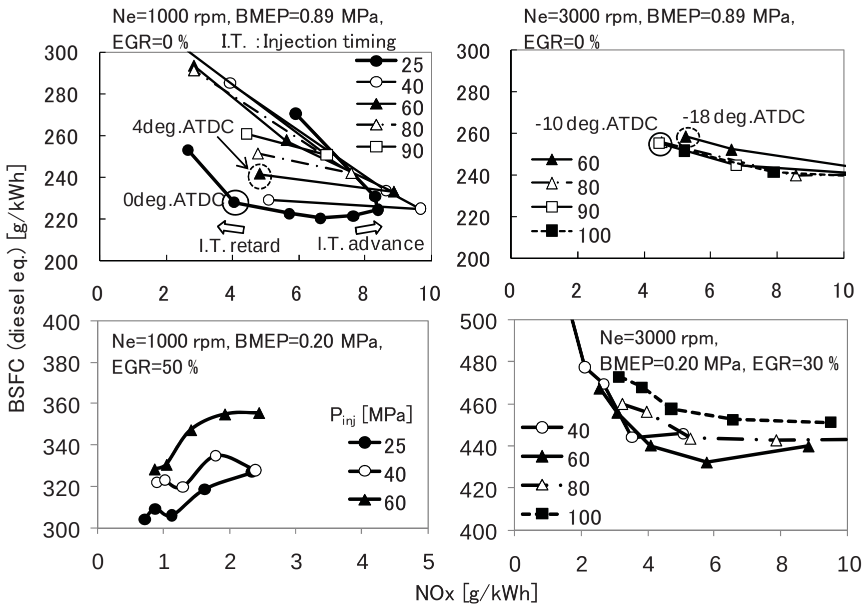

Fig. 4. Influence of injection pressure and injection timing on BSFC and NOx trade off

A combustion analysis is used to investigate the reasons for improvement of the trade-off.

Figure 5 shows the results of a comparison between 25 MPa and 60 MPa at 1000 rpm and BMEP of 0.89 MPa (the circles in Fig. 4). The combustion duration of 25 MPa is longer than that of 60 MPa, and peak of the rate of heat released at 25 MPa is lower. The maximum cylinder pressure of 25 MPa is also lower. It can be considered from these that NOx of 25 MPa decreases, though injection timing advanced. The indicated thermal efficiency of 25 MPa is about 1.5% higher than that of 60 MPa, the mechanical efficiency of 25 MPa also is about 4.5% higher from the fuel consumption analysis. The indicated thermal efficiency of 25 MPa improved compared with that of 60 MPa because the combustion efficiency and the cooling loss fraction of 25 MPa are better, though the degree of constant volume is worse. In case of 25 MPa, the maximum cylinder pressure is lower, and the drive work of the high pressure pump is assumed to be lower due to low injection pressure. Those suggest that the mechanical efficiency of 25 MPa is higher than that of 60 MPa.

Therefore, the fuel consumption and NOx at 1000 rpm were better when the injection pressure of 25 MPa was used, because the combustion duration is long and the mechanical efficiency increases with the decreasing injection pressure.

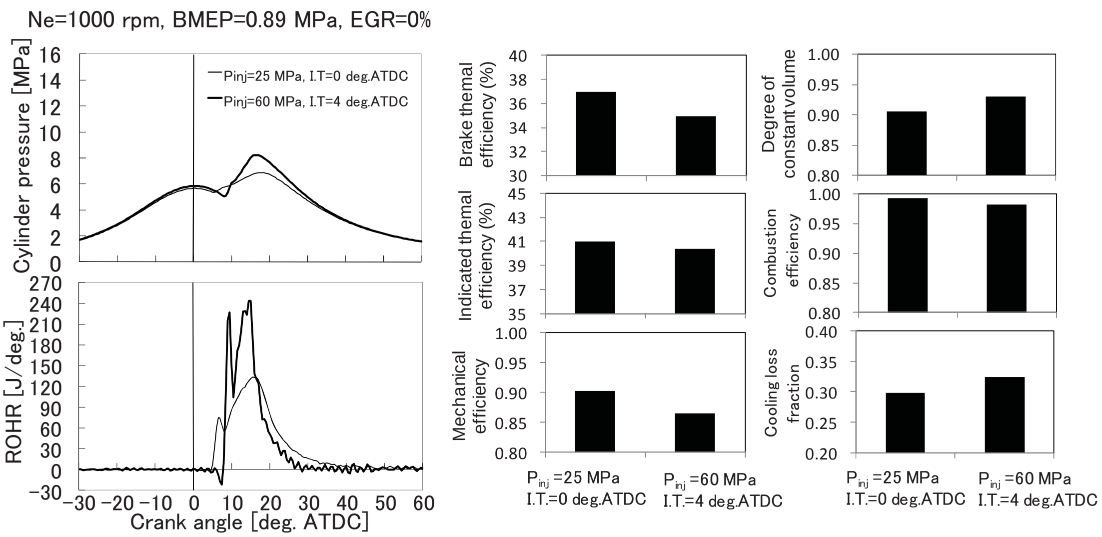

Fig. 5. Analysis of influence of injection pressure on BSFC-NOx trade-off at 1000 rpm and BMEP of 0.89 MPa

Figure 6 shows the results of the combustion analysis comparing 60 MPa with 90 MPa at 3000 rpm and a BMEP of 0.89 MPa (the circles in Fig. 4). The end of combustion at 90 MPa is close to that of 60 MPa due to the short combustion duration, through the injection timing is retarding compared with 60 MPa. It can be considered from these that the NO of 90 MPa was lower due to a lower maximum cylinder pressure. The indicated thermal efficiency of 90 MPa is about 1.5% higher than that of 60 MPa and the mechanical efficiency of 90 MPa is close to that of 60 MPa. The degree of constant volume of 90 MPa is higher than that of 60 MPa, and the cooling loss fraction of 90 MPa is also little higher than that of 60 MPa.

Therefore, the fuel consumption and NOx at 3000 rpm was better when an injection pressure of 90 MPa was used, because the degree of constant volume was higher and injection timing became retarded. However, the better fuel consumption at 3000 rpm and BMEP of 0.2 MPa was achieved when an injection pressure of 60 MPa was used. This reason is assumed that the mechanical loss is too high when an injection pressure is over 80 MPa.

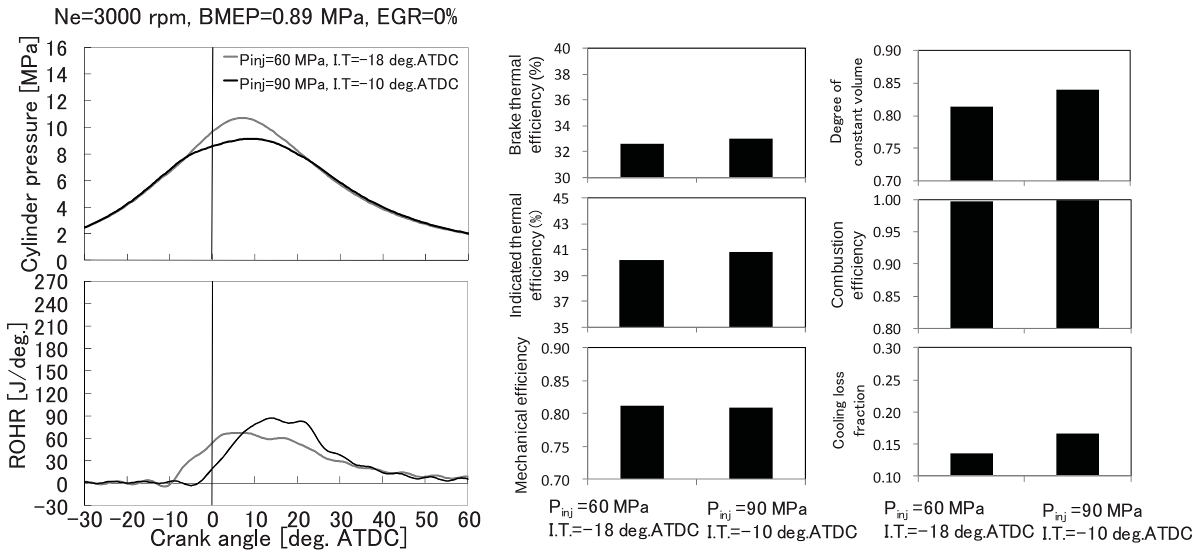

Fig. 6. Analysis of influence of injection pressure on BSFC-NOx trade-off at 3000rpm and BMEP of 0.89 MPa

3.2. Influence of injection pressure and injection timing on the trade-off between un-burned gas and NOx

This section discusses optimum injection pressures for the trade-off between CO, NMHC and NOx.

Figure 7 shows the trade-off between CO and NOx when the injection pressure and injection timing varied. The trade-off at 1000 rpm and a BMEP of 0.89 MPa tends to improve by decreasing injection pressure and the best results are achieved when an injection pressure of 25 MPa was used. This tendency is similar to the result of the trade-off between BSFC and NOx under the same conditions.

Figure 8 shows change of the excess air fuel ratio. The excess air ratio in 1000 rpm and a BMEP of 0.89 MPa decreases below 1.5 with increasing injection pressure. It is well known about diesel engine performance that CO increases rapidly when the excess air ratio becomes below 1.5, so that relatively high concentration CO is emitted when injection pressure increased. The tendency of the influence of injection pressure for CO is similar to the tendency of BSFC, because the excess air ratio decreases due to the increasing BSFC. On the other hand, CO at 3000 rpm and a BMEP of 0.89 MPa is much lower than that of 1000 rpm at any injection pressure. The main reasons are that the excess air ratio at this condition is around 1.7, and the mixing of DME and air was improved by the increasing the engine speed. The boost pressure increased with the increasing engine speed because the exhaust gas volume per unit time and exhaust gas temperature increased, so that the excess air ratio at 3000 rpm was higher than that of 1000 rpm. The CO at 3000 rpm and BMEP of 0.89 MPa improves while keeping a constant NOx level when the injection pressure increases from 60 MPa to 80 MPa, this assume that the air induction into DME splay was accelerated. The effect of injection pressure on CO in 1000 rpm and BMEP of 0.2 MPa was not observed because it was compensated by decreasing excess air ratio with the effect of improving the DME-fuel mixing by high pressure injection. The best result at 3000 rpm and a BMEP of 0.2 MPa is achieved when injection pressure of 60 MPa is used.

Therefore, the optimum injection pressure for the trade-off between CO and NOx is high injection pressure in the high speed and high load condition, and it has a low pressure injection with low engine speeds.

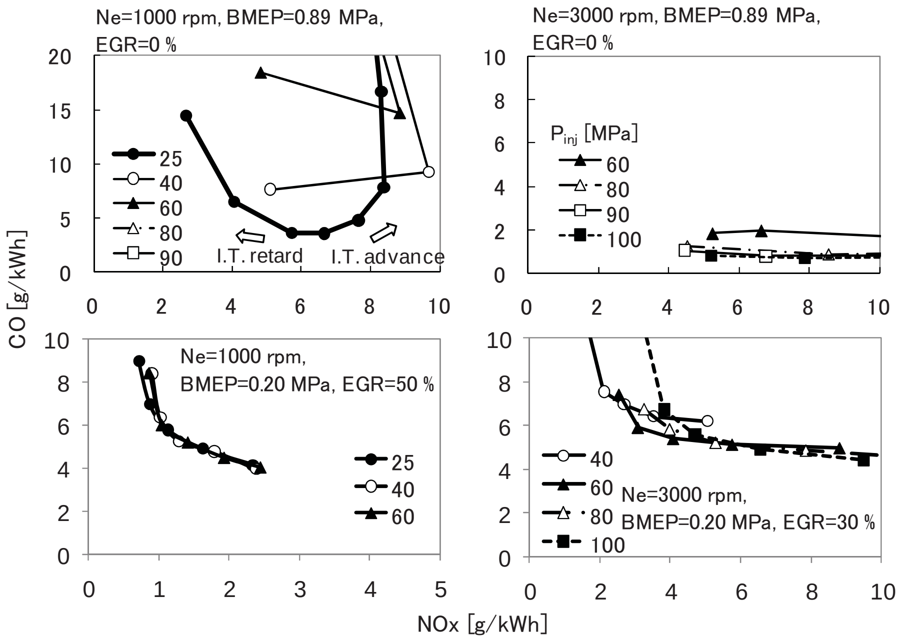

Fig. 7. Influence of injection pressure and injection timing on CO and NOx trade off

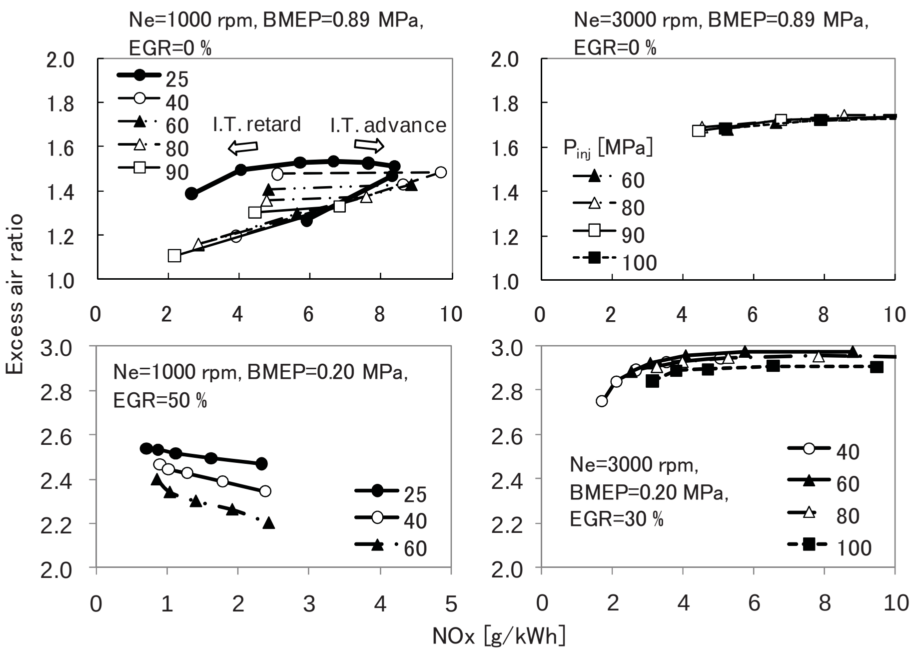

Fig. 8. Change of excess air ratio

Figure 9 shows the trade-off between NMHC and NOx when injection pressure and injection timing varied. NMHC at BMEP of 0.2 MPa is relatively higher than that at the BMEP of 0.89 MPa, because the cylinder wall temperature was lower and the ignition delay was longer. The best result at 1000 rpm and a BMEP of 0.89 MPa is achieved when an injection pressure of 25 MPa is used, and the NMHC in the 25 MPa was reduced by retarding the injection timing. The NMHC at 3000 rpm and BMEP of 0.89 MPa improves when injection pressure increases from 60 MPa to 80 MPa, these NMHC tend to reduce with retarding injection timing. This main reason is that the ignition delay becoming shorter. The best result at 3000 rpm and a BMEP of 0.2 MPa is achieved when an injection pressure of 60 MPa is used, this tendency is the same as the tendency of the BSFC or CO. The NMHC at 1000 rpm and BMEP of 0.2 MPa reduces when the injection pressure increases from 25 MPa. If the injection pressure increased over 60 MPa, the NMHC can be reduced. An NMHC with low speeds and low load condition were conspicuously influenced by the shortening ignition delay and increasing cylinder temperature by high pressure injection.

Therefore, it can be concluded that the optimum injection pressure for the trade-off between NMHC and NOx is around 25 MPa at 1000 rpm with a BMEP of 0.89 MPa, and it is over 80 MPa at 3000 rpm with a BMEP of 0.89 MPa. In 1000 rpm and a BMEP of 0.2 MPa, NMHC can be reduced when the injection pressure increases from 25 MPa.

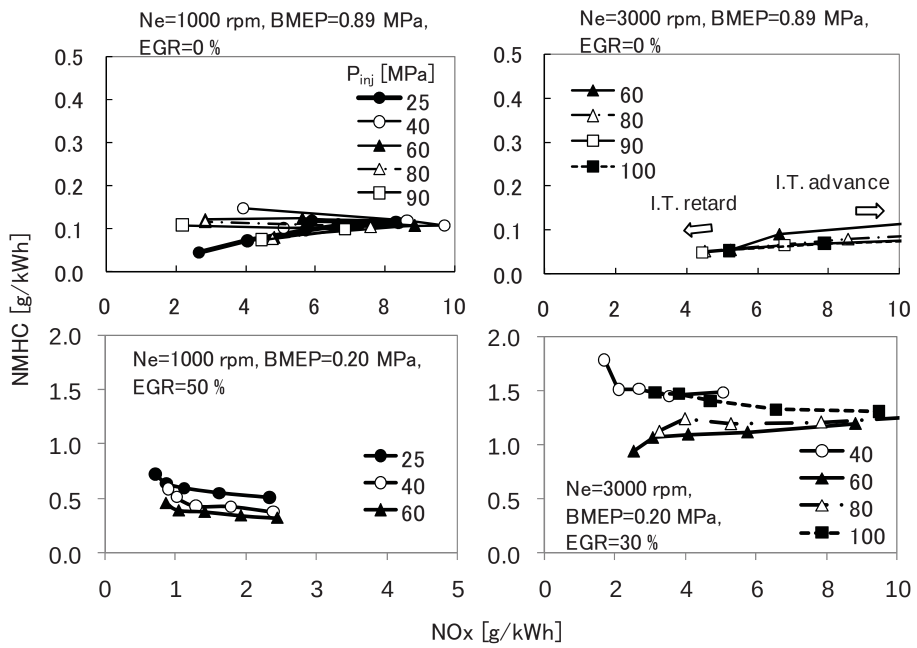

Fig. 9. Influence of injection pressure and injection timing on NMHCand NOx trade off

3.3. The influence of injection pressure and injection timing on the trade-off between un-burned gas and NOx with diesel oxidant catalyst

The activation of the DOC is varied by injection pressure due to the different exhaust gas temperature. Therefore, it can be expected that optimum injection pressures with DOC is difficult than it is without DOC. This section discusses optimum injection pressures for the trade-off between CO, NMHC and NOx with DOC.

Figure 10 shows the trade-off between CO and NOx after DOC when injection pressure and injection timing varied. CO is almost reduced to level zero by DOC at any operating points and any injection pressure. For the reason, Platinum has a very high conversion characteristic for CO.
Figure 11 shows the trade-off between NMHC and NOx after DOC when the injection pressure and injection timing varied. NMHC at the high load BMEP of 0.89 MPa can be reduced to nearly zero by DOC at any injection pressure. However, while the effect of the NMHC reduction at 1000 rpm and

BMEP of 0.2 MPa is not observed, the values after DOC is almost the same as that before DOC in Fig. 9. Figure 12 shows the conversion rate of the DOC. The horizontal axis is the exhaust gas temperature near the entrance side of the DOC. The conversion rate of NMHC is worse than that of CO in the low temperature range. NMHC can be reduced by the DOC when the exhaust gas temperature is over 500 K. The condition at 1000 rpm and a BMEP of 0.2 MPa, the exhaust gas temperature is below 450 K, so that the NMHC could not be decreased. At 3000 rpm and a BMEP of 0.2 MPa, the NMHC of 40 MPa greatly decreases compared with that of before DOC in Fig. 9. This main reason is that the activation of the DOC of 40 MPa increased, because the exhaust gas temperature with a low injection pressure is higher than that of a high injection pressure due to the increasing combustion duration.

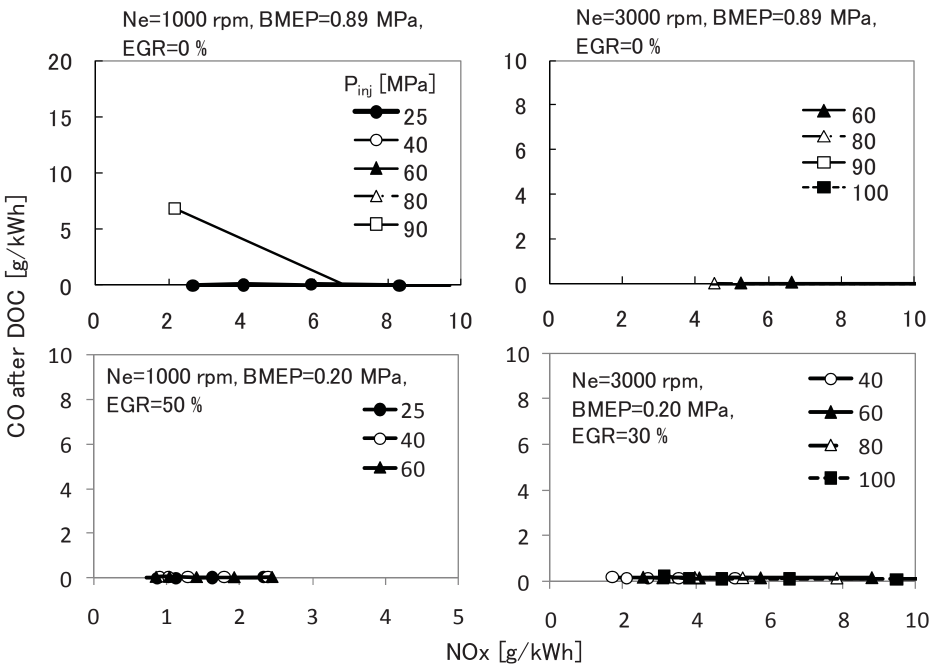

Fig. 10. Influence of injection pressure and injection timing on CO and NOx trade off after diesel oxide catalyst

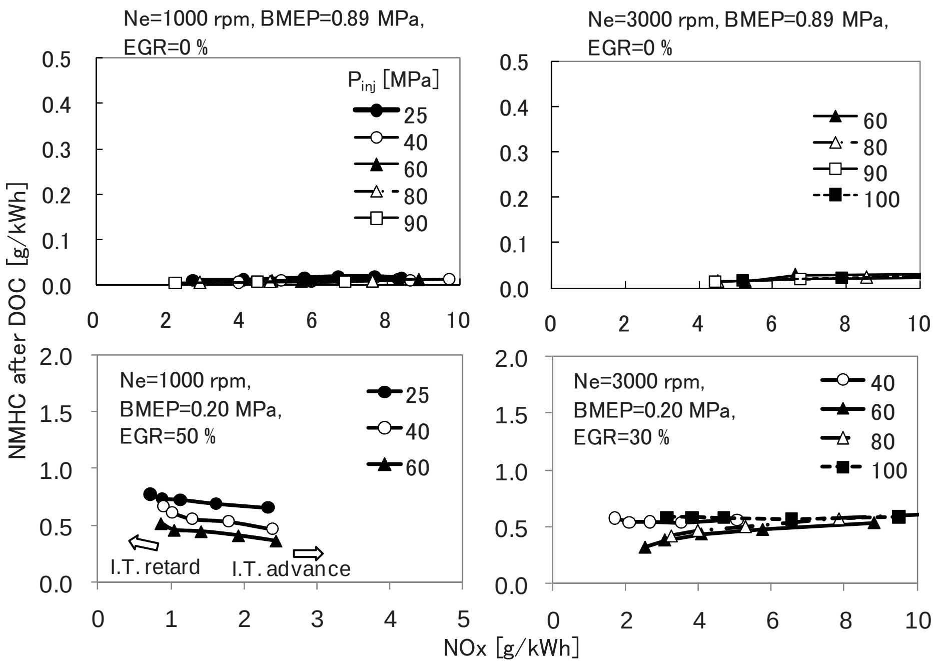

Fig. 11. Influence of injection pressure and injection timing on the NMHC and NOx trade off after the diesel oxide catalyst

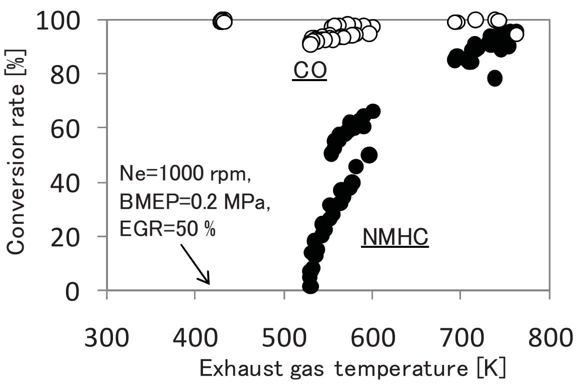

Fig. 12. Conversion rate of diesel oxidant catalyst

Therefore, in the case of using DOC, CO can be reduced to almost level zero at any of the operating points. However, NMHC at a low speed and low load cannot be reduced by the DOC, and high injection pressure is effective in reducing NMHC under this condition.

3.4. Optimized injection pressure for fuel consumption and exhaust emissions

It can be found from the above facts that there are optimum injection pressures for fuel consumption, NOx, CO and NMHC at each operating point. Both optimum injection pressures for fuel consumption and exhaust emissions are almost commonly expected at 1000 rpm and a BMEP of 0.2 MPa. The problem of the increasing NMHC at 1000 rpm with a BMEP of 0.2 MPa still remained. The NMHC at 1000 rpm and BMEP of 0.2 MPa can be expected to reduce in the future by refinements of the composition of the oxidant catalyst, the specification of the injection nozzle and the combustion chamber. This operating condition makes a point about the minimizing fuel consumption and NOx. Fig. 13 shows the optimized injection pressure and the results of the engine performances.

Therefore, the optimum injection pressure at 1000 rpm is around 25 MPa at any of the engine loads. The injection pressure has to increase with the increasing engine speed, the optimum injection pressure is around 90 MPa at 3000 rpm and BMEP 0.89 MPa, and is around 60 MPa at 3000 rpm and a BMEP of 0.2 MPa.

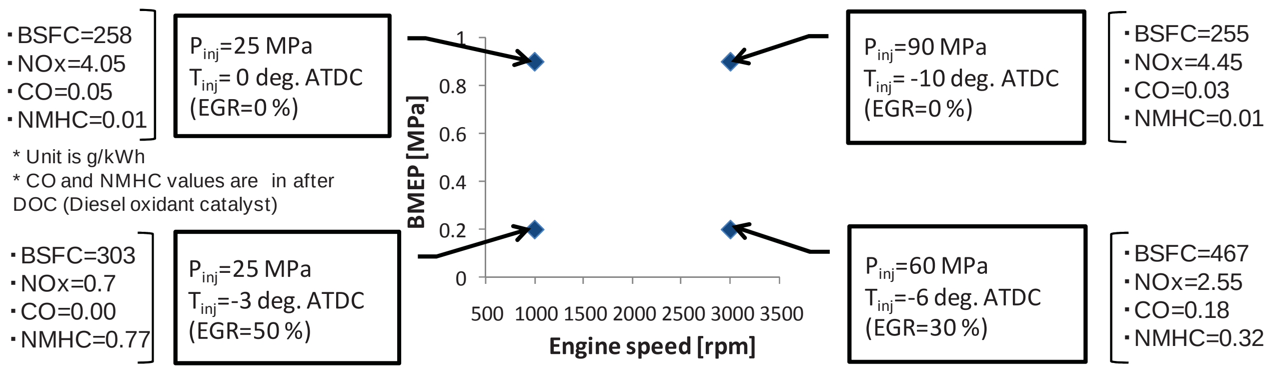

Fig. 13. Optimized injection pressure, injection timing and the results of the engine performances

4. Conclusions

This study is intended to improve the fuel consumption and exhaust emission of a DME engine. The influence of injection pressure on fuel consumption, NO, CO, NMHC with EGR and a diesel oxidation catalyst have been investigated using injection pressure up to 100 MPa, and the optimum injection pressures have been discussed from these results.

1) The trade-off between BSFC and NOx at the low engine speed of 1000 rpm tend to improve when an injection pressure decrease, because the cylinder pressure decrease and the mechanical efficiency increase simultaneously. The trade-off at high engine speed of 3000 rpm tend to improve when injection pressure increase, because the degree of constant volume increase and the injection timing can be retarded.
2) The optimum injection pressure for the trade-off between CO and NOx is high injection pressure in the high speed and high load condition, and it has a low pressure injection with low engine speeds. The optimum injection pressure for the trade-off between NMHC and NOx is around 25 MPa at 1000 rpm with a BMEP of 0.89 MPa, and it is over 80 MPa at 3000 rpm with a BMEP of 0.89 MPa, NMHC at 1000 rpm and a BMEP of 0.2 MPa can be reduced when the injection pressure increases from 25 MPa.
3) In the case of using DOC, CO can be reduced to almost level zero at any of the operating points. However, NMHC at a low speed and low load cannot be reduced by the DOC, and high injection pressure is effective in reducing NMHC under this condition. The NMHC at a low speed and low load range must be reduced in the future by refinements of the composition of the oxidant catalyst, the specification of the injection nozzle and the combustion chamber.
4) It can be concluded from the results as follows; the optimum injection pressure at 1000 rpm is around 25 MPa at any engine load. The injection pressure has to increase with the increasing engine speed. The optimum injection pressure is around 90 MPa at 3000 rpm with a BMEP of 0.89 MPa, and is around 60 MPa at 3000 rpm with a BMEP of 0.2 MPa.

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