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An experimental study on the performance parameters of an experimental CI engine fueled with diesel–methanol–dodecanol blends

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Abstract

The effects of using diesel–methanol–dodecanol blends including methanol of various proportions on a CI engine performance are experimentally investigated. The methanol concentration in the blend has been changed from 2.5% to 15% with the increments of 2.5%, and 1% dodecanol was added into each blend to solve the phase separation problem. Experimental study has been conducted on a single-cylinder, water-cooled CI engine. The engine has been operated at different compression ratios (19, 21, 23 and 25) and the engine speed was varied from 1000 to 1600 rpm at each compression ratio. The performance parameters such as torque, effective power, specific fuel consumption and effective efficiency for each blend at various conditions are calculated depending on the experimental data. It was concluded that among the different blends, the blend including 10% methanol (DM10) is the most suited one for CI engines from the engine performance point of view. Improvements obtained up to 7% in performance parameters with this blend without any modification to engine design and fuel system are very promising.

Keywords: Compression-ignition engines; Alcohols; Diesel–methanol blends

1. Introduction

Compression-ignition (CI) or diesel engines are widely used in the fields of commercial transportation, automotive, agricultural applications and industrial sector due to its high fuel conversion efficiency and ease of operation [1]. The existing CI engines operate with conventional diesel fuel derived from crude oil. It is well known that the world petroleum resources are limited and the production of crude oil is becoming more difficult and more expensive. On the other hand, although CI engines emit low CO_2 emission, they are the most important source of particulate emissions, soot, smoke and oxides of nitrogen (NO_X) which are harmful for the environment [2]. Oxides of nitrogen can also cause smog production, reacting with hydrocarbon compounds in the presence of sunlight [2]. In this connection, improving the performance and reducing emissions of these engines will be helpful to improve fuel economy and to reduce environmental pollution.

Using alternative fuels produced from non-petroleum resources in CI engines is suggested as one of the most attractive methods for improving their performance and emissions. These fuels include alcohols (such as ethanol and methanol), ethers, vegetable oils, animal fats, gaseous fuels (hydrogen, natural gas, liquefied petroleum gas) and bio-diesel [1,3–7]. Among these fuels, despite its very low cetane rating and poor solubility in diesel fuel, methanol has advantages of low cost, being non-sooting fuel, and high oxygen content [8,9]. The advantages and disadvantages of methanol as a CI engine fuel will be discussed in the next section.

The results reported in the numerous published papers relating the effect of using methanol in CI engines on the

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Nomenclature

B	mass flow rate of the fuel, kg/h
F/A	fuel/air ratio, dimensionless
LHV	lower heating value of the fuel, MJ/kg
$\dot{m}$	mass flow rate, kg/h
n	engine speed, rpm
N_e	effective power, kW
P	pressure, Pa
SFC	specific fuel consumption, kg/kW h
T	temperature, K
X	volume percentage, %

Greek symbols

ε	compression ratio, dimensionless
ϕ	fuel/air equivalence ratio, dimensionless
η	efficiency, %
ρ	density, g/cm^3

Subscripts

a	air
act	actual
bl	blend
e	effective
0	standard atmospheric conditions
s	stoichiometric

exhaust emissions and engine performance are very prospective [3,4,8,10,11]. It was also concluded from the literature review that studies in the matter of using methanol in CI engines have been mainly related to the combustion and exhaust emissions and studies concerning the effects of methanol on engine performance and fuel economy are at limited number. This topic needs to be further investigated for widespread use of methanol. For this reason, in the present study, it is aimed to investigate the performance of a CI engine fueled with various diesel–methanol blends. For this purpose, different diesel–methanol blends including 1% 1-dodecanol (which is commonly named as dodecanol or lauryl alcohol) as solvent are examined in a single-cylinder experimental CI engine and the effects on engine performance are evaluated.

2. Properties of methanol as a fuel for CI engines

Methanol is manufactured from any material that can be decomposed into carbon monoxide (or carbon dioxide) and hydrogen [12]. In this regard, it may be produced from the sources which are independent of petroleum. The primary feedstocks for methanol production are natural gas, lignite, coal and the renewable resources such as wood, agricultural materials, biomass, waste biomass and municipal wastes [12–14]. In the methanol production, raw material is firstly converted to synthesis gas ($CO + H_2$) by gasification and finally CO and H_2 are catalytically combined by means of a catalyst like Cu–Zn–Cr to yield methanol [15].

Properties of diesel, methanol and dodecanol are listed in Table 1. Methanol has lower heating value than diesel due to its partially oxidized state therefore much more fuel is needed to obtain same performance with that of diesel-fueled engine. Its high stoichiometric fuel/air ratio, high oxygen content and high H/C ratio may be beneficial for improving the combustion and reducing the soot and smoke. However, it was reported that engines operating with methanol and other oxygenated fuels emit more aldehyde emissions such as formaldehyde [2,12,15]. Formaldehyde can cause eye irritation and the formation of smog [12]. The use of an exhaust catalyst is helpful to reduce these emissions. Methanol has higher latent heat of vaporization than diesel so that it extracts much more heat as it vaporizes, therefore can lead to cooling effect on the cylinder charge [18]. As a result of cooling effect on the charge, cylinder temperature may decrease and therefore emissions of nitrogen oxides would be reduced. Methanol has the poor ignition behavior due to its low cetane number, high latent heat of vaporization and high ignition temperature therefore it can produce longer ignition delay. As shown in Table 1, methanol has very low viscosity compared to diesel fuel therefore it can easily be injected, atomized and mixed with air. In this case, a lubricant additive should be added to the fuel to improve the lubrication [19].

The possible benefits and shortcomings of methanol as a fuel for CI engines are summarized above. Methanol can be used in CI engines as pure or by blending with conventional diesel fuel. Problems concerning the use of methanol in diesel engines can be removed by different approaches which are briefly described below. Using it in CI engines as diesel–methanol blends is the simplest method. The most important problem encountered in this case is the phase separation. This problem can be prevented by adding some solvent into mixture [20]. Moreover, an ignition improver like diethyl ether [5] can be added in the blended fuel to compensate the cetane number. This application requires no modification on engine design and fuel system if concentrations of methanol in the blends are at low levels. In the second way, methanol can be used as pure in CI engines by the methods such as dual-fueling [1], blending with an ignition improver [21], spark-assisted method [22] and fumigation [23]. Although these methods eliminate the

Table 1
Properties of diesel, methanol and dodecanol [4,8,16,17]

Property	Diesel	Methanol	1-Dodecanol
Molecular formula	$C_{14.342}H_{24.75}$	CH_3OH	$C_{12}H_{26}O$
Molecular weight (kg/kmol)	197.21	32.042	186.339
Stoichiometric fuel/air ratio	0.06924	0.15393	0.07462
Cetane number	45–55	3–5	–
Flash point (°C)	78	11	107
Ignition temperature (°C)	235	470	527
Viscosity at 298.15 K (mPa s)	3.35	0.59	16.136
Density ($g\,cm^{-3}$)	0.83	0.79	0.83
Lower heating value (MJ/kg)	42.740	20.270	39.860
Heat of vaporization (MJ/kg)	0.270	1.110	–

disadvantage of poor ignition quality of methanol, using additional equipments such as spark plug, fuel-injection system, carburetor or vaporizer and storage system results in additional costs.

As a consequence of the above discussions, using methanol in CI engines as diesel–methanol blends including methanol at low concentrations is the most practical and economic way. For this reason, the present study is mainly concentrated on the use of diesel–methanol blends in CI engines. Murayama [20] examined 10 different solvents and found that dodecanol was the best one among them for diesel–methanol blends. For this reason, in the present study, dodecanol has been used as solvent to stabilize the fuel mixture. Dodecanol is a fatty alcohol and commonly produced from coconut. It has nearly same heating value with diesel, has high viscosity and high ignition temperature [16,17]. Although the high viscosity and high ignition temperature can cause difficulties in fuel atomization and ignition, the present author's opinion is that adding small amounts of dodecanol into the fuel blends will not lead to such problems.

3. Experimental study

3.1. Fuel properties

The molecular formulae of diesel, methanol and dodecanol used in this study are given in Table 1. The lower heating value of each fuel has been calculated using the Mendeleyev's formula given by Khovakh [24] as follows:

$$\mathrm{LHV} = 34.013c' + 125.6h' - 10.9o' - 2.512(9h' + w'), \quad (1)$$

where c', h', o' and w' refer to the elementary composition of fuel that is, the amounts of separate elements (carbon, hydrogen, oxygen and water) in unit mass of the fuel. The values of LHV for diesel, methanol and dodecanol have been determined from Eq. (1) as 42.74, 20.27 and 39.86 MJ kg^{-1}, respectively. The densities and stoichiometric fuel/air ratios of diesel, methanol and dodecanol used in this study are given in Table 1 and the properties of blends have been determined from the following equations:

$$\rho_{\mathrm{bl}} = \frac{\sum X_i \rho_i}{100}, \quad (2)$$

$$(F/A)_{\mathrm{sbl}} = \frac{\sum X_i \rho_i (F/A)_{si}}{\sum X_i \rho_i}, \quad (3)$$

$$\mathrm{LHV}_{\mathrm{bl}} = \frac{\sum X_i \rho_i \mathrm{LHV}_i}{\sum X_i \rho_i}. \quad (4)$$

Here, subscript i refer to diesel, methanol or dodecanol. The other properties of fuels given in Table 1 have been taken from the cited literature [4,8,16,17].

3.2. Experimental setup and procedure

The engine used in the experiments is a single-cylinder, four-stroke, water-cooled, variable compression, direct injection (DI) compression-ignition engine having a swept volume of 763 cm^3. Dimensions of the engine are: the bore $D = 90$ mm and stroke $H = 120$ mm. The shaft of the engine is coupled to the rotor of an electric dynamometer which is used to load engine by increasing the field voltage and to measure the engine output torque. A calibrated burette and a stopwatch were employed to measure the mass flow rate of fuel. Diesel–methanol–dodecanol blends have been prepared by blending the analysis-grade anhydrous methanol having a purity of 99.8%, with diesel in concentrations of 2.5 through 15% (by volume) with the increments of 2.5% and have been directly injected in to the combustion chamber. Dodecanol of 1% (by volume) was added into each blend to obtain stabilized mixture. Above the 15% methanol, engine could not run smoothly therefore experimental results obtained up to this percentage of methanol are presented. The engine has been operated at full throttle setting and at the compression ratios of 19, 21, 23 and 25. At each compression ratio, engine speed has been changed from 1000 to 1600 rpm with the increments of 100 rpm. All the experiments have been performed at the fixed fuel-injection timing condition. The engine performance parameters when using diesel and different diesel–methanol–dodecanol blends have been comparatively determined.

The torque exerted by the engine is measured from the stator of dynamometer by balancing it with weight of 200 N and a spring. The engine effective power is corrected for the standard atmospheric conditions by Eq. (5), and then the specific fuel consumption and effective efficiency are calculated from Eqs. (6) and (7) as follows:

$$N_{eo} = N_e \frac{0.1013}{p_0} \frac{T_0}{293.15}, \tag{5}$$

$$\mathrm{SFC} = \frac{B}{N_e}, \tag{6}$$

$$\eta_e = \frac{3.6}{\mathrm{LHV} \times \mathrm{SFC}}. \tag{7}$$

Finally, the actual fuel/air ratio $(F/A)_{act}$ and the fuel/air equivalence ratio ϕ_{bl} are calculated from Eqs. (8) and (9), respectively

$$(F/A)_{act} = \frac{B}{\dot{m}_a}, \tag{8}$$

$$\phi_{bl} = \frac{(F/A)_{act}}{(F/A)_{sbl}}. \tag{9}$$

4. Results and discussion

The fuel/air equivalence ratio of the blends ϕ_{bl} has been calculated from Eqs. (3), (8) and (9). Methanol has the higher stochiometric fuel/air ratio than diesel due to its partially oxidized state or it is an oxygenated fuel, therefore blending it into diesel leads to the leaner operation. This effect is clearly shown in Figs. 1 and 2: fuel/air equivalence ratio of the diesel–methanol blend ϕ_{bl} decreases as the proportion of methanol increases. As shown from the figures, blending methanol with diesel leads to a leaning effect of about 18% on the blended fuel. The leaner operation can result in some improvements in engine performance parameters as will be explained below.

Since methanol has lower cetane number, higher ignition temperature and higher latent heat of vaporization than the conventional diesel fuel, it has poorer ignition quality. Consequently, difficulties in the ignition of fuel–air mixture may occur and the duration of ignition delay could increase. Increasing ignition delay to some extent may be beneficial for combustion and engine performance.

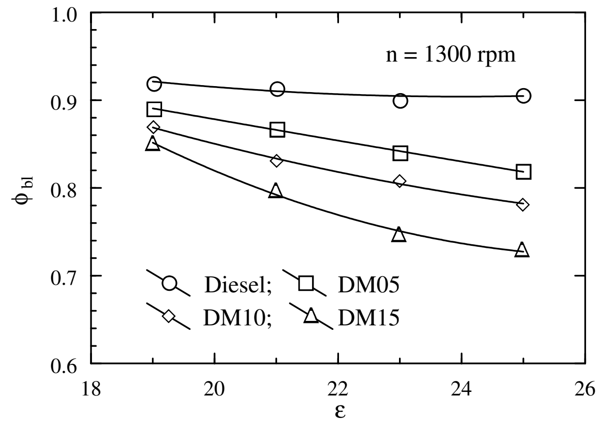

Fig. 2. Variation of ϕ with compression ratio for diesel and different blends.

The amount of fuel burned in the premixed burning phase can increase due to the longer ignition delay therefore higher cylinder pressures would be obtained with diesel–methanol blends. Meanwhile, blending methanol with diesel fuel can supply additional oxygen for diffusive combustion phase and can cause improvements in combustion process; namely, more efficient and more complete combustion. Moreover, as stated in Section 2, methanol addition in diesel fuel leads to cooling effect on the cylinder charge because of its lower heating value, higher latent heat of vaporization and higher stoichiometric fuel to air ratio. As a result, the peak cylinder temperature decreases and the engine knock could be prevented. Therefore, improvements in engine performance can be expected when diesel–methanol blends are used in CI engines. This is clearly shown in Fig. 3–6 which indicate the engine output powers obtained with different blended fuels at various engine operating conditions. Here, "DM" designates the diesel–methanol blend and numbers next to "DM" refer to the volume percentage of methanol in the blended fuel. In all cases, the blend including 10% methanol (DM10) gives the better values for the effective power N_e. The maximum increment in effective power obtained with this blend is

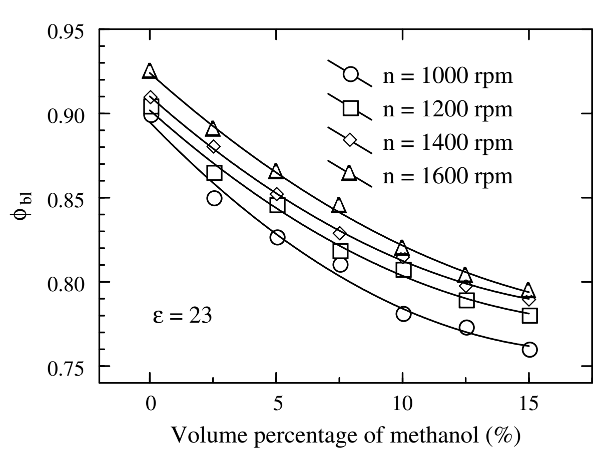

Fig. 1. The effect of methanol amount on ϕ at different engine speeds.

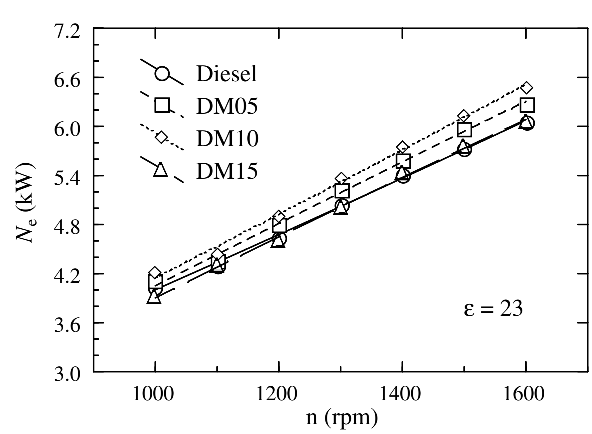

Fig. 3. Variation of N_e with engine speed for diesel and different blends.

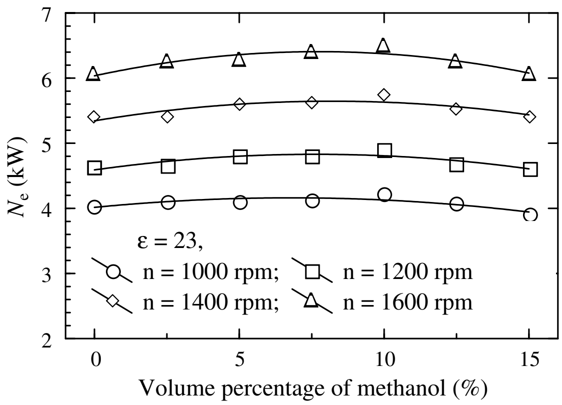

Fig. 4. The effect of methanol concentration on N_e at different engine speeds.

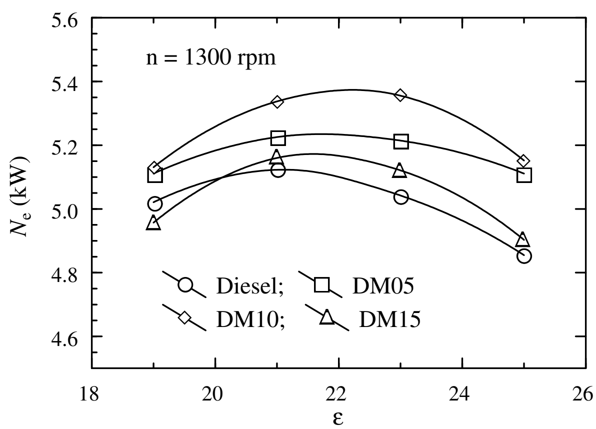

Fig. 5. Variation of N_e with ε for diesel and different blends.

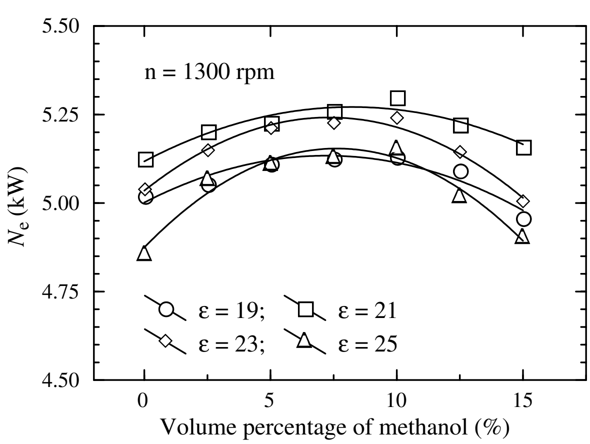

Fig. 6. The effect of methanol concentration on N_e at different compression ratios.

about 7% at the operating conditions of $n = 1500$ rpm and $\varepsilon = 23$. These figures also show that the blends having methanol proportions above 10% cause deteriorations in the engine performance. This can be explained with the following reasons: The cetane number of the blended fuel decreases and the auto ignition temperature and vaporization heat of blended fuels increase as methanol concentration increases. When the cetane number drops below the levels required for CI engine operation and the charge temperature extremely decreases, very long ignition delays could occur because of the difficulties in ignition, so that combustion may start during the expansion process and the fuel cannot be completely burned within this limited time remaining for the combustion. Another reason for decreasing power can be attributed to the decreasing lower heating value of diesel–methanol blends with increasing methanol concentration.

The improvement in combustion enhances the fuel conversion efficiency and results in decreases in specific fuel consumption SFC. On the other hand, SFC is inversely proportional to the effective power as in Eq. (6). Due to such reasons, SFC decreases with increasing methanol percentage up to 10% as shown in Figs. 7–10. A maximum improvement of about 7% were obtained with DM10 at $n = 1300$ rpm and $\varepsilon = 25$. As stated above, the methanol content higher than 10% affects the combustion and therefore SFC, negatively.

The exhaust temperature may decrease with the increasing methanol amount owing to the decreasing peak cylin-

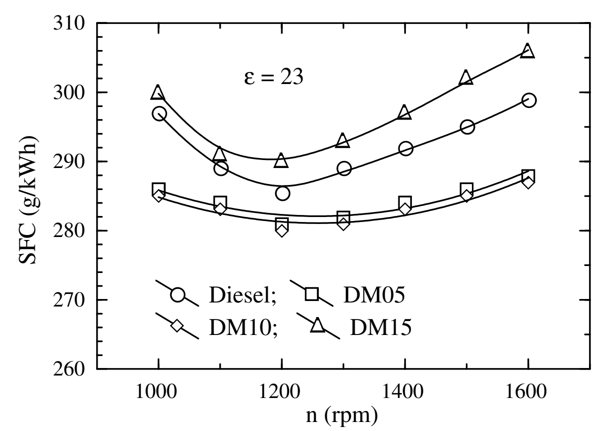

Fig. 7. Variation of SFC with rpm for diesel and different blends.

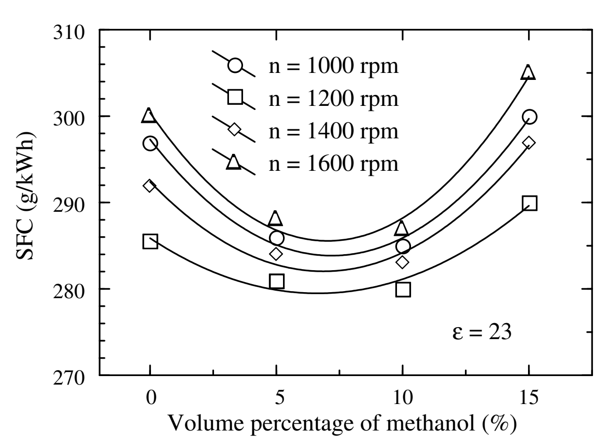

Fig. 8. The effect of methanol concentration on SFC at different engine speeds.

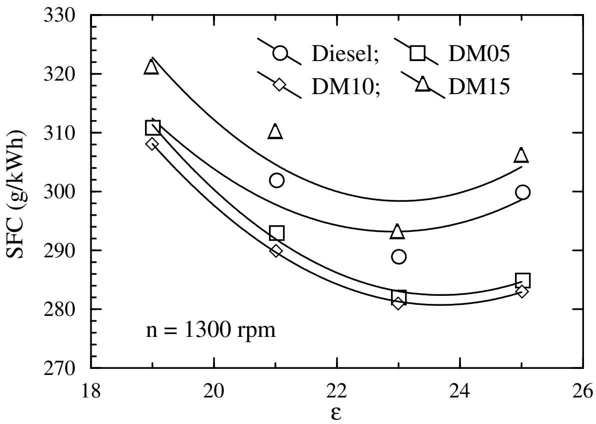

Fig. 9. Variation of SFC with compression ratio for diesel and different blends.

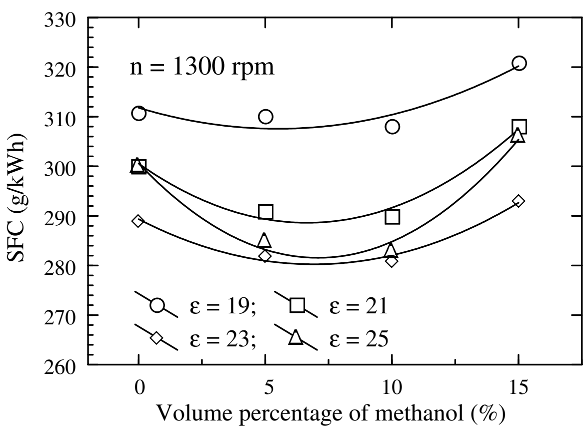

Fig. 10. The effect of methanol concentration on SFC at different compression ratios.

der temperature resulting from both raising latent heat of vaporization and the leaner operation. According to the second law of thermodynamics, the engine thermal efficiency increases due to the reduced heat loss from the engine through heat transfer to the coolant and to atmosphere. Moreover, the effective efficiency is inversely pro-

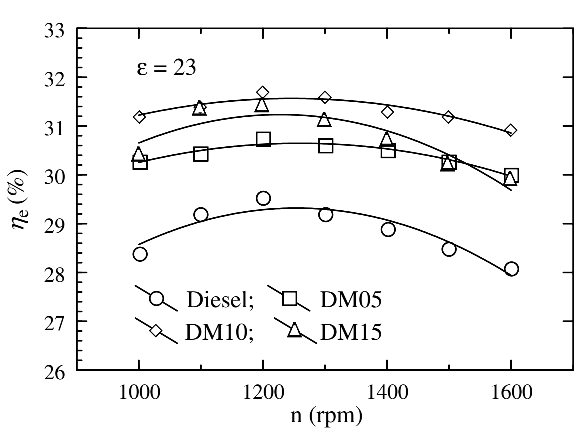

Fig. 11. Variation of η_e with engine speed for diesel and different blends.

portional to the SFC and LHV as in Eq. (7). Consequently, decreases in the heat loss, SFC and LHV improve the effective efficiency. As shown in Figs. 11–14, η_e reaches maximum at DM10 and then decreases. The maximum increase in effective efficiency obtained with this blend is about 7%.

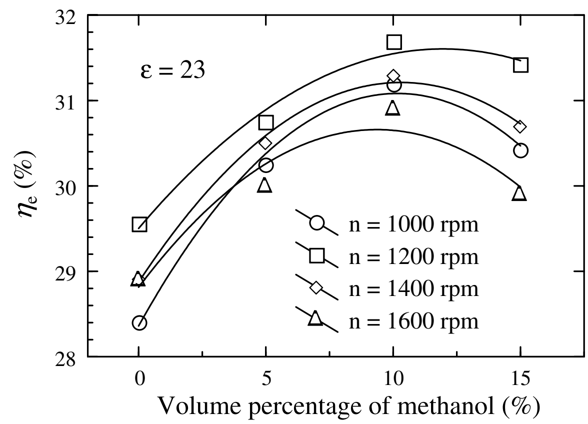

Fig. 12. The effect of methanol concentration on η_e at different engine speeds.

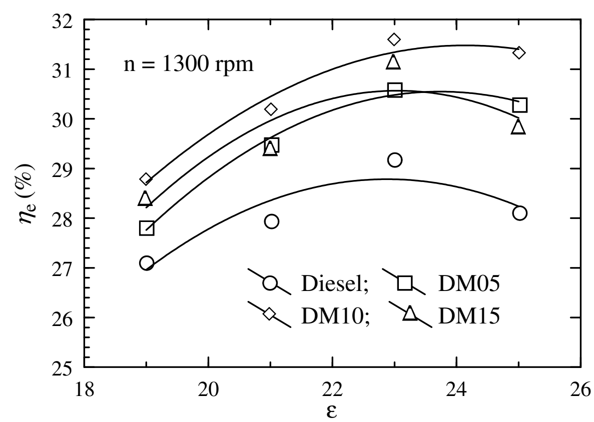

Fig. 13. Variation of η_e with ε for diesel and different blends.

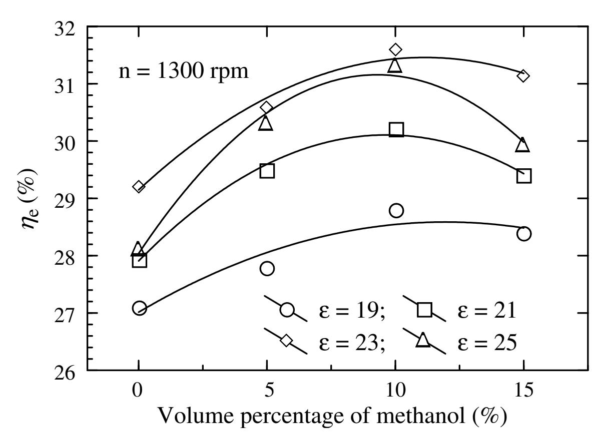

Fig. 14. The effect of methanol concentration on η_e at different compression ratios.

5. Conclusions

In the present study, a detailed experimental investigation has been conducted on the effects of methanol addition up to 15% (by volume) into diesel fuel on the engine performance parameters. The phase separation problem has been prevented by adding 1% dodecanol to each diesel–methanol blend. The engine has been operated with each blend at different compression ratios at which the engine speed has been changed between 1000 and 1600 rpm. The main conclusions can be summarized as follows:

1. The fuel–air equivalence ratio of diesel–methanol blend decreases with the increasing methanol amount. The blend of 15% methanol results in a leaning of about 18% in the fuel–air mixture.
2. Methanol causes improvement in engine effective power. The maximum improvement of about 7% in N_e was obtained with the blend of DM10.
3. The specific fuel consumption SFC decreases and the engine effective efficiency increases when using the diesel–methanol blends. An improvement of about 7% in both parameters was obtained with the blend of DM10.
4. Consequently, it can be concluded that methanol can be used in CI engines without any modification on the engine design and fuel system by blending it with diesel fuel at low concentrations (up to 10%), and the engine performance can be improved by this way. In this case, phase separation which is the most important problem encountered can be prevented by adding some solvent (here dodecanol) into the blend. The optimum percentage of methanol was determined as 10%. The better results would be obtained if the cetane rating of the blends is raised by blending some cetane improver in to the blends.

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