Available online at www.sciencedirect.com

ScienceDirect

Procedia Engineering 100 (2015) 1013 – 1022

Procedia Engineering

www.elsevier.com/locate/procedia

25th DAAAM International Symposium on Intelligent Manufacturing and Automation, DAAAM 2014

A Comparison of Ethanol and Methanol Blending with Gasoline Using a 1-D Engine Model

Simeon Iliev*

Department of Engines and Vehicles, University of Ruse, 8 Studentska Str.,7017 Ruse, Bulgaria

Abstract

The world's fossil fuel reserves are limited and there has been intensive research to find out alternatives to fossil fuels. Hence, there is a progressive interest related to using non-fossil sources in vehicles. The biofuel is a major renewable energy source to supplement declining fossil fuel resources. Alcohols are an important category of bio-fuels. The ethanol and methanol have been good candidates as alternative fuels for the vehicles because they are liquid and have several physical and combustion properties similar to gasoline. That is why this study is aimed to develop the 1-D model of a four-stroke spark ignited engine for predicting the effect of various fuel types on engine performances and fuel consumption on various engine operating conditions. AVL Boost was used as a simulation tool to analyze the performance and emissions characteristics for different blends of ethanol, methanol and gasoline (by volume). The results obtained from the simulation of different fuel blends were compared to those of gasoline fuel. The results indicated that when alcohol–gasoline fuel blends were used, the brake power decreased and the brake specific fuel consumption increased compared to those of gasoline fuel. When fuel blends percentage increases, the CO and HC concentration decreases and there is a significant increase NOx emissions when fuel blends percentage increases up to 30% E30 (M30).

Peer-review under responsibility of DAAAM International Vienna

Keywords: Engine simulation; Alternative fuels; Ethanol blends; Methanol blends; Spark-ignition engine

1. Introduction

The problem with crude oil depletion has arisen in the last years. There has been intensive research to find out alternatives to fossil fuels. Alternative fuels are derived from resources other than petroleum. When these fuels are used in internal combustion engines, they produce less air pollution compared to gasoline and most of them are more economically beneficial compared to oil. Last but not least, they are renewable. The most common fuels that are used as alternative fuels are natural gas, propane, ethanol, methanol and hydrogen. Lots of works have been written on engines operating with these fuels individually; but a small number of publications have compared some of these fuels together in the same engine [1–4]. The idea of adding low contents of ethanol or methanol to gasoline is not new, extending back at least to the 1970s, when oil supplies were reduced and a search for alternative energy carriers began in order to replace gasoline and diesel fuel. Initially, methanol and ethanol were considered the most attractive alcohols to be added to gasoline. Methanol and ethanol can be produced from natural products or waste materials, unlike gasoline which is a non-renewable energy resource [5, 6]. One of the important features is that the ethanol and methanol can be used directly without requiring any major changes in the structure of the engine. Among the various alcohols, ethanol and methanol are known as the most suitable fuels for spark ignited (SI) engines.

* Corresponding author. Tel.: +359-82-888-331.
E-mail address: spi@uni-ruse.bg

Peer-review under responsibility of DAAAM International Vienna
doi:10.1016/j.proeng.2015.01.461

The use of fuel additives is very important because many of these additives can be added to fuel in order to improve its efficiency and its performance. Some of the most important additives to improve fuel performance are oxygen containing organic compounds (oxygenates). Several oxygenates have been used as fuel additives, such as methanol, ethanol, tertiary butyl alcohol and methyl tertiary butyl ether [7]. The use of oxygenated fuel additives provides more oxygen in the combustion chamber and has a great potential to reduce emissions from SI engines.

Nomenclature

SI	spark ignition
PFI	port fuel injection
ICEs	internal combustion engines
CFD	computational fluid dynamics
AFR	air-fuel ratio
CO	carbon monoxide
CO_2	carbon dioxide
HC	hydrocarbon
NOx	nitrogen oxides
UHC	unburned hydrocarbon
BMEP	brake mean effective pressure
BSFC	brake specific fuel consumption
BTDC	before top death centre
ABDC	after bottom death centre
BBDC	before bottom death centre
ATDC	after top death centre

On the combustion characteristics, the auto-ignition temperature and flash point of ethanol and methanol are higher than those of gasoline, which makes it safer for transportation and storage. The latent heat of evaporation of ethanol is between three and five times higher than that of gasoline; this makes the temperature of the intake manifold much lower, and increases the volumetric efficiency. The heating value of ethanol is lower than that of the gasoline. Therefore, we need 1.6 times more alcohol fuel to achieve the same energy output. The stoichiometric AFR (air–fuel ratio) of ethanol is about 2/3 that of the gasoline, so the required amount of air for complete combustion is lesser for alcohol [8]. Ethanol has some advantages over gasoline, such as the reduction of CO, unburned HC emissions and better anti-knock characteristics [9]. Methanol and ethanol have much higher octane number than gasoline [10]. This allows engines to have much higher compression ratios, thus increasing thermal efficiency [11]. Methanol can be produced from natural gas at no great cost, and is quite easy to blend with gasoline, so this alcohol was seen as an attractive additive. However, when methanol was used in practice, it became clear that precautions had to be taken when handling it and that methanol is aggressive to some materials, such as plastic components and even metals in the fuel system [12].

There is plenty of literature to various blends of ethanol, methanol and gasoline. Palmer [13] studied the effect of using various blend rates of ethanol–gasoline fuels in engine tests. Results indicated that 10% ethanol addition increases the engine power output by 5%, and the octane number can be increased by 5% for each 10% ethanol added. He indicated that 10% of ethanol addition to gasoline could reduce the concentration of CO emissions up to 30%. Bata et al. [14] studied different blend rates of ethanol–gasoline fuels in engines, and found that the ethanol could reduce the CO and UHC emissions to some degree. The reduction of CO emissions are apparently caused by the wide flammability and oxygenated characteristic of ethanol. Kim et al. [15] estimated that the potential for ethanol production is equivalent to about 32% of the total gasoline consumption worldwide, when used in 85% ethanol in gasoline for a midsize passenger vehicle. Shenghua et al. [16] used a three-cylinder SI engine with different blends of methanol (10%, 15%, 20%, 25% and 30%) in gasoline under full load condition. Results indicated that engine power and torque decreased, while the brake thermal efficiency improved with the methanol blends increase in the fuel blend. Bilgin and Sezer [17] investigated the influence of methanol addition to gasoline on the engine performance. They reported that the maximum brake mean effective pressure (bmep) was obtained from M5 fuel blend. Altun et al. [18] studied the effect of 5% and 10% ethanol and methanol blending in unleaded gasoline on engine performance and exhaust emission. Results indicated that M10 and E10 blended fuels demonstrated the best result in exhaust emission. The HC emission of M10 and E10 are reduced by 13% and 15% and the CO emissions by 10,6% and 9,8%, respectively. Increased CO_2 emission for M10 and E10 compared with unleaded gasoline was observed. The ethanol and methanol addition to unleaded gasoline demonstrated an increase of BSFC (brake specific fuel consumption) and a decrease of break thermal efficiency in comparison to unleaded gasoline.

The reviewed literature shows that the emissions for methanol-gasoline and ethanol-gasoline blends are lower than that of pure gasoline fuel. The engine performance and exhaust emissions with ethanol-gasoline blends are similar to those with methanol-gasoline blends.

From the literature review, it was concluded that the emission and performance characteristics of different blends of ethanol and methanol in different engines have not been investigated sufficiently. For this reason, this study investigated the effects of ethanol–gasoline and methanol–gasoline fuel blends on the performance and emissions characteristics of a SI engine at different engine speed and compared them with those of gasoline.

The gasoline engine performance theory linked together with computer modelling of the engine thermodynamics in engine simulations is a great challenge, as the latter make the most complete use of the former and the models used are becoming widespread. Engine modelling is a very large subject, in part because of the range of engine configurations possible and the variety of alternative analytical techniques or sub-models, which can be applied in overall engine models. Engine modelling is a fruitful research area and as a result many research laboratories have produced their own engine thermodynamics models with varying degrees of complexity, scope and ease to use [19].

Engine simulation is becoming an increasingly important engineering tool for time and cost efficiency in the development of internal combustion engines (ICEs). Most of the results that are obtained by simulation are rather difficult to be obtained experimentally. The use of Computational Fluid Dynamics (CFD) simulations allow researchers to understand flow behaviour and quantify important flow parameters such as mass flow rates or pressure drops, provided that the CFD tools have been properly validated against experimental results. For reasons such as the aforementioned, CFD simulations have become a valuable tool in helping both the analysis and design of the intake and exhaust systems of an ICEs. Many processes in the engine are 3-D but it requires greater knowledge and large computational time. Thus simplified 1-D simulation is often used. There are several components that manifest a complex three-dimensional flow behaviour, such as turbo machinery or manifolds which cannot be simulated properly by 1-D codes, and thus require viscous, 3-D codes.

Hence, it is a right choice to save computational time by simulating the complex components by means of a 3-D code and modelling the rest of the system with a 1-D code, i.e. the ducts. In this way, a coupling methodology between the 1-D and the 3-D code in the respective interfaces is required, and has become the objective of numerous authors [20–22].

In 1-D simulation, equations for conservation of mass, momentum, and energy are solved in time and in one dimension along the main flow direction in the engine pipes. Additional models, correlations, or measurements are needed in 1-D capturing 3-D phenomena such as flow over valves and combustion [23, 24].

2. Theoretical study

The present paper aims to develop the 1-D combustion model of four-stroke port fuel injection (PFI) gasoline engine for predicting the effect of ethanol–gasoline (E0, E5, E10, E20, E30 and E50) and methanol–gasoline (M0, M5, M10, M20, M30 and M50) fuel blends on the performance and emissions of SI engine. For this purpose, a simulation of calibrated gasoline engine model was used as basic operating condition and the laminar burning velocity correlations of ethanol–gasoline and methanol–gasoline fuel blends for calculating the changed combustion duration. The engine performances: torque and specific fuel consumption were compared and discussed.

2.1. Simulation setup

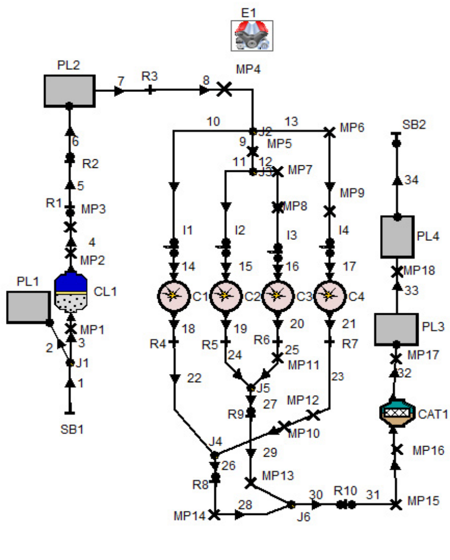


Fig. 1. Layout of gasoline engine model.

The 1-D engine simulation model is developed by using the software AVL BOOST and has been employed to study the performance of an engine working on ethanol-gasoline and methanol-gasoline blends.

The pre-processing step of AVL Boost enable the user to model a 1-Dimensional engine test bench setup using the predefined elements provided in the software toolbox. The various elements are joined by the desired connectors to establish the complete engine model using pipelines.

In Fig.1, E1 represents the engine while C1, C2, C3 and C4 represent the number of cylinders of the engine. MP1 to MP18 represent the measuring points. PL1, PL2, PL3 and PL4 represent the plenum. SB1 and SB2 are for the system boundary. The flow pipes are numbered 1 to 34. CL1 represents the cleaner. R1 to R10 represent flow restrictions, CAT1 represents catalyst and I1 to I4 represent fuel injectors.

The engine model used in this simulation was performed on a four stroke, four cylinder spark ignition engine with port fuel injection. The gasoline engine model was calibrated and described by Iliev [24] and its layout is shown in Fig. 1 with engine specification shown in Table 1.

Table 1. Engine specification.

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

2.2. Combustion model

For the current study Vibe two zone model was selected for the combustion analysis. This model divides the combustion chamber into unburned and burned gas regions [18]. However the assumption that burned and unburned charges have the same temperature is dropped. Instead, the first law of thermodynamics is applied to the burned charge and unburned charge respectively.

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

$$\frac{dm_u u_u}{d\alpha} = -p_c \frac{dV_u}{d\alpha} - \sum \frac{dQ_{Wu}}{d\alpha} - h_u \frac{dm_B}{d\alpha} - h_{BB,u} \frac{dm_{BB,u}}{d\alpha} \tag{2}$$

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

In addition the sum of the volume changes must be equal to the cylinder volume change and the sum of the zone volumes must be equal to the cylinder volume.

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

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

The amount of mixture burned at each time setup is obtained from the Vibe function. For all other terms, like wall heat losses etc., models similar to the single zone models with an appropriate distribution on the two zones are used [25].

2.3. Emission model

The NOx formation model in AVL Boost is based on Pattas and Hafner [25] which incorporates the well-known Zeldovich mechanism [26]. The rate of NOx production was estimated by using the following equation (5):

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

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

In the above equation, C_{PPM} denotes Post Processing Multiplier, C_{KM} denotes Kinetic Multiplier, C denotes molar concentration in equilibrium and r_i denotes reactions rates of Zeldovich mechanism

The NOx formation model in AVL Boost is based on Onorati et al. [27].

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

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

In the above equation, C denotes molar concentration in equilibrium and r_i denotes reactions rates based on the model.

In a spark ignition engine the unburned hydrocarbons have different sources. A complete description of their formation process cannot yet be given and definitely the achievement of a reliable predictive model within a thermodynamic approach is prevented by the fundamental assumptions and the requirement of reduced computational times. Nevertheless 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 major sources of unburned hydrocarbons can be identified in spark ignition engines [22]:

1. A fraction of the charge enters the crevice volumes and is not burned since the flame quenches at the entrance.
2. Fuel vapor is absorbed into the oil layer and deposits on the cylinder wall during intake and compression. The following desorption takes place when the cylinder pressure decreases during the expansion stroke and complete combustion cannot take place any more.
3. Quench layers on the combustion chamber wall which are left as the flame extinguishes prior to reaching the walls.
4. Occasional partial burning or complete misfire occurring when combustion quality is poor.
5. Direct flow of fuel vapor into the exhaust system during valve overlap in PFI 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. Quench layer and partial burn effect cannot be physically described in a quasidimensional approach, but may be included by adopting tunable semiempirical correlations.

The process of formation of unburned hydrocarbons 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 given by equation (7):

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

In the above equation, $m_{crevice}$ is mass of unburned charge in the crevice, p is cylinder pressure, $V_{crevice}$ is total crevice volume, M is unburned molecular weight, R gas constant and T_{piston} piston temperature.

A second significant source of hydrocarbon is the presence of lubricating oil in the fuel or on the walls of the combustion chamber. In fact, during compression, the fuel vapor pressure increases so, by Henry's law, absorption occurs even if the oil was saturated during the intake. During combustion the fuel vapor concentration 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. The oil film is at the same temperature as the cylinder wall.
2. Fuel is constituted by a single hydrocarbon species, completely vaporized in the fresh mixture.
3. Oil is represented by squalane ($C_{30}H_{62}$), whose characteristics are similar to the SAE5W20 lubricant.
4. Traverse flow across the oil film is negligible.
5. Diffusion of the fuel in the oil film is the limiting factor, since the diffusion constant in the liquid phase is 104 times smaller than the corresponding value in the gas phase.

Under these hypotheses the radial distribution of the fuel mass fraction in the oil film can be determined by solving the diffusion equation (8):

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

In the above equation, w_F denotes mass fraction of the fuel in the oil film, t denotes time, r denotes radial position in the oil film (distance from the wall), D denotes relative (fuel-oil) diffusion coefficient.

AVL Boost evaluates the position of the flame front at every time-step and accumulates only HCs that are desorbed into the burned gases, since any HC released into the unburned mixture would be burned by the propagating flame front.

For the current study Vibe two zone model was selected for the combustion analysis. This model divides the combustion chamber into unburned and burned gas regions

3. Result and discussion

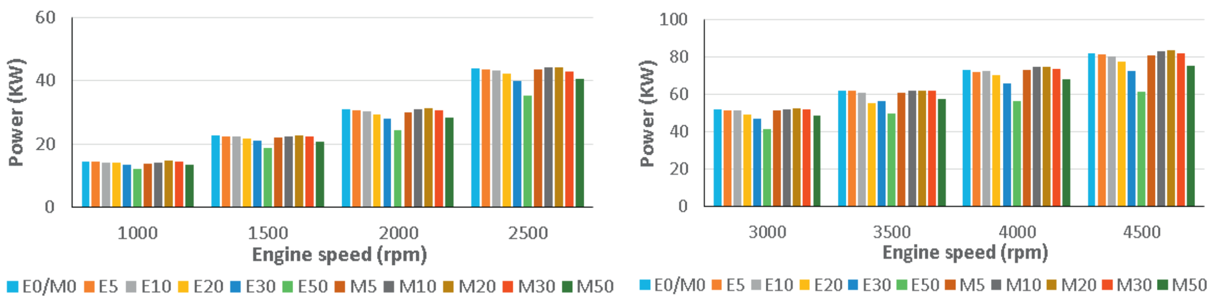
The present study concentrated on the emission and performance characteristics of the ethanol and methanol-gasoline blends. Different concentrations of the blends 0% Ethanol (Methanol) E0 (M0), 5% Ethanol (Methanol) E5 (M5), 10% Ethanol (Methanol) E10 (M10), 20% Ethanol (Methanol) E20 (M20), 30% Ethanol (Methanol) E30 (M30), 50% Ethanol (Methanol) E50 (M50) and 85% Ethanol (Methanol) E85 (M85) by volume were analysed using AVL BOOST at full load conditions for the speeds ranging from 1000 - 6500 rpm in the steps of 500rpm. The results are divided into different subsections based on the parameter analysed.

3.1. Engine performance characteristics

The results of the brake power, and specific fuel consumption for ethanol and methanol gasoline blended fuels at different engine speeds are presented here.

Fig. 2 shows the influence of ethanol gasoline blended fuels on engine brake power.

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 (M5), E10 (M10), E20 (M20), E30 (M30), E50 (M50) and pure gasoline E0 (M0), using the CFD results.



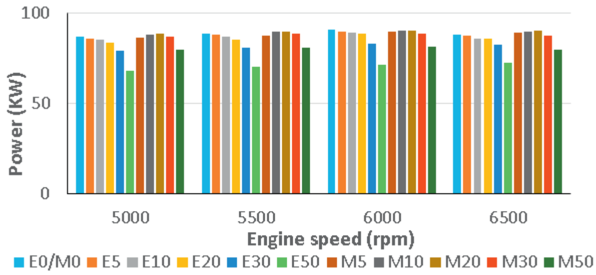


Fig. 2. Influence of ethanol and methanol gasoline blended fuels on engine brake power.

When the ethanol content in the blended fuel was increased, the engine brake power decreased for all engine speeds. The brake power of gasoline was higher than those of E5-E50 for all engine speeds. The heating value of ethanol is lower than that of gasoline and heating value of the blended fuel decreases with the increase of the ethanol content. As a result, a lower power output is obtained [23, 24].

When the methanol content in the blended fuel was increased (M5 and M10), the engine brake power slightly increased. This can be explained by the fact that oxygenated fuels have a better combustion efficiency. When the methanol content in the blended fuel was increased (M30 and M50), the engine brake power decreased for all engine speeds. The heating value of the blended fuel decreases with the increase of the methanol content. As a result, a lower power output is obtained. The brake power of gasoline was higher than those of M50 for all engine speeds.

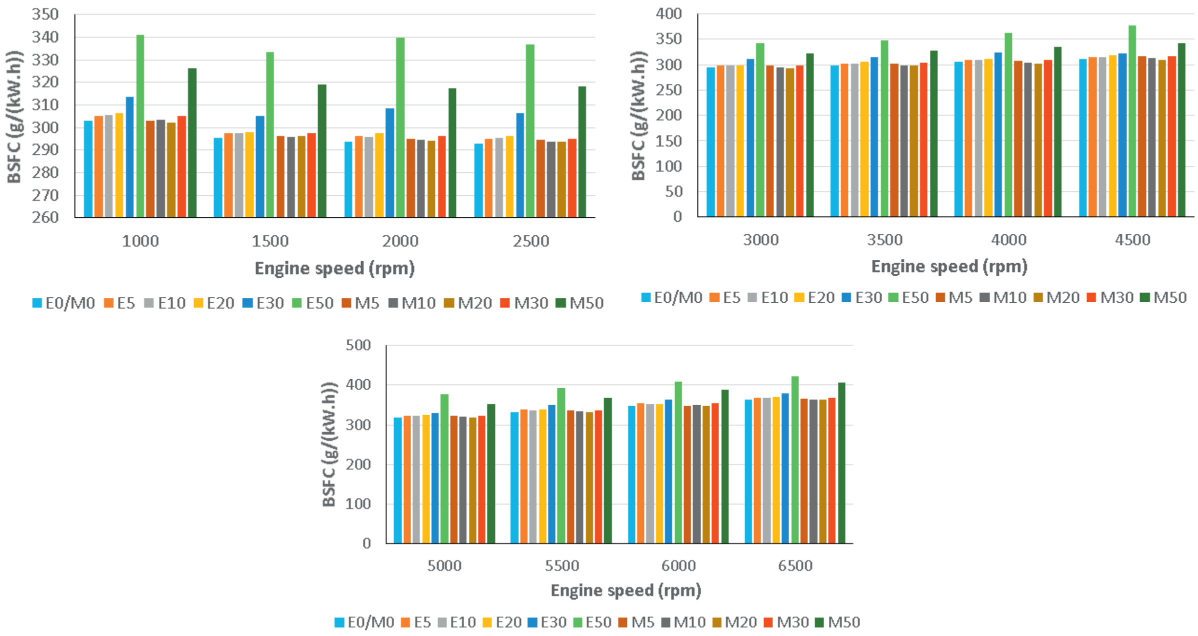


Fig. 3. Influence of ethanol and methanol gasoline blended fuels on brake specific fuel consumption.

Fig. 3 indicates the variations of the BSFC for ethanol and methanol gasoline blended fuels under various engine speeds. As shown in this figure, the BSFC increased as the ethanol percentage increased. The reason is well known: the heating value and stoichiometric air-fuel ratio are the smallest for this fuel, which means that for specific air-fuel equivalence ratio, more fuel is needed. The highest specific fuel consumption is obtained at E50 (M50) blended fuel.

Also, a slight difference exists between the BSFC when using gasoline and when using ethanol and methanol gasoline blended fuels (E5 (M5), E10 (M10) and E20 (M20)). The lower energy content of ethanol gasoline blended fuels causes some increment in BSFC of the engine when it is used without any modification.

3.2. Engine emissions characteristics

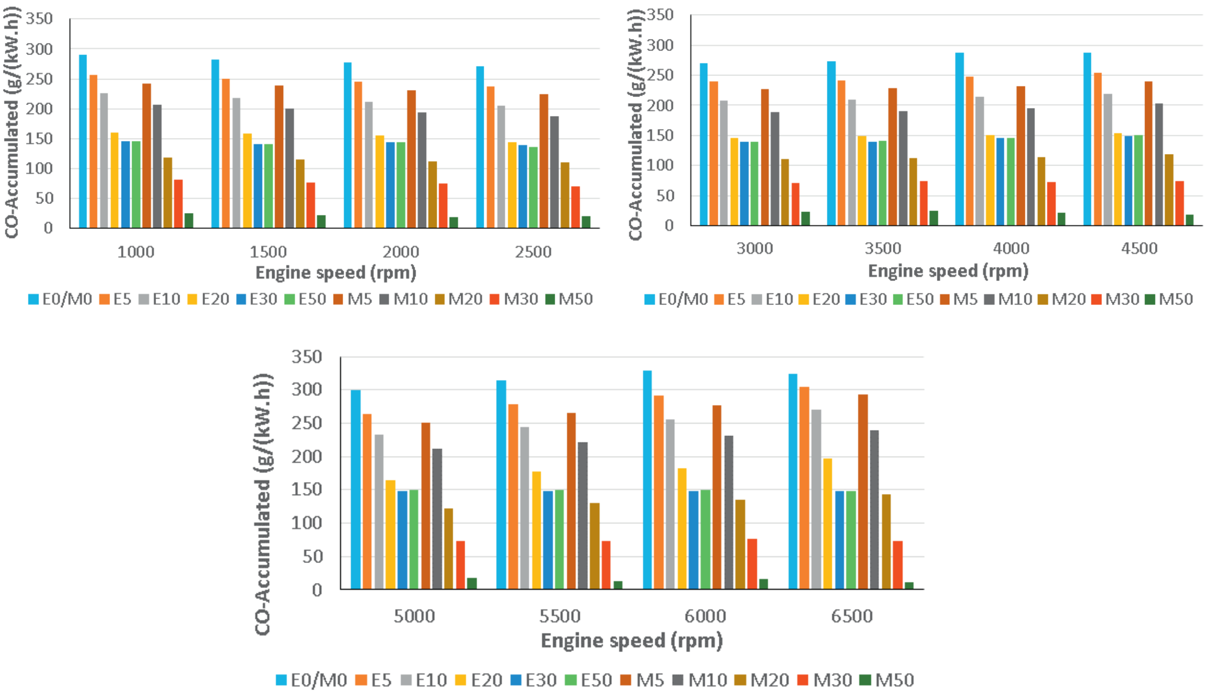


Fig. 4. Influence of ethanol and methanol gasoline blended fuels on CO emissions.

The effect of the ethanol and methanol gasoline blends on CO emissions for different engine speeds is shown in Fig. 4. It can be seen that when ethanol and methanol percentage increases, the CO concentration decreases. This can be explained by the enrichment of oxygen owing to 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 significant reason for this reduction is that ethanol (C_2H_5OH) and methanol (CH_3OH) has less carbon than gasoline (C_8H_{18}). The lowest CO emissions are obtained with blended fuel containing methanol (M50).

The effect of the ethanol and methanol gasoline blends on HC emissions for different engine speeds is shown in Fig. 5. It can be seen that when ethanol and 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 is similar to that of CO concentration described above. The comparison of decrease of HC emissions among the blended fuels indicates that methanol is more effective than ethanol. The lowest HC emissions are obtained with blended fuel containing methanol (M50). When the complete combustion is more, the HC emission is lower.

The effect of the ethanol and methanol gasoline blends on NOx emissions for different engine speeds is shown in Fig. 6. It can be seen that when ethanol and methanol percentage increases up to 30% E30 (M30), the NOx concentration increases after which it decreases with increasing the ethanol (methanol) percentage. This can be explained by the improved combustion inside the cylinder resulting in an increased in-cylinder temperature. The higher percentage of ethanol (methanol) in gasoline reduces the in-cylinder temperature. The reasons for the reduction in temperature are: 1. Latent heat of evaporation of ethanol (methanol), which decreases the in-cylinder temperature when they vaporizes, 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 in-cylinder temperature can also lead to an increment in the unburned combustion product.

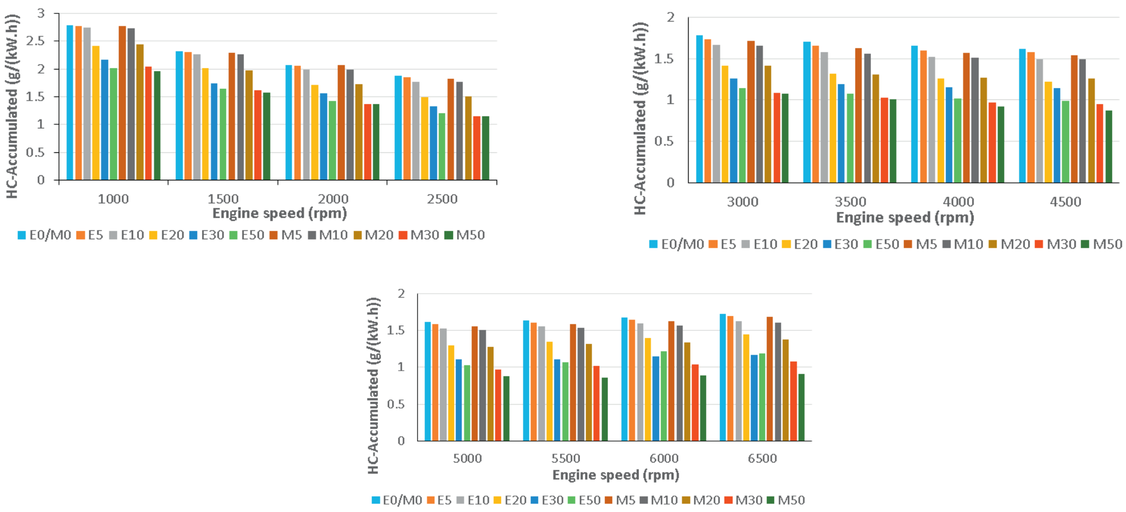


Fig. 5. Influence of ethanol and methanol gasoline blended fuels on HC emissions.

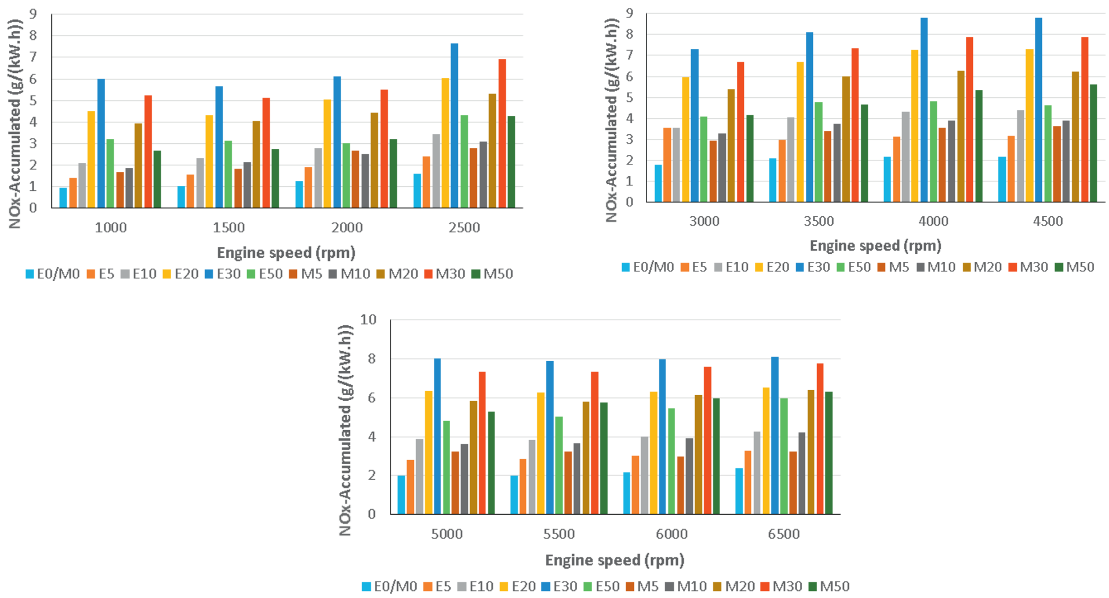


Fig. 6. Influence of ethanol and methanol gasoline blended fuels on NOx emissions.

Conclusion

The present paper demonstrates the influences of ethanol and methanol addition to gasoline on SI engine performance and emission characteristics. General results concluded from this study can be summarized as follows:

When the ethanol content in the blended fuel was increased, the engine brake power decreased for all engine speeds. When the methanol content in the blended fuel was increased (M5 and M10), the engine brake power slightly increased and when the methanol content in the blended fuel was increased (M30 and M50), the engine brake power decreased for all engine speeds.

The BSFC increased as the ethanol (methanol) percentage increased. Gasoline blended fuels show lower brake power and higher BSFC than those of gasoline. Also, a slight difference exists between the BSFC when using gasoline and when using gasoline blended fuels (E5 (M5), E10 (M10) and E20 (M20)).

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

Ethanol and methanol gasoline blends the significant increase NOx emissions with the increase of ethanol and methanol percentage. When ethanol and methanol percentage increases up to 30% E30 (M30), the NOx concentration increases, followed by a decrease, after which it decreases with increasing ethanol (methanol) percentage. The lowest NOx emissions are obtained with gasoline.

Acknowledgements

The present document has been produced with the financial assistance of the European Social Fund under Operational Programme "Human Resources Development". The contents of this document are the sole responsibility of the "Angel Kanchev" University of Ruse and can under no circumstances be regarded as reflecting the position of the European Union or the Ministry of Education and Science of the Republic of Bulgaria. Project № BG051PO001-3.3.06-0008 "Supporting Academic Development of Scientific Personnel in Engineering and Information Science and Technologies"

We are also eternally grateful to AVL-AST, Graz, Austria for granting use of AVL-BOOST under the university partnership program.

References

[1]. B. Hakan and D. Orhan. Investigating the effects of LPG on spark ignition engine combustion and performance, Energy Conversion and Management, 46:2317–33, 2005.
[2]. T. Melo, G. Machado, RT. Machado, Jr CR. Pereira Belchior, PP. Pereira, Thermodynamic modeling of compression, combustion and expansion processes of gasoline, ethanol and natural gas with experimental validation on a flexible fuel engine, SAE World Congress, 2007-24-0035, 2007.
[3]. Y. Varol, C. Oner, HF. Oztop, S. Altun Comparison of methanol, ethanol, or n-butanol blending with unleaded gasoline on exhaust emissions of an SI engine, Energy Sourse, Part A 36:938-948, 2014.
[4]. A. Pourkhesalian, A. Shamekhi, and F. Salimi, Alternative fuel and gasoline in an SI engine: A comparative study of performance and emissions characteristics, Fuel, vol. 89, 2010, pp. 1056 – 1063, 2010.
[5]. C. Chen, P. Rao, J. Delfino, Oxygenated fuel induced to solvent effect on the dissolution of polynuclear aromatic hydrocarbons from contemned soil, Chemosphere, 60: 1572–82, 2005.
[6]. M. Canaksi, AN. Ozsezen, E. Alptekin, Impact of alcohol-gasoline fuel blends on exhaust emission on an SI engine, Renewable Energy, 52: 111–117, 2013.
[7]. T. Cavalcante Cordeiro de Melo, G. Bastos Machado, RT. Machado, Pereira Belchior Jr CR, PP. Pereira, Thermodynamic modeling of compression, combustion and expansion processes of gasoline, ethanol and natural gas with experimental validation on a flexible fuel engine. SAE World Congress, 2007-24-0035; 2007.
[8]. A. Pikunas, S. Pukalskas, J. Grabys, Influence of composition og gasoline-ethanol blends on parameters of internal combustion engines, Journal of KONES Internal Combustion Engines, vol. 10, 3-4, 2003.
[9]. R. Bata, A. Elord, R. Rice, Emissions from IC engines fueled with alcohol–gasoline blends: a literature review. Transactions of the ASME1989; 111:424–31.
[10]. R. Silva, R. Cataluna, EW. Menezes, D. Samios, CMS. Piatnicki, Effect of additives on the antiknock properties and Reid vapor pressure of gasoline, Fuel 2005; 84:951–9.
[11]. K. Raveendran, A. Ganesh, Heating value of biomass and biomass pyrolysis products, Fuel 1996; 75:1715–20.
[12]. K. Egebäck, M. Henke, B. Rehnlund, M. Wallin and R. Westerholm, Blending of ethanol in gasoline for spark ignition engines – Problem inventory and evaporative measurements, Rapport MTC 5407, May 2005.
[13]. F. Palmer, Vehicle performance of gasoline containing oxygenates. International conference on petroleum based and automotive applications, Institution of Mechanical Engineers Conference Publications, MEP, London, UK, pp. 33–46, 1986.
[14]. R. M. Bata, A.C. Elord, R. W. Rice, Emissions from IC engines fueled with alcohol–gasoline blends: a literature review, Transactions of the ASME 111, 424–431, 1989.
[15]. S. Kim, B.E. Dale, Global potential bioethanol production from wasted crops and crop residues, Biomass and Bioenergy, Vol.26, No.4, pp. 361-375, 2004
[16]. L. Shenghua, E. R. C. Clemente, H. Tiegang, W. Yanjv, Study of spark ignition engine fueled with methanol/gasoline fuel blends, Appl. Therm. Eng. 27:1904–1910, 2007.
[17]. A. Bilgin, I. Sezer, Effects of methanol addition to gasoline on the performance and fuel cost of a spark ignition engine, Energy Fuel 2008;22: 2782–8.
[18]. S. Altun, H. Oztop, C. Oner, Exhaust, Y. Varol, Emissions of methanol and ethanol-unleaded gasoline blends in a spark ignition engine, Thermal science, 17 (2013), 1, pp. 291-297.
[19]. A. Shamekhi, N. Khtibzade and AH. Shamekhi, Performance and emissions characteristics of a bi-fuel SI engine fueled by CNG and gasoline. ASME paper, ICES2006-1387, 2006.
[20]. A. Onorati, G. Montenegro and G. D'Errico, Prediction of the attenuation characteristics of IC engine silencers by 1-D and multi-D simulation models, SAE Technical Paper Series 2006, Tech. Rep. 2006-01-1541, doi:10.4271/2006-01-1541, 2006.
[21]. G. Montenegro and A. Onorati, Modeling of silencers for IC engine intake and exhaust systems by means of an integrated 1D-multiD Approach, SAE International Journal of Engines 1 (1), 466, doi:10.4271/2008-01-0677, 2009.
[22]. G. Montenegro, A. Onorati, F. Piscaglia, G. D'Errico, Integrated 1D-Multi-D Fluid Dynamic Models for the Simulation of ICE Intake and Exhaust Systems, SAE Technical Paper Series 2007, Tech. Rep. 2007-01-0495, doi:10.4271/2007-01-0495, 2007.
[23]. S. Iliev, Developing of a 1-D combustion model and study of engine performance and exhaust emissions using ethanol-gasoline blends, IAENG Transaction of Engineering Technologies, Netherlands, Springer, 2014, unpublished.
[24]. S. Iliev, Developing of a 1-D combustion model and study of engine characteristics using ethanol-gasoline blends, Proceedings of the World Congress on Engineers 2014, Vol II, WCE 2014/ 978-988-19253-5-0.
[25]. AVL List Gmbh, AVL Boost – Theory, 2013.
[26]. C. T. Bowman, Kinetics of pollutant formation and destruction in combustion, Progress in energy and combustion science, 1(1), 33-45, 1975.
[27]. A. Onorati, G. Ferrari, G.D'Errico, 1D unsteady flows with chemical reactions in the exhaust duct-system of S.I. engines: Predictions and Experiments, SAE Paper No. 2001-01-0939.