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Comparison of Ethanol and Methanol Blending with Gasoline Using Engine Simulation

Simeon Iliev

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Abstract

During the last years, concerns regarding climate change, decline of energy security, and hydrocarbon reserves have resulted in a wide interest in renewable alternative sources for transportation fuels. Methanol and ethanol have been possible candidates as alternative fuels for the internal combustion engines because they are liquid and have several physical and combustion properties which resemble those of gasoline. Therefore, the aim of this study is to develop the one-dimensional model of a gasoline engine for predicting the effect of various fuel types on engine performances, specific fuel consumption, and emissions. Commercial software AVL BOOST was used to examine the engine characteristics for different blends of methanol, ethanol, and gasoline (by volume). A comparison was made between the results gained from the engine simulation of different fuel blends and those of gasoline. They show that when blended fuel was used, the engine brake power decreased and the BSFC increased compared to those of gasoline fuel. When blended fuel increases, the CO and HC emissions decrease, and there is a major increase in NOx emissions when blended fuel increases up to 30% M30 (E30). Increase in the percentage of ethanol and methanol leads to a significant increase in NOx emissions.

Keywords: alternative fuels, ethanol blends, methanol blends, engine simulation, spark-ignition engine, emissions

1. Introduction

In the last years, the problem with crude oil depletion has arisen. Intensive research has been carried out to find out alternative to fossil fuels. Alternative fuels are derived from resources different from petroleum. When used in internal combustion engines (ICE), these fuels generate lower air pollutants compared to petrol fuel, and a majority of them are more economically

beneficial compared to fossils fuels. They are also renewable. The most common fuels that are used as alternative fuels are natural gas, propane, methanol, ethanol, and hydrogen. Regarding engine operating with blended fuels, a lot of papers have been written about these blended fuels; but a small number of works have compared some of these fuels together in the same engine [1–4]. Low contents of ethanol or methanol have been added to gasoline since at least the 1970s, when there was a reduction in oil supplies and scientists began searching for alternative energy carriers in order to replace petrol fuels. In the beginning, ethanol and methanol were thought to be the most attractive alcohols to be added to gasoline. Ethanol and methanol can be manufactured from natural products or waste materials, whereas gasoline fuel which is a nonrenewable energy resource cannot be manufactured [5, 6]. An important feature is that methanol and ethanol can be used without requiring any significant changes in the structure of the engine. Being part of the various alcohols, ethanol and methanol are known as the most suitable fuels for spark-ignition (SI) engines.

The use of blended fuels is crucial since many of these blends can be used in engines with the aim to improve its performance, efficiency, and emissions. The oxygenates are one of the most important fuel additives to improve fuel efficiency (organic oxygen-containing compounds). A few oxygenates have been used as fuel additives, such as ethanol, methanol, methyl tertiary butyl alcohol, and tertiary butyl ether [7]. The process of using oxygenates makes more oxygen available in the combustion process and has a great potential to reduce SI engine exhaust emissions.

Regarding the combustion process, the flash point and autoignition temperature of methanol and ethanol are higher than pure gasoline, which makes it safer for storage and transportation. The latent heat of ethanol of evaporation is three to five times higher than pure gasoline; this leads to increase the volumetric efficiency because temperature of the intake manifold is lower. The heating value of ethanol is lower than gasoline. Consequently, 1.6 times more alcohol fuel is needed to achieve the exact same energy output. The stoichiometric air-fuel ratio of ethanol is around two-third of the pure gasoline; therefore, for complete combustion, the needed amount of air is lesser for ethanol [8]. Ethanol has several advantages compared to gasoline, e.g., lowering of unburned HC emissions, CO, and much better antiknock characteristics [9]. Ethanol and methanol have a lot higher octane number compared to pure gasoline fuel [10]. This enables higher compression ratios of engines and, as a result, increases its thermal efficiency [11]. The production of methanol can be from natural gas at no great cost and is easy to blend with gasoline fuel. These properties of methanol make it as an attractive additive. Methanol is aggressive to some materials, like plastic components and some of the metals in the fuel system. When using methanol it is necessary that precautions had to be taken when handling it [12].

There are many publications with different blends of alcohols and gasoline fuel. For example, Palmer [13] examined the influence of blends of ethanol and gasoline in spark-ignition engine. The obtained results pointed out that ethanol addition (10%) leads to 5% increase in the engine power and 5% octane number increase for each 10% ethanol added. The result showed that 10% of ethanol addition to gasoline fuel lead to reduction the emissions of CO up to 30%. In another study, Bata et al. [9] examined different blends of ethanol and gasoline and discovered that ethanol reduced the UHC and CO emissions. The lowered CO emissions are caused by the oxygenated characteristic and wide flammability of ethanol. Other

researchers [14] studied that the potentialities for ethanol production are equivalent to about 32% of the total gasoline consumption worldwide, when used in 85% ethanol in gasoline for a passenger vehicle. In another study, Shenghua et al. [15] examined a gasoline engine with various percentages of methanol blends (from 10 to 30%) in gasoline. The results showed that engine torque and power decreased, whereas the brake thermal efficiency improved with the increase of methanol percentage in the fuel blend. Other authors [16] have studied the influence of methanol-gasoline blends on the gasoline engine performance. The results showed that the highest brake mean effective pressure (BMEP) was obtained from 5% methanol-gasoline blend. In another study, Altun et al. [17] researched the influence of 5 and 10% methanol and ethanol blending in gasoline fuel on engine performance and emissions. The best result in emissions showed blended fuels. The HC emissions of E10 and M10 are reduced by 13 and 15% and the CO emissions by 10.6 and 9.8%, respectively. An increased CO_2 emission for E10 and M10 was observed. The methanol and ethanol addition to gasoline showed an increase in the brake-specific fuel consumption (BSFC) and a decrease in break thermal efficiency compared to gasoline.

It can be seen in the literature survey that the exhaust emissions for ethanol-gasoline and methanol-gasoline blends are lower than that of pure gasoline fuel [9, 13, 14, 17]. The engine performance and exhaust emissions with ethanol-gasoline blends resemble those with methanol-gasoline blends.

From the reviewed literature, a conclusion was made that the exhaust emission and engine performance of various blends of methanol and ethanol in gasoline engines have not been investigated sufficiently. Therefore, the objective of this work is to investigate the effects of methanol-gasoline and ethanol-gasoline fuel blends on the performance and exhaust emissions of a gasoline engine under various engine speeds, comparing them with those of pure gasoline.

The simulation tools are the most used in recent years owing to the continuous increase in computational power. The use of engine simulations enables optimization of engine combustion, geometry, and operating characteristics toward improving specific fuel consumption and exhaust emissions and reducing engine development time and costs. Consequently, it can be expected that the use of engine simulations during engine construction will continue to increase. Engine modeling is a fruitful research area, and therefore many laboratories have their own engine thermodynamic models with varying degrees of complexity, scope, and ease to use [18].

Computer simulation is becoming an important tool for time and cost efficiency in engine's development. The simulation results are challenging to be obtained experimentally. Using computational fluid dynamics (CFD) has allowed researchers to understand the flow behavior and quantify important flow parameters such as mass flow rates or pressure drops, under the condition that the CFD tools have been properly validated against experimental results. Many processes in the engine are three-dimensional; however, it requires greater knowledge and large computational time. Thus, simplified one-dimensional simulation is occasionally used. Hence, simulating the complex components by means of a three-dimensional code and modeling the rest of the system with a one-dimensional code are the right choice to save computational time, i.e., the ducts. This way, a coupling methodology between the one-dimensional and the three-dimensional codes in the respective interfaces is necessary and has become the aim of numerous authors [19–21].

2. Research methodology

The aim of the present chapter is to develop the one-dimensional model of four-stroke port fuel injection (PFI) gasoline engine for predicting the effect of methanol-gasoline (M0–M50) and ethanol-gasoline (E0–E50) addition to gasoline on the exhaust emissions and performance of gasoline engine. For this, simulation of gasoline SI engine (calibrated) as the basic operating condition and the laminar burning velocity correlations of methanol-gasoline and ethanol-gasoline blends for calculating the changed combustion duration was used. The engine power, specific fuel consumption, and exhaust emissions were compared and discussed [22, 23].

2.1. Simulation setup

The one-dimensional SI engine model is created by using the AVL BOOST software and has been employed to examine the performance and emissions working on gasoline, ethanol-gasoline, and methanol-gasoline blends.

In **Figure 1**, PFIE symbolizes the engine, while C1–C4 is the number of cylinders of the SI engine. The cylinders are the main element in this model, because they have many very important parameters to settle: the internal geometry, bore, stroke, connecting rod, length and compression ratio, as well as the piston pin offset and the mean crankcase pressure. The measuring points are marked with MP1–MP18. PL1–PL4 symbolizes the plenum. System boundary stands for SB1 and SB2. CL1 represents the cleaner. R1–R10 stands for flow restrictions. CAT1 symbolizes catalyst and fuel injectors—I1–I4. The flow pipes are numbered 1–34.

The calibrated gasoline engine model was described by Iliev [23], and its layout is shown in **Figure 1** with engine specification shown in **Table 1**.

Table 2 presents a comparison between the properties of gasoline, ethanol, and methanol. As shown in **Table 2**, compared with gasoline and ethanol, methanol has a higher elemental oxygen content and a lower heating value, molecular weight, elemental carbon, hydrogen content, and stoichiometric air/fuel ratio (AFR).

2.2. Combustion model description

In this research, two-zone model of Vibe was chosen for the combustion simulation and analysis. The combustion chamber was divided into two regions: unburned gas region and burned gas regions [17]. For the burned charge and unburned charge, the first law of thermodynamics is applied:

$$\frac{dm_b u_b}{d\alpha} = -p_c \frac{dV_b}{d\alpha} + \frac{dQ_F}{d\alpha} - \sum \frac{dQ_{Wb}}{d\alpha} + h_u \frac{dm_b}{d\alpha} - h_{BB,b} \frac{dm_{BB,b}}{d\alpha} \tag{1}$$

$$\frac{dm_u u_u}{d\alpha} = -p_c \frac{dV_u}{d\alpha} - \sum \frac{dQ_{Wu}}{d\alpha} - h_u \frac{dm_B}{d\alpha} - h_{BB,u} \frac{dm_{BB,u}}{d\alpha} \tag{2}$$
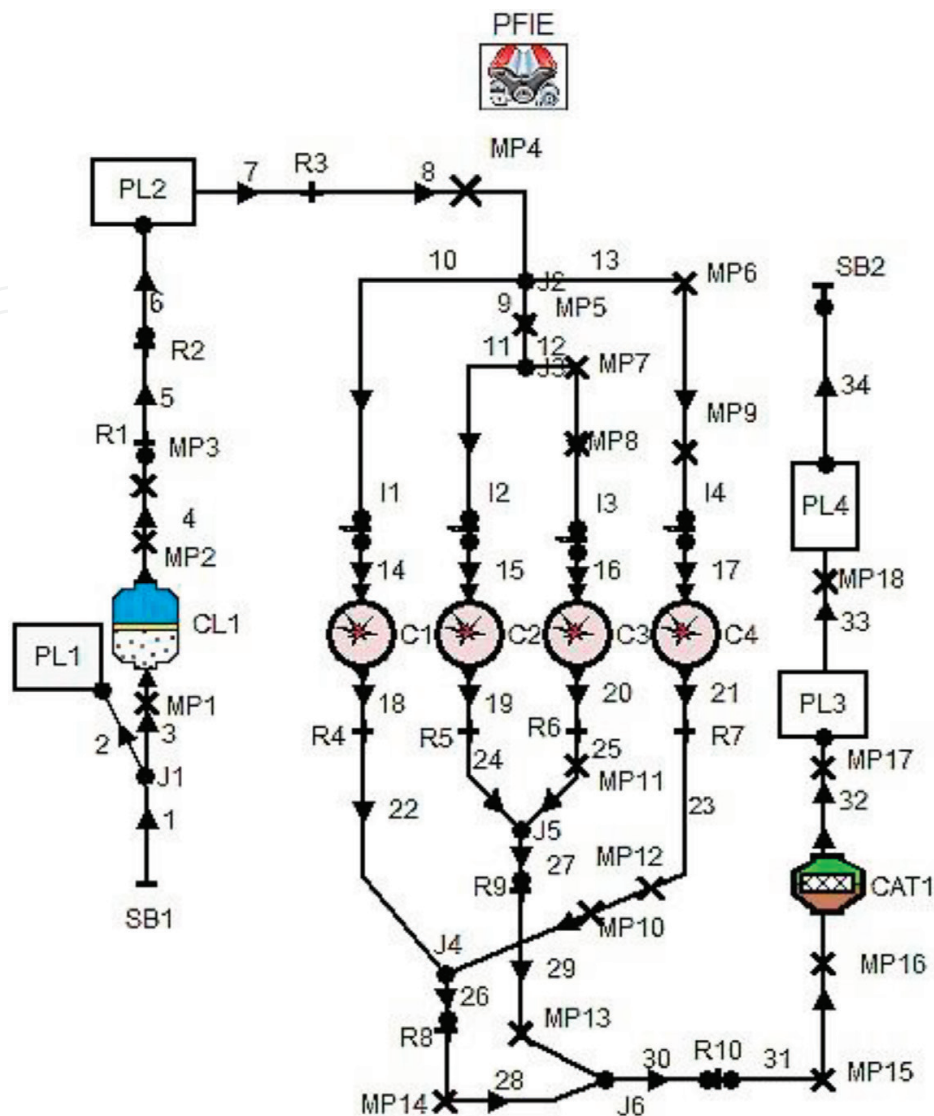
Figure 1. Schematic of the gasoline PFI engine model.

where dm_u represents the change of the internal energy in the cylinder, $p_c \frac{dV}{da}$ is the piston work, $\frac{dQ_F}{da}$ stands for the fuel heat input, $\frac{dQ_W}{da}$ is the wall heat loses, $h_u \frac{dm_b}{da}$ represents the enthalpy flow from the unburned to the burned zone due to the conversion of a fresh charge to combustion products. The heat flux between the two zones is neglected. $h_{BB} \frac{dm_{BB}}{da}$ is the enthalpy flow due to blow-by, **u** and **b** in the subscript are unburned and burned gas.

Moreover, the sum of the zone volumes must be equal to the cylinder volume, and the sum of the volume changes must be equal to the cylinder volume change:

$$\frac{dV_b}{d\alpha} + \frac{dV_u}{d\alpha} = \frac{dV}{d\alpha} \tag{3}$$

$$V_b + V_u = V \tag{4}$$

Engine parameters	Value
Bore	86 (mm)
Stroke	86 (mm)
Compression ratio	10.5
Connection rod length	143.5 (mm)
Number of cylinder	4
Piston pin offset	0 (mm)
Displacement	2000 (cc)
Intake valve open	20 BTDC (deg)
Intake valve close	70 ABDC (deg)
Exhaust valve open	50 BBDC (deg)
Exhaust valve close	30 ATDC (deg)
Piston surface area	5809 (mm^2)
Cylinder surface area	7550 (mm^2)
Number of stroke	4

Table 1. Engine specification.

Properties	Gasoline	Methanol	Ethanol
Chemical formula	C_8H_{15}	CH_3OH	C_2H_5OH
Molecular weight	111.21	32.04	46.07
Oxygen content (wt%)	—	49.93	34.73
Carbon content (wt%)	86.3	37.5	52.2
Hydrogen content (wt%)	24.8	12.5	13.1
Stoichiometric AFR	14.5	6.43	8.94
Lower heating value (MJ/kg)	44.3	20	27
Heat of evaporation (kJ/kg)	305	1.178	840
Research octane number	96.5	112	111
Motor octane number	87.2	91	92
Vapor pressure (psi at 37.7 OC)	4.5	4.6	2
Destiny (g/cm^3)	0.737	0.792	0.785
Normal boiling point (OC)	38–204	64	78
Autoignition temperature (OC)	246–280	470	365

Table 2. Comparison of fuel properties.

The amount of burned mixture at each time setup is obtained from the Vibe function. For all other terms, for instance, wall heat losses, etc., models similar to the single-zone models with an appropriate distribution on the two zones are used [24].

2.3. A description of exhaust emission model

In AVL BOOST, the model of formation on NOx is based on AVL List Gmbh [24], which incorporates the Zeldovich mechanism [25]. The rate of NOx production was obtained using Eq. (5):

$$r_{NO} = C_{PPM} C_{KM}(2,0).(1-\alpha^2).\left(\frac{r_1}{1+\alpha AK_2}+\frac{r_4}{1+AK_4}\right).$$ (5)

where $\alpha = \frac{C_{NO.act}}{C_{NO.equ}}.\frac{1}{C_{KM}}$, $AK_2 = \frac{r_1}{r_2+r_3}$, $AK_4 = \frac{r_4}{r_5+r_6}$.

In the above equation, C_{PPM} represents post-processing multiplier, C_{KM} denotes kinetic multiplier, C stands for molar concentration in equilibrium, and r_i represents reaction rates of Zeldovich mechanism.

The NOx formation model in AVL Boost is based on Onorati et al. [26]:

$$r_{CO} = C_{Const}(r_1+r_2).(1-\alpha)$$ (6)

$$\text{where } \alpha = \frac{C_{CO.act}}{C_{CO.equ}}.$$

In Eq. (6), C represents molar concentration in equilibrium and r_i represents reaction rates based on the model.

The unburned HC has different sources. A complete description of HC formation still cannot be given, and the achievement of a reliable model within a thermodynamic approach is definitely prevented by the fundamental assumptions and the requirement of reduced computational times. Still, a phenomenological model which accounts for the main formation mechanisms and is able to capture the HC trends as function of the engine operating parameter may be proposed. The following important sources of unburned HC can be identified in SI engines [21]:

1. During the intake and compression stroke, fuel vapor is absorbed into the oil layer and deposits on the cylinder walls. The following desorption occurs when the cylinder pressure decreases during the expansion stroke and complete combustion cannot take place anymore.

2. A fraction of the charge enters the crevice volumes and is not burned since the flame quenches at the entrance.

3. Occasional complete misfire or partial burning takes place when combustion quality is poor.

4. Quench layers on the combustion chamber wall which are left as the flame extinguishes prior to reaching the walls.

5. The flow of fuel vapor into the exhaust system during valve overlap in gasoline engines.

The first two mechanisms and in particular the crevice formation are considered to be the most important and need to be accounted for in a thermodynamic model. Partial burn and quench layer effect cannot be physically described in a quasi-dimensional approach, but may be included by adopting tunable semiempirical correlations.

The formation of unburned HC in the crevices is described by assuming that the pressure in the cylinder and in the crevices is the same and that the temperature of the mass in the crevice volumes is equal to the piston temperature.

The mass in the crevices at any time is described by Eq. (7):

$$m_{crevice} = \frac{p V_{crevice} M}{R T_{piston}} \tag{7}$$

In Eq. (7), $m_{crevice}$ represents the mass of unburned charge in the crevice, p denotes cylinder pressure, $V_{crevice}$ stands for total crevice volume, M represents unburned molecular weight, T_{piston} is the temperature of the piston, and R denotes gas constant.

The second important source of HC is the presence of lubricating oil in the fuel or on the walls of the combustion chamber. During the compression stroke, the fuel vapor pressure increases so, by Henry's law, absorption occurs even if the oil was saturated during the intake. During combustion the concentration of fuel vapor in the burned gases goes to zero so the absorbed fuel vapor will desorb from the liquid oil into the burned gases. Fuel solubility is a positive function of the molecular weight, so the oil layer contributed to HC emissions depending on the different solubility of individual hydrocarbons in the lubricating oil.

The assumptions made in the development of the HC absorption/desorption are the following:

1. Fuel is constituted by a single hydrocarbon species, completely vaporized in the fresh mixture.

2. The oil film temperature is at the same as the cylinder wall.

3. Traverse flow across the oil film is negligible.

4. Oil is represented by squalane ($C_{30}H_{62}$), whose characteristics resemble those of the SAE5W20 lubricant.

5. Diffusion of the fuel in the oil film is the limiting factor, for the diffusion constant in the liquid phase which is 104 times smaller than the corresponding value in the gas phase.

The radial distribution of the fuel mass fraction in the oil film can be determined by solving the diffusion Eq. (8):

$$\frac{\partial w_F}{\partial t} - D \frac{\partial^2 w_F}{\partial r^2} = 0 \tag{8}$$

In Eq. (8), w_F represents fuel's mass fraction in the oil film, t is the time, r stands for radial position in the oil film (distance from the wall), and D is relative (fuel-oil) diffusion coefficient.

3. Result and discussion

The present research focused on the performance and emission characteristics of the methanol and ethanol-gasoline blends. Various concentrations of the blends 0% methanol (ethanol) M0 (E0), 5% methanol (ethanol) M5 (E5), 10% methanol (ethanol) M10 (E10), 20% methanol (ethanol) M20 (E20), 30% methanol (ethanol) M30 (E30), 50% methanol (ethanol) M50 (E50), and 85% methanol (ethanol) M85 (E85) by volume were analyzed.

3.1. Engine performance characteristics

The results of the brake power and specific fuel consumption for ethanol-gasoline blended fuels at different engine speeds are shown on **Figures 2** and **3**.

The brake power is one of the important factors that determine the performance of an engine. The variation of brake power with speed was obtained at full load conditions for E5, E10, E20, E30, E50, and pure gasoline E0. The ethanol content in the blended fuel increased, and the brake power decreased for all engine speeds. The gasoline brake power was higher than E5–E50 for all engine speeds. The ethanol's heat of evaporation is higher in comparison to gasoline fuel, providing air-fuel charge cooling and increasing the density of the charge. The blended fuel causes the equivalence ratio of blend approaches to stoichiometric condition which can lead to a better combustion. However, the ethanol heating value is lower compared to gasoline, and it can neutralize the previous positive effects. Consequently, a lower power output is obtained.

Figure 3 shows the changes of the BSFC for ethanol-gasoline blends under various engine speeds. The figure shows that the BSFC increased as the ethanol percentage increased. Heating value and stoichiometric air-fuel ratio are the smallest for these two fuels, which means that for specific air-fuel equivalence ratio, more fuel is needed. The highest specific fuel consumption is obtained at E50 ethanol-gasoline blend.
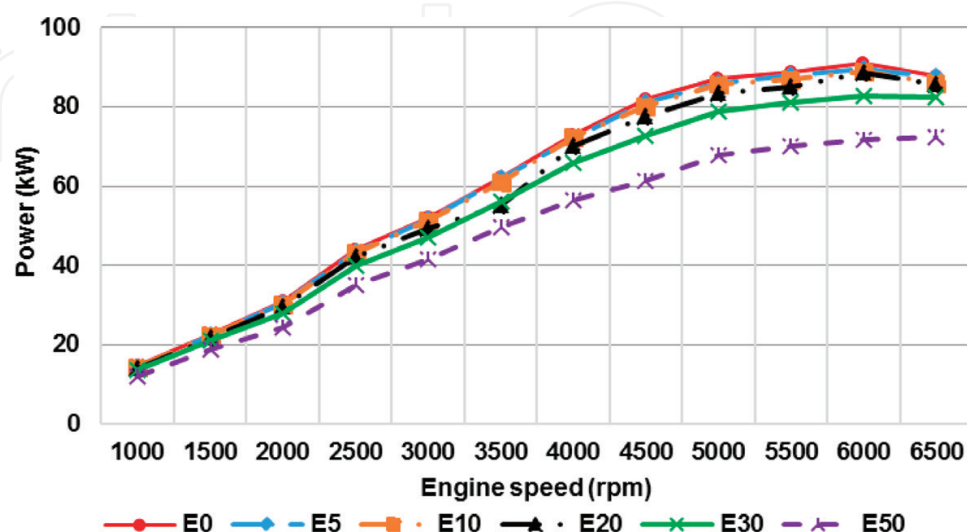

Figure 2. Influence of ethanol-gasoline blended fuels on brake power.
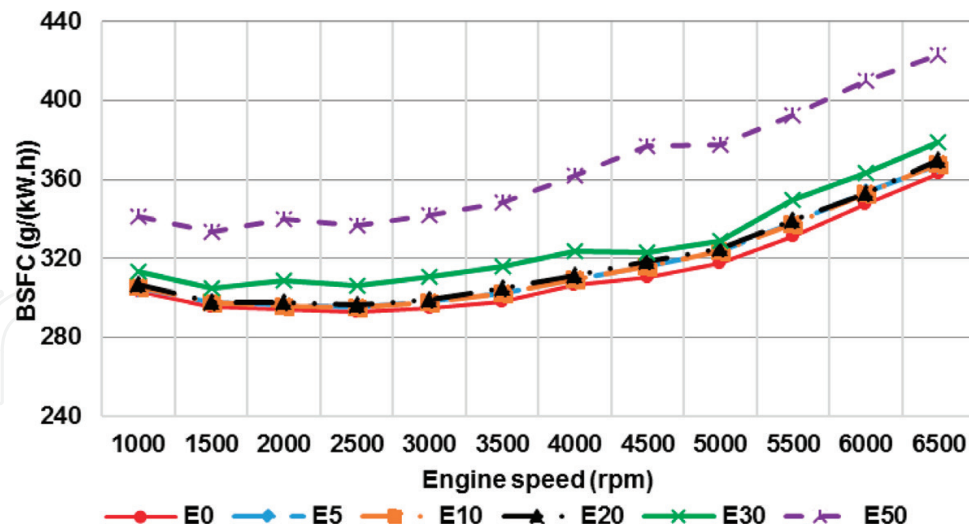
Figure 3. Influence of ethanol-gasoline blended fuels on brake-specific fuel consumption.

Moreover, there is a slight difference between the BSFC when using pure gasoline and when using blends (E5, E10, and E20). The lower energy content of blended fuels causes some increment in BSFC of the engine.

Figure 4 shows the influence of methanol-gasoline blended fuels on engine brake power. The variation of brake power with speed was obtained at full load conditions for M5, M10, M20, M30, M50, and pure gasoline M0. When the methanol content in the blended fuel was increased (M10, M20, and M30), there was not a significant increase in engine brake power.

The engine brake power may be due to the increase of the indicated mean effective pressure for higher methanol content blends. The methanol's heat of evaporation is higher compared to that of gasoline, thus providing air-fuel charge cooling and increasing the density of the charge. Therefore, a higher power output is obtained. The engine brake power was higher in operation with gasoline in comparison to M50 for all engine speeds.
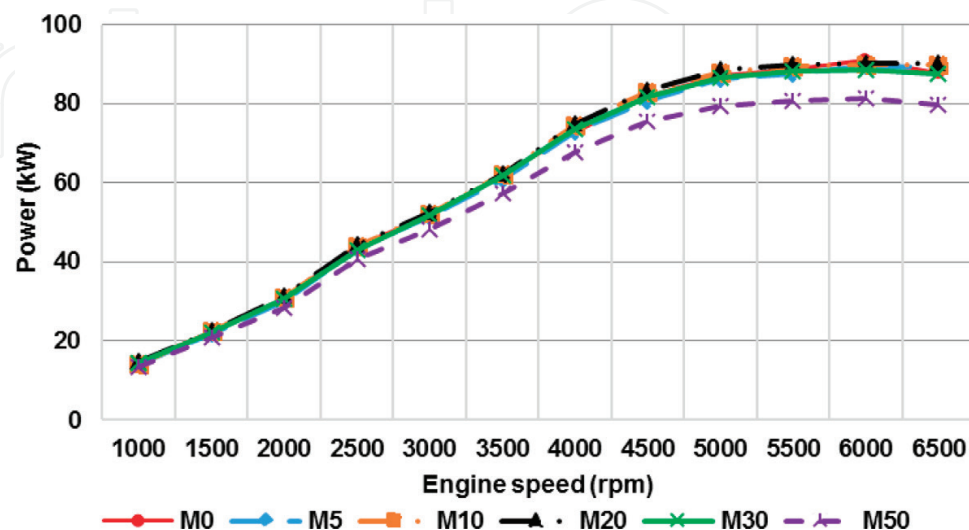
Figure 4. Influence of methanol-gasoline blended fuels on brake power.

Figure 5 shows the variations of the BSFC for methanol-gasoline blended fuels under various engine speeds. As shown in this figure, the BSFC increased as the methanol percentage increased. This can be described with heating value, and stoichiometric air-fuel ratio is the smallest for these two fuels, which means that for specific air-fuel equivalence ratio, more fuel is needed. The specific fuel consumption of M50 methanol-gasoline blend was highest compared to those of gasoline for all engine speeds.

Furthermore, there is a small difference between the BSFC when using gasoline and when using methanol-gasoline blended fuels (M5–M30). As engine speed increased reaching 2000 rpm, the BSFC decreased reaching its minimum value.

The results of the brake power and specific fuel consumption for ethanol- and methanol-gasoline blended fuels at different engine speeds are presented in **Figures 6** and **7**.

When there was an increase in the ethanol content in the blended fuel, the brake power decreased for all engine speeds. The brake power of gasoline fuel was higher than those of E5–E50. The heating value of ethanol is lower than pure gasoline fuel, and the heating value of the blends decreases with the increase of the ethanol percentage. Consequently, a lower power output is obtained [22, 23].

By increasing the percentage of methanol in the blends (M5 and M10), the brake power slightly increased, which can be explained by better combustion efficiency of oxygenated fuels. By increasing the methanol content in the blends (M30 and M50), the engine brake power decreased for all engine speeds. The blended fuel heating value decreases with the increase of the percentage of methanol. This results in a lower power output. The gasoline brake power was higher compared to blend M50.

Figure 7 shows the changes of the BSFC for blended fuels under different engine speeds. The BSFC increased as the ethanol and methanol percentage increased. The reason has been known—the heating value and stoichiometric air-fuel ratio are the smallest for this fuel, which
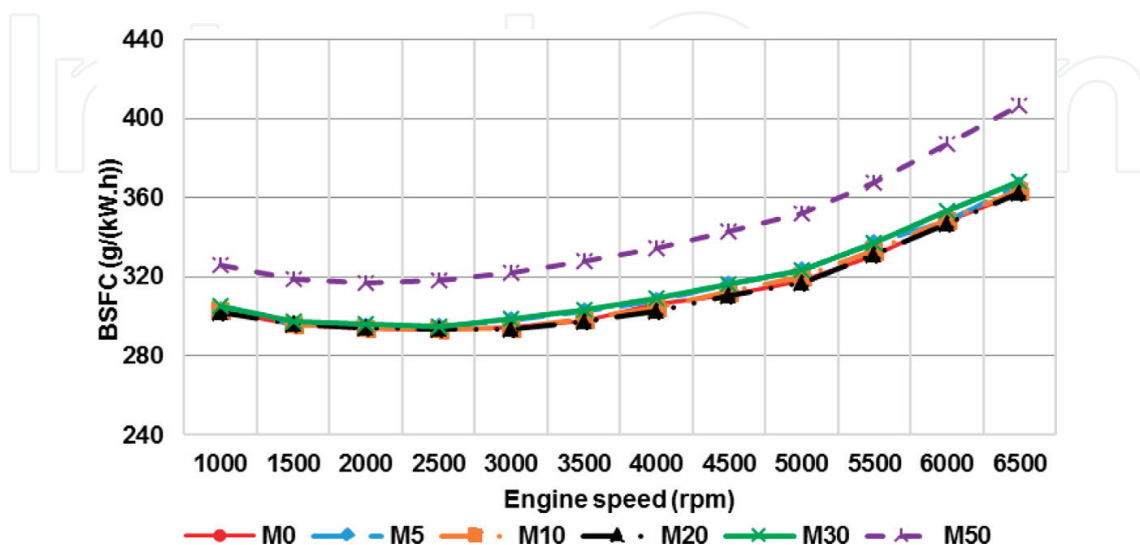

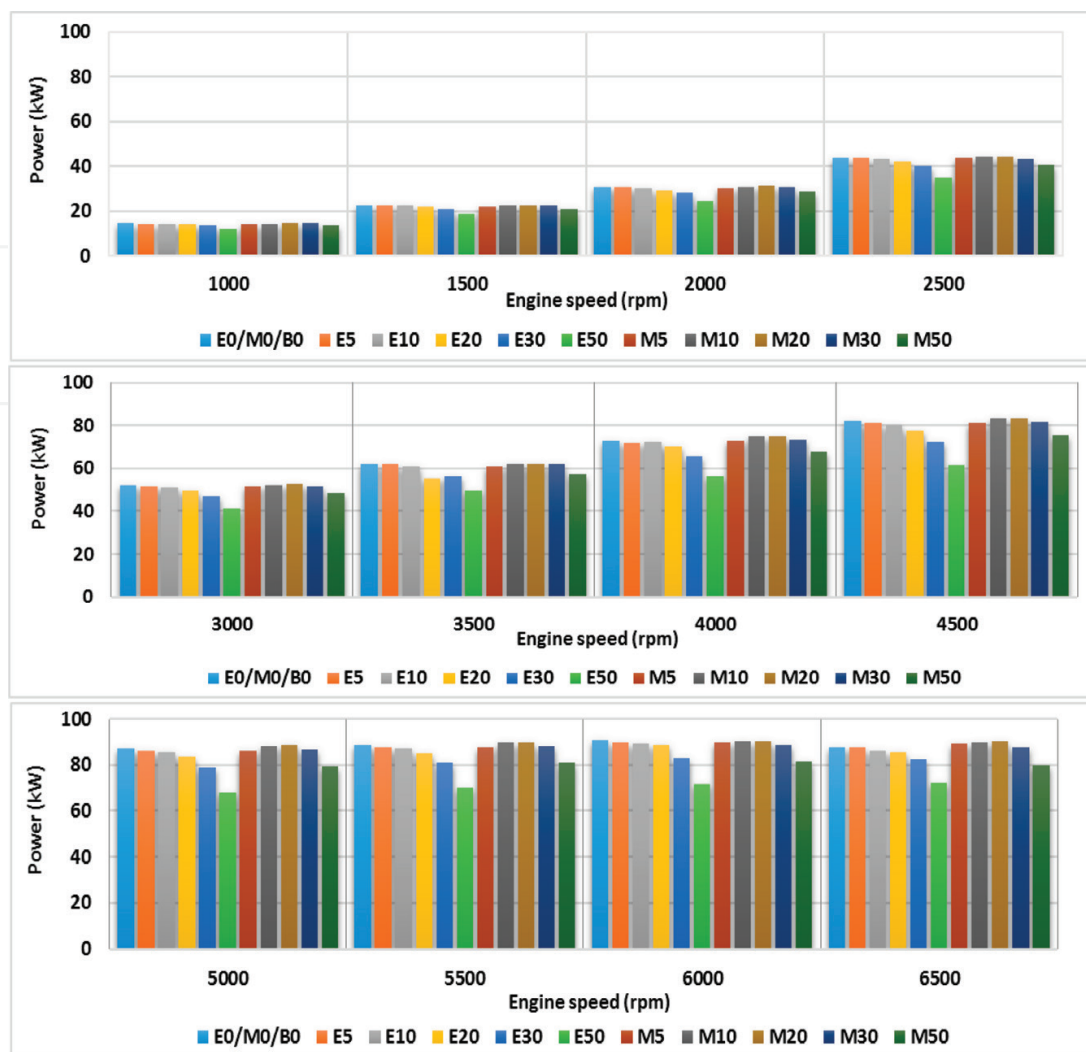**Figure 5.** Influence of methanol-gasoline blended fuels on engine brake power.

Figure 6. Effect of blended fuels on engine brake power.

means that more fuel is needed for specific air-fuel equivalence ratio. The highest specific fuel consumption is obtained at E50 (M50) blended fuel.

What is more, there is small difference between the BSFC when using pure gasoline and blended fuels (E5 (M5), E10 (M10), and E20 (M20)). The lower energy content of ethanol blended fuels makes some increment in BSFC.

3.2. Emission characteristics

The result of the ethanol-blended fuels on CO emissions is shown in **Figure 8**.

A conclusion, which can be made by **Figure 8**, is that when ethanol content increases, the CO emission decreases. The reason for this could be explained with the enrichment of oxygen owing to the ethanol, in which an increase in the proportion of oxygen will promote the further oxidation of CO during the engine exhaust process. One of the other significant reasons for this reduction is that ethanol (C_2H_5OH) has less carbon than gasoline (C_8H_{18}).
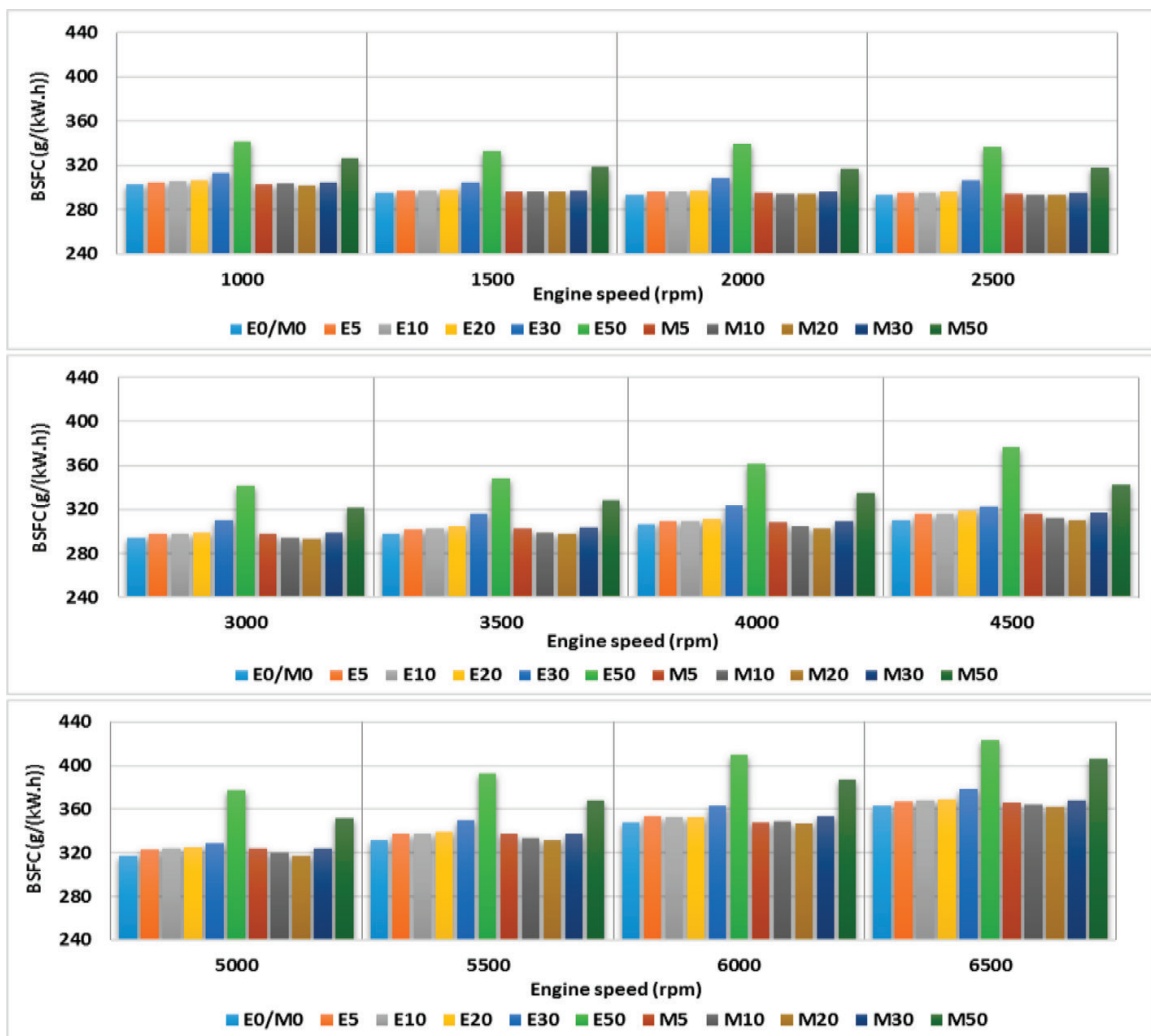
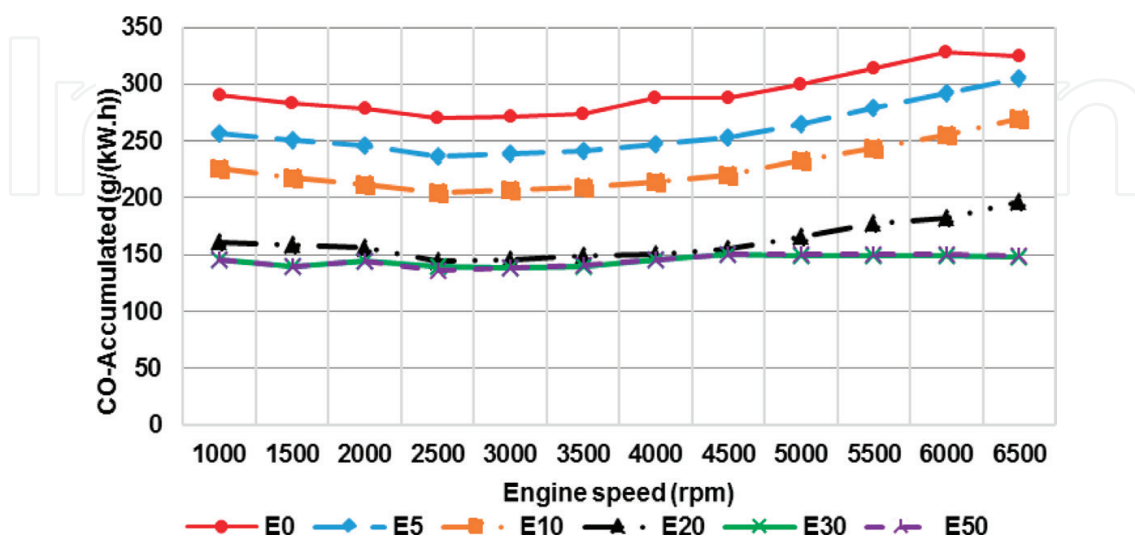**Figure 7.** Influence of blended fuels on engine fuel consumption.

Figure 8. Influence of ethanol-gasoline blended fuels on CO emissions.

The result of the ethanol gasoline blends on HC emissions is shown in **Figure 9.** The figure shows that when ethanol percentage increases, the HC concentration decreases. The HC emission decreases with the increase of the relative air-fuel ratio. The decrease of HC can be explained similarly to that of CO concentration described above.

The effect of the ethanol gasoline blends on NOx emissions for various engine speeds is shown in **Figure 10**. When the combustion process is closer to stoichiometric, flame temperature increases. As a result, the NOx emissions are increased.

The effect of the methanol-gasoline blends on CO emissions for various engine speeds can be seen in **Figure 11**. When methanol percentage increases, the CO concentration decreases. This can be explained with the enrichment of oxygen because of the methanol and less carbon of methanol than gasoline.

The effect of the methanol-gasoline blends on HC emissions is visible in **Figure 12**. When methanol percentage increases, the HC concentration decreases. The concentration of HC emissions decreases with the increase of the relative air-fuel ratio. The reason for the decrease of HC concentration resembles that of ethanol.

The effect of the methanol-gasoline blends on NOx emissions can be seen in **Figure 13**. When methanol percentage increases, the NOx concentration increases. When combustion process is closer to stoichiometric, flame temperature increases and the NOx emissions increase as well.

The effect of the ethanol- and methanol-gasoline blends on CO emissions can be viewed in **Figure 14**. By increasing the methanol and ethanol content in the blended fuel, the CO emission decreases. The reason can be the enrichment of oxygen because of the ethanol and methanol, in which an increase in the proportion of oxygen will promote the further oxidation of CO during the engine exhaust process. Another major reason for this reduction is that ethanol (C_2H_5OH) and methanol (CH_3OH) have less carbon than gasoline (C_8H_{18}). The lowest CO emissions are obtained with blended fuel containing methanol (M50).
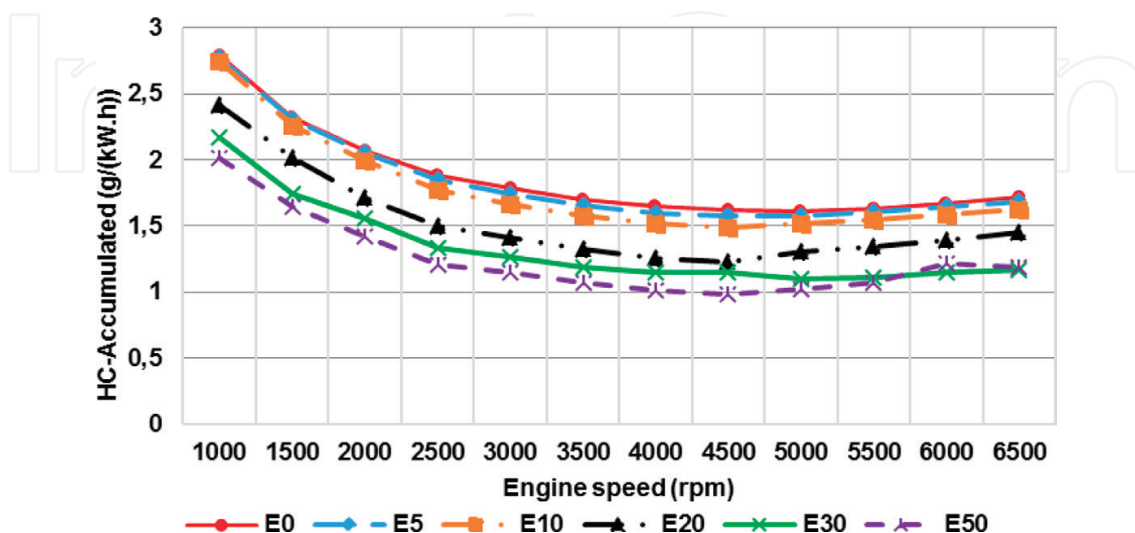

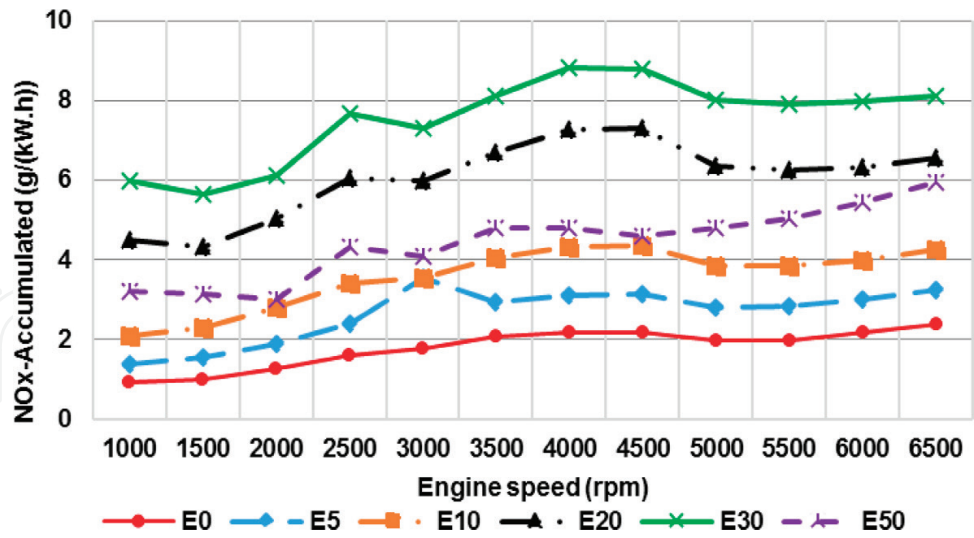**Figure 9.** Influence of ethanol-gasoline blended fuels on HC emissions.

Figure 10. Influence of ethanol-gasoline blended fuels on NOx emissions.
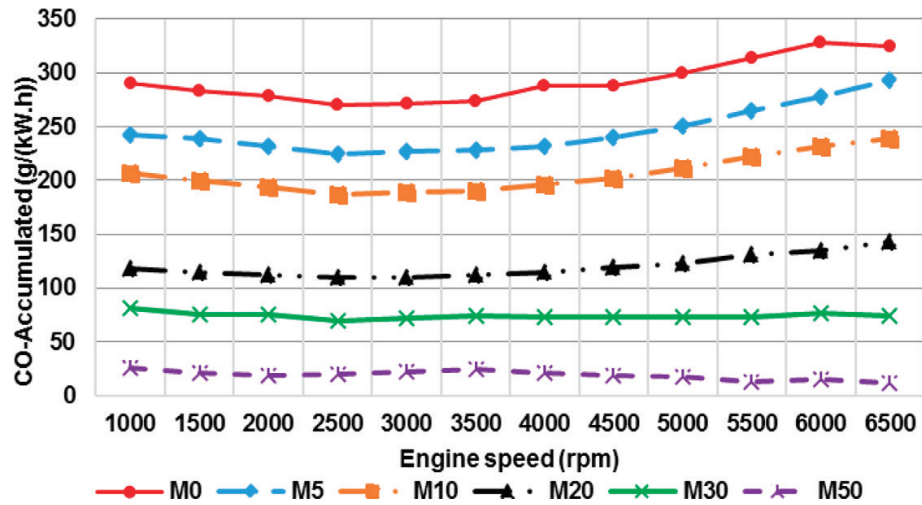

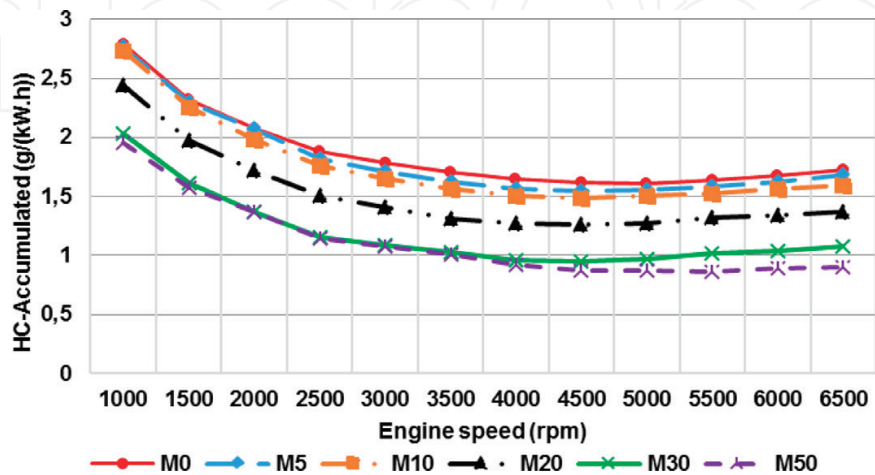**Figure 11.** Influence of methanol-gasoline blended fuels on CO emissions.

Figure 12. Influence of methanol-gasoline blended fuels on HC emissions.
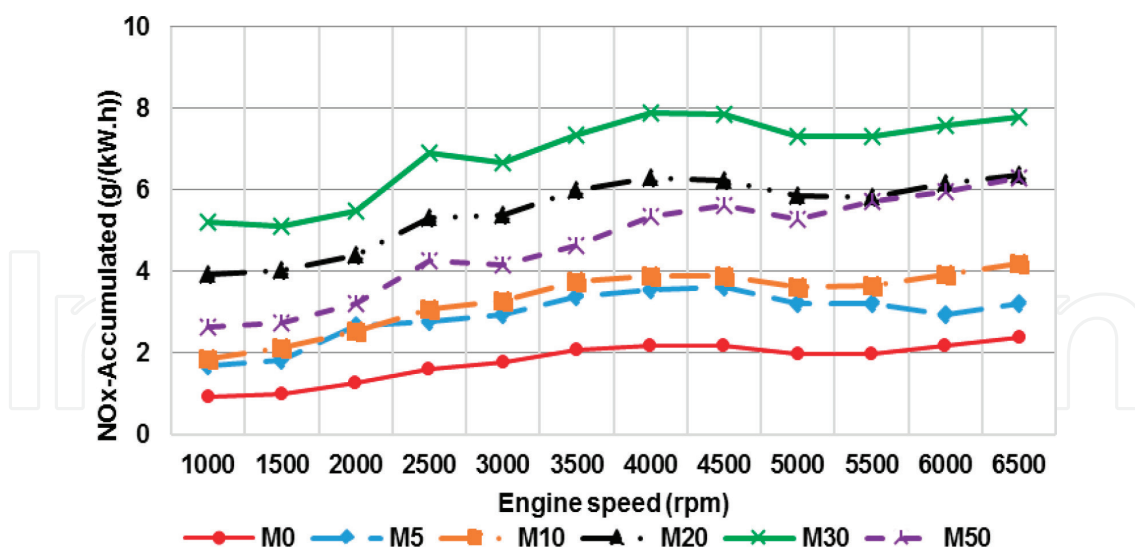
Figure 13. Influence of methanol-gasoline blended fuels on NOx emissions.
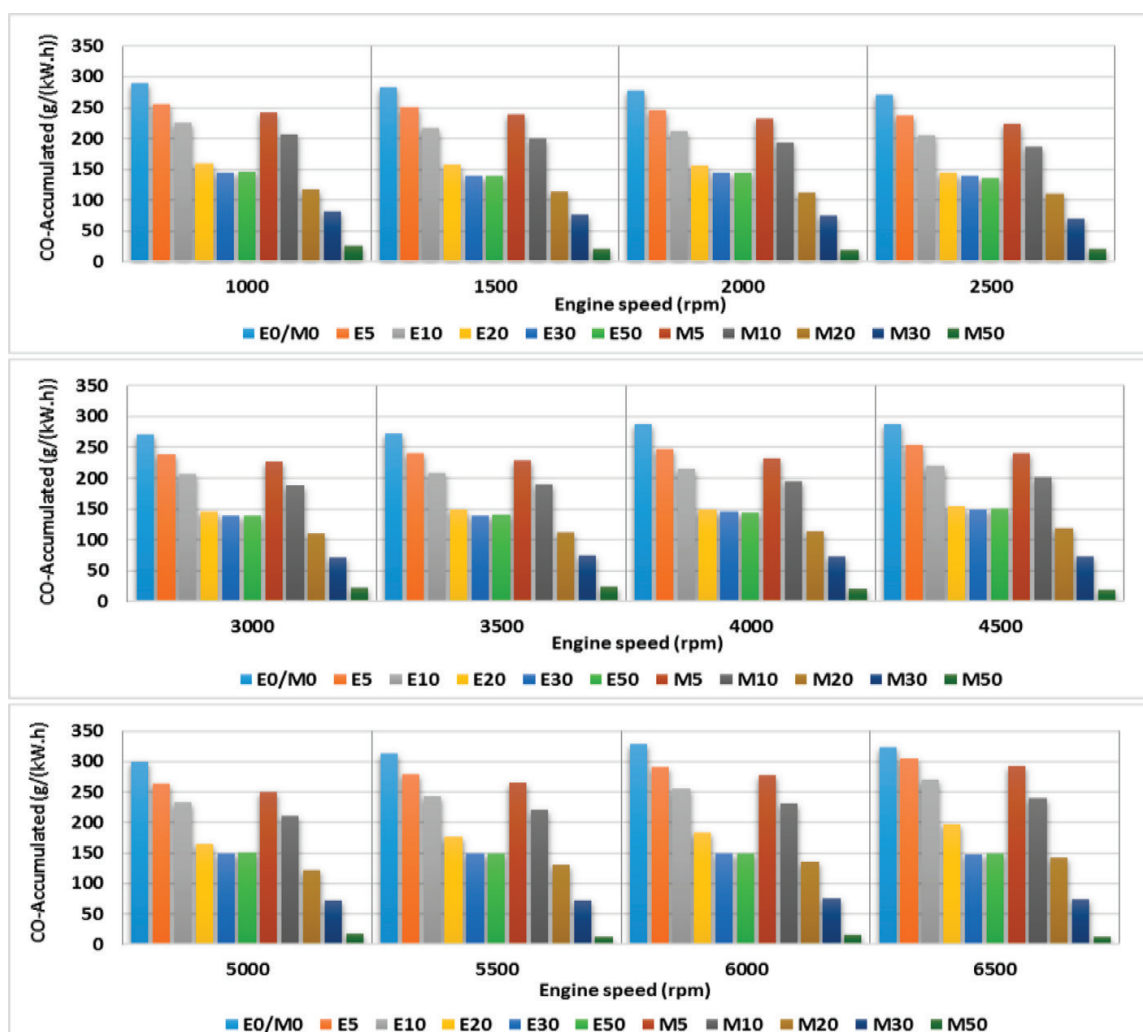

Figure 14. Influence of ethanol- and methanol–gasoline blended fuels on CO emissions.

The effect of the ethanol- and methanol-gasoline blends on HC emissions is visible in **Figure 15**. When there is an increase in the ethanol and methanol percentage, the HC concentration decreases.
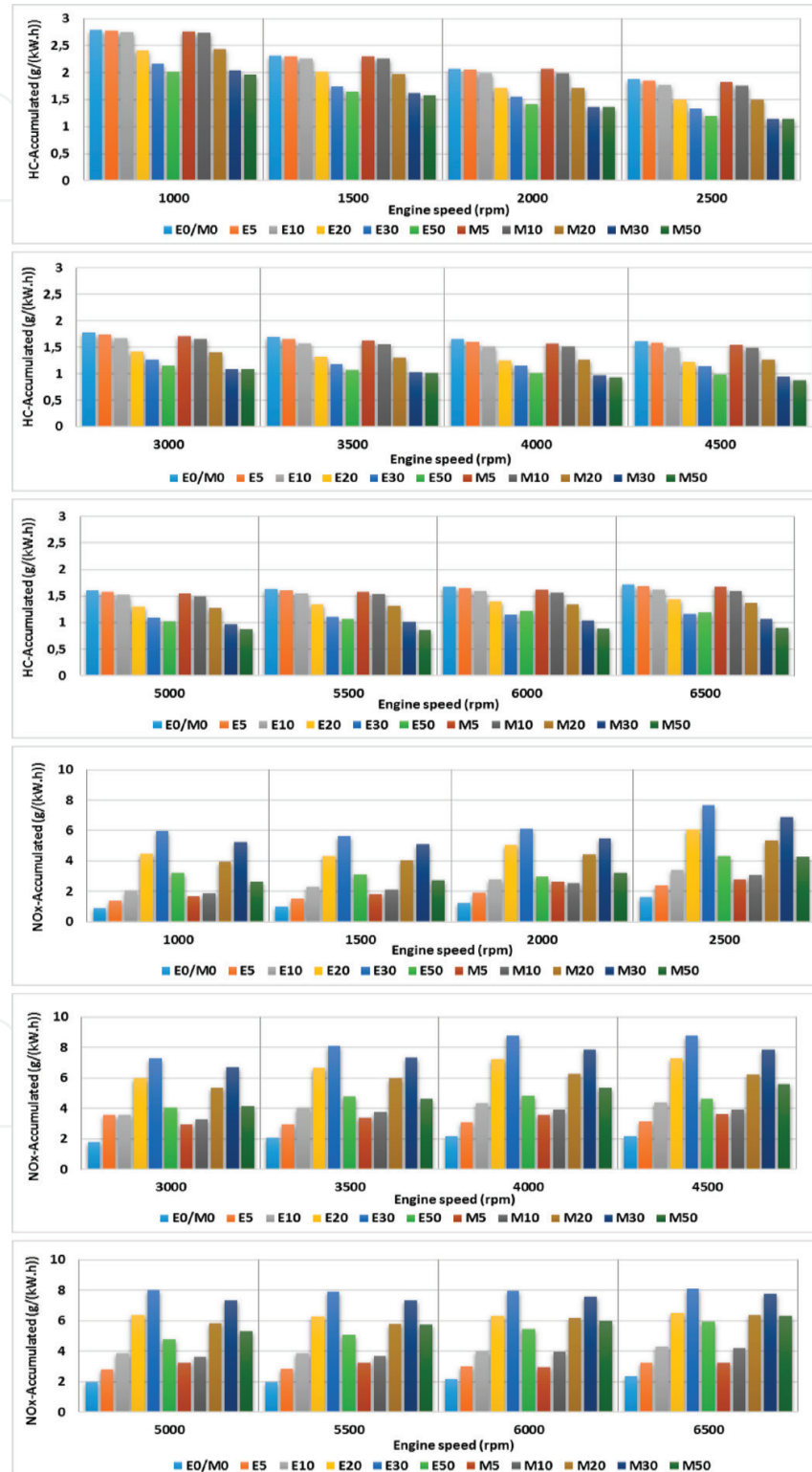
Figure 15. Influence of blended fuels on HC and NOx emissions.

When the relative air-fuel ratio increases, the concentration of HC emissions decreases. The reason for the decrease in HC emissions is similar to that of CO described above. The comparison between the decrease in HC emissions and the blended fuels indicates that methanol is more effective than ethanol. The lowest HC emissions are obtained with methanol-blended fuel (M50). When more combustion is complete, it will result in lower HC emissions.

Figure 15 shows the influence of the blended fuels on NOx emissions. It is noticeable that when methanol and ethanol percentage increases up to 30% E30 (M30), the NOx emission increases, after which it decreases with increasing the percentage of the methanol (ethanol).

The reason is that the improved combustion results in increased temperature in combustion chamber. The higher methanol (ethanol) content in the blends lowers the temperature in combustion chamber. The lower temperature is due to:

1. Latent heat of evaporation of alcohols, which decreases the temperature in combustion chamber during the vaporization.

2. The more triatomic molecules are produced: the higher the gas heat capacity and the lower the combustion gas temperature will be. However, the low temperature in combustion chamber can also lead to an increment in the unburned combustion product.

4. Conclusions

The purpose of the present chapter is to demonstrate the influence of ethanol and methanol addition to gasoline on spark-ignition engine performance and emission characteristics. The summarized results from this study are the following:

With the increase of the percentage of ethanol in the blended fuel, the engine brake power decreased for various engine speeds.

With the increase of the percentage of methanol in the blends M5 and M10, the brake power slightly increased, and with the increase of the percentage of methanol in the blends M30 and M50, the brake power decreased.

As the ethanol (methanol) percentage increased, the BSFC increased. The blended fuels show higher BSFC and lower engine brake power than pure gasoline. Furthermore, there is a slight difference between the BSFC in comparison of gasoline and gasoline blended fuels (E5, E10, and E20 and M5, M10, and M20).

When there is an increase in ethanol and methanol percentage, the CO and HC concentration decreases. The lowest CO and HC emissions are obtained with blended fuel containing methanol (M50).

Increasing the percentage of ethanol and methanol leads to a significant increase in NOx emissions.

When there is an increase in the ethanol and methanol percentage up to 30% E30 (M30), there is an increase in the NOx concentration, followed by a decrease, after which it decreases with increasing ethanol (methanol) percentage. The lowest NOx emissions are obtained with gasoline.

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