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A Study on Engine Performance and Emission Reduction by Ethanol Addition in Compression Ignition Diesel Engine

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ABSTRACT: *Using alcohol fuels instead of fossil fuels is encouraging for alternative fuels. However, the use of compression ignition engines has been limited by its low viscosity and cetane number. In this study, fumigation combustion was performed using a dual fuel supply system that supplies diesel fuel through a compression ignition engine and ethanol through a carburetor. As the ethanol feed rate increased compared to pure diesel fuel, Torque, BMEP and BHP were slightly decreased. As the latent heat of vaporization of ethanol is higher than that of diesel and oxygen is sucked due to the role of ethanol as an oxygenate, the generation of CO, HC, and Smoke is less as the ethanol mixture increases compared with the operation of diesel fuel . Ethanol fuel has the effect of lowering the combustion temperature because it has larger latent heat of vaporization than diesel fuel. Therefore, it was found that the effect of reducing NOx is great.*
Keywords *-Torque, BHP (Brake Horse Power), Brake Thermal Efficiency, Smoke, CO, HC, NOx*

I. INTRODUCTION

The problem of depletion of fossil fuels is a longstanding problem. There are studies that fossil fuels survive, but fossil fuels are still being discovered in unexpected places. More fear than the depletion of fossil fuels is that exhaust gases from fossil fuels are deteriorating the quality of life [1, 2]. Currently, many researches have been conducted to solve the problem of air pollution emitted from automobiles. Hydrogen, natural gas, electric vehicles, hybrid vehicles and so on are being studied extensively. You may wonder if you should re-research alcohol, but it is also a field that is spurring new research. Methanol and ethanol can be produced from sugar cane, fermentation materials, starchy materials, biomass, natural gas, etc., and they are also renewable fuels. Alcohol becomes a fuel that does not matter the exhaustion of fossil fuels. It can also be used as a fuel that can be economically removed from the fossil fuel category [3-5].

An easy way to use alcohol as a fuel for the future was to mix methanol or ethanol with spark ignition engines. In fact, in the United States, gasoline and about 10% alcohol are mixed to be called gasohol and used for exhaust gas reduction [6, 7]. However, compression ignition engines are not easy to mix. Compression-ignition engines should be ignited by compression. Fuel should be used at about 45-55, which is the usual cetane value for ignition. However, the cetane number of alcohol fuels is less than 10. Due to the low cetane number, operation in a compression ignition engine is almost impossible, and viscosity of the fuel is lower than that of diesel oil, which causes a problem of lubrication of the fuel injection pump [8-10].

When the diesel fuel and ethanol fuel are mixed, there is a problem of phase separation phenomenon. Therefore, studies to prevent such phase separation and studies on mixing stability and spray characteristics should be carried out. Particularly, an experiment is performed in which the mixing ratio of ethanol to the water content and the temperature, which are the main parameters for phase separation, are different. As a result, as the mixing ratio of ethanol increases, the surface tension and density of the mixed fuel become lower, which leads to better mixing formation, and the distribution of the Sauter mean diameter (SMD) decreases sharply as the mixing ratio of ethanol increases. Ethanol improves spray atomization [11-13].

It is known that the addition of the anti-segregation agent and ethanol to the diesel fuel can significantly reduce the exhaust gas generated from the diesel engine. Especially, if alcohol fuels can solve exhaust emissions from diesel cars, it is a worthwhile study. However, it is difficult to solve the problem of low cetane number and lubricant even if the engine is mixed with the alcohol fuel and diesel fuel by adding the anti-segregation agent. Therefore, most of the research that can be done is about 10% alcohol fuel mixture [14-16].

Therefore, this study adopts a method of performing combustion by adopting a dual fuel supply system which differs the fuel system rather than the mixture of diesel and alcohol fuel.

In this study, a diesel engine for compression ignition engine was used. Experiments were conducted using a dual fuel supply system in which ethanol was injected into a combustion chamber through a carburetor installed in a diesel engine and the supply of diesel fuel was injected into a combustion chamber by a common fuel injection device.

The aim of this study is to evaluate the performance of engine and the effect of exhaust gas (Smoke, CO, HC, NOx) through diesel fuel and ethanol by dual method.

II. EXPERIMENTAL APPARATUS AND METHOD, AND CHARACTERISTICS OF FUEL

Table 1 shows the engine specifications used in this study. The engine used in this study was a four-cycle single cylinder diesel engine. The cooling method is water-cooled.

The compression ratio is 21, which is a compact engine designed with a high compression ratio that allows self-ignition in compression ignition engines.

Figure 1 shows a schematic diagram of the experimental apparatus. A DC dynamometer capable of motoring was used to control and measure the engine load applied to the test laboratory. A thermocouple was inserted to measure the temperature of the required parts such as the intake / exhaust pipe, cooling water and engine oil.

Table 1.Specifications of engine used.

Items	**Specifications**
Cooling system	Water-Cooled
Displacement	632 cc
Bore × stoke	92 × 95 mm
Compression ratio	21.0
Cylinder number	single
Combustion chamber	Pre-combustion chamber
Fuel injection pump	Bosch A-type
Injection nozzle	Pintle type
Nozzle opening pressure	11,765kPa
Fuel injection timing	16° BTDC static
Ignition timing	MBT (Minimum Spark Advance for the Best Torque)

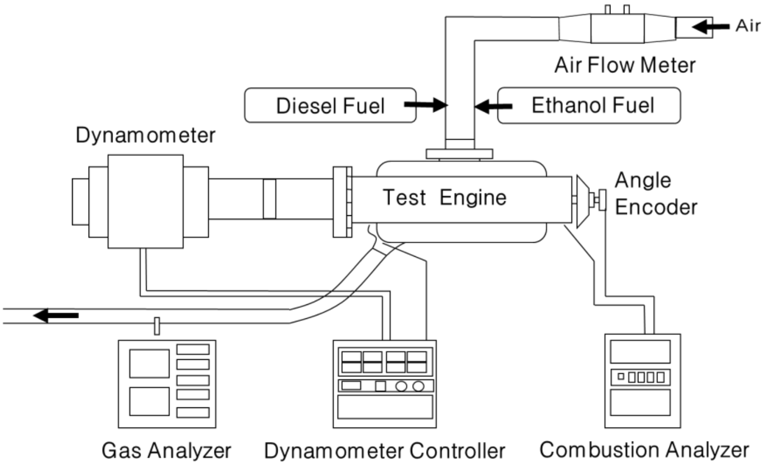

Figure 1. Schematic diagram of experimental apparatus**.**

In order to measure the amount of intake air, the suction system is equipped with an orifice flow meter and a suction surge tank for the safety of the intake air.

Ignition lead angle confirmation device was installed to confirm ignition advance angle. In order to detect the injection timing, a 7cm needle was placed on a nozzle holder, and an ignition coil was connected with a contact breaker through an overflow needle to measure the injection timing.

In order to obtain the indicated diagram, a pressure sensor was used for pressure measurement. The crank angle was measured using an encoder. The pressure obtained was analyzed by taking an average of 100 cycles. The pressure sensor was inserted into the combustion chamber head with a groove of 4 mm in diameter and input to the combustion analysis field via the charge amplifier. From this, it is possible to obtain pressure line, pressure rise rate diagram, pressure-volumetric diagram, heat release rate, mass fraction rate, etc.

The engine speed was performed at 800 rpm, 1000 rpm, 1200 rpm, 1400 rpm, 1600 rpm, 1800 rpm, and 2000 rpm, and the injection timing was constantly performed in MBT (Minimum spark advance for the best torque).Also, smoke, CO, HC and NOx were measured to investigate the effect on exhaust gas.

Table 2 shows the basic characteristics of diesel fuel and ethanol. Ethanol has a structure close to water. When ethanol and diesel are mixed, phase separation occurs, so unleaded gasoline is used to avoid phase separation. However, the fuel supply of this study is not a mixture of ethanol and diesel, but adopts fumigation method to supply separately.

The fumigation method, which belongs to the compound fuel system, is expected to be effective even in fuel such as ethanol which has low ignitability. In this experiment, a fumigation method was introduced to improve the usability of air by using carburetor. A carburetor was attached at a distance of 10 cm from the engine combustion chamber. In the fuel supply method, ethanol is supplied through a separate fuel line through carburetor, and the conventional fuel injection device supplies only diesel fuel.

The mixing ratio of diesel fuel to ethanol was measured by using a fuel flow meter. The ethanol mixing ratio was 10%, 20%, 30%, 40% and 50%. The notation is expressed as EF10, EF20, EF30, EF40, and EF50, respectively.

Table 2. Comparison of diesel and ethanol components.

	Diesel	**Ethanol**
Formula	$C_{16}H_{34}$	C_2H_5OH
Specific gravity	0.82-0.85	0.79
Lower heating value (MJ/kg)	42,600	26,808
Cetane number	45-60	8
Boiling point (℃)	210-325	78.4
Viscosity (cSt) at 25℃	2.79	1.1
Latent heat of evaporation (MJ/kg)	310	863
Theoretical air-fuel ratio	14.6	9.0

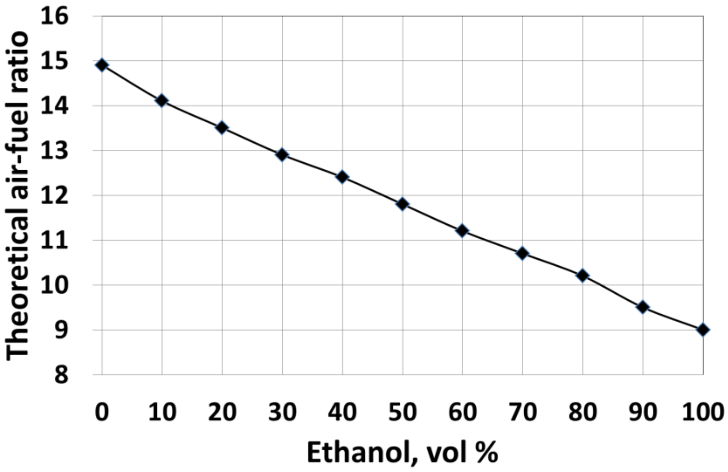

Figure 2. Theoretical air-fuel ratio versus ethanol volume %.

The dual fuel supply system was operated at the same air fuel ratio by adjusting the needle valve of the vaporizer and the fuel injector rack of the existing engine to operate under the same conditions as the method of mixing ethanol with the diesel fuel. Experiments were also carried out with a separate rack control system to control the amount of fuel injected to the rack adjustment side.

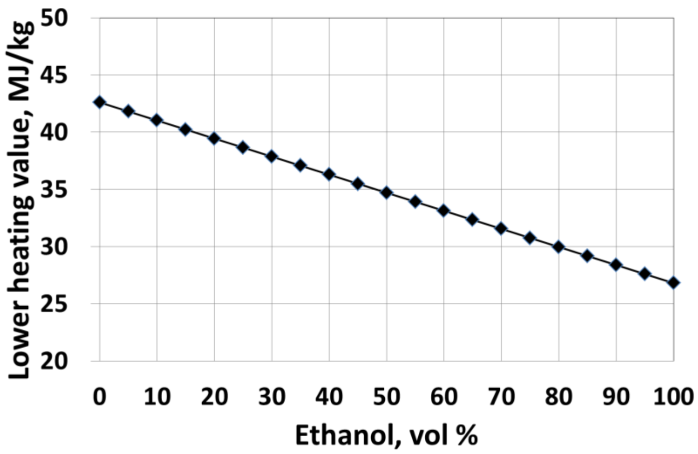

Figure 3. Lower heating value versus ethanol volume %.

The operation conditions of the vaporizer were fixed with the throttle valve of the vaporizer fully open and the flow rate of ethanol was controlled by the needle valve.

Experiments were carried out while keeping the atomized flow rate of the diesel fuel at an excess air ratio of 1.4.

The latent heat of vaporization of ethanol is about 2.8 times higher per unit mass than diesel fuel, so the evaporative cooling effect is quite large. This increases the evaporative cooling effect by increasing the amount of ethanol. When the amount of ethanol is increased, liquid ethanol which is not vaporized flows into the cylinder through the intake pipe, which causes engine knock and combustion fluctuation, which is the limit of operation.Figure 2 is the result of air fuel ratio when ethanol vol% is increased. The air fuel ratio is 14.6 for diesel and 9 for ethanol. Although the fuel consumption rate will increase due to low low calorific value of ethanol and low air fuel cost, it would not be economically problematic if a small amount of ethanol is mixed.
Figure 3 shows the results of the lower calorific value when ethanol was added in vol%. In the figure, the low calorific value (MJ / kg) of diesel is 46,600, while that of ethanol is 26,808, which is only 63% of diesel.
Cetane number is 45-60 diesel while ethanol is 8. Low cetane number is considered to be a negative factor in viscosity, lubrication, and injection pressure to operate compression ignition engine.
The latent heat of vaporization (MJ / kg) has a vaporization latency of about 2.8 times that of diesel (310) while that of ethanol is 863, which is considered to be a positive factor by the action of NOx reduction and smoke reduction.

III. RESULTS AND DISCUSSION

Figure 4 is the experimental result showing the torque with respect to the engine speed change. The fuel used is pure diesel fuel, 10% ethanol (EF10), 20% (EF20), 30% (EF30), 40% (EF40) and 50% (EF50). The magnitude of the torque depends on the magnitude of the low calorific value as shown in Table 2 and Fig. 3. That is, the 100% of the diesel fuel has the highest value for the engine speed variation, and the ethanol mixing ratio is 10% (EF10), 20% (EF20), 30% (EF30) 50% (EF50).
However, as the amount of ethanol is increased, the torque is not so much lower than expected, because the latent heat of vaporization of ethanol is so large that the effect of lowering the air temperature is large. As a result, the density of the air becomes larger and the influence on the torque becomes larger.

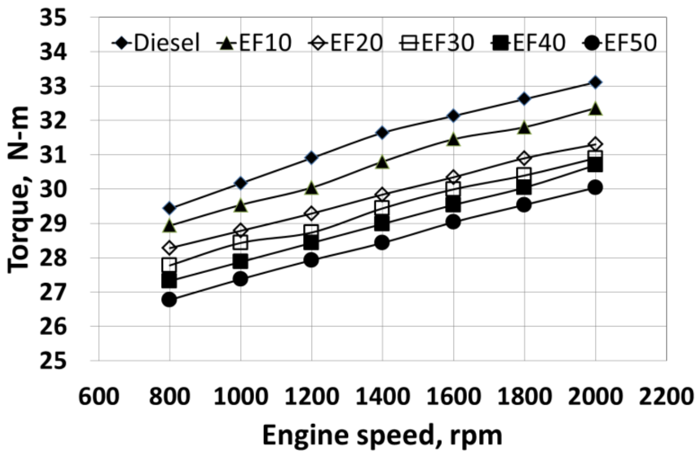

Figure 4. Torque versus engine speed for various fuels.

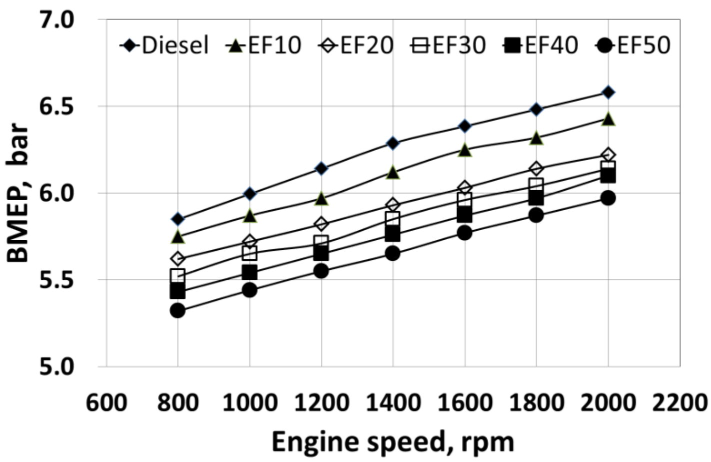

Figure 5. Brake mean effective pressure versus engine speed for various fuels.

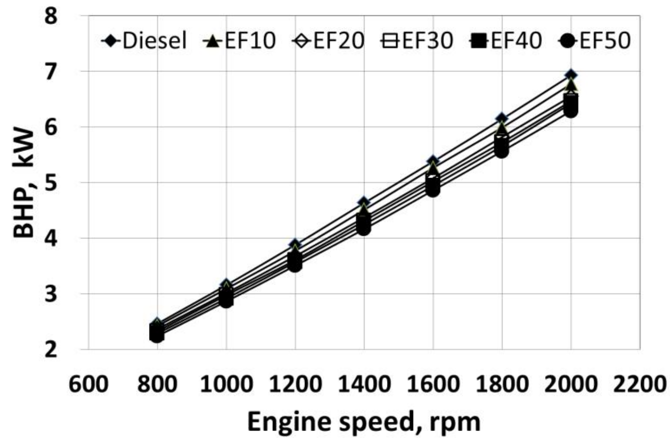

Figure 6. Brake horse power versus engine speed for various fuels.

Figure 5 shows BMEP (brake mean effective pressure). The average effective pressure is proportional to the torque when the displacement is constant. Figure 5 shows the same trend as the torque curve in Figure 4. Figure 6 shows the experimental results of engine power versus fuel change for engine speed variation. Since the engine output is affected by the torque and the engine speed, the engine output is rapidly increasing with respect to the engine speed increase. Pure diesel fuel has the highest BHP among diesel, ethanol 10% (EF10), 20% (EF20), 30% (EF30), 40% (EF40) and 50%. However, the difference is not very large. The BHP is affected by the value of torque and engine speed. However, changing the value of the engine speed relative to the torque value may have the greatest effect on the BHP. Compared with the torque test results shown in Figure 4, the influence of the torque value is large. However, since the latent heat of vaporization of ethanol is large, the effect of lowering the temperature of the air is large. As the ethanol growth rate increases, the difference in BHP is less than that of pure diesel fuel.

Figure 7 shows the results of the experiment showing the brake specific fuel consumption (BSFC) for the engine speed change. It is common that the fuel consumption rate is proportional to the amount of fuel supplied and is in general inversely proportional to the engine output, and the tendency of the curve should also exhibit a tendency to be inversely proportional to the torque curve. However, the amount of fuel supplied is different from the order of EF50, EF40, EF30, EF20, EF10, and diesel, which is the opposite of the order of the torque curves, diesel, EF10, EF20, EF30, EF40, and EF50. This is thought to be the result of experimentation caused by the difference in calorific value even if the amount of supplied fuel is kept constant.

Figure 8 shows the experimental results with respect to the fuel composition for the thermal efficiency versus engine speed variation. The thermal efficiency has an inverse relationship with the brake specific fuel consumption rate and the low calorific value. It is seen that the thermal efficiency is relatively high near the engine speed of 1600 rpm to 1800 rpm. Figure 9 shows the results of smoke test for changes in engine speed. Smoke is generated when the compression ignition engine is in a state where the excess air ratio is not homogeneous or when the fuel injected from the injection pump is not uniformly mixed with air. Even if air is left unevenly due to such uneven combustion, smoke is generated.

In particular, the supplied fuel, diesel (C16H34), produces dehydrogenation in the presence of hot oxygen depletion and produces smoke if the liberated carbon is released. As shown in Figure 5, as the engine speed increases, BMEP increases, indicating that the smoke concentration increases. In addition, the latent heat of vaporization of ethanol (C2H5OH) is 2.8 times higher than that of diesel, so that as the hydrogen atom in the fuel molecules weakens the binding force with oxygen and the mixing ratio of ethanol increases, the amount of unburned carbon decreases. Figure 10 shows the experimental results of the CO emissions for increasing engine speed. CO generation is increasing with increasing engine speed, and CO is decreasing with increasing ethanol addition. However, the difference in the rate of increase of ethanol addition is not significant. In general, CO is incomplete combustion due to insufficient mixing.

Compressed ignition engines, however, generate CO less than spark ignition engines that use gasoline because the excess air ratio can be operated at relatively high operating ranges with spark ignition engines. As shown in the figure, as the proportion of ethanol is increased, the oxygen is sucked due to the role of ethanol as the oxygenate, so that as the ethanol mixture increases, the generation of CO becomes less as compared to the operation of the diesel fuel. Figure 11 shows the experimental results showing the HC emissions to the engine speed change. Generally, HC refers to unburned hydrocarbons generated from incomplete combustion. HC emissions increase in rich areas, and emissions increase in lean areas where high emissions and combustion are unstable. As shown in the figure, HC emissions increase with increasing engine speed. However, it can be seen that as the mixing ratio of ethanol increases, the HC emission amount is sharply reduced. Hydrocarbons such as gasoline and diesel fuel are already paraffinic and contain HC. However, since ethanol has a structure containing oxygen, it is considered that HC is less generated than diesel fuel [17, 18].

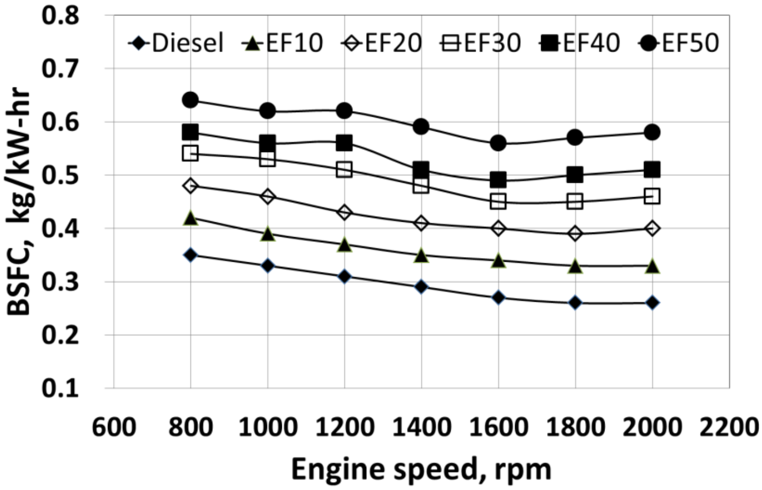

Figure 7. Brake specific fuel consumption versus engine speed for various fuels.

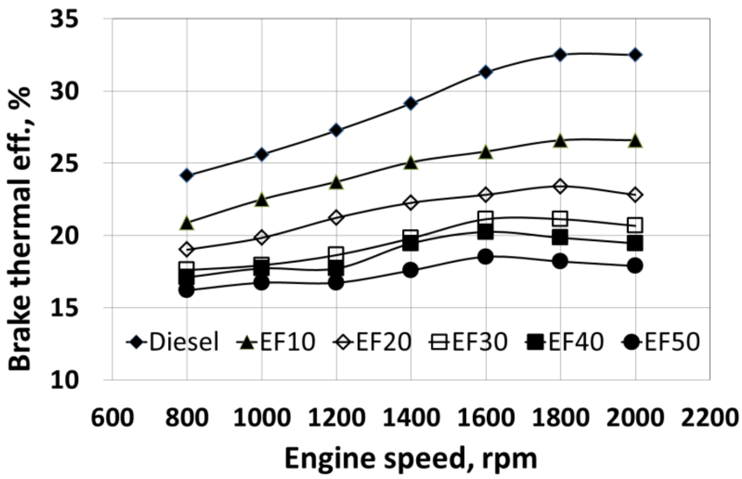

Figure 8. Brake thermal efficiency versus engine speed for various fuels.

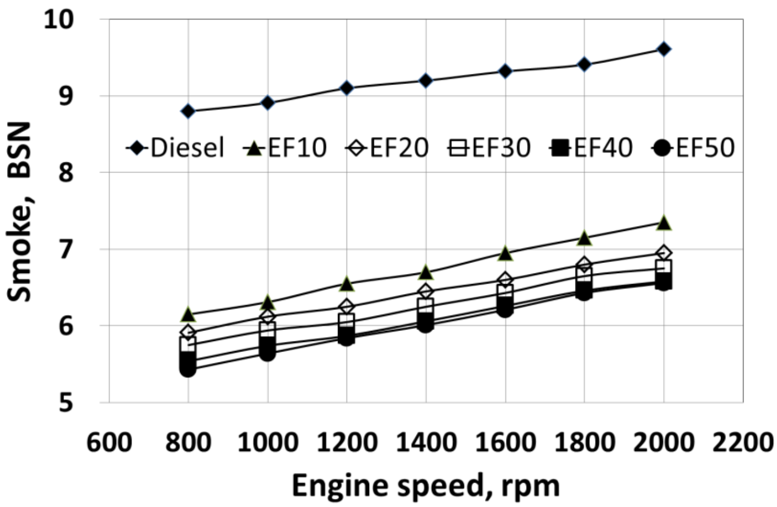

Figure 9. Smoke versus engine speed for various fuels.

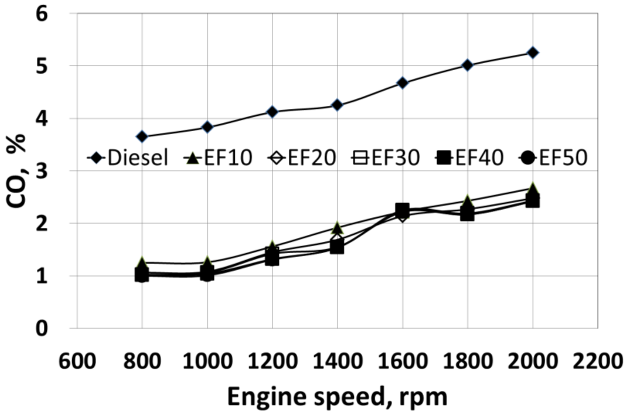

Figure 10. Carbon monoxide versus engine speed for various fuels.

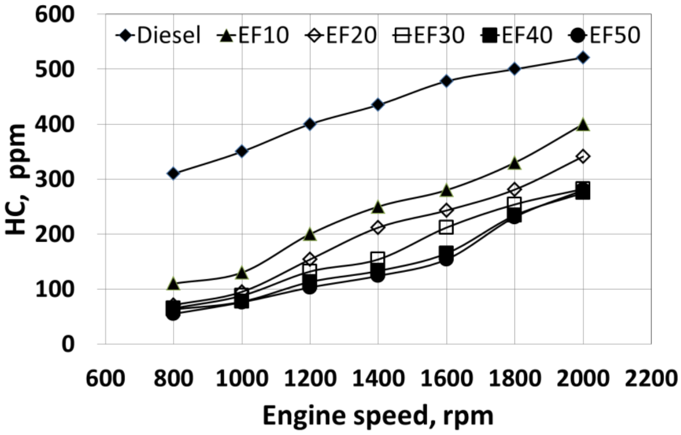

Figure 11.Hydrocarbon versus engine speed for various fuels.

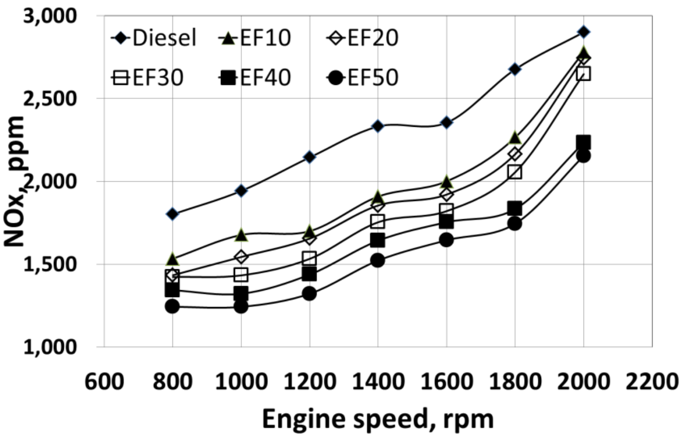

Figure 12. Nitrogen oxide versus engine speed for various fuels.

Especially, when the ethanol mixing ratio is increased with EF10, EF20, EF30, EF40 and EF50, the HC generation rate is also reduced. However, it can be seen that the difference between EF40 and EF50 in the operation area is almost the same as the number of engine revolutions increases. Figure 12 shows the results of experiments showing NOx emissions with engine speed variation. Generally, it is known that the higher the combustion temperature, the higher the generation of NOx, and the lower the combustion temperature, the less the generation of NOx. Ethanol fuel has the effect of lowering the combustion temperature because it has larger latent heat of vaporization than diesel fuel. Therefore, it is considered that the effect of reduction of NOx is great because the diesel combustion temperature becomes low when ethanol is supplied with diesel fuel. As with CO and HC, NOx generation is reduced as the ethanol mixing ratio is higher than that of diesel fuel.

IV. CONCLUSION

Fumigation combustion was performed using a dual fuel supply system that supplies diesel fuel through a compression ignition engine and ethanol through a carburetor. The following results were obtained.

1) The fuel used is pure diesel fuel, ethanol feed rate of 10% (EF10), 20% (EF20), 30% (EF30), 40% (EF40) and 50% (EF50).Torque, and BMEP are dependent on the magnitude of the low calorific value. That is, 100% of the diesel fuel has the highest value for the engine speed change, and the ethanol feed rate is 10% (EF10), 20% (EF20), 30% (EF30) 50% (EF50).
2) Despite the low calorific value of ethanol, the large latent heat of vaporization of ethanol has a great effect of lowering the temperature of the air, which causes the density of the air to increase and the influence on the BHP becomes large. It can be seen that BHP is not so small compared to pure diesel fuel as the ethanol increase rate increases.
3) BSFCs tend to be proportional to the amount of fuel supplied and inversely proportional to engine power. However, it was found that the amount of supplied fuel increases as the amount of ethanol supplied increases due to the difference in calorific value of the mixed fuel.
4) Because the latent heat of vaporization of ethanol is 2.8 times higher than that of diesel, the hydrogen atoms in the fuel molecules weaken the binding force with oxygen, and as the mixing ratio of ethanol increases, the amount of unburned carbon decreases and smoke is gradually decreased as compared with operation of pure diesel fuel.
5) As the ratio of ethanol addition increased, the oxygen uptake was sufficient due to the role of ethanol as the oxygenate. As the ethanol mixture increased, the generation of CO was decreased as compared with the operation of diesel fuel.
6) HC emissions increase with increasing engine speed. However, it can be seen that as the mixing ratio of ethanol increases, the HC emission amount is sharply reduced. Hydrocarbons such as gasoline and diesel fuel are already paraffinic and contain HC. However, ethanol has a structure that already contains oxygen, so that HC generation is less than diesel fuel.
7) Ethanol fuel has a higher latent heat of vaporization than diesel fuel, so it has an effect of lowering combustion temperature. Therefore, it can be concluded that the effect of reduction of NOx is increased because the diesel combustion temperature is lowered when ethanol is supplied with diesel fuel.

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