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*Full Length Research Paper*

# An experimental investigation on effects of methanol blended diesel fuels to engine performance and emissions of a diesel engine

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**Considering strict restrictions on exhaust emissions of newly produced diesel engines, in this study, the effects of methanol and diesel fuel blends on compression ignition engine performance and exhaust emissions of a four cylinder, four stroke, direct injection, turbocharged diesel engine were experimentally investigated. Methanol-blended diesel fuels were ranged from 0 to 15% volumetric methanol content with an increment of 5%. The tests were performed by varying engine speed between 1000 min$^{-1}$ to 2700 min$^{-1}$ by an engine testing dynamometer. Results indicated that brake specific fuel consumption and nitrogen oxide emissions increased while brake thermal efficiency, carbon monoxide and hydrocarbons decreased relative to single diesel fuel operation with increasing amount of methanol in the fuel mixture. Effects can be visualized by data which were 49 and 47.5 kW for power, 169 and 190 g/kWh for brake specific fuel consumption, 33 and 30% for brake specific thermal efficiency, 0.21 and 0.18% for carbon monoxide, 7.15 and 8.1% for carbon dioxide, 8.02 and 6.1 ppm for hydrocarbons, 385 and 418 ppm for nitrogen oxides at 1600 min$^{-1}$ in order of standard diesel fuel operation and fuel blend with 10% methanol content.**

**Key words:** Methanol, diesel fuel, compression ignition, duel fuel, exhaust emission, emission reduction, combustion.

## INTRODUCTION

Compression ignition (CI) engines are widely used for transportation, automotive, agricultural applications and industrial sectors because of their high fuel conversion efficiencies and relatively easy operation. These wide fields of usage lead to increasing requirements of petroleum-derived fuels. The depletion of petroleum reserves and increasing demand also induce a steep rise in fuel prices. On the other hand, their exhaust emissions, such as soot and nitrogen oxide ($NO_x$) are harmful to natural environment and living beings (Yao et al., 2008). Much effort is being paid worldwide to reduce the soot, carbon monoxide (CO), hydro-carbon (HC) and $NO_x$ emissions from diesel engines. Recently, changing the engine operating parameters such as valve timing, injection timing, and atomization ratio has been carried out in many studies on the internal combustion engines (ICE) aiming to reduce the exhaust emissions (Canakci et al., 2009). At the same time, depletion of fossil fuels and environmental considerations has led to investigations on the renewable fuels such as methanol, ethanol, hydrogen, and biodiesel. Particularly, methanol can be obtained from many fossil and renewable sources. These include coal, petroleum, natural gas, biomass, wood, landfills and even the ocean (Sayin et al., 2009).

Common technologies for internal combustion engines, especially CI engines have certain specifications for their fuel systems. This situation restricts renewable energy sources to be directly used in these engines. Thus, these sources are blended with fossil fuels to be used with ICE which are already being on field and to reduce petroleum derived fuels costs and environmental harms. An instance

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and a widely investigated application is methanol addition into diesel fuel for CI engines. Duel fuel operation with methanol and diesel fuel brings following advantages and disadvantages;

The relative advantages of methanol comparing with conventional diesel fuel include:

1) High stoichiometric fuel to air ratio
2) High oxygen content, high hydrogen to carbon ratio and low sulfur content
3) Higher latent heat of vaporization
4) Improving the combustion and reducing the soot and smoke
5) Higher cooling by evaporation of methanol blended diesel fuel relative to single diesel fuel.

Thus required work input in the compression stroke is reduced.

The disadvantages are:

1) Poor ignition behavior due to its low cetane number, high ignition temperature. Therefore it can produce longer ignition delay
2) More corrosive than diesel fuel on copper, brass, aluminum, rubber, and many plastics
3) Methanol has lower energy content and much lower flash point comparing with diesel engine (Bayraktar, 2008).

The possible benefits and shortcomings of methanol as a fuel for CI engines are summarized above. Methanol can be used in diesel engines either by blending it with the diesel fuel or by injecting into charge air (Zhang et al., 2010). Using it in CI engines as diesel fuel–methanol blends is the simplest method. The most important problem encountered in this case is the separation of phases. This problem can be prevented by adding some solvent into mixture. Moreover, an ignition improver like diethyl ether can be added into the blended fuel to increase the cetane number. This application doesn't require modification on engine design and fuel system if concentrations of methanol in the blends are at low levels (Bayraktar, 2008). On the other hand, the fumigation method requires minor modification of the engine so that the methanol can be injected into the air intake using low-pressure fuel injectors. This approach allows a larger percentage of methanol to be used. Moreover it allows variation of the diesel/methanol ratio for different operating conditions while the premixed fuel can only operate at a fixed diesel/methanol ratio (Zhang, 2009).

A major disadvantage of using the fumigation methanol method is the increase of HC and CO emissions. However, diesel oxidation catalysts (DOC) can be used for the oxidation of HC and CO, as well as particular matter (Zhang, 2009). Since using methanol-blended diesel fuel can reduce the air pollution and depletion of petroleum fuels simultaneously, many researchers have studied the influence of this alternative fuel on the exhaust emissions of ICE. Cheng et al. (2008) reported the effects of the fumigated methanol to engine performance, exhaust emissions, and particulates. It is expressed that methanol was fumigated and injected up to 10, 20 and 30% engine loads under different engine operating conditions. The experimental results showed a decrease in brake thermal efficiency (BTE) when fumigated methanol is applied, except at the highest load of 0.67 MPa. At low loads, the BTE is decreased with the increase in fumigation methanol; but at high loads, it increased with the increase in the fumigation methanol. The fumigated methanol resulted in a significant increase in unburned hydro-carbons (UHC), CO, and $NO_x$ emissions.

Çanakçi et al. (2009), showed effects of injection pressure to engine performance, exhaust emissions and combustion characteristics with a series of experiment when using methanol-blended diesel fuel from 0 to 15% with an increment of 5%. The tests were conducted at three different injection pressures (180, 200 and 220 bar) by decreasing or increasing shim number. The experimental test results proved that brake thermal efficiency, heat release rate, peak cylinder pressure, smoke number, carbon monoxide and unburned hydrocarbon emissions reduced as brake-specific fuel consumption, brake specific energy consumption, combustion efficiency, and nitrogen oxides and carbon dioxide emissions increased with increasing amount of methanol in the fuel blend. Yao et al. (2008), explained effects of diesel-methanol compound combustion (DMCC) on diesel engine combustion. The emissions were studied and experiments were conducted on a four cylinder CI engine, which had been modified to operate in diesel fuel/methanol compound combustion. Experiments were conducted at idle and five engine loads at two levels of engine speeds to compare engine exhaust emissions from operating on pure diesel fuel and on operating with DMCC, with and without the oxidation catalytic converter. The experimental results show that the diesel engine operating with the DMCC could simultaneously reduce the soot and $NO_x$ emissions while increasing the HC and CO emissions compared with the standard diesel engine. However, using the DMCC coupled with an oxidation catalyst, the CO, HC, $NO_x$ and soot emissions could all be reduced. According to Bayraktar (2008), the effects of using diesel–methanol–dodecanol blends including methanol of various proportions on a CI engine performance are found as 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. Methanol concentration in the blend was changed from

**Table 1.** Technical specifications of the test engine.

| | |
|---|---|
| Cylinder number | 4 |
| Cylinder bore | 104 mm |
| Stroke | 115 mm |
| Total cylinder volume | 3908 ml |
| Compression ratio | 17:1 |
| Maximum torque | 295 Nm (at 1600 $min^{-1}$) |
| Maximum power | 62.5 kW (at 2500 $min^{-}$) |
| Maximum speed | 2700 $min^{-1}$ |
| Cooling system | Water Cooling |
| Injection advance | 18 (Crank shaft angle) |
| Injection pressure | 230 Bar |

2.5 to 15% with the increments of 2.5 and 1% dodecanol was added into each blend to solve the separated phases problem. The engine was operated at different compression ratios (19, 21, 23 and 25) and the engine speed was varied from 1000 to 1600 $min^{-1}$ at each compression ratio. Chao et al. (2001), investigated the emission characteristics of a six cylinder, naturally aspirated, direct injection diesel engine using diesel fuel blended with up to 15% volume of a methanol containing additive. They conducted steady state tests as well as transient cycle tests. They found a decrease in $NO_x$ emissions but an increase in CO and HC emissions as the methanol content in the blended fuel was increased. Regarding particulate matter (PM), the results are mixed: PM emission could increase or decrease, depending on the operating conditions.

During this study, methanol and diesel fuel dual operation is selected to be one of the solutions for both air pollution and combustion efficiency. Scientific literature about the dual fuel operation was investigated and blending method was determined to be the proper way because it's easy application without any modification in ICE and its performance characteristics. However, to make certain suggestions about the application and its results the research team decided to study specific circumstances of dual fuel operation. Therefore, in this study, methanol was blended with diesel fuel at rates of 0, 5, 10 and 15% diesel fuel volume and their effects on the engine performance and exhaust emissions were experimentally investigated using a four cylinder, turbocharged, direct injection diesel engine. Results were evaluated, interpreted and as a result some suggestions were made at the end of the study about duel fuel operation and its application to ICE.

## EXPERIMENTAL SETUP AND PROCEDURE

The present study was conducted on a "4DT 39T/185B-217299" turbocharged diesel engine of TUMOSAN (Konya, TURKEY). The engine used in the study has four- cylinder, four-stroke, direct injection swept volume of 3.908 liter, compression ratio of 17:1 and was turbocharged and water cooled. The general specifications of the engine are given in Table 1. The shaft of the engine is couple to the rotor of a hydraulic dynamometer which is used to load engine to measure the engine output torque and calculate power. A load sensor was employed to determine the load of dynamometer. The engine speed was measured by rotation sensor installed on the dynamometer. A calibrated burette and a stopwatch were employed to measure the volumetric flow rate of fuel. The schematic view of the test equipment is show in Figure 1. Exhaust emissions ($CO_2$, CO, HC and $NO_x$) were measured with a Italo plus – spin exhaust emission device. The analyzer was calibrated with standard gases and zero gas before each experiment. The general specifications of the device are given in Table 2.

The fuels used in this study were euro-diesel and methanol fuels. The major properties of these fuels are shown in Table 3. Before the test process, standard diesel engine (SDE) test were carried out according to Turkish Standards 1231 (TS-1231). Euro diesel was purchased from OPET (İstanbul, TURKEY). Methanol, with a purity of 99% was purchased from a commercial supplier. The volume percentages of test fuels were 0, 5, 10 and 15% of methanol with 100, 95, 90 and 85% diesel fuel respectively, which were named as SDE, M5, M10 and M15. The fuel blends were prepared just before starting experiments to provide homogenous mixture. A mixer was mounted inside fuel tank in order to prevent phase separation. The experiments were conducted at steady state for ten different engine speeds (1000 to 2700 $min^{-1}$) at full load. The values of engine coolant water temperature, mass flow rate of air, exhaust pollutants such as CO, $CO_2$, UHC, and $NO_x$ were recorded during the experiments. All data were collected after the engine stabilized. All the gaseous emissions were continuously measured for 5 min and the average results were presented. The steady state tests were repeated to ensure that the results are repeatable.

## EXPERIMENTAL RESULTS

Results which are engine performance parameters such as engine power, engine torque, brake specific fuel consumption (BSFC), brake thermal efficiency (BTE) and exhaust emissions such as $NO_x$, HC, CO, and $CO_2$ are provided further. The power output variation of the tested engine with different engine speeds at full load due to

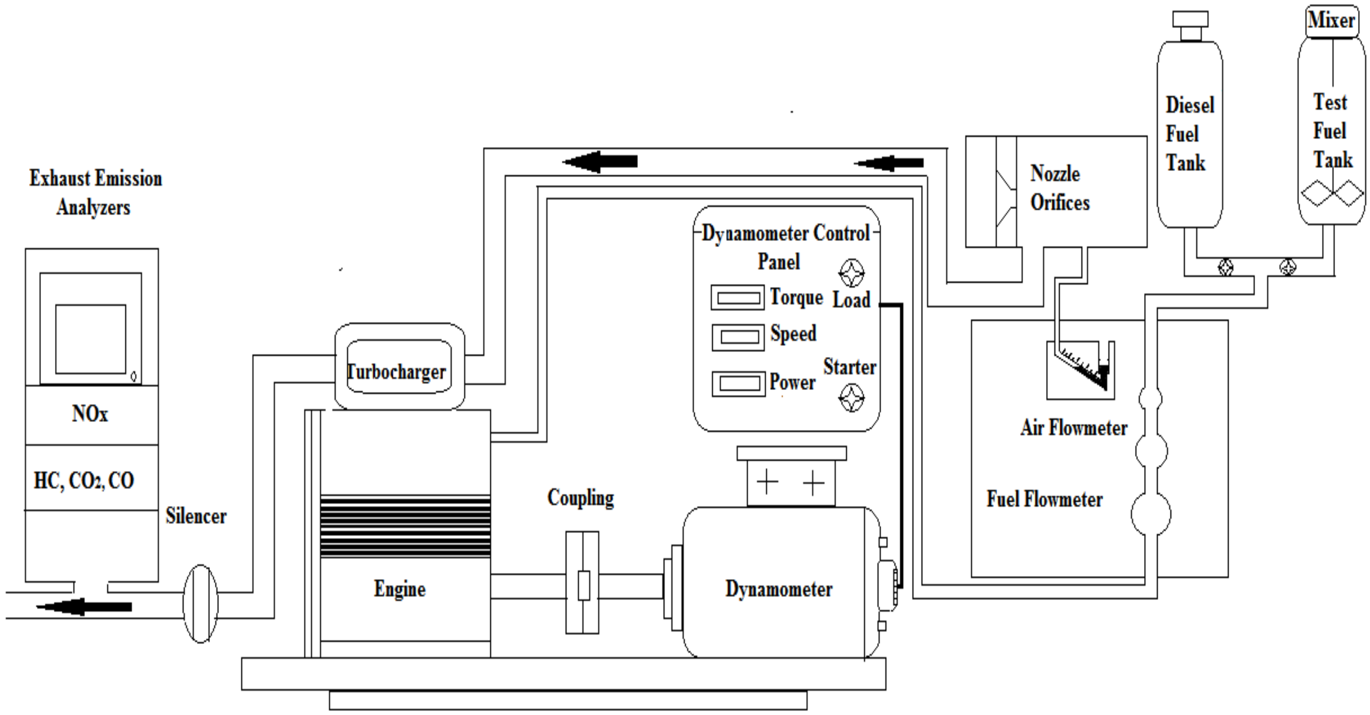


**Figure 1.** Experimental setup.

**Table 2.** Specifications of italo plus – spin exhaust emission analyzer device.

| | Unit | Measure range |
|---|---|---|
| CO | % | 0-9.99 |
| $CO_2$ | % | 0.19.99 |
| HC | ppm | 0-2500 |
| COK | % | 0-9.99 |
| λ | % | 0-1.99 |
| $O_2$ | % | 0-20.8 |
| NOx | ppm | 0-2000 |
| Operation temperature | °C | 5-40 |
| Storage temperature | °C | (-20)-(+60) |
| Feed voltage | V | 12 DC |

dual fuel strategies is shown in Figure 2. BTEs are shown in Figure 3, for diesel fuel and fuel blends. BTE indicates the ability of the combustion system to accept the experimental fuel and provides comparable means of assessing how efficiently the energy in the fuel was converted to mechanical output (Sayin, 2010). Figure 4 shows BSFC according to different engine speeds. The BSFC is defined as the ratio of mass fuel consumption to the brake power. As shown in Table 3, the maximum lower heating value (LHV) (42.74 MJ/kg) belongs to diesel fuel, lowest LHV (20.27 MJ/kg) belongs to methanol. CO emission results are given in Figure 5 for different engine speed at full load. At maximum torque (1600 min-1), CO percentages were found as 14, 18, 19 and 21% for M10, M15, M5 and SDE, respectively. The changes on the $NO_x$ emissions at different engine speeds are shown at Figure 6 for diesel fuel and fuel blends. Figure 7 shows $CO_2$ emission behavior of different fuel blends at different engine speeds. When the methanol amount was increased in the fuel mixture, maximum $CO_2$ was observed to be 7.91, 8.1, 7.88 and 7.15% at M15 M10, M5 and SDE for the full load of engine at 1600 min$^{-1}$.

**Table 3.** Fuel properties of euro-diesel and methanol fuels.

| | Euro-diesel | Methanol |
|---|---|---|
| Chemical formula | $C_{14}H_{28}$ | $CH_3OH$ |
| Flame speed rate (cm/s) | 33 | 35 |
| Boiling temperature (℃) | 190-280 | 64.7 |
| Density (g/cm$^3$, at 2 ℃) | 0.84 | 0.79 |
| Flash point (℃) | 52 | 11 |
| Auto ignition temperature(℃) | 254 | 464 |
| Lower heating value (MJ/kg) | 42.74 | 20.27 |
| Cetane number | 56.5 | 4 |
| Stoichiometric air/fuel ratio | 14.7 | 6.66 |
| Octane number | Not applicable | 109 |
| C/H ratio | 0.50 | 0.25 |
| Kinematic viscosity at | $2.5*10^{-6}$ | $75*10^{-6}$ |
| Carbon content (wt.%) | 86 | 37.5 |
| Hydrogen content (wt.%) | 14 | 12.5 |
| Oxygen content (wt.%) | 0 | 50 |
| Sulfur content (wt.%) | <50 | - |
| Heat of vaporization (MJ/kg) | 0.27 | 1.11 |

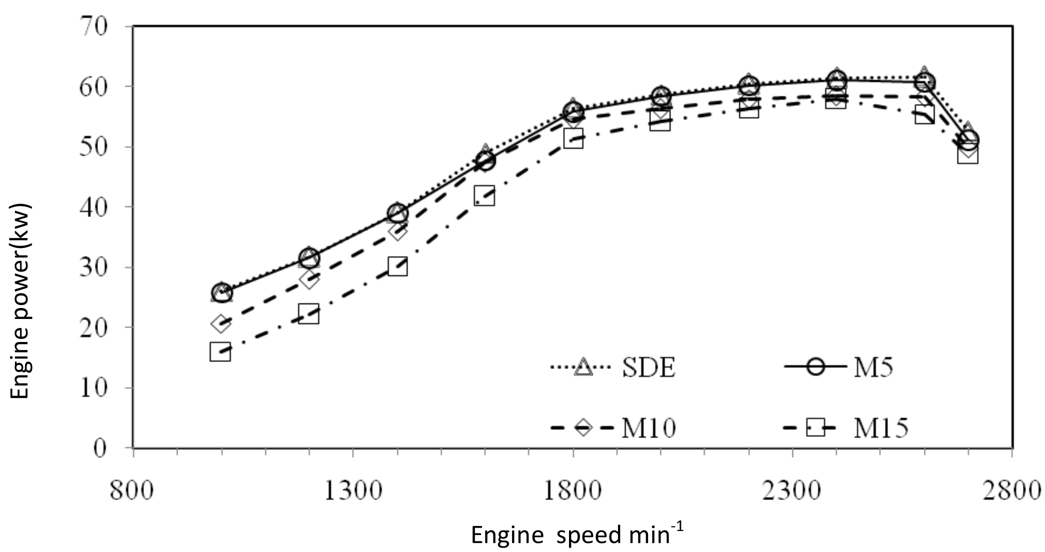


**Figure 2.** Engine power values at different engine speeds and full load with diesel methanol blends.

HC emissions are shown in Figure 8, for diesel fuel and fuel blends. In comparison with SDE, the decrease in HC emissions were 7.15, 6.1 and 5.85 ppm for M5, M10, M15, respectively, at 1600 min$^{-1}$ which is the engine speed of maximum torque.

## DISCUSSION

Interpretations for experimental results are maintained in four sub section. Graphic interpretations were made considering maximum torque value as base value.

### Engine power

As shown at Figure 2, it can be seen that the power values of M5, M10 and M15 are lower than regular diesel and decrease with the increase of methanol addition into the blends. The differences of power indicate the

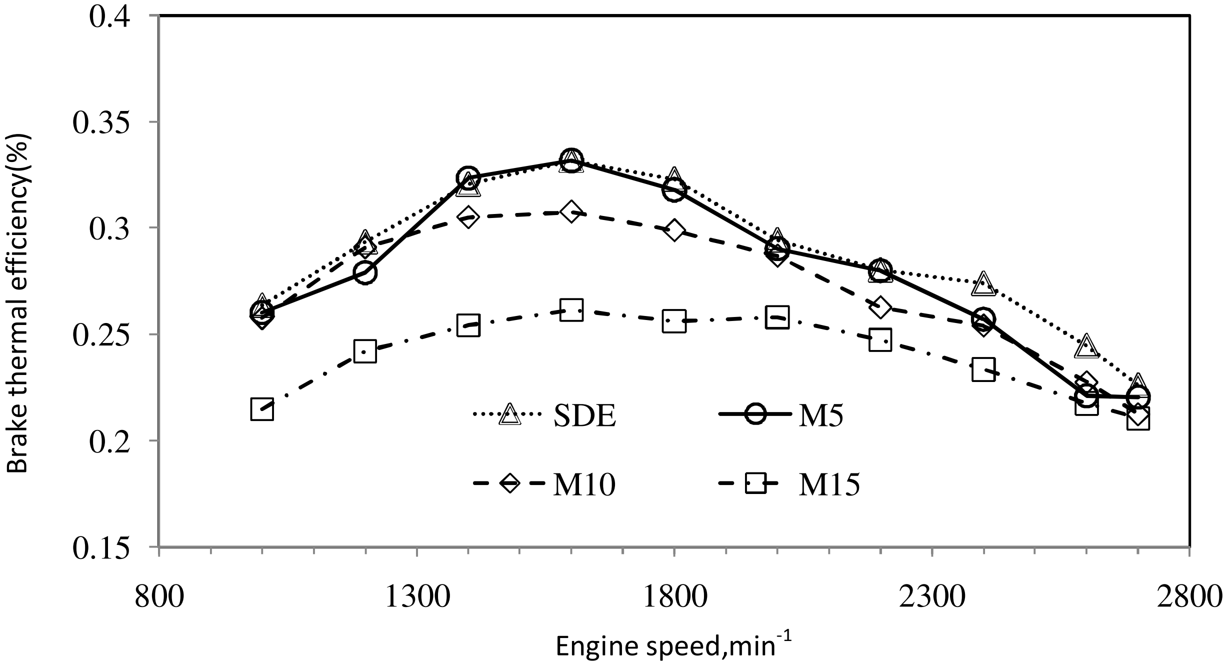


**Figure 3.** Brake thermal efficiency of the test engine at different engine speeds and full load with diesel methanol blends.

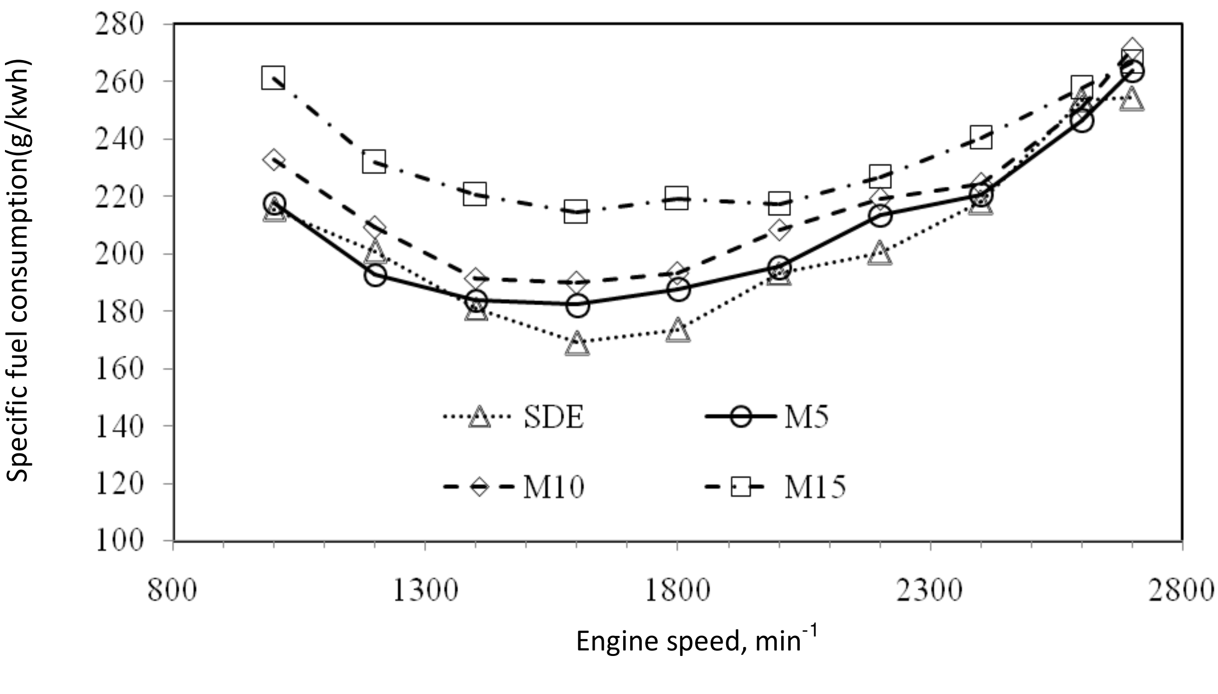


**Figure 4.** Specific fuel consumption amounts of the test engine at different engine speeds and full load with diesel methanol blends.

differences in some of physical properties of the fuel such as density and lower calorific values. The densities of M5, M10 and M15 are slightly lower than SDE which are 0.8375, 0.835 and 0.8325 kg/l respectively. Calorific

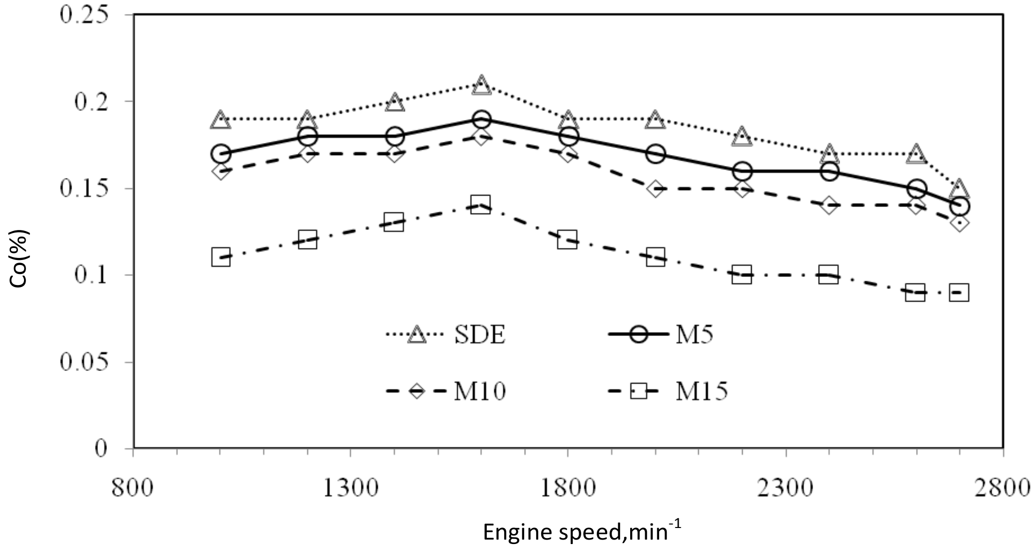


**Figure 5.** CO emission values of the test engine at different engine speeds and full load with diesel methanol blends.

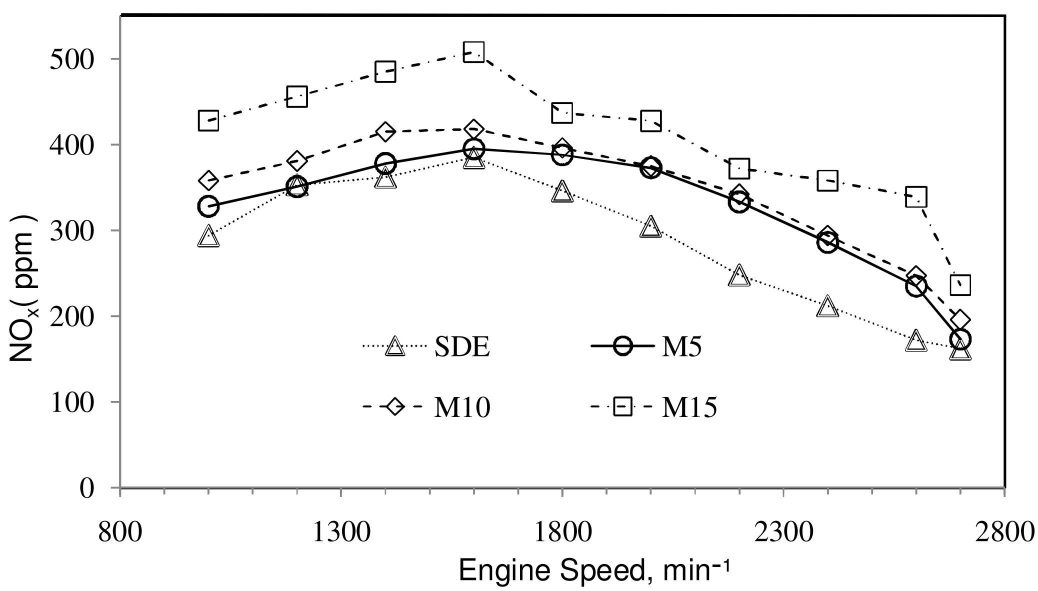


**Figure 6.** $NO_x$ emission values of the test engine at different engine speeds and full load with diesel methanol blends.

values decrease similarly with densities.

When Figure 2 is investigated, two obvious facts can be observed. First one is that methanol increase in diesel blends caused a decrease in engine power output. This decrease isn't directly proportional with methanol percentage in fuel blend also forms second fact. For instance, power decrease of M15 is five times bigger than M10 at 1600 $min^{-1}$ while it is nearly same at 2700 $min^{-1}$. In other words, much more methanol is required at high engine speeds due to short combustion time despite increased fuel amount. It can be determined that engine speed interval at intensively operation area should be carefully decided to select proper blend. For example, M15 usage is more advantageous at higher engine speeds because of its exhaust emission behaviors with small power loss relative to SDE while M5 usage is more

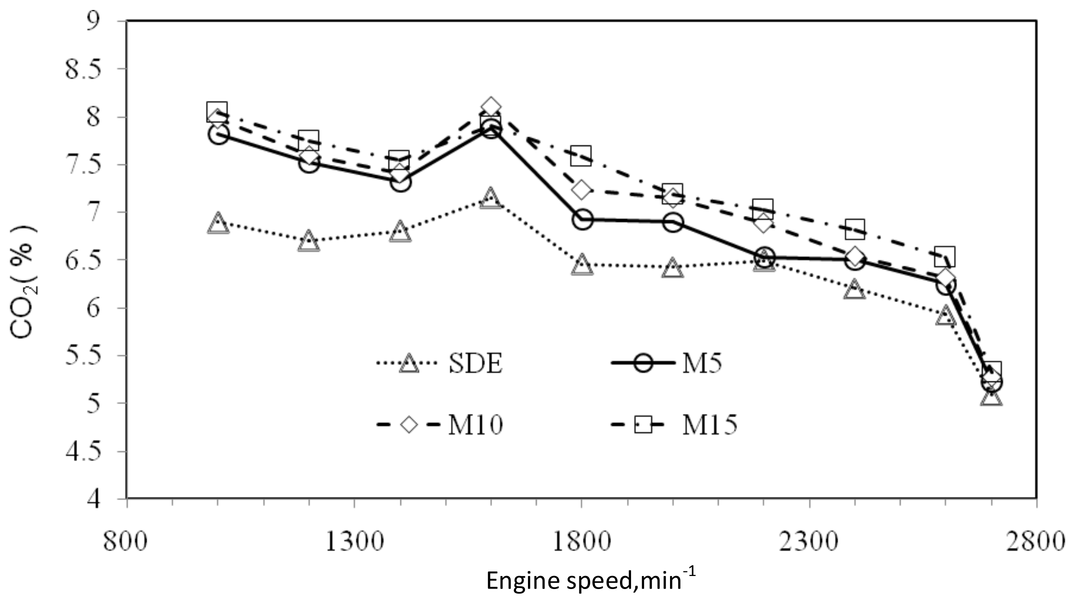


**Figure 7.** $CO_2$ emission values of the test engine at different engine speeds and full load with diesel methanol blends.

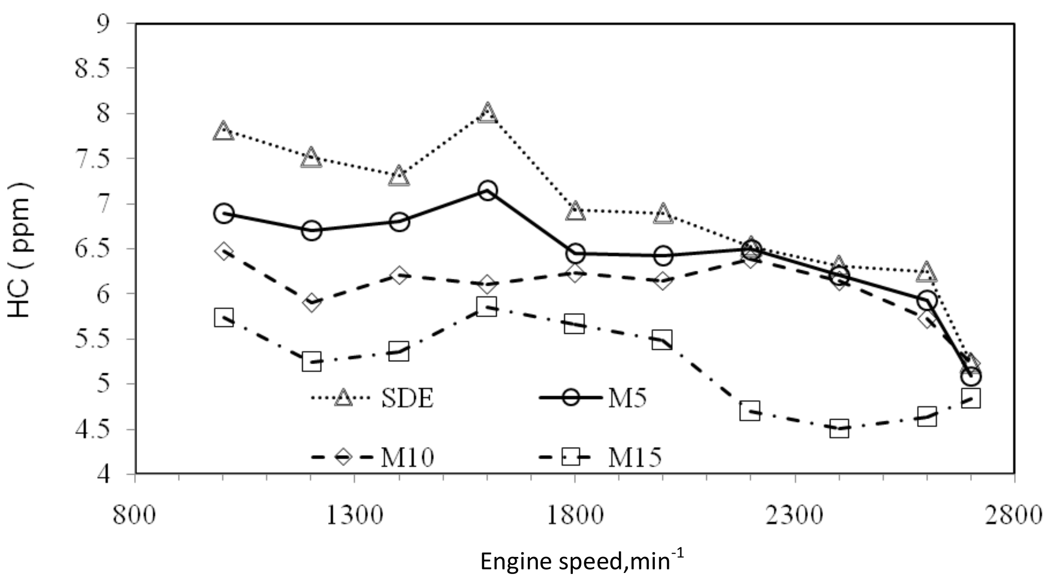


**Figure 8.** HC emission values of the test engine at different engine speeds and full load with diesel methanol blends.

advantageous at lower engine speeds because of nearly same power output with SDE.

## Brake specific fuel consumption (BSFC)

As shown in Figure 4, increasing methanol rate causes a decrease in LHV value of the blend which in return increases BSFC. For instance, at 1600 $min^{-1}$, BSFC amounts are 169.21, 182.28, 189.87 and 214.43 g/kWh for SDE, M5, M10 and M15 respectively. According to the figure, pure diesel fuel presents the best BSFC among other blends. However at high speeds of engine, the differences between BSFC values of fuel blends become smaller due to short combustion period in spite of increased fuel amount. In other words, methanol content increase in fuel blend leads to an oxygen increase because of its oxygen atoms in its molecules. By excess oxygen and fast burning methanol molecules, combustion temperature increases. All these factors affect combustion in a better way. As a result of this, BSFC values of methanol blends become closer to pure diesel fuel BSFC at high engine speeds.

### Brake thermal efficiency (BTE)

From the previous discussion, it could be concluded that as the methanol amount increases in the fuel blend, the BSFC increases, since the LHV value of the blend decreases. As mentioned above, BTE is a function of BSFC and LHV of the blend for a constant effective power. It is clear that BSFC is more effective than LHV with regard to increasing BTE. Therefore the BTE decreased as the methanol content increased (Sayin et al., 2009). The maximum BTE was recorded with SDE for the speed of the maximum torque of the engine ($1600 min^{-1}$). This is particularly due to higher LHV of SDE comparing with methanol blends. It can be concluded from the Figure 3 that M5 fuel blend is more preferable from BTE aspect.

### Exhaust emissions

Diesel engine emissions can be improved by adding methanol to diesel fuel. Because methanol has higher stoichiometric fuel/air ratio than diesel fuel due to its partially oxidized state or it is an oxygenated fuel, therefore blending it into diesel leads to the leaner operation. The leaner operation can result in some improvements in engine performance parameters (Sayin et al., 2009). Also methanol has different physical and chemical properties that are affect combustion such as density, LHV, flame speed and etc. Measured exhaust gasses from experiments at full engine load and different engine speeds at steady state circumstances were recorded and explained thus.

#### *Carbon monoxide (CO) emissions*

CO is a colorless, odorless, poisonous gas, and it must be restricted. CO results from incomplete combustion of fuel and is emitted directly from vehicle tailpipes. Besides the ideal combustion process that combines carbon (C) and oxygen ($O_2$) to $CO_2$, incomplete combustion of carbon leads to the formation of CO. The formation of CO takes place when the oxygen presents during combustion is insufficient to form $CO_2$ (Canakci et al., 2009). The results showed that when the methanol ratio in the blends increased, the CO concentrations in the exhaust emissions were decreased. This was the result of improving combustion process since oxygen content in the methanol caused better combustion. Comparing with the SDE there was an average of reduction in CO emission 33.3, 14.2 and 4.76% for M15, M10 and M5, respectively, at 1600 $min^{-1}$. At maximum torque (1600 min-1), CO percentages were found as 14, 18, 19 and 21% for M10, M15, M5 and SDE, respectively. Since methanol has lower carbon and higher oxygen content, more methanol in the blend results in less CO in exhaust emissions. A falling trend of CO emission can be observed from the figure as the engine speed increases. This is particularly due to turbo charging system which effectively takes part in volumetric efficiency. For instance volumetric efficiency was found as 70.05% for M10 at 1600 $min^{-1}$ while it was found as 76.59% for M10 at 2700 $min^{-1}$. As a result of improved combustion, volumetric efficiency was increased. Similar CO reduction with methanol-diesel fuel blends is also reported by some investigators (Sayin, 2010).

### Nitrogen oxides ($NO_x$) emissions

One of the most critical emissions from CI engines is $NO_x$ emissions. The oxides of nitrogen in the exhaust emissions contain nitric oxide (NO) and nitrogen dioxide ($NO_2$). The formation of $NO_x$ is highly dependent on in-cylinder temperature, oxygen concentration and residence time for the reaction to take place (Sayin et al., 2009). The changes on the $NO_x$ emissions at different engine speeds are shown at Figure 6, for diesel fuel and fuel blends. The experimental results indicate that $NO_x$ values of M15 fuel blend are higher than the others. Maximum $NO_x$ was observed as 508 ppm with M15 and 418, 395 and 385 ppm with M10, M5 and SDE, respectively at 1600 $min^{-1}$. By a different expression, $NO_x$ concentration in exhaust emissions were increased 38, 8 and 2.5% for M15, M10 and M5 respectively according to SDE. In a similar work, same trends were obtained as methanol content was increased (Kulakoğlu, 2009). $NO_x$ concentration generally increased with increased engine speed. However, after the speed of the maximum torque of the engine (1600 $min^{-1}$), they started to decrease.

Methanol contains 34% oxygen and its cetane number is lower than diesel fuel, which increases peak temperatures in the cylinder. On the other hand, the LHV of methanol is nearly two times lower than diesel fuel and the latent heat of vaporization of methanol is about of four times greater than diesel fuel, which decreases peak temperature in the cylinder (Sayin et al., 2009). As seen from the figure, the former effects are more effective than latter ones. Because $NO_x$ formation takes place at higher in cylinder temperatures and the highest $NO_x$ emission value is belong to M15 fuel blend.

### *Carbon dioxide ($CO_2$) emissions*

$CO_2$ is a normal product of combustion. Ideally, combustion of a hydrocarbon fuel should produce only $CO_2$ and water (H2O) (Sayin et al., 2009). Hence, $CO_2$ amount increases while HC and CO emission amounts decreasing and this is an indication of a successful combustion. Consequently, a $CO_2$ chart can be used to present a different side of view to combustion process of the specific fuel although its graphic and tendency tells the same with HC and CO graphics for experienced eyes. Another result that can be drawn from the figure is difference between methanol blends and base fuel. Especially at early engine speeds (800 to 1000 $min^{-1}$), methanol increases $CO_2$ rate among other exhaust emissions. Between 2000 to 2700 $min^{-1}$ the difference become smaller due to insufficient combustion time.

### *Hydrocarbon (HC) emissions*

Most of the UHC are caused by unburned fuel air mixture, whereas the other sources are the engine lubricant and incomplete combustion. The term UHC means organic compounds in the gaseous state; solid HCs are the part of the particulate matter. It can be named as total hydrocarbon emission (THC) here. Typically, HCs are a serious problem at low loads in CI engines. At low loads, the fuel is less apt to impinge on surfaces; but, because of poor fuel distribution, large amounts of excess air and low exhaust temperature, lean fuel air mixture regions may survive to escape into the exhaust (Canakci et al., 2009).

When methanol is added to the diesel fuel, it provides more oxygen for the combustion process and leads to the improving combustion. Improved combustion in rich mixture areas leads to higher temperatures which inreturn affects the whole combustion. For instance, particulate matters which occur in lean fuel air mixtures due to low temperatures, become lesser than before because of increasing temperature. In addition, methanol molecules are polar and cannot be absorbed easily by the non-polar lubrication oil; and therefore methanol can lower the possibility of the production of UHC emissions (Canakci et al., 2009). It might be hypothesized that addition of methanol to diesel fuel improves THC oxidation due to the high temperature in the cylinder to make the fuel be easier to react with oxygen. At the same time, laminar flame speed is seen to increase for alcohols compared with diesel fuel. The increase in the flame speed will reduce combustion duration but increase combustion temperature. The higher combustion temperature promotes more complete combustion and hence there are less THC emissions (Sayin, 2010).The results obtained in this study confirm these statements. The engine speed is also important since it affects combustion process. In the case of short combustion process at high speeds, the amount of HC will also decrease (Sayin, 2010). For instance, HC emission was found 6.48 ppm with M10 at 1000 rpm, and it was 5.9 at 2550 rpm for current work. Previous research showed that HC amount decreased similarly when methanol amount was increasing (Ilhan, 2007).

From the Figure 8, HC emissions seem to be decreasing while methanol amount are increasing. There should be three main reasons for the explanation of this event. First and the most important one is hydrogen and carbon amount in methanol and diesel molecules. Methanol molecules have 7 times less hydrogen and 14 times less carbon than diesel molecules. Thus increasing methanol amount causes a decrease in HC amount which in return decrease HC emission. The second effect occurs due to relatively high combustion temperature of blended fuel. Lean HC-air mixture areas start to be burned when sufficient temperature is attained. After these, oxygen amount of methanol provide required oxygen to fuel rich areas of combustion chamber. Hence, combustion is improved and HC amount tends to be decreased.

## Conclusions

In this work, duel fuel opportunities of methanol/diesel fuel blends in compression ignition engines were investigated. For this purpose, a four cylinder, four liter compression ignition diesel engine was utilized. Experiments were performed at steady state conditions of 10 different point of engine speed for attaining performance and exhaust emission values.The main results are summarized as follows:

1) Engine power is decreasing while methanol content increasing in fuel blend. However this decrease becomes smaller at high engine speeds.
2) When methanol is applied to the diesel engine, there is a decrease in brake thermal efficiency at low engine speeds but there is no significant change at medium to high engine speeds.
3) The BSFC with the all fuel blends increased mainly due to the lower LHV of methanol. The increase in BSFC is proportional with methanol content.
4) Increasing methanol mass fraction in the diesel/methanol blends resulted in a decrease in HC and CO emissions. Inversely $NO_x$ emissions are increased with increasing methanol rate.

## ACKNOWLEDGEMENT

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**Abbreviations: BSFC,** Brake specific fuel consumption (g/kWh); **LHV,** lower heating value (MJ); **BTE,** brake thermal efficiency (%); **MIN,** minute; **CI,** compression ignition; **M5,** 5% methanol content; **CO,** carbon monoxide (%); **M10,** 10% methanol content; **$CO_2$,** carbon dioxide (%); **M15,** 15% methanol content; **DI,** direct injection; **NOx,** nitrogen oxides (ppm); **DMCC,** diesel-methanol compound combustion; PM, particulate matter; **DOC,** diesel oxidation catalyst; **SDE,** standard diesel engine; **HC,** hydro-carbons (ppm); **THC,** total hydro-carbons; **ICE,** internal combustion engines; **UHC,** unburned hydro-carbons.

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