A Study on Effect of Environmental Characteristics by Intake Mixture Temperature in Scrubber EGR System Diesel Engines

MYUNG-WHAN BAE, CHANG-SUNG RYU

School of Mechanical and Aerospace Engineering, Gyeongsang National University,
900, Gajwa-dong, Chinju, Gyeongnam, 660-701, Korea
Phone : +82-55-751-6071, Fax : +82-55-762-0227
E-mail : mwbae@nongae.gsnu.ac.kr

ABSTRACT

The effects of intake mixture temperature on performance and exhaust emissions under four kinds of engine loads were experimentally investigated by using a four-cycle, four-cylinder, swirl chamber type, water-cooled diesel engine with scrubber EGR system operating at three kinds of engine speeds. The purpose of this study is to develop the scrubber exhaust gas recirculation(EGR) control system for reducing NO_x and soot emissions simultaneously in diesel engines. The EGR system is used to reduce NO_x emissions. And a novel diesel soot-removal device of cylinder-type scrubber with five water injection nozzles is specially designed and manufactured to reduce soot contents in the recirculated exhaust gas to the intake system of the engine. The influences of cooled EGR and water injection, however, would be included within those of scrubber EGR system. In order to survey the effect of intake mixture temperature on performance and exhaust emissions, the intake mixtures of fresh air and recirculated exhaust gas are heated by a heating device with five heating coils made of a steel drum. It is found that the specific fuel consumption rate is considerably elevated by the increase of intake mixture temperature, and that NO_x emissions are markedly decreased as EGR rates are increased and intake mixture temperature is dropped, while soot emissions are increased with increasing EGR rates and intake mixture temperature. Thus one can conclude that the performance and exhaust emissions are considerably influenced by the cooled EGR.

Key Words : Intake Mixture Temperature, Diesel Engine, Scrubber EGR System, Exhaust Gas Recirculation (EGR), NO_x and Soot Emissions, Cooled EGR

1. INTRODUCTION

The diesel engine has the greatest heat-efficiency among the existing heat engines, with more than 50% of thermal efficiency in two-cycle, low speed, large-sized marine diesel engines and with more than 45% of maximum thermal efficiency in buses and trucks powered by diesel engines.

Despite its advantages, however, the major environmental problem with the diesel engine is that it generates vast amounts of NO_x and soot emissions. Naturally, the top priority of diesel combustion engineering has been to find ways to reduce hazardous exhaust gases without downgrading the engine's excellent thermal efficiency. Though NO_x and soot emissions, among harmful exhaust emissions, do not lend themselves to simultaneous reduction due to trade-off effect, there have been numerous experimental studies based on combinations of high pressure spray, intercooler system turbocharging, cooled EGR, oxidation catalyst, etc. These technologies, however, presuppose improvements in combustion engineering, such as lower sulfates and oxygen-containing fuel and at the same time after-treatment must also be developed, such as lean burn, catalyst and DPF(diesel particulate filter trap).[1, 2]

Though EGR is a widely known method that can reduce NO_x emissions drastically without additionally mounted devices or renovations, it tends to increase other exhaust gases, especially soot emissions. Controlled EGR systems, however, are widely adopted in small-sized diesel engines, and reliability and endurance tests are performed to make this system feasible in medium- and large-sized engines. The application of EGR to these larger engines is delayed for the same reason: increase in soot emissions and other exhaust gases.[3 ~ 7]

Bae et al. employed a simple EGR system to investigate the effect of recirculated exhaust gases on fuel consumption rate, NO_x and soot emissions, with intake and exhaust oxygen concentrations, and equivalence ratio as parameters for increases in EGR rate.[8]

Recently, Bae et al. devised a scrubber EGR system that eliminates soot emissions from exhaust gases by spraying water onto these gases; this EGR system incorporated an EGR system for reducing NO_x and a scrubber-type water-spray system for eliminating soot contents in order to investigate the effect of EGR rates in recirculated exhaust gases on the engine's internal wear, fuel consumption rate, heat release rate and exhaust emissions.[9, 10] Pure EGR effect, however, could not be determined, because exhaust gases contained moisture and recirculated exhaust gases mixed with fresh air at the surge tank underwent cooled EGR due to water sprayed for eliminating soot contents regardless of intention.

The purpose of this study is to investigate the effects of moisture and cooled EGR in a diesel engine with the scrubber EGR system. For this purpose, mixed gases(recirculated exhaust gases + fresh air) were heated in a cylinder-shaped electrical heater to eliminate moisture and to ensure hot EGR, in order to examine the effect of temperature variations in intake mixtures on fuel economy, combustion, and exhaust characteristics.

2. EXPERIMENTAL APPARATUS AND PROCEDURES

2.1 Apparatus

The schematic diagram of apparatus used in the experiment is shown in Fig. 1. The test engine employed for the experiment is a four-stroke, four-cylinder, water-cooled, swirl chamber type, automobile diesel engine manufactured in Korea. The major specifications of the test engine are presented in Table 1.

The engine power was measured by DC eddy current dynamometer. The piezo-electric pressure transducer(Kistler 6061B) was installed onto the head part of the first cylinder to monitor the cylinder pressure in the combustion chamber. Fuel consumption rate was calculated by measuring the amount of fuel consumption per time unit on a mass flow meter(HF-2000GD). The intake air amount was measured by a laminar flow elements. Exhaust emissions were measured by exhaust analyzers; CO and CO_2 emissions by the NDIR exhaust analyzer(CGT-7000); O_2 emissions by an O_2 analyzer employing Zirconia method; NO_x emissions by a NO_x analyzer employing chemiluminescent method(Signal 4000VM); THC emissions by a THC analyzer employing HFID(Signal 3000 HM); and soot emissions by Reflection Photometry-Filter type diesel smoke meter(DST-210).

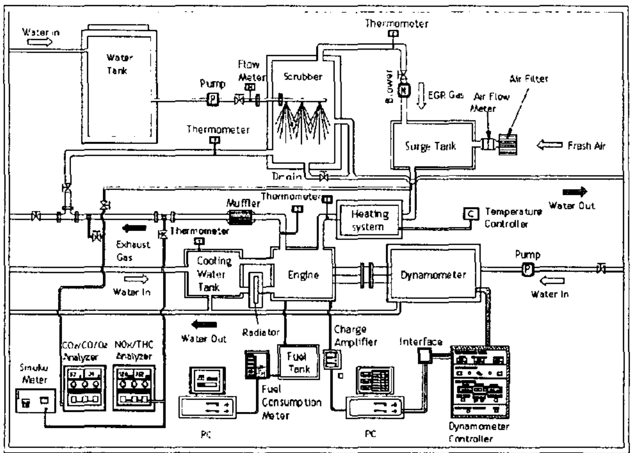


Fig. 1 Schematic of experimental apparatus

Table 1 Specifications of test engine

Item	Specification
Type	4 Cylinder, 4 Cycle, Water-cooled, Swirl chamber, Natural aspiration
Piston Displacement (cc)	2476
Bore (mm) × Stroke (mm)	91.1 × 95
Max. Power	58.82 kW/4200 rpm
Fuel Injection Timing	ATDC 4°
Compression Ratio	21 : 1

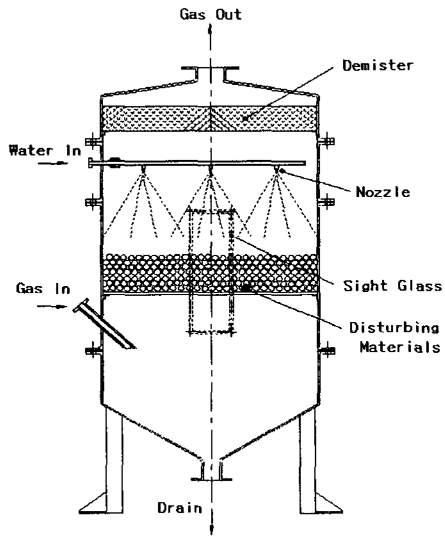


Fig. 2 Cross-section of a novel diesel soot-removal system with a cylinder-type scrubber

A surge tank was installed on the side of the engine intake manifold, not only to ensure the effective mixing of fresh air and recirculated gas but also to minimize the surging. A digital thermometer/hygrometer was installed to measure the temperature and humidity of intake mixtures(fresh air + recirculated exhaust gas). In order to allow an adequate supply of recirculated exhaust gas and minimize the back pressure in the engine due to the soot-removal device, a Roots blower was connected between the scrubber outlet and the surge tank inlet.

2.2 Soot-removal System and Intake Mixture Heater

Figure 2 shows the soot-removal system with a cylinder-type scrubber that is installed between the exhaust pipe and the surge tank. The exhaust gases emitted from the engine are directed into the inlet port in the lower side of the scrubber, and collide with agitating materials, finally its soot contents being eliminated by water sprayed from five conical nozzles. To ensure a constant and continuous water flow, sprayed water is controlled by a flow-control valve installed next to a pump. An orifice flow meter was installed to measure the water supply amount. The resultant soot-free exhaust gas is passed through a demister to remove water, which is fanned through the outlet port into a surge tank, where it is mixed with incoming fresh air before being fed back into the cylinders.

Water-spray for removing the soot contents is predicted to cause cooled EGR on recirculated exhaust gases, which may be further increased by water-spray for NO_X emissions reduction. To examine the effects of these two factors, an electrical heater was installed to eliminate moisture from intake mixtures. A 850 mm × ø 560 mm steel plate was shaped into a cylinder, inside of which was installed by five circular heating coils with an electrical current of 6.7 A. Recirculated exhaust gases and fresh air mixed in the surge tank are fed into the heating cylinder through a ø 100 mm pipe and then is heated before flowing into the intake manifold. The mixed gases were heated up to 80℃ in normal operation; it was found that it would be difficult to heat at temperatures higher than this particular temperature.

2.3 Procedures

At engine speeds of 1800, 2800 and 3800 rpm, the experiments were conducted over engine load ranges of 25 and 100% with a 25% interval. The EGR rates of 0, 10 and 20% were tried on intake mixtures heated to 40, 60 and 80℃(with 25℃ as baseline). 25℃ means the intake mixture temperature mixed in the surge tank passed through a cylinder-type scrubber without heating. At the engine speed of

3800 rpm, however, the gases could not be heated to higher than 40℃, because a higher temperature would lead to unstable engine operation due to unsatisfactory intake charging efficiency. Of engine speed, load, EGR rate and intake mixture temperature, three parameters were kept constant and only one parameter was varied. Of all the engine speeds, however, the data involving with 2800 rpm were adopted in the discussion of this study because the whole data were too vast for presenting and because there was a similar tendency for the other engine speeds. Though the engine was operated in the same conditions, the variation of EGR rate is less than ±3%; the variation rate of engine load, less than ±5%; and the temperature variation rate of heated intake mixture, less than ± 1℃.

A water-cooled heat exchanger was employed to control the temperature of cooling water and lubricating oil, and K-type thermocouples(CA) were inserted into the engine and its components(exhaust manifold, the inlets and outlets of the scrubber and the surge tank, and the inlets and outlets of cooling water and lubricating oil) to check regularly the engine operation and combustion. The amount of water sprayed in the soot-removal device was fixed at 30 ℓ /min.

CO_2 concentration was measured at the intake and exhaust manifolds to calculate the EGR rate by using the following equation.[9, 10]

$$\text{EGR Rate(\%)} = \frac{[CO_2]_{EGR} - [CO_2]_{w/o\ EGR}}{[CO_2]_{EXH}} \times 100$$

where $[CO_2]_{EGR}$ is the CO_2 concentration in intake air with EGR; $[CO_2]_{W/O\ EGR}$ is CO_2 concentration in intake air without EGR; and $[CO_2]_{EXH}$ is CO_2 concentration in the exhaust manifold with EGR. Though the fuel injection timing is usually one of experimental parameters in applying EGR, this study fixes it at ATDC 4° , as shown in Table 1.

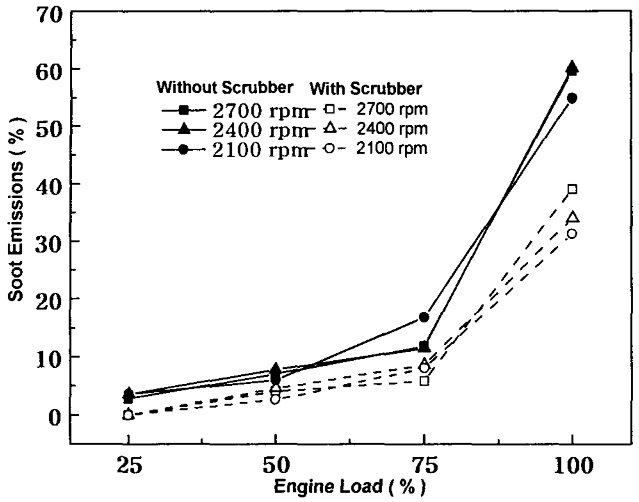
3. RESULTS AND DISCUSSION

3.1 Efficiency of Soot-removal Device

To investigate the performance of the soot-removal device, the cylinder-type scrubber manufactured for this study, engine speed was varied from 1200 to 3900 rpm at an interval of 300 rpm, with engine load increased at an interval of 25%. Fig. 3 shows the measured soot concentrations as a function of engine load when the whole exhaust gases go through with and without the scrubber at the typical engine speeds of 2100, 2400 and 2700 rpm. Though showing variations depending on engine speeds and loads, when passed through the scrubber, soot emissions were found to be reduced by 30 ~ 44% at 100% engine load; by 40 ~ 70% at 75% engine load; by 40 ~ 100% at 50% engine load; and by 100% at 25% engine load. The reduction rate of 100% here means that there is no soot concentrations detected by the smoke meter. To sum up the results, the reduction rate induced by the scrubber was 30% greater than without the scrubber; at the engine speed of 25%, 100% soot removal was achieved regardless of engine speeds. At higher engine speeds, in particular, lower engine loads produced more remarkable soot removal rates, and lower engine speeds generated much the same removal effect regardless of engine loads.

3.2 Specific Fuel Consumption Rate

To investigate the effect of intake mixture temperature with EGR application on the specific fuel consumption rate, the engine speeds of 1800, 2800 and 3800 rpm were applied to the engine, with engine load, intake mixture temperature and EGR rate as parameters. Fig. 4 shows typically the measured results at the engine speed of 2800 rpm. As indicated in the figure, in case of the same engine speed and load, the increase or decrease in the specific fuel consumption rate with increasing EGR rates



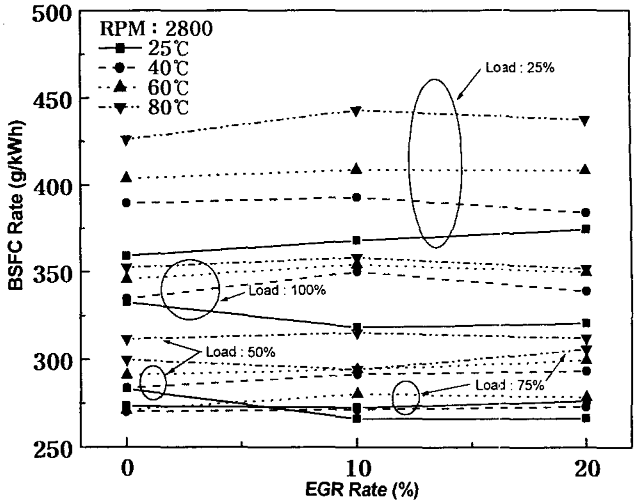


Fig. 3 Comparison of the soot reduction rate between without and with scrubber in a given engine speed 2100, 2400 and 2700 rpm

Fig. 4 Effect of EGR rate on brake specific fuel consumption rate as a parameter of intake temperature at an engine speed of 2800 rpm

was not significantly great up to the EGR rate of 20%. Moreover, the variation was not significant as compared with a serious effect on air pollutions, and the variation rate was below 2 to 3%.

Previous research results of the effect of EGR rate on the fuel consumption rate are classified into the following categories: increased EGR rate producing 1) greater fuel consumption rate at a certain threshold of EGR rate[(11, 12)]; 2) decreased fuel consumption rate[(13, 14)]; and 3) no significant variation in fuel consumption rate.[(15, 16)] To generalize the results of Bae et al. and other researchers, however, there was no significant variation in fuel consumption rate within 40% EGR rate, beyond which the rate was found to increase due to decrease in intake oxygen concentration.[(8, 9)] Under the same conditions, higher temperatures of intake mixture generally produced greater fuel consumption rate in hot EGR than in cooled EGR; when these gases were heated from 25 to 80℃, in this study, maximum increases in fuel consumption rate were found to be 20.3% at 25% engine load; 18.4% at 50% engine load; 10.7% at 75% engine load; and 13.8% at 100% engine load. The effect of increased EGR rate proved to be insignificant.

The reason why increases in intake mixture temperature led to greater fuel consumption rates may be attributed to the possibility that instead of enhancing combustion by increasing flame temperature, higher intake mixture temperature caused local incomplete combustion due to relative decrease in intake oxygen concentration.[(17)]

Though fuel consumption rate was significantly variable depending upon experimental conditions, increased EGR rate generated a slight increase or decrease in fuel consumption, whereas increase in intake mixture temperature produced significant increases in fuel consumption.

3.3 Combustion Characteristics

To investigate the effects of EGR rate and temperature variation in intake mixture on combustion characteristics of the engine, the combustion pressure in the cylinder was measured at each of the engine loads and at the engine speeds of 1800, 2800 and 3800 rpm, and the measured pressures were used to calculate the rate of heat release. Fig. 5 shows typically the result obtained at the engine speed of 2800 rpm, with 100% engine load, with EGR rate at 0, 10 and 20%, and with intake mixture temperatures at 25, 40, 60 and 80℃. When all the other conditions were equal, increased EGR rate led to a slight increase in ignition delay. In addition, the rate of heat release with higher temperatures of intake mixture tended to increase at 20% EGR rate, and ignition delay increased slightly.

In their study of EGR and supercharging based on a single-cylinder, four-cycle, direct injection diesel engine, Uchida et al.[18] found that ignition delay increased with higher EGR rate, with great variation in the peak value of premixed combustion and with little or no effect on the one of diffusion combustion. The reasons they suggested are local combustion temperature decrease due to increase of inert contents in charging and reduction in oxygen concentration.

Saito et al.[19] in their research of EGR based on a single-cylinder, four-cycle, direct injection, non-turbocharger diesel engine reported that the ignition delay was hardly changed with increasing EGR, and that the duration of diffusion combustion grew longer because the peak value of premixed combustion decreased. The reason they explained was that decrease of oxygen concentration in intake mixture led to reduced mixing probability between fuel and oxygen, which resulted in decreased combustion rate at a certain time point or at a certain spot in the combustion chamber, thereby reducing the rate of heat release and deterring combustion temperature from rising.

Dünholz et al.[20]have showed that an inconsistent intake air temperature has no serious effect on ignition delay with increasing EGR rate(up to an EGR rate of 40%), while ignition delay increases with the EGR rate at a constant intake air temperature.

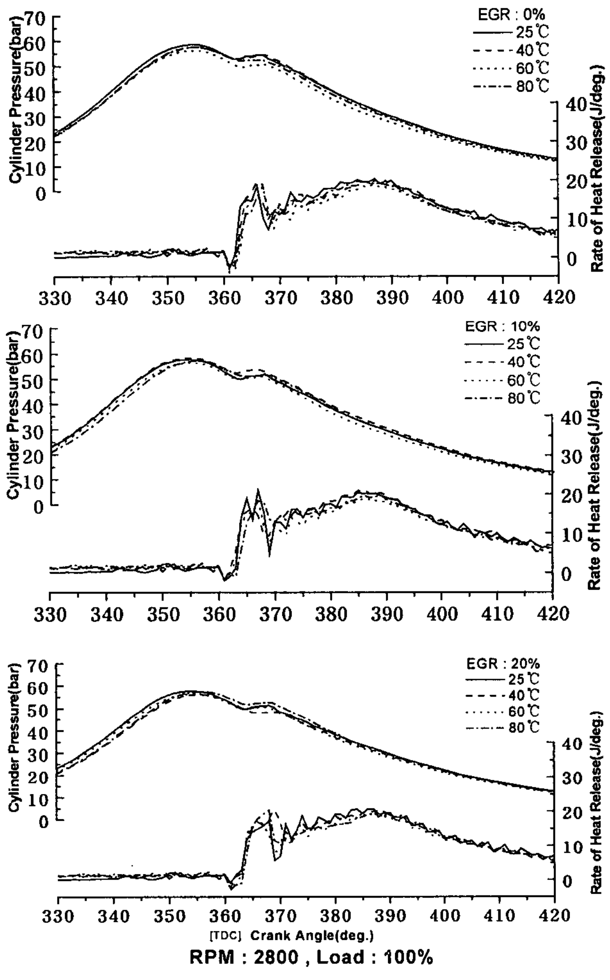


Fig. 5 Comparisons of cylinder pressure and rate of heat release as a parameter of intake mixture temperature at an engine speed of 2800 rpm and an engine load of 100%

It is difficult to draw an exact conclusion on combustion characteristics including the rate of heat release, since results are significantly dependent upon experimental conditions. According to the previous research results of Bae et al.,[21] however, the reason why ignition delay increases with the EGR rate may be attributed to reduced oxygen concentration in the intake air. The higher temperature in compressed mixture during ignition due to increased temperature in intake mixture may leads to the increase in ignition delay.

3.4 Exhaust Emissions

3.4.1 NO_x Emissions

To measure NO_x emissions, the engine speeds of 1800, 2800 and 3800 rpm were applied to the engine, with engine load, intake mixture temperature and EGR rate as parameters. Fig. 6 shows typically the measured values at the engine speed of 2800 rpm. As shown in the figure, at the same engine load and intake mixture temperature, NO_x emissions decreased with higher EGR rate, in particular, with greater engine loads producing even higher reduction in NO_x emissions. At the same engine load, hotter EGR led to higher intake mixture temperature, which causes higher flame temperature, resulting in greater NO_x emissions.

When intake mixture temperature was increased from 25 to 80℃ at 100% engine load, NO_x emissions at the engine speed of 1800 rpm increased by 6, 13.5 and 18.7% with regard to the EGR rates of 0, 10 and 20%; at 2800 rpm, by 17.6, 30.3 and 67.5%. When intake mixture temperature was increased from 25 to 40℃ at the engine speed of 3800 rpm, NO_x emissions increased by 1.5, 2 and 4.6% with regard to the EGR rates of 0, 10 and 20%. All these results demonstrated that the temperature had a significant effect on increase in NO_x emissions, and that intake mixture temperature had a greater effect with increases in EGR rate and engine speed.

The reason why higher EGR rates caused a remarkable reduction in NO_x emission may be attributed to the possibility that inert gases admixed into mixture may have caused increase in the heat capacity of these gases, which lowered maximum combustion temperature, at the same time leading to longer ignition delay due to decrease in intake oxygen concentration.[22]

3.4.2 Soot Emissions

At the engine speeds of 1800, 2800 and 3800 rpm, measurements were taken of soot concentrations as affected by the variations of intake mixture temperature and EGR rates at each of the engine loads for a given condition. Fig. 7 shows typically the measured values at the engine speed of 2800 rpm. At the same engine load, higher EGR rates caused greater increases in soot emissions; at less than 75% engine load, EGR rates did not have a significant effect on soot emissions, whereas EGR rates had a considerable influence at 100% engine load. In addition, at the same EGR rate, greater engine loads led to higher soot emissions, showing a drastic increase at 100% engine load.

The reason why increased EGR rates caused increase in soot emissions may be attributed to the possibility that increased EGR rates lowered intake oxygen concentration for combustion, thereby decreasing oxygen amounts when sprayed fuel was premixed with recirculated exhaust gases. And since decrease in intake oxygen concentration reduced oxygen amounts necessary for oxidizing soot during expansion stroke, soot emissions increased and the range of EGR rate at higher loads in diesel engines tended to be considerably restrict.[23]

Fig. 7 also shows the effect of temperature variation in intake mixture; when intake mixture temperature was increased from 25 to 80℃ at 100% engine load, soot emissions at 1800 rpm increased by 35.8, 31.7 and 21.1% with regard to the EGR rates of 0, 10 and 20%; at 2800 rpm, by 12, 7.1 and 9.2%; and at 3800 rpm, when temperature of intake mixture was increased from 25 to 40℃, soot emissions increased by 7.5, 3.3 and 4.5% with regard to the EGR rates of 0, 10 and 20%. These results demonstrated that at the same engine load, hotter EGR resulted in greater soot emissions. This increase rate, however, was shown to be lower with higher EGR rates. The reason why soot emissions increased with higher temperature of intake mixture may be that these hotter gases lowered intake charging efficiency, thereby decreasing intake oxygen concentration. The observation that influences due to higher EGR rates at more than 50% engine loads became weak may be

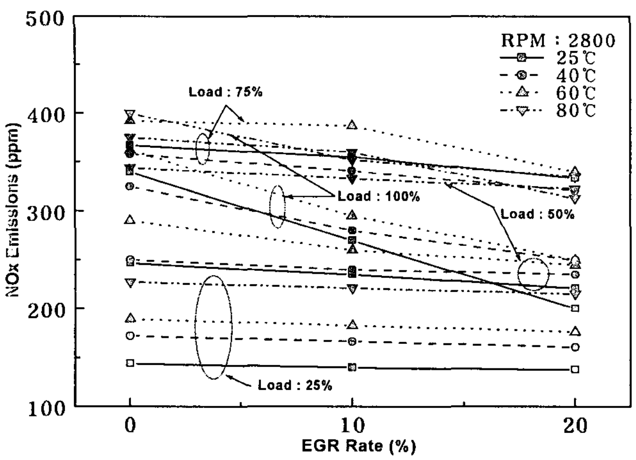


Fig. 6 Effect of EGR rate on NO_x emissions as parameters of intake mixture temperature and engine load at an engine speed of 2800 rpm

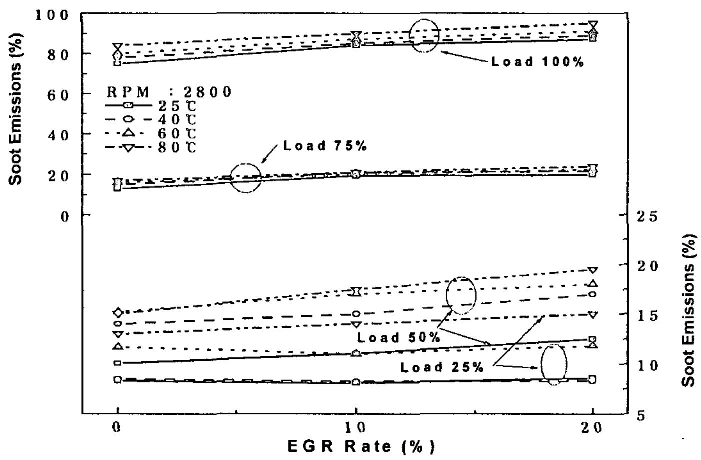


Fig. 7 Effect of EGR rate on soot emissions as parameter of intake mixture temperature and engine load at an engine speed of 2800 rpm

attributed to the possibility that because hotter temperatures in the combustion chamber had been generated by combustion, the effect of increased intake mixture temperature was weakened in the first place.

3.4.3 CO Emissions

In this study, unlike the others by Bae et al.,[8, 9, 21] CO and THC emissions were measured and discussed to examine the effect of heated intake mixture on incomplete combustion and unburned fuels. The measurements of these two emissions were made at each of the engine speeds for a given condition, but those data only at 2800 rpm were presented in Fig. 8 for reasons mentioned earlier.

As is shown in Fig. 8, at the same engine load, higher EGR rates led to greater CO emissions, and at higher engine load incomplete combustion resulted in higher CO emissions. At around 50% engine load, however, the emissions began to decrease with dropping engine load until they increased significantly with greater engine loads, with CO emissions increasing sharply at the full engine load. Fig. 8 also shows the effect of intake mixture temperature; when the temperature was increased from 25 to 80℃, CO emissions at 1800 rpm engine speed and 100% engine load increased by 29.5, 27.6 and 30% with regard to the EGR rates of 0, 10 and 20%, while at 25% engine load it did decrease by 41.8, 40.6 and 38%. CO emissions increased by 20, 11.2 and 8.3% with regard to the EGR rates of 0, 10 and 20% when the engine was operated at 2800 rpm and 100% engine load, whereas at 25% engine load, they decreased by 15, 17.8, and 20.5 %. In addition, when intake temperature was increased from 25 to 40℃ at 3800 rpm engine speed and 100% engine load, the EGR rates of 0, 10 and 20% lead to increases in CO emissions of 0.3, -1.8 and 1.5%, but at 25% engine load, they decreased by 9.1, 8.7 and 12.2%. In other words, while at 25% engine load, cooler EGR produced increased CO emissions, at engine loads higher than 50% hotter EGR increased CO emissions.

The reason for this may be that even though lower temperatures in the combustion chamber at lower loads caused greater incomplete combustion, higher temperatures of intake mixture, despite worse intake charging efficiency, helped increase temperatures in the combustion chamber, which led to enhancement in combustion at lower loads, thereby reducing CO emissions.[17]

At higher engine loads, the characteristics of CO emissions, like those of soot emissions, showed similar tendency as affected by EGR rates, regardless of engine speeds. Plee et al.[23] suggested that the CO emissions may be in almost direct proportion to soot emissions, which may mean that the effect of EGR on CO emissions can be correlated in the same fashion as that on soot emissions, and that these two emissions, in addition, tend to indicate a similar reaction kinetic process involved in determining them. They also observed that at EGR rates up to 50%, decreases in intake oxygen concentration and burnt gas temperature may be attributed to increase in CO emissions.

Bae et al.[17] have shown that CO emissions occur largely due to insufficient oxygen concentration at a combustion state, and that though CO emissions increase drastically at the full load in diesel engines, in some cases partial loads did lead to increased CO emissions. In other words, even though the combustion in a diesel engine is in excessive oxygen concentration, its diffusion spray combustion tends to cause local poor mixture, which may result in insufficient oxygen concentration, thereby increasing CO concentration in exhaust emissions.[24]

3.4.4 THC Emissions

To measure THC emissions, the engine speeds of 1800, 2800 and 3800 rpm were applied to the engine, with variations in engine load and intake mixture temperature and EGR rate as parameters. Fig. 9 shows typically the measured values at the engine speed of 2800 rpm. As shown in the figure, at the same engine load, THC emissions increased with higher EGR rates; the increase tended to be greater with hotter EGR, and even greater increase rates were observed in lower loads.

Fig. 9 also includes the effect of intake mixture temperature; when the temperature was increased from 25 to 80℃ at 100% engine load, THC emissions at 1800 rpm increased by 16.3, 15.4 and 16.6% with regard to the EGR rates of 0, 10 and 20%, and at 2800rpm, by 10.5, 21.3 and 21.7%. In addition, when the temperature was increased from 25 to 80℃ at 3800 rpm, THC emissions increased by 0.5, 11.2 and 2% with regard to the EGR rates of 0, 10 and 20%.

According to the result of previous studies,[13 ~ 17] THC emissions tended to increase gently with greater EGR rates. The reason was attributed to the possibility that the increase rate of the emissions was suppressed at a value lower than the concentration increase rate.

THC emissions, substances produced by incomplete combustion due to misfire, quenching, and poor and uneven injection, occurs profusely during cold starting and low load operation.[17] When EGR is applied to diesel engines, it will lower oxygen concentration and increased heat capacity of burnt gasses will lead to lower flame temperatures, thereby increasing THC emissions. This tendency will aggravate in lower engine loads.[17, 24] The reason for this may be that lower loads tend to inject relatively smaller amounts of fuel into unnecessarily greater amounts of air, which will create an area with inflammability dropping sharply below its lower limit in the combustion chamber; higher EGR rates will lower intake oxygen concentration, which will increase local insufficient oxygen concentration, with incompletely burned HC fuel molecules exhausted as THC emissions.[25 ~ 29]

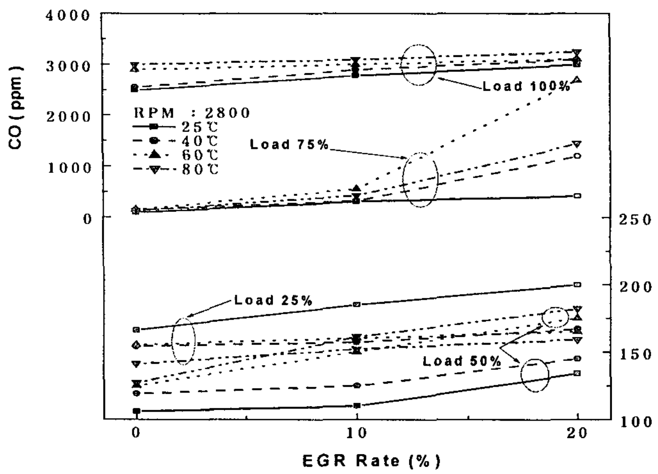


Fig. 8 Effect of EGR rate on CO emissions as parameters of intake mixture temperature and engine load at an engine speed of 2800 rpm

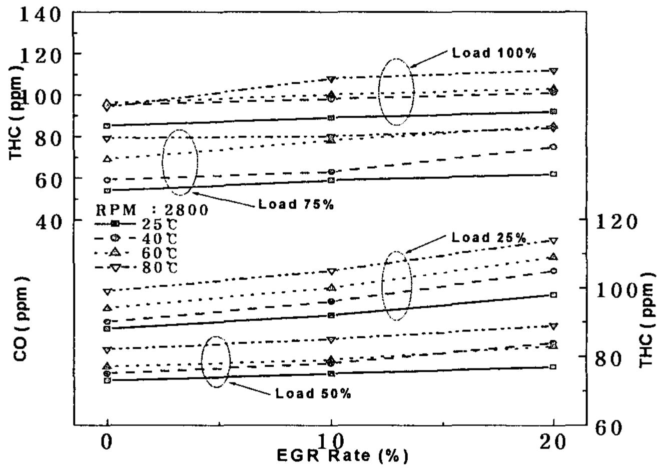


Fig. 9 Effect of EGR rate on THC emissions as parameters of intake mixture temperature and engine load at an engine speed of 2800 rpm

4. CONCLUSIONS

A four-stroke, four-cylinder, water-cooled, swirl chamber type diesel engine was experimentally investigated with engine speeds, engine loads, EGR rates and intake mixture temperatures as a parameter, in order to clarify the effect of water-spray and cooled EGR on the characteristics of combustion and exhaust emissions. The major results obtained are as follows:

1) The soot removal efficiency of the scrubber, though variable depending upon engine speeds and loads, ranged roughly from 30 to 100%.

2) Though variable depending upon heated intake mixture temperatures and EGR rates, the variation in fuel consumption rate was affected roughly within 3% by increased EGR rates, and the variation due to increased temperature of intake mixture was found to be 20.3% at the maximum.

3) In the case of heat release rate, increase in EGR rates and higher temperatures of intake mixture led to a slight increase in ignition delay.

4) NO_x emissions decreased with greater EGR rates, but hotter EGR(greater temperature of intake mixture) increased NO_x emissions.

5) Soot emissions increased with higher EGR rates and hotter EGR.

6) CO emissions increased with greater EGR rates; at 25% engine load, the emissions increased with cooler EGR, and at 50% engine load, the emissions increased with hotter EGR.

7) THC emissions increased with greater EGR rates and hotter EGR, with lower engine loads producing even greater increase rate.

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