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3Bio-Based Additives to Improve Diesel Engine Performance and Emissions

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Keywords：Diesel engine, exhaust emissions, ethanol blends, diethyl ether.

ABSTRACT

This study investigates addition of ethanol and diethyl ether into diesel fuel. Base fuel and three different blends were used in the tests. The experiments were performed at engine speed of 1500 rpm and under various loads by single cylinder diesel engine. Injection, combustion, performance and emissions parameters were evaluated for base fuel and blends. Results show that injection pressure decreases for all blends, and DEE addition causes delay in injection timing. Burnt mass fraction is lower for ethanol and DEE blends during early combustion phase, while it increases with DEE blends during progressive combustion phase. Ethanol and DEE blends also give reductions in cylinder pressure and temperature. Generally, air excess coefficients increase but exhaust gas temperature decreases when using ethanol and DEE blends. Although ethanol blend causes decrements in engine power and torque, DEE blends give closer values to diesel fuel. Both ethanol and DEE blends yield improvement in brake thermal efficiency and reduction in brake specific fuel consumption. When using fuel blends, generally, decrements in NO_x and CO_2 emissions and increments in HC emission occur, while CO emission shows fluctuations depending on engine load.

INTRODUCTION

Diesel engines are widely used in several areas such as construction, transportation, and agricultural due to they are more efficient than gasoline engine and diesel fuel is cheaper than gasoline (Koc and Abdullah, 2013). However, continuously fluctuating petroleum prices and the problems about environmental pollution have forced the researchers to develop alternative fuels which ensure the future energy security and reduction harmful emissions (Ajav et al, 1999). Biodiesel and alcohol especially ethanol are promising among the various alternatives, as they are produced from renewable bio-sources which are available locally (Agarwal, 2007; Rakopoulos et al, 2008). Biodiesel is one of the oxygen containing and sulfur-free alternatives to petroleum-based diesel fuel, which are the fatty acids (or triglycerides) obtained from various plain vegetable oils (Kala and Masjuki, 2002; Raheman and Phadatare, 2004; Banapurmath et al, 2008) or the recycled waste oil (Najafi et al, 2007; Rao et al, 2008) and animal fat (Reyes and Sepulveda, 2006; Cunha et al, 2008). Biodiesels can be used as pure or as additives in diesel engines because of their high cetane number and for having calorific value very close to diesel fuel (Srivastava and Prasad, 2000). However, it has been reported that, they leaded to operational problems, such as engine deposits under long-term use because of the high viscosity and low volatility, besides the reduction in engine power and efficiency (Kumar et al, 2003; Demirbas and Karslioglu, 2007). Alcohol and its derivative ethers are the other alternatives as oxygenated fuel or fuel additives for diesel engines. The addition of oxygenates into conventional fuel is one of the effective methods to decrease the consumption of limited petroleum resources and hazardous engine emissions, by little or no modification in engine configuration (Ren et al, 2008). Ethanol and its derivation diethyl ether (DEE) are good choices as oxygenates, since they are produced from renewable biomass. Therefore, the effects of ethanol on diesel engine's performance and emissions have been investigated in numerous studies (Ajav et al, 1999; Hansen et al, 2005; Rakopoulos et al, 2008; Park et al, 2010). However, the works are limited about DEE as an additive alone (Sachuthananthan and Jeyachandran, 2007; Rakopoulos et al, 2012; Swaminathan and Sarangan, 2012; Sivalakshmi and Balusamy, 2013) or with ethanol (Qi et al, 2011; Nagdeote and Deshmukh, 2012; Iranmanesh, 2013), since the interest on DEE has recently risen. DEE is produced from ethanol by dehydration process, so it can be considered as a

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renewable fuel. DEE is liquid under ambient conditions, which makes it attractive for fuel storage and handling. DEE also has several favorable properties such as, exceptional cetane number, reasonable energy density, high oxygen content, low autoignition temperature and high volatility. Therefore, it can assist to improve the engine performance and reduce the cold starting problem and emissions when using as an additive into diesel fuel (Sezer, 2011). Ethanol affects the self-ignition characteristics of diesel fuel negatively because of having very low cetane index (Hansen et al, 2005). Therefore, DEE is considered as a cetane improver for this study and a critical blending ratio (15%) for ethanol is preferred because several researches have reported that, up to 10% ethanol no noticeable differences occur in the engine performance, compared to diesel fuel (Hansen et al, 2005). Thus, baseline fuel and three different fuel blends containing 15% ethanol, 15% ethanol and 2% DEE and 15% ethanol and 4% DEE have been used in this study.

MATERIALS AND METHODS

Test Fuels

The commercial diesel fuel used in the tests was obtained locally. Anhydrous ethanol with 99.7% purity and JT Baker pure grade DEE with 99.5% purity were used in the tests. The ethanol is considered an oxygenated additive and DEE can compensate for the reduction in the cetane number sourced from the ethanol addition. Moreover, DEE is a good oxygenate to increase the oxygen content of ethanol-diesel blends. Table 1 tabulates the properties of the diesel, ethanol and DEE.

Table 1. The properties of the fuels

Property	Diesel	Ethanol	DEE
Chemical Formula	C_xH_y	C_2H_6O	$C_4H_{10}O$
Molecular weight	190–220	46.04	74.12
Density of liquid (kg/L) @ NTP*	~0.83	0.789	0. 713
Viscosity (cP) @ 40°C	2.6–4.1	1.19	0.23
Bulk modulus (GPa) @ 20°C	1.39	1.32	0.69
Surface tension (mN/m) @ 20°C	25.2	22.3	17
Oxygen content (wt %)	–	34.7	21
Sulfur content (ppm)	~250	–	–
Boiling temperature (°C)	180–360	78.4	34.6
Auto ignition temperature in air (°C)	315	235	160
Flammability limit in air (vol %)	0.6–6.5	3.3–19	1.9–9.5
Stoichiometric air-fuel ratio (AFR_s)	14.6	9	11.1
Heat of vaporization (kJ/kg) @ NTP*	250	825	356
Lower heating value (MJ/kg)	42.5	26.8	33.9
Cetane number (CN)	40–55	5–8	>125

* NTP: Normal temperature and pressure

The ethanol blend was prepared with the addition of 15% ethanol to the diesel fuel. DEE blends were prepared by mixing of 2% and 4% DEE to the ethanol blend on a volume basis. Thus, baseline fuel and three different blends, namely D, E15, E15DEE2 and E15DEE4, were used in this study. The blends were prepared just before starting the experiments to obtain homogenous mixture and prevent the phase separation. The density, stoichiometric air-fuel ratio and lower heating values of the blends required for the test set-up software are determined from the following equations.

$$\rho_{\mathrm{blend}} = \sum y_{\mathrm{i}} \rho_{\mathrm{i}} \tag{1}$$

$$AFR_{\mathrm{s,blend}} = \frac{\sum y_{\mathrm{i}} \rho_{\mathrm{i}} AFR_{\mathrm{s,i}}}{\sum y_{\mathrm{i}} \rho_{\mathrm{i}}} \tag{2}$$

$$LHV_{\mathrm{blend}} = \frac{\sum y_{\mathrm{i}} \rho_{\mathrm{i}} LHV_{\mathrm{i}}}{\sum y_{\mathrm{i}} \rho_{\mathrm{i}}} \tag{3}$$

The subscript i refers to the diesel fuel, ethanol or DEE, and y_i is the volume ratio of the every fuel in the blend.

Test Setup and Procedure

The full schematic layout of the test bench with the instruments and data logging systems is shown in Fig. 1. The experimental set-up consists of three main parts, namely dynamometer, engine and control panel. An eddy current dynamometer is connected to the engine and it is integrated with a data acquisition system to store the measured data. The set-up has the air box, fuel tank, U manometer, fuel measuring unit (burette), transmitters for the air and fuel flow measurements.

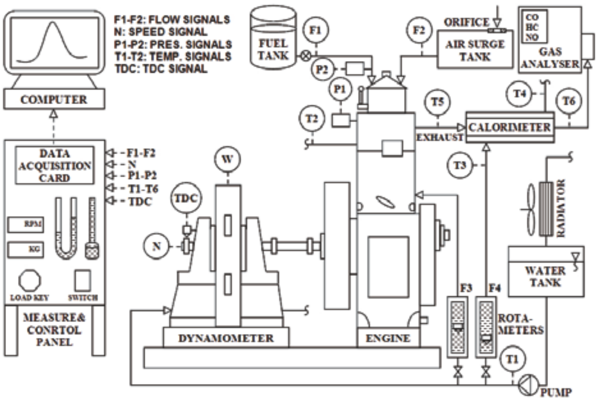


Fig. 1. The schematic layout of the test setup

A big water tank and an electrically-driven pump assure the coolant water circulation. The radiator cools the returning hot cooling water by means of an electrically-driven air-fan. The rotameters are fitted to the set-up for the engine and calorimeter cooling water flow measurements. Thermocouples are located at the related points for the measurement of the exhaust gas and cooling water temperatures. An exhaust gas analyser (CAPALEC-CAB) is employed to measure the NO_x, HC, CO, and CO_2 emissions on the line. The Labview based namely the EngineSoft software is used on the test set-up for the engine performance and combustion analysis. The four-strokes, water-cooled, naturally-aspirated, direct injection diesel engine is placed on the test bench, and its main parameters are given in Table 2.

Table 2. The specifications of the test engine

Make & Model	Kirloskar TV1 engine
Number of cylinder	1
Bore	87.5 mm
Stroke	110 mm
Swept volume	661 cc
Compression ratio	17:1
Rated power	3.5 kW @ 1500 rpm
Speed	1500 rpm, constant
Dynamometer	Eddy current, water cooled
Injection timing	23° before TDC
Injection pressure	210 bar

The engine has the ability to operate on the spark ignition (SI) or the compression ignition (CI) engine by changing the cylinder head, and the compression ratio is varied via the tilting cylinder block. The cylinder pressure is measured by the piezoelectric pressure transducer fitted on the engine cylinder head and a crank angle encoder fitted on the flywheel. The fuel injection pressure is also recorded with a pressure transducer on the injector fuel line. The tests are conducted at the rated speed of 1500 rpm at different loads equals to 0 kg, 3 kg, 6 kg, 9 kg and 12kg in the load cell readings. The load of 12 kg is equivalent to the full load on the engine, so the engine was loaded between 0% and %100 with %25 increments. The measurements are recorded after the engine reaches at a stable regime for each engine load. The injection, combustion and performance characteristics of the engine are gained from the stored data. The engine emission parameters such as the carbon monoxide (CO), carbon dioxide (CO_2), hydrocarbon (HC), nitric oxides (NO_x) are also measured in all the tests.

Uncertainty Analysis

The uncertainties in the experiments can arise from the instruments, condition, calibration, environment, observation, reading and test planning. Therefore, an uncertainties analysis is required to prove the accuracy of the experiments. An uncertainty analysis is performed using the method described by Holman (Holman, 2001). According to this method, if the result R is a function of the independent variables x_1, x_2,…,x_n, it can be expressed as follow.

$$R = R(x_1,\ x_2,\cdots,x_n) \tag{4}$$

$$U_R = \sqrt{U_{R,1}^2 + U_{R,2}^2 + \cdots + U_{R,n}^2} = \sqrt{\sum_{i=1}^{n} U_{R,i}^2} \tag{5}$$

$$U_{R,i} = \left|\frac{\partial R}{\partial x_i}\right| U_i \qquad (1 \le i \le \mathrm{n}) \tag{6}$$

This approximation is also called as partial uncertainty in the result because of the dependence on a measured quantity x_i and its uncertainty U_i. The percentage of uncertainty for various parameters, i.e. speed (N), brake power (BP), brake specific fuel consumption (BSFC), brake thermal efficiency (BTE), exhaust gas temperature (EGT), pressure pickup (PP) and emission measurements were calculated using the percentage uncertainties of instruments. Accordingly, the total percentages of uncertainty (TPU) in the experiments are determined as 2,56 %.

RESULTS AND DISCUSSIONS

The injection pressure of the base fuel and blends is given in Fig. 2 for the specified condition in the figure. As it is seen in the figure, the injection pressure is lower for the ethanol and DEE blends. It is considered that this variation is sourced from the physical properties of the fuels. The fuel properties, like density, viscosity, bulk modulus and surface tension have significant effects on the start of injection and the other injection characteristics. As shown in Table 1, the values of the density, viscosity, bulk modulus and surface tension are lower for the ethanol and DEE than those of the diesel fuel. These differences cause to the decrements in the injection pressure and retardation in the injection timing when using the ethanol and DEE containing blends. For example, a reduction in viscosity and bulk modulus of elasticity lead to the lower injection pressure and delay in the injection timing (Kegl, 2006).

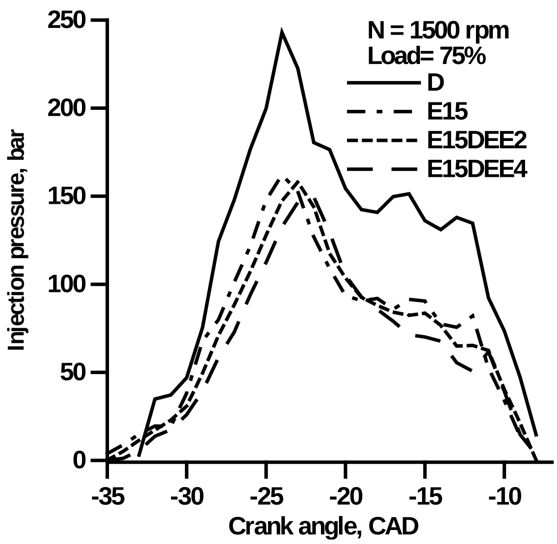
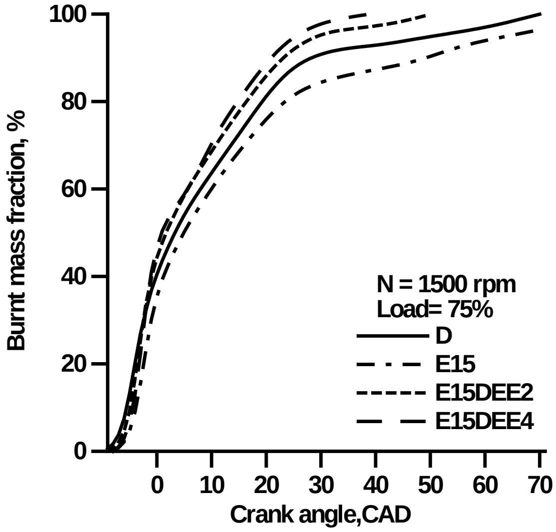


Fig. 2. The injection pressure for the tested fuels

Fig. 3 shows the variation of the burnt mass fraction and the heat release rate for the tested fuels.

In Fig. 3(a), the burnt mass fraction is lower for the ethanol and DEE blends compared to the diesel fuel during the onset of the combustion phase. It is considered that, this is sourced from the extended auto-ignition period for the ethanol blend, because ethanol has an extremely lower cetane number as seen in Table 1. Additionally, although the DEE improves the cetane number of the blends, the late injection of the DEE blends observed in Fig. 2 causes a reduction in the burnt mass fraction in the early combustion period. The burnt mass fraction is higher with the DEE blends, while it is still lower with the ethanol blend during the advancing of the combustion period. The variations can be attributed to the fuel properties such as the oxygen content and cetane number. The existing oxygen in the chemical structure of the ethanol and DEE assists to the rapid completion of the combustion by giving a homogenous mixture in the combustion chamber.



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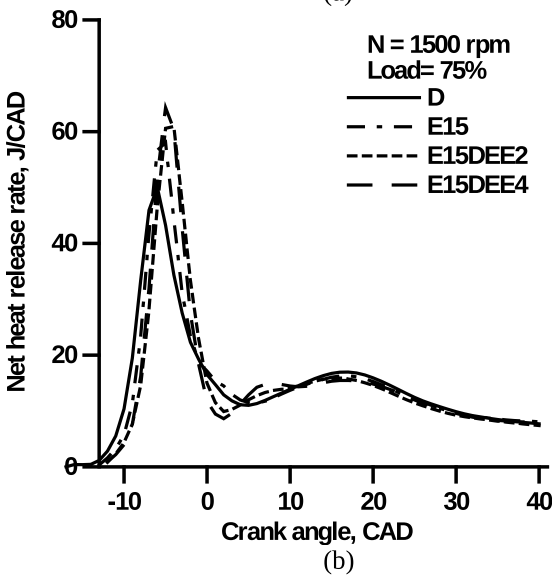
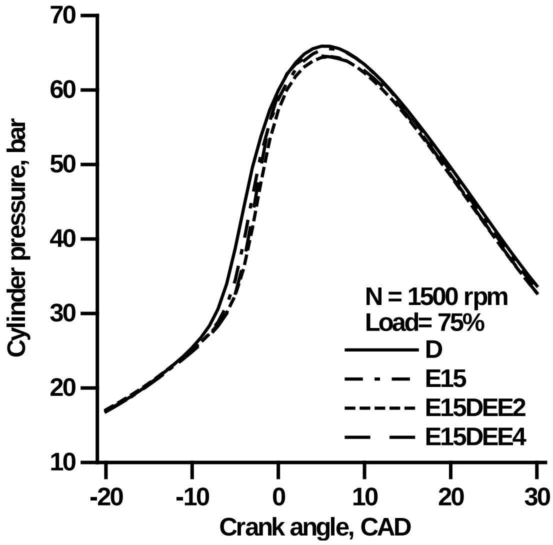


(b)

Fig. 3. Burnt mass fraction and heat release rate for the tested fuels

The lowest cetane number of ethanol among tested fuels as seen in Table 1, also affects the increment of the burnt mass fraction negatively. Heat release rate in Fig. 3(b) is lower for blended fuels during the early stage of combustion, while it increases for ethanol and DEE blends with progressing of the combustion. The variations in heat release rate can be attributed to properties of fuel such as the cetane number, oxygen content and heating value. The late injection of ethanol and DEE blends for reasons cited above causes a delay in the start of the combustion, so lower heat release rates are obtained with blends during early combustion phase. Low cetane index of ethanol is also another reason for the late combustion start and the lower heat release rate. However, the oxygen content of the ethanol and DEE increases the heat release rate by improving the combustion in the later stage of the combustion. The ethanol blend has lower heat release values than DEE blends during this phase due to the lowest heating value among the fuels.



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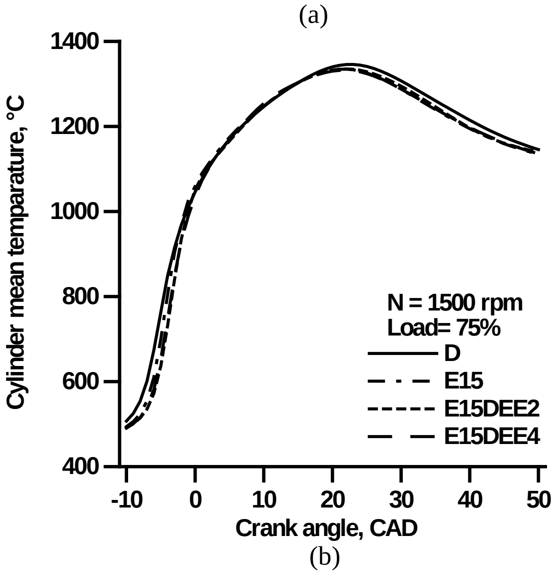
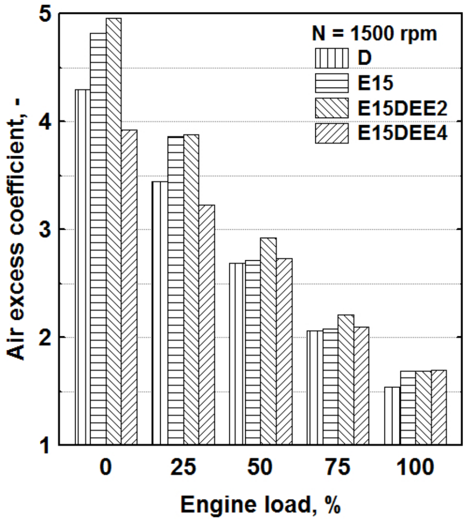


(b)

Fig. 4. Cylinder pressure and cylinder mean temperature for the tested fuels

Fig. 4 (a) and (b) illustrate the variation of the cylinder pressure and cylinder mean temperature for the tested fuels, respectively. As it can be seen in the figures, the ethanol and DEE blends have given a little lower pressure and fewer temperature values. It is believed that the main factor of the reductions in the pressure and temperature is the heating values of the tested fuels. The calorific values of the ethanol and DEE are considerably lower than that of the diesel fuel, as seen in Table 1. However, the blended fuels achieve to obtain the pressures and temperatures closer to the diesel fuel by providing the improvement in the combustion efficiency because of the oxygen in the chemical structures.



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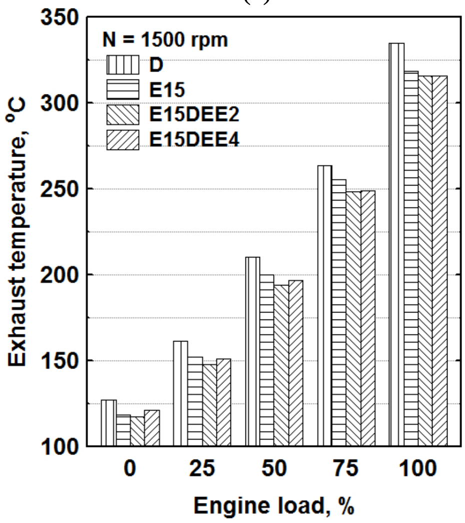
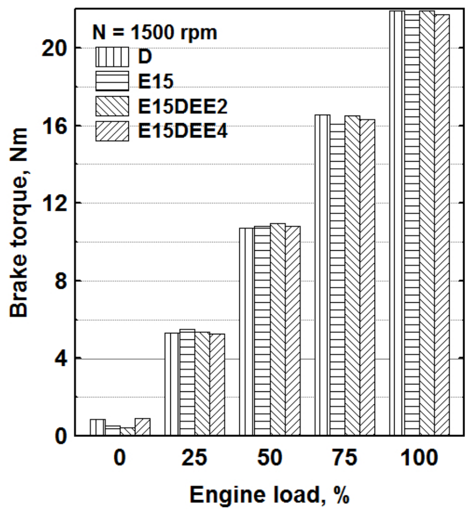


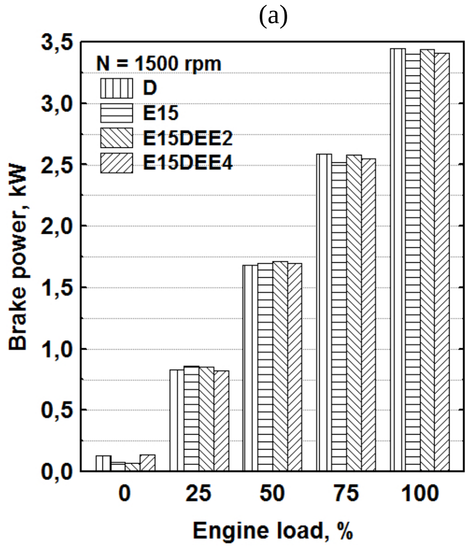
(b)

Fig. 5. Air excess coefficient and exhaust gas temperature for the tested fuels

The air excess coefficient (AEC) in Fig. 5(a) generally has an increment trend for the ethanol and DEE blends compared to the diesel fuel. The lower stoichiometric air-fuel ratios of the ethanol and DEE, as tabulated in Table 1, cause to increments in AEC by delivering more air for the per unit fuel mass when using the blended fuels. Oxygenated fuels also give an additional contribution to the increment of the AEC by providing extra oxygen in the combustion chamber. These are also called as the leaning effects which increase the AEC by leaning the air-fuel mixture in the cylinder. Maximum increments in the AEC obtained with E15, E15DEE2 and E15DEE4 are 9.3%, 9.7% and 10.4% at full load condition, respectively. Exhaust gas temperature (EGT) in Fig 5(b) is lower with the blended fuels for all engine loads. Lower EGT's are related to the lower cylinder (combustion) temperatures, as seen in Fig. 4(b), and they result from the lower heating values of the ethanol and DEE. The higher heat of the vaporization of ethanol and DEE, as given in Table 1, is the other reason for both the lower combustion and EGT's, because these fuels cause a reduction in cylinder temperature by absorbing the higher heat during vaporization. The most reductions in EGT obtained with E15, E15DEE2 and E15DEE4 are 4.8%, 5.6% and 5.5% at full load condition, respectively.



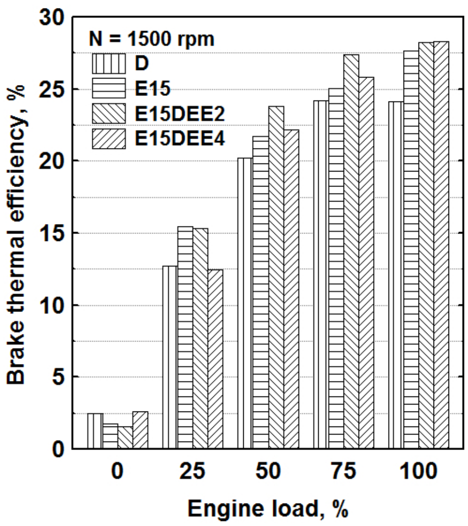
(a)



(b)

Fig. 6. Brake torque and power for the tested fuels

As it is seen in the figure, the brake torque in Fig. 6(a) and brake power in Fig. 6(b) have similar variations due to the engine operating at a constant speed of 1500 rpm. Both the torque and power increase with the increasing engine load, and the ethanol and DEE blends have given a little lower torque and power values than those of the diesel fuel especially at the high engine loads. The reductions in the torque and power when using the blended fuels can be attributed to the lower heating values of the oxygenated fuels as cited above.



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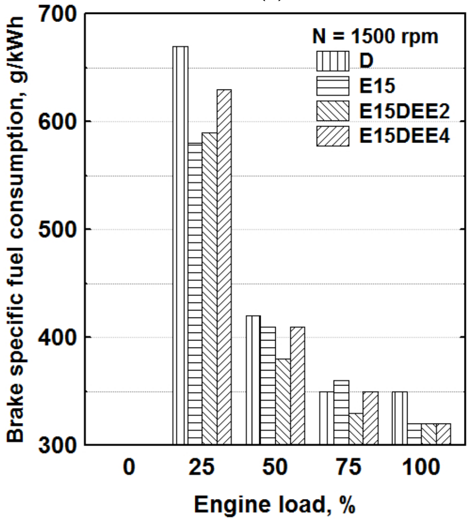
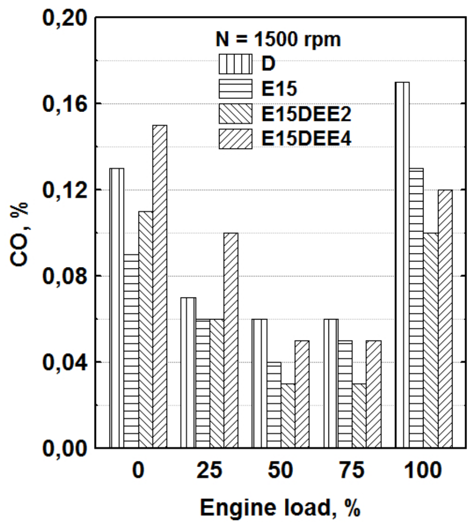


(b)

Fig. 7. Brake thermal efficiency and specific fuel consumption for the tested fuels

Fig. 7(a) shows the variations of the brake thermal efficiency (BTE) for the tested fuels. The BTE increases with the engine load, and the ethanol and DEE blends generally have higher BTE values than those of the diesel fuel. The increments in the BTE are the result of the improvements in the combustion via the oxygenated fuels as mentioned before. The addition of the DEE also slightly advances the BTE especially at high engine loads, compared to the ethanol blends. The E15DEE2 blend usually serves the best values among the blends, as seen in the figure. The maximum increments in the BTE obtained with E15, E15DEE2 and E15DEE4 are about 14.7%, 17.1% and 17.2% at the full load condition, respectively. On the other hand, the BSFC in Fig. 7(b) exhibits an opposite characteristic compared to the BTE namely, the BSFC decreases with the engine load and the blended fuels generally present the lower BSFC values than those of the diesel fuel. Thus, the blended fuels have given the lower BSFC values in general and the minimum BSFC is obtained with E15DEE2 at medium and high load conditions. The most reduction in the BSFC is about 8.5% for E15, E15DEE2 and E15DEE4 at the full load condition, respectively.



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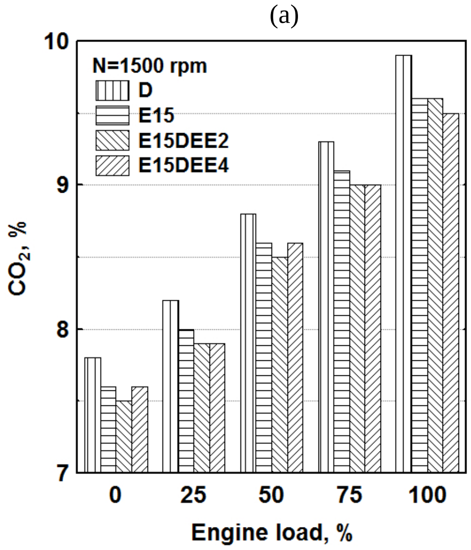
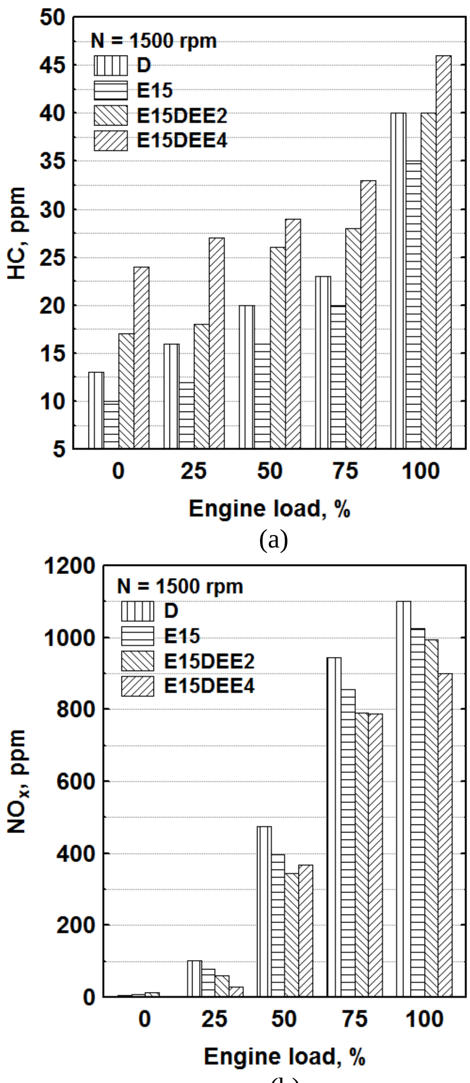


(b)

Fig. 8. CO and CO_2 emissions for tested fuels

The CO emission in Fig. 8(a) has lower values at the medium engine loads, while it is raising the low and high engine loads for all the tested fuel. The variation in CO for the blended fuels is fluctuating. Although the E15 and E15DEE2 blends have given the lower CO than those of the diesel fuel, an increment trend occurs when using the E15DEE4 blend at all the engine loads. Thus, the minimum CO emissions are generally obtained with the E15DEE2 blend. The reduction in CO can be explained as follows: the oxygenated fuels achieve the more complete combustion via the leaning effects, and thus the CO which is one of the incomplete combustion products, is reduced. On the other hand, the increment in the CO when increased the DEE ratio can be explained with the fact that the less time is available for the oxidization process and for leaving more CO in the exhaust because of the injection timing delay, and consequently retarding of the onset of the combustion, as given in Fig. 2 as well as the effect of the high heat of vaporization that leads to temperature reduction. Moreover, the erratic operation of the engine with the increased DEE ratio may be another reason for the CO formation, due to rough burning and misfiring because of too much leaning. The best reductions in the CO obtained with E15, E15DEE2 and E15DEE4 are about 23.5%, 41.2% and 29.4% at the full load condition, respectively. The CO_2 emission in Fig. 8(b) shows an increment trend with the engine load, and this is a result of the improvement in combustion efficiency, as can be seen in Fig. 7(a). On the other hand, the ethanol and DEE blends serve the lower CO_2 due to the ethanol and DEE having the lower amount of carbon atoms as tabulated in Table 1. This means that, a lower amount of carbon enters to the combustion chamber, and thus the CO_2 is reduced when using the blended fuels. The most reductions in CO_2 are about 10.3%, for E15 and E15DEE2 and 13.8% for E15DEE4 at full load condition, respectively. Thus, the best choice in terms of CO_2 is the E15DEE4.

The unburned HC emission in Fig. 9(a) increases with the engine loads for all the tested fuel and the ethanol blend has the lower HC emissions, while the DEE blends usually give the higher ones than those of the diesel fuel. The reductions in the HC with the ethanol addition can be based upon their leaning effects that improve the combustion by supplying more oxygen in the cylinder. However, the increase of the HC with the addition of the DEE, as well as ethanol, may result from the following reasons: the first is the higher heat of the evaporation of the ethanol or DEE in the blends which tends to produce slow vaporization and poorer fuel-air mixing which leading to incomplete combustion of the mixture. Another reason is the increased spray penetration causing undesired fuel impingement on the chamber walls hence the flame quenching and cushioning in the ring land areas. The final reason is related to the lean flame-out region. This region is referred to a region near the outer edge of the spray in which the mixture is often observed to be too lean to ignite or to support a stable combustion. The lower combustion temperatures in this region increase the HC (Iranmanesh, 2013). The reduction in the HC is about 12.5% for E15, and E15DEE2 has the same value with the diesel fuel, and the increment is about 15% for E15DEE4 at the full load condition, respectively. Thus, the best performer in terms of HC is E15.



(a)

(b)

Fig. 9. HC and NO_x emissions for the tested fuels

The NO_x emissions in Fig. 9(b) continuously increases with the engine load, and this is a result of the increment in the combustion temperatures which is directly related to the exhaust gas temperatures given in Fig. 5(b). The ethanol and DEE blends present the lower NO_x as seen in the figure. The NO_x emissions are affected from the oxygen content, adiabatic flame temperature and spray characteristics (Sivalakshmi and Balusamy, 2013). The formation of NO_x emissions significantly depends on the combustion temperatures and the blended fuels generate the lower cylinder temperatures because of their higher heat of vaporization and lower heating values that are given in Fig. 4(b). The effect of the fuel properties on the injection characteristics may be another reason for the decrease in the NO_x. The retardation of the injection timing because of the fuel properties of the oxygenated fuels cited above achieves a decrease in NO_x by shifting the start of the combustion to later and reducing the peak cylinder temperatures (Devan and Mahalakshmi, 2009). For these reasons, the NO_x emissions are reduced when using both ethanol and DEE blends, and the addition of the DEE results in additional decreases. The best reductions in the NO_x obtained with E15, E15DEE2 and E15DEE4 are about 6.8%, 9.7% and 18.2% at the full load condition, respectively. Thus, the best performer in terms of the NO_x is E15DEE4.

CONCLUSIONS

In this study, the effects of diethyl ether, as well as ethanol, addition to the diesel fuel on the engine performance and exhaust emissions have been investigated experimentally. A commercial diesel fuel and three different fuel blends, namely E15, E15DEE2 and E15DEE4 have been used in the experimental study. The following conclusions can be summarized as the results of the study.

1. Both ethanol and diethyl ether addition to diesel fuel causes some reductions in fuel injection pressure, and diethyl ether addition also results in retardation in injection timing.
2. The lower cylinder pressure and temperatures are obtained with ethanol and diethyl ether blends.
3. The increments in air excess coefficient and the decrements in exhaust gas temperatures occur when using ethanol and diethyl ether blends.
4. The ethanol and diethyl ether blends provide improvements in fuel economy and brake thermal efficiency, with a negligible engine power output reduction.
5. The ethanol addition serves the reductions in the emissions of the CO, CO_2, HC and NO_x and the diethyl ether, as well as ethanol, addition has undesirable effects on CO and HC emissions, depending on the blending ratio, but it is still better than diesel fuel except for HC.
6. The E15DEE2 blend is recommended as the best choice with the improvements of 17.1% in brake thermal efficiency, 8.5% in brake specific fuel consumption, 41.2% in CO, 10.3% in CO_2 and 9.7% in NO_x and a reduction in the power output by 0.3 percent.

ACKNOWLEDGMENT

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NOMENCLATURE

AFR air-fuel ratio (kg air/kg fuel)
BP brake power (kW)
BSFC brake specific fuel consumption (kg fuel/kWh)
BTE brake thermal efficiency, (%)
EGT exhaust gas temperature (°C)
LHV lower heating value (kJ/kg)
N engine speed (rpm)
TPU total percentages of uncertainty (%)
U uncertainty (%)
y volume percentage of each fuel in blend (%)

Greek letters

ρ density (kg/m^3)

Subscripts

b blend
s stoichiometric

Abbreviations

D diesel fuel
E15 a blend consist of 15% ethanol
E15DEE2 a blend consist of 15% ethanol and 2% DEE
E15DEE4 a blend consist of 15% ethanol and 4% DEE