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Effect of ethanol–unleaded gasoline blends on engine performance and exhaust emission

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Abstract

This paper investigates the effect of using unleaded gasoline–ethanol blends on SI engine performance and exhaust emission. A four stroke, four cylinder SI engine (type TOYOTA, TERCEL-3A) was used for conducting this study.

Performance tests were conducted for equivalence air–fuel ratio, fuel consumption, volumetric efficiency, brake thermal efficiency, brake power, engine torque and brake specific fuel consumption, while exhaust emissions were analyzed for carbon monoxide (CO), carbon dioxide (CO_2) and unburned hydrocarbons (HC), using unleaded gasoline–ethanol blends with different percentages of fuel at three-fourth throttle opening position and variable engine speed ranging from 1000 to 4000 rpm. The results showed that blending unleaded gasoline with ethanol increases the brake power, torque, volumetric and brake thermal efficiencies and fuel consumption, while it decreases the brake specific fuel consumption and equivalence air–fuel ratio. The CO and HC emissions concentrations in the engine exhaust decrease, while the CO_2 concentration increases. The 20 vol.% ethanol in fuel blend gave the best results for all measured parameters at all engine speeds.

Keywords: Fuel additive; Ethanol–unleaded gasoline blend; Alternative engine fuel; Exhaust emissions

1. Introduction

Fuel additives are very important, since many of these additives can be added to fuel in order to improve its efficiency and its performance. One of the most important additives to improve fuel

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Nomenclature

$(AFR)_{act}$ actual air–fuel ratio of fuel blend
$(AFR)_{st.b}$ stoichiometric air–fuel ratio of fuel blend
$(AFR)_{st.i}$ molar stoichiometric air–fuel ratio of fuel blend
B_p brake power, kW
BSFC brake specific fuel consumption, $kg\,kW^{-1}\,h^{-1}$
$(LHV)_b$ lower heating value of fuel blend, $kJ\,kg^{-1}$
$(LHV)_i$ lower heating value of given component in fuel blend, $kJ\,kg^{-1}$
$\dot{m}_a$ air mass flow rate, $kg\,h^{-1}$
$\dot{m}_f$ mass flow rate of fuel, $kg\,h^{-1}$
N engine speed, rpm
P atmospheric pressure, Pa
Q_f volume flow of fuel, cm^3
Ra air constant, $J\,kg^{-1}\,K^{-1}$
T engine torque, N m
t time required to consume 100 cm^3 of fuel, s
T_a charge temperature at end of induction process, K
T_h temperature difference between charge and engine parts, K
T_0 ambient temperature, K
T_v temperature difference between charge and vapor, K
ΔT total change in charge temperature, K
v_i volume fraction of given component in fuel blend, vol.%
V_s swept volume of engine, m^3

Greek letters
$\eta_{b.th}$ brake thermal efficiency
η_v volumetric efficiency, %
ρ_i density of given component in fuel blend, $g\,cm^{-3}$
ρ_b density of fuel blend, $g\,cm^{-3}$
ϕ equivalence air–fuel ratio

performance is oxygenates (oxygen containing organic compounds). Several oxygenates have been used as fuel additives, such as methanol, ethanol, tertiary butyl alcohol and methyl tertiary butyl ether.

Ethanol was the first fuel among the alcohols to be used to power vehicles in the 1880s and 1890s. Henry Ford presented it as the fuel of choice for his automobiles during their earliest stages of development [1].

Presently, ethanol is prospective material for use in automobiles as an alternative to petroleum based fuels. The main reason for advocating ethanol is that it can be manufactured from natural products or waste materials, compared with gasoline, which is produced from non-renewable natural resources. In addition, ethanol shows good anti-knock characteristics. However, economic reasons still limit its usage on a large scale. At the present time and instead of pure ethanol, a blend of ethanol and gasoline is a more attractive fuel with good anti-knock characteristics.

Palmer [2] reported that all oxygenated blends gave a better anti-knock performance during low speed acceleration than hydrocarbon fuels of the same octane range. Goodger [3] reported the comparisons with hydrocarbon fuels made by Ricardo at a fixed compression ratio, in which there was a 5% improvement in efficiency using ethanol. Winnington and Siddiqui [4] studied the effect of using ethanol–gasoline blends: A (15% ethanol, 41% premium and 44% regular) and B (20% ethanol, 54% premium and 26% regular), as a fuel on the performance of spark ignition engines, such as the Ricardo and Peugeot 504 GR engines. The Ricardo engine, over the test range of 8:1 to 10:1 compression ratio, showed an average drop in power compared to premium gasoline of 2.5% on blend A and 7.5% on blend B. The specific fuel consumption of the ethanol–gasoline blend showed an increase compared to premium gasoline of around 0.5% and 4% on blends A and B, respectively. The Peugeot engine tests showed that the power was down, overall, by around 1% and 2.5% on blends A and B, respectively, and the specific fuel consumption was increased by about 0.5% for blend A and 1% for blend B.

Hamdan and Jubran [5] using the ATD 34 engine conducted performance tests using different ethanol–gasoline blends. The maximum percentage of ethanol (E%) used was 15%. The best performance was achieved when the 5% ethanol–gasoline blend was used, with thermal efficiency increasing by 4% under low speed conditions and 20% at the high speed condition.

El-Kassaby [6] studied the effect of ethanol–gasoline blends on SI engine performance. The performance tests were conducted using different percentages of ethanol–gasoline up to 40% under variable compression ratio conditions. The results showed that the engine indicated power improves with ethanol addition, the maximum improvement occurring at the 10% ethanol and 90% gasoline fuel blend.

The effect of using ethanol with unleaded gasoline on exhaust emissions (carbon monoxide, CO; carbon dioxide, CO_2 and unburned hydrocarbons, HC) have been experimentally investigated by Bata and Roan [7]. The concentration of CO was reduced by about 40–50% at an equivalence ratio on the lean side near stoichiometry. Also, the concentration of CO decreases as the percent of ethanol increases in the blend.

Gulder [8] performed a series of engine tests using ethanol–gasoline blends. The results of this study were summarized as follows.

It is possible to obtain a lead-free, high octane fuel by adding 20–30% ethanol to unleaded gasoline. Pure ethanol yields a higher engine thermal efficiency than gasoline.

Taljaard et al. [9] studied the effect of oxygenate in gasoline on exhaust emission and performance in a single cylinder, four stroke SI engine. They concluded that oxygenates significantly decreased the CO, NO_x and HC emissions at the stoichiometric air–fuel ratio.

Unzelman [10] studied the influence of gasoline composition on air quality. The results indicated that oxygenates can improve air quality by reducing the amount of exhaust emission.

The objective of the present paper is to investigate the effect of ethanol–unleaded gasoline blends on spark ignition engine performance and CO, CO_2 and HC at three-fourth throttle opening position and variable engine speed operating conditions.

2. Experimental apparatus and procedure

2.1. Engine and equipment

The experiments were conducted on a four cylinder, four stroke spark ignition engine (type Toyota-Tercel-3A). The engine has a swept volume of 1452 cm^3, a compression ratio of 9:1 and a maximum power of 52 kW at 5600 rpm. The engine was coupled to a hydraulic dynamometer (type—HPA engine dynamometer 203), which is equipped with an instrument cabinet (column mounted) fitted with a torque gauge, electric tachometer and switches for the load remote control. In addition, it is fitted with a damper valve for the torque gauge. Fig. 1 shows a schematic diagram of the engine and its instrumentation.

Fuel consumption was measured by using a calibrated burette and a stopwatch (type Herwins) with an accuracy of 0.2 s. The concentrations of the exhaust emissions (CO, CO_2 and HC) and air–fuel ratio were measured using a "Sun Gas Analyzer" MGA 1200. The analyzer has a non-dispersive infrared module for CO, CO_2 and HC. The sample line tube is fitted to the tailpipe 300 mm away from the exhaust port in order to allow sufficient mixing of the exhaust gases. The MGA automatically requires a 15 min warm up period and then goes into auto-calibration. The concentration of each gas is measured continuously and digitally.

2.2. Fuels

Two different fuel samples were experimentally investigated during this study. Unleaded gasoline was obtained from the Jordan Petroleum Refinery Corporation. Ethanol, with a purity of 99%, was used in preparing the blends.

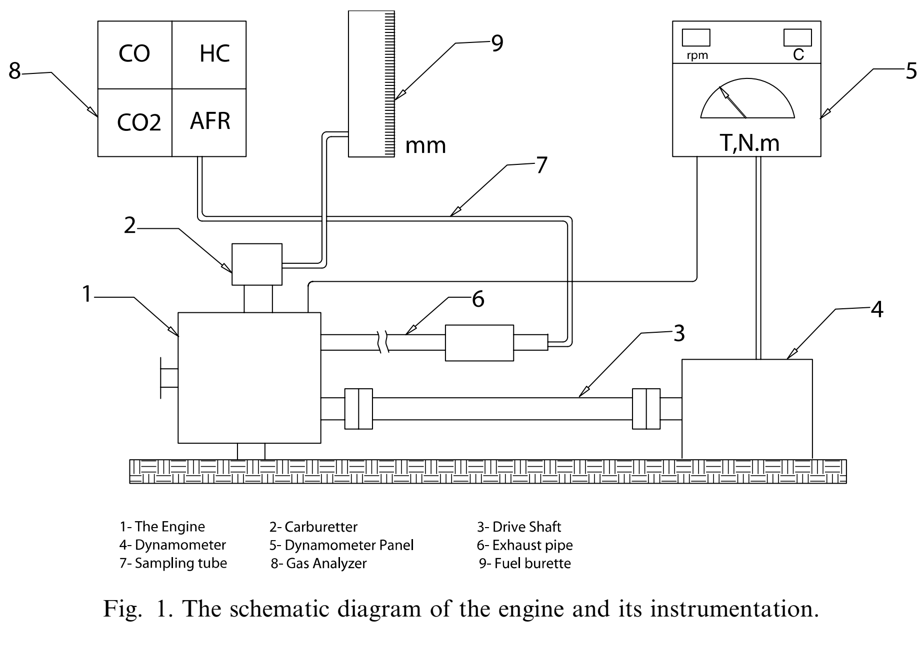

Fig. 1. The schematic diagram of the engine and its instrumentation.

The unleaded gasoline was blended with ethanol to get 10 test blends ranging from 0% to 25% ethanol with an increment of 2.5%. The fuel blends were prepared just before starting the experiment to ensure that the fuel mixture is homogenous and to prevent the reaction of ethanol with water vapor. The fuel types properties are shown in Appendix A.

2.3. Procedures

The engine was started and allowed to warm up for a period of 20–30 min. The air–fuel ratio was adjusted to yield maximum power on unleaded gasoline. Engine tests were performed at 1000, 2000, 3000 and 4000 rpm engine speed at three-fourth throttle opening position. The lowest desired speed is maintained by the load adjustment. The required engine load was obtained through the dynamometer control.

Before running the engine to a new fuel blend, it was allowed to run for sufficient time to consume the remaining fuel from the previous experiment.

For each experiment, three runs were performed to obtain an average value of the experimental data. The variables that were continuously measured include engine rotational speed (rpm), torque, time required to consume 100 cm^3 of fuel blend (s), air–fuel ratio, CO, CO_2 and HC emissions.

The parameters, such as fuel consumption rate, equivalence air–fuel ratio, volumetric efficiency, air consumption, brake power, brake specific fuel consumption, brake thermal efficiency, density, stoichiometric air–fuel ratio and lower heating value (LHV) of the fuel blends, were estimated using the following equations.

The fuel consumption is estimated by measuring the fuel consumed per unit time and the calculated values of the density for different fuel blends through Eqs. (1) and (2):

$$\dot{m}_f = \frac{3.6 Q_f \rho_b}{t} \tag{1}$$

$$\rho_b = \sum \rho_i v_i \tag{2}$$

The volumetric efficiency is defined as follows:

$$\eta_v = \frac{\dot{m}_a \mathrm{Ra} T_0}{30 P V_s N} \tag{3}$$

where

$$\dot{m}_a = (\mathrm{AFR})_{\mathrm{act}} \dot{m}_f \tag{4}$$

The equivalence air–fuel ratio is defined as

$$\phi = \frac{(\mathrm{AFR})_{\mathrm{st.b}}}{(\mathrm{AFR})_{\mathrm{act}}} \tag{5}$$

where

$$(\mathrm{AFR})_{\mathrm{st.b}} = \sum (\mathrm{AFR})_{\mathrm{st.i}} v_i \tag{6}$$

The brake power is calculated by measuring the engine speed and the engine torque and is given by Eq. (7). The specific fuel consumption is defined as the ratio of the fuel consumption to the brake power, as shown in Eq. (8). The brake thermal efficiency is defined as the ratio of the brake power to the heat input for each blend, as shown in Eq. (9)

$$B_{\mathrm{p}} = \frac{\mathrm{NT}}{9549.29} \tag{7}$$

$$\mathrm{BSFC} = \frac{\dot{m}_{\mathrm{f}}}{B_{\mathrm{p}}} \tag{8}$$

$$\eta_{\mathrm{b.th}} = \frac{3600 B_{\mathrm{p}}}{\dot{m}_{\mathrm{f}}(\mathrm{LHV})_{\mathrm{b}}} \tag{9}$$

where

$$(\mathrm{LHV})_{\mathrm{b}} = \sum \left(\frac{\rho_i v_i}{\rho_{\mathrm{b}}} \right) (\mathrm{LHV})_i \tag{10}$$

3. Results and discussion

The effects of ethanol addition to unleaded gasoline on SI engine performance and exhaust emissions at three-fourth throttle opening at variable engine speeds were investigated. The average changes and the mean of the average changes in the values of the parameters of engine performance and exhaust emissions for all fuel blends and the four different engine speeds obtained from the experimental runs are summarized in Table 1.

3.1. Fuel consumption ($\dot{m}_f$)

The effect of the ethanol–unleaded gasoline blends on the fuel consumption is shown in Fig. 2. From Fig. 2, the $\dot{m}_{\mathrm{f}}$ increases as the E% increases for all engine speeds. This behavior is attributed

Table 1
The summarized results of the experimental runs at variable speeds for all fuel blends

Engine parameters	Speed (rpm)				Mean average value (%)
	1000	2000	3000	4000	
B_{p}	8.9	9.1	7.76	7.6	8.3
$\eta_{\mathrm{b.th}}$	9.2	9.5	8.5	8.6	9.0
η_{v}	7.1	6.9	7.6	6.6	7.0
$\dot{m}_{\mathrm{f}}$	6.0	5.8	5.9	5.0	5.7
BSFC	−2.7	−2.9	−1.8	−2.3	−2.4
ϕ	−3.4	−3.5	−4.0	−3.8	−3.7
CO	−44.6	−41.4	−54.7	−45.4	−46.5
HC	−23.0	−24.4	−26.3	−23.5	−24.3
CO_2	7.4	6.9	7.9	7.9	7.5

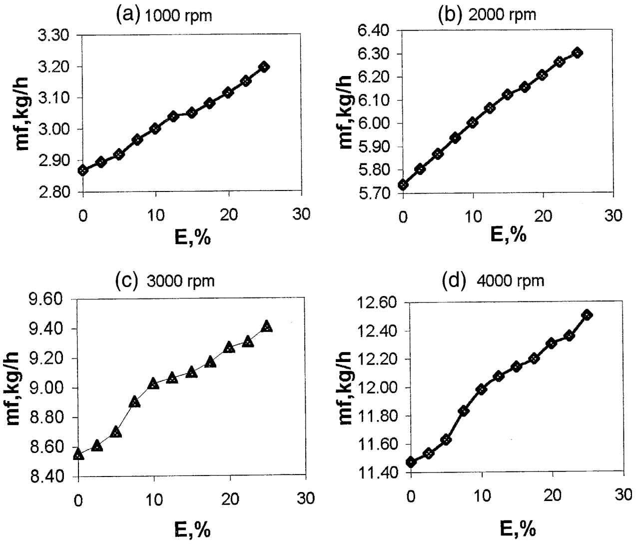

Fig. 2. The effect of ethanol addition on the fuel consumption rate.

to the LHV per unit mass of the ethanol fuel, which is distinctly lower than that of the unleaded gasoline fuel. Therefore, the amount of fuel introduced into the engine cylinder for a given desired fuel energy input has to be greater with the ethanol fuel. Fig. 2a and d, for example, show that at the engine speeds of 1000 and 4000 rpm, the relative increase of $\dot{m}_f$ is approximately 11% and 9%, respectively. In addition, $\dot{m}_f$ increases about 4.5 times as the engine speed increases from 1000 to 4000 rpm. This increase in $\dot{m}_f$ could be explained by the fact that as the engine speed increases, the air velocity increases and the pressure decreases at the carburetor venturi. Consequently, the pressure drop between the pressure at the carburetor venturi and the pressure (atmospheric) inside the float chamber increases, which causes more fuel consumption.

3.2. Equivalence air–fuel ratio (ϕ)

The effect of the ethanol–unleaded gasoline blends on the equivalence air–fuel ratio is shown in Fig. 3. As shown from Fig. 3 the equivalence air–fuel ratio decreases as the E% increases to 20%. This effect is attributed to two factors: (1) the decrease in the stoichiometric air–fuel ratio of the fuel blends, since the stoichiometric air–fuel ratio of ethanol fuel is usually lower than that of the unleaded gasoline fuel and (2) the increase of actual air–fuel ratio of the blends as a result of the oxygen content in ethanol. For E% exceeding 20%, the behavior is reversed because the actual air–fuel ratio decreases. It is obvious from Fig. 3 that as the engine speed increases to 3000 rpm, ϕ decreases, since the amount of air introduced into the engine cylinder increases i.e., the air–fuel ratio increases. This is due to the increase of the amount of air introduced to the engine cylinder (i.e., the actual air–fuel ratio increases). This is due to the increase in the pressure drop from atmospheric pressure to cylinder pressure. Therefore, a greater decrease in the cylinder pressure occurs. With a further increase in the engine speed beyond 3000 rpm, ϕ increases, since the air

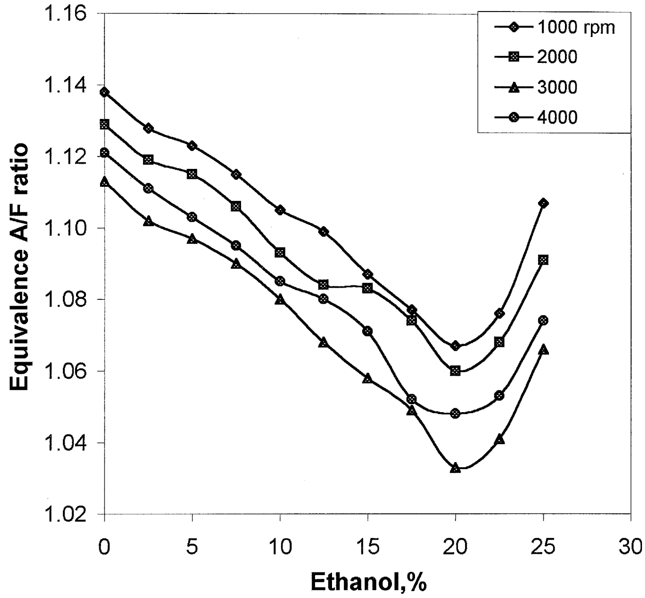

Fig. 3. The effect of ethanol addition on the equivalence air–fuel ratio.

flow into the cylinder, during at least part of the induction process becomes choked. Thus, the amount of air decreases (i.e., the actual air–fuel ratio decreases).

3.3. Volumetric efficiency (η_v)

Fig. 4 shows an increase in the volumetric efficiency as the percentage of ethanol in the fuel blends increases. This is due to the decrease of the charge temperature at the end of the induction process (T_a). This decrease is attributed to the increase in the charge temperature by an amount T_h as a result of the heat transfer from the hot engine parts and the residual gases in the charge. At the same time, the charge temperature drops by an amount T_v due to vaporization of the fuel blend in the inlet manifold and engine cylinder. Therefore, the total change in the charge temperature (ΔT) could be expressed by the following simple equation:

$$\Delta T = T_h - T_v \quad \text{and} \quad T_a = T_h + \Delta T$$

It has been reported by Holger and Bernd [11] that as the E% in the fuel blend increases, the volatility and the latent heat of the fuel blend increases. Meanwhile, with increasing volatility and latent heat of the fuel blend, the drop of the charge temperature T_v increases. At the same conditions, the total heat capacity of the charge increases, since the specific heat of the ethanol fuel is higher than that of the unleaded gasoline fuel, and this led to decreases in the drop of the charge temperature T_v. Therefore, increasing the ethanol in the fuel blend has two contradicting effects on T_v. Hence, the value of T_v depends upon which effect is more dominant. As the quantity of ethanol in the fuel blend increases to 20%, the effect of the increasing volatility and latent heat of the fuel blend is more significant, resulting in T_v increasing. With further increase, the effect of increasing

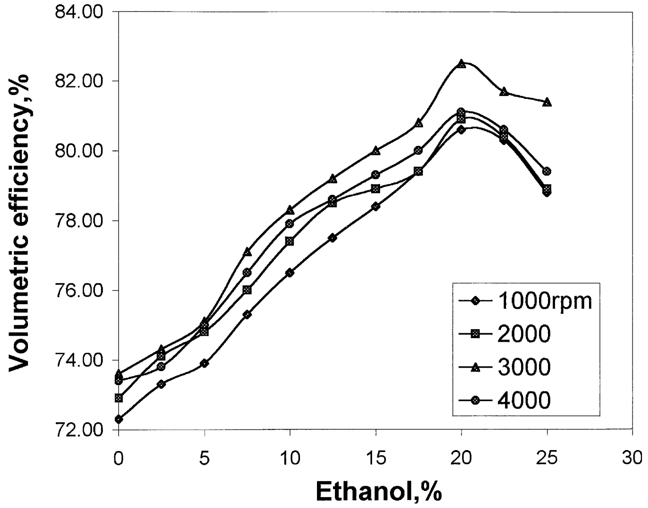

Fig. 4. The effect of ethanol addition on the volumetric efficiency.

the total heat capacity of the charge is more pronounced, and hence, T_v decreases. For a given engine speed and atmospheric temperature, the difference in temperature between the charge and the hot engine parts and residual gases is constant, i.e., $T_h =$ constant. Therefore, T_v changes with E% in the fuel blend as ΔT. This means that T_v changes with the E% in the fuel blends.

From the previous discussion, it is clear that as the E% in the fuel blend increases from 0% to 20%, the volumetric efficiency increases due to the ΔT decrease and T_v increase. Conversely, as the E% changes from 20% to 25%, the volumetric efficiency decreases as ΔT increases and T_v decreases.

The effect of engine speed on η_v can be also explained from Fig. 4. As the engine speed increases to 3000 rpm, η_v increases, as the amount of air introduced to the engine cylinder increases. Further increase in the engine speed results in a decreasing η_v, where the amount of air decreases as a result of choking in the induction system.

3.4. Brake thermal efficiency ($\eta_{b.th}$)

Fig. 5 presents the effect of using ethanol–unleaded gasoline blends on brake thermal efficiency. As shown in the figure, $\eta_{b.th}$ increases as the E% increases. The maximum $\eta_{b.th}$ is recorded with 20% ethanol in the fuel blend for all engine speeds. To discuss the nature of the previous result, it is necessary to discuss the nature of the compression and combustion processes.

The vaporization of fuel continues during the compression stroke. This tends to decrease the temperature of the working charge (i.e., reduces the compression work) and increase the quantity of vapor in the working charge (i.e., increases the compression work). When the latent heat of the fuel used is low, as in the case of unleaded gasoline, the effect of cooling is not sufficient to overcome the effect of additional vapor. Increasing the latent heat of the fuel blend used by increasing the E% increases the effect of cooling (i.e., reduces the compression work).

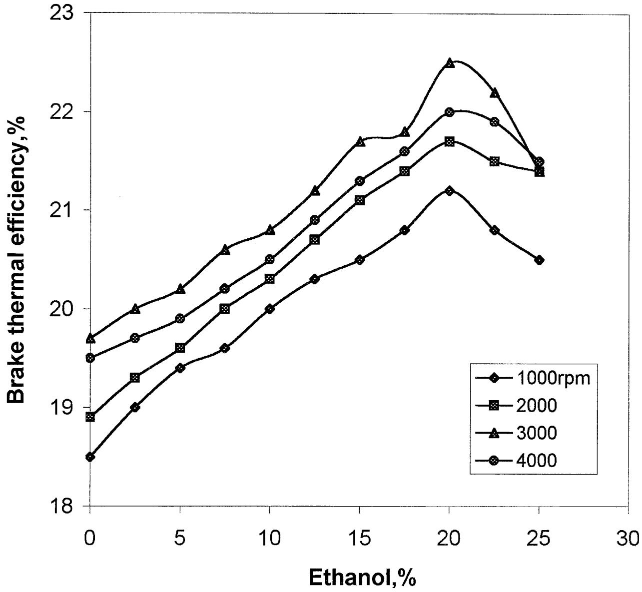

Fig. 5. The effect of ethanol addition on the brake thermal efficiency.

On the other hand, as E% increases in the fuel blend, the pressure and temperature decrease at the beginning of combustion (i.e., the delay period increases or the crank angle at which maximum pressure is achieved increases). However, increasing E% increases the air–fuel ratio, i.e., decreases the heat transfer to the cylinder walls (heat losses) due to incomplete combustion, and therefore, increases the value of maximum pressure.

From the previous discussion, it could be concluded that as the E% increases in the fuel blend, the indicated work increases (i.e., increases the indicated efficiency η_i). Since the mechanical efficiency η_m is a function of engine speed only, the effect of increasing E% on brake thermal efficiency is the same as that on indicated efficiency ($\eta_{b.th} = \eta_i\eta_m$).

A further increase in the E% beyond 20% results in decreasing $\eta_{b.th}$. This behavior has the same explanation as that of the η_v variation with E%.

The effect of engine speed on $\eta_{b.th}$ can be explained through its effect on the equivalence air–fuel ratio (ϕ) and volumetric efficiency (η_v). As the engine speed increases to 3000 rpm, $\eta_{b.th}$ increases, whereas ϕ decreases and η_v increases. Further increases in engine speed beyond 3000 rpm, lead to a decrease in $\eta_{b.th}$, whereas ϕ increases and η_v decreases. This behavior validates the fact that at points were ϕ is minimum (i.e., leaner mixture), the brake thermal efficiency is maximum.

3.5. Brake torque (T) and brake power (B_p)

The effects of ethanol–unleaded gasoline blends on brake torque and brake power is illustrated in Figs. 6 and 7, respectively. It is clear in these two figures that both T and B_p increase as the E% increases for all engine speeds. This increase continues until the E% reaches 20%. After this point, T and B_p start to decrease. This behavior agrees with that of the volumetric and brake thermal efficiencies shown in Figs. 4 and 5.

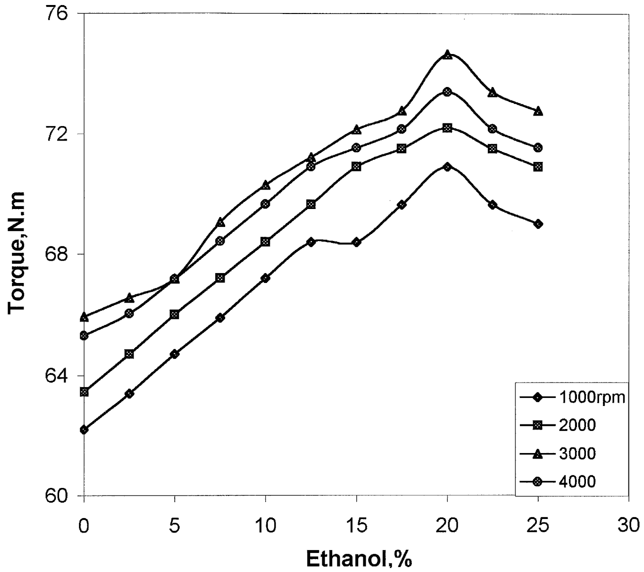

Fig. 6. The effect of ethanol addition on the brake torque.

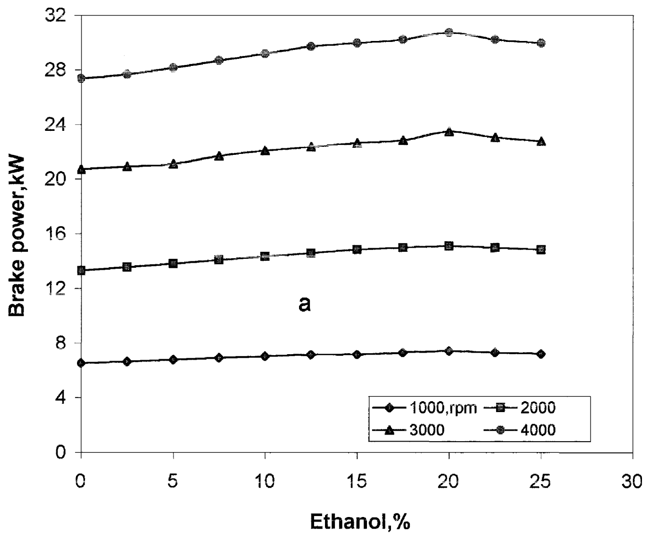

Fig. 7. The effect of ethanol addition on the brake power.

Generally, the brake torque has a significant dependence on the volumetric efficiency and only a slight dependence on the engine speed. As a consequence, the influence of engine speed on T is similar to its influence on the volumetric efficiency. However, as Eq. (7) shows, the brake power is proportional to the product of the engine torque and speed, which suggests that B_p increases as the engine speed increases.

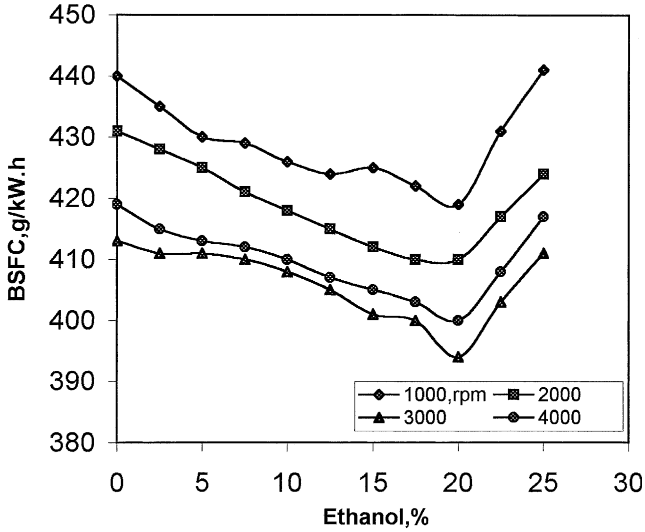

Fig. 8. The effect of ethanol addition on the brake specific fuel consumption.

3.6. Brake specific fuel consumption

Fig. 8 shows the effect of using ethanol–unleaded gasoline blends on brake specific fuel consumption. As shown in this figure, the BSFC decreases as the E% increases up to 20%. This is a normal consequence of the behavior of the engine brake thermal efficiency shown in Fig. 5. On the other hand, as the engine speed increases to 3000 rpm, the BSFC decreases. This is due to the increase in $\eta_{b.th}$ and decreases in ϕ (Figs. 3 and 5). A further increase in engine speed results in increasing BSFC, since the $\eta_{b.th}$ decreases and ϕ increases.

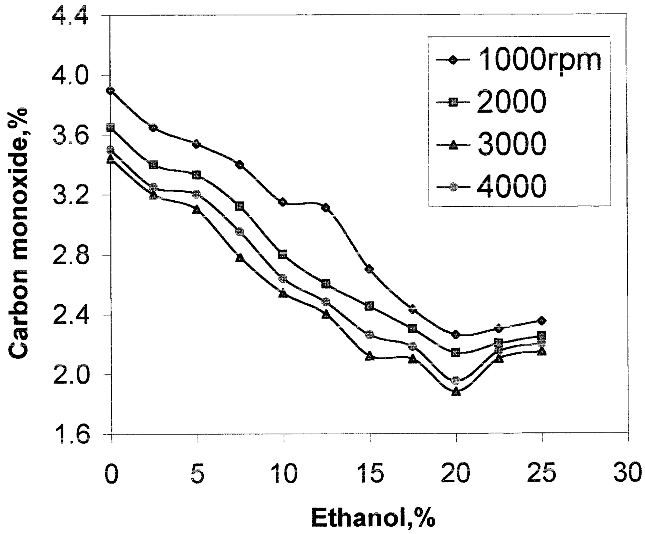

Fig. 9. The effect of ethanol addition on CO emission.

3.7. Exhaust emissions

Figs. 9–11 show the effect of the E% in the fuel blend on the CO, CO_2 and HC. From Figs. 9 and 11, it can be seen that as the E% increases to 20%, the CO and HC concentrations decrease and then increases for all engine speeds. This agrees with the behavior shown in Fig. 3. The CO_2 concentrations have an opposite behavior when compared to the CO concentrations, and this is

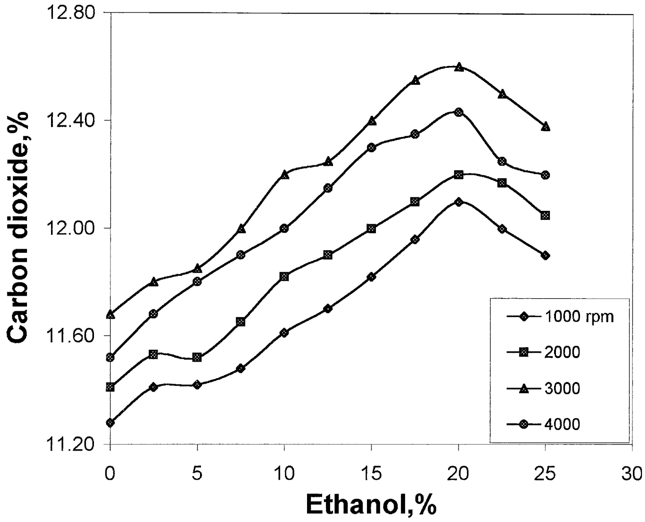

Fig. 10. The effect of ethanol addition on CO_2 emission.

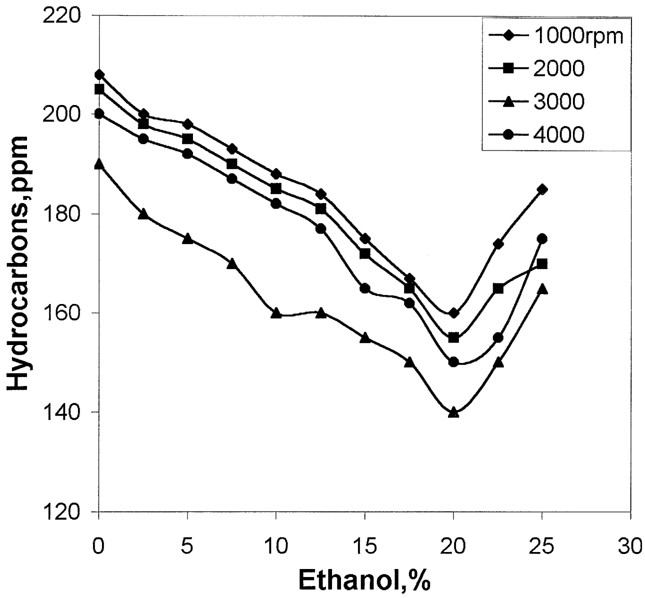

Fig. 11. The effect of ethanol addition on HC emission.

clear in both Figs. 9 and 10. This is due to improving the combustion process as a result of the oxygen content in the ethanol fuel.

4. Conclusions

From the results of the study, the following conclusions can be deduced:

1. Using ethanol as a fuel additive to unleaded gasoline causes an improvement in engine performance and exhaust emissions.
2. Ethanol addition results in an increase in brake power, brake thermal efficiency, volumetric efficiency and fuel consumption by about 8.3%, 9.0%, 7% and 5.7% mean average values, respectively. In addition, the brake specific fuel consumption and equivalence air–fuel ratio decrease by about 2.4% and 3.7% mean average value, respectively.
3. Using an ethanol–unleaded gasoline blend leads to a significant reduction in exhaust emissions by about 46.5% and 24.3% of the mean average values of CO and HC emission, respectively, for all engine speeds. On the other hand, CO_2 emissions increase by about 7.5%.
4. The 20% ethanol fuel blend gave the best results of the engine performance and exhaust emissions.
5. The addition of 25% ethanol to the unleaded gasoline is achieved in our experiments without any problems during engine operation.

Acknowledgements

The author would like to thank the technical staff of the ICE laboratory at the Mechanical Engineering Department of Al-Balqa Applied University and of the chemical laboratory at the Jordan Petroleum Refinery Corporation.

Appendix A. Fuel specification

Property		Gasoline	Ethanol
Formula (liquid)		C_8H_{18}	C_2H_6O
Molecular weight ($kg\,kmol^{-1}$)		114.15	46.07
Density ($kg\,m^{-3}$)		765	785
Heat of vaporization ($kJ\,kg^{-1}$)		305	840
Specific heat ($kJ\,kg^{-1}\,K^{-1}$)	Liquid	2.4	1.7
	Vapor	2.5	1.93
LHV ($kJ\,kg^{-1}$)		44,000	26,900
Stoichiometric air–fuel ratio		15.13	9.00
Enthalpy of formation ($MJ\,kmol^{-1}$)	Liquid	−259.28	224.10
	Gas	−277.0	−234.6

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