Theoretical and experimental investigation of SI engine performance and exhaust emissions using ethanol-gasoline blended fuels

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Abstract—**In this study, potato waste bioethanol was evaluated as an alternative fuel for gasoline engines. The pollutant emissions and performance of a four stroke SI engine operating on ethanol-gasoline blends has been investigated experimentally and theoretically. In the theoretical study, a quasi-dimensional SI engine cycle model has been adapted for spark ignition engines running on gasoline-ethanol blends. A mathematical model using Matlab software was developed using the first law of thermodynamics and conservation equations to predict the SI engine performance for different blend ratios. The model was also used to evaluate the engine emissions and the mechanical and heat losses in the engine which is not included in this study. Experiments were performed with the blends containing 5, 10, 15 and 20 vol% ethanol. The results show that increasing ethanol-gasoline blended will marginally increase the power and torque output of the engine. For ethanol blends it was found that the brake specific fuel consumption (bsfc) was decreased using 5% and 10% ethanol while the brake thermal efficiency and the volumetric efficiency were increased. Exhaust gas emissions were measured and analyzed for unburned hydrocarbons (UHC), carbon dioxide (CO2), carbon monoxide (CO), Oxygen (O2) and Oxide of Nitrogen NOx at engine speeds ranging from 1000 to 5000 rpm. The concentration of CO and UHC emissions in the exhaust pipe were found to be decreased when ethanol blends were introduced. The concentration of CO2 and NOx was found to be increased when ethanol is introduced. Results obtained from both theoretical and experimental studies were compared. The simulation results have been validated against data from experiments and it results to a good agreement between the trends in the predicted and experimental results.**

I. INTRODUCTION

Using renewable energy resources has become an important feature of worldwide energy policy which aims to reduce greenhouse gas emissions caused by fossil fuel usage. Alternative transport fuels such as hydrogen, natural gas and biofuels are seen as an option to help the transport sector in decreasing its dependency on oil and reducing its environmental impact.

According to reference [1], using ethanol-gasoline blend fuel in SI engines lead to higher engine torque in comparison with gasoline fuel. Using E40 and E60 blends led to a significant reduction of CO and HC emissions. It was also reported by reference [2] that blends with ethanol allowed the compression ratio to increase by 50% without knock. The most suitable ethanol-gasoline fuel blend in terms of performance and emissions was E50 in a small gasoline engine with low efficiency [3]. Engine power increased by about 29% running with E50 fuel at high compression ratio compared to running with E0 fuel. The specific fuel consumption, CO, CO2, HC emissions were reduced by approximately 3%, 53%, 10% and 12% respectively. Reference [4] reported that with increasing the ethanol content in gasoline fuel, the heating value of the blended fuels is decreased, while the octane number of the blended fuels increases. NOx emissions are more dependent on the engine operating condition than the ethanol content of the fuel. Reference [5] found that NOx concentrations are adversely affected because of the cylinder temperature increases with increasing ethanol percentage. Ethanol is reported to be an important contributor to decreased engine-out regulated emissions and decreased brake specific energy consumption. The 20 % (vol.) ethanol in the fuel blend gave the best results for all measured parameters at all engine speeds whereas Ceviz et al. (2005) reported that the 10 % (vol.) ethanol in the fuel blend gave the best results [6].

II. EXPERIMENTAL STUDY

A. *Description of the Experimental Setup and Testing Procedure*

In this study, the experiments were performed on a KIA 1.3 SOHC, four cylinder, four-stroke, and spark ignition (SI) gasoline engine. The engine specification is given in Table 1. A 190 kW SCHENCK-WT190 type eddy-current dynamometer was used in the experiments. Fuel consumption rate was measured in the range of 0.4-45 kg/hr by using laminar type flow meter, Pierburg model. Air consumption was measured using air flow meter. The relative air fuel ratio, emission parameters and the exhaust gas temperature from an online and accurately calibrated exhaust gas analyzer DIGAS 4000 type were recorded.

Five separate fuel tanks were fitted to the gasoline engine and these contained gasoline and the bioethanol-gasoline blends. The engine control unit (ECU) that used in this engine is a Johnson Controls JCAE S2000. ECU function is to control the quantity of fuel, injection timing, ignition timing and engine speed by receiving signals from seven sensors. These sensors are oxygen sensor, knock sensor, manifold air pressure sensor, intake air temperature sensor, throttle position sensor, water temperature sensor and engine speed sensor.

Multi point fuel injection (MPFI) system with the top-feed injectors is used to inject the fuel into the combustion chamber. The ignition system was semi-static distributor less ignition (DLI). A schematic diagram of the experimental setup is shown in Fig. 1.

The performance and emission parameters of ethanol derived from potato waste and its blends with gasoline (E5, E10, E15 and E20) were evaluated and compared with gasoline fuel. The properties of ethanol fuel are given in Table 2. Above 20% ethanol, engine could not run smoothly, therefore, experimental results obtained up to this percentage of ethanol will be presented. The fuel blends were prepared just before starting the experiment to ensure that the fuel mixture is homogenous and to avoid of the reaction of ethanol with water.

A series of experiments were carried out using gasoline, and the various bioethanol blends. All the blends were tested under varying engine speed conditions. The engine was started using gasoline fuel and it was operated until it reached the steady state condition. The engine speed, fuel consumption, and load were measured, while the brake power, brake specific fuel consumption (bsfc), brake thermal efficiency and volumetric efficiency were computed. After the engine reached the stabilized working condition, emission parameters such as CO, CO2, HC, NOx and the exhaust gas temperature from an online and accurately calibrated exhaust gas analyzer were recorded. All experiments have been carried out at full throttle setting.

To adjust ignition timing, electronic ignition system was used. The experiments were performed on a multi-point port injection four-cylinder electronic fuel injection. Before running the engine with a new blended fuel, it was allowed to use the new fuel to cleanout the remaining fuel from the pipeline of the engine to avoid the leftover interfering each other.

Fuel properties were determined at the laboratories of Research Institute of Petroleum Industry (RIPI) in Iran. In this study, the legend EX presents a blend including X% bioethanol by volume, i.e. E5 indicates a blend including 5% ethanol in 95% gasoline.

The properties of the five fuels have been summarized in Table 3.

Table 1. THE PROPERTIES OF POTATO'S WASTES ETHANOL

Property	Method	Ethanol
Density (kg/m3)	ASTM D 4052	785
Viscosity (cSt)	ASTM D 88	1.1
Calorific value (KJ/kg)	ASTM D 240	27000
Research octane number	ASTM D 2699	108.6
Pour point (° C)	ASTM D 97	<<-50
Flash point (° C)	ASTM D 93	14
Ash content (mass %)	ASTM D 482	0

Table 2. MAIN CHARACTERISTICS OF THE TEST ENGINE

Engine Type	8 Valves – 4 Cylinder
Combustion Order	1-3-4-2
Bore ×Stroke(mm)	71 * 83.6
Displacement Volume (cc)	1323
Compression Ratio	9.7
Max. Torque (Nm /rpm)	103 / 2750
Max. Power(kW/rpm)	47 / 5200
Max. Speed (rpm)	6200
Cooling System	Liquid, cooling fluid

Table 3. PROPERTIES OF DIFFERENT ETHANOL/GASOLINE-BLENDED FUELS

Property	Test fuel					Test Method
	E0	E5	E10	E15	E20	
Vapor pressure (KPa), reid@37.8°C	48	55	55	55	55	ASTM D323
Octane number (research)	85	90	92	94	99	ASTM-D2699
Gross heat combustion (MJ/kg)	45	44	43	42	40	ASTM-D340
Distillation range (°C) @760mmHg						ASTM-D86
IBP	36	41	39	44	41	
10 vol%	59	54	53	58	55	
50 vol%	93	94	72	71	72	
90 vol%	146	148	144	145	142	
End Point	177	184	175	182	177	

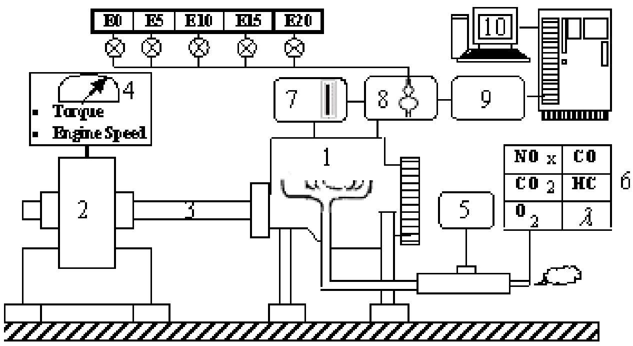

Figure 1. Schematic diagram of experimental setup.1-Engine, 2-Dynamometer, 3- Drive shaft, 4- Dynamometer control unit, load and speed indicator, 5- Temperature indicator, 6- Gas

analyzer, 7- Air flow meter, 8- Fuel measurement system, 9- Measuring boom, 10- Computer.

B. Engine Performance Parameters

When the ethanol content in the blend fuel is increased, the engine brake power is slightly increased for all engine speeds (Fig. 2). The gain of the engine power was due to the increase of the indicated mean effective pressure and the in cylinder pressure due to the higher ethanol content in the blends. The heat of evaporation of ethanol is higher than that gasoline, this provides fuel-air charge cooling and increases the density of the charge, and thus higher power output is obtained. The increase of ethanol content will increase the torque of the engine. Added ethanol will produce lean mixture that increases the relative air-fuel ratio (λ) to a higher value and makes the burning more efficient. The improved anti-knock behavior (due to the addition ethanol, which raised the octane number) allowed a more advanced timing that result in higher combustion pressure and thus higher torques (Fig. 3).

The brake thermal efficiency increased as the ethanol percentage increased. The maximum brake thermal efficiency was approximately 35% when 20% ethanol is in the fuel blend (Fig. 4).

The volumetric efficiency increased as the ethanol percentage increased for all engine speeds (Fig. 5).

The bsfc decreased as the ethanol percentage increases. This is a normal consequence of the behavior of the engine brake thermal efficiency (Fig. 6).

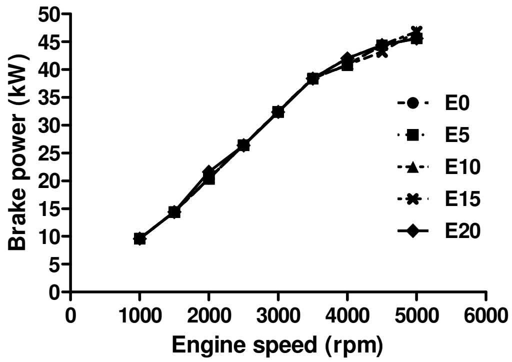

Figure 2. Experimental results of brake power at different fuel blends and engine speeds

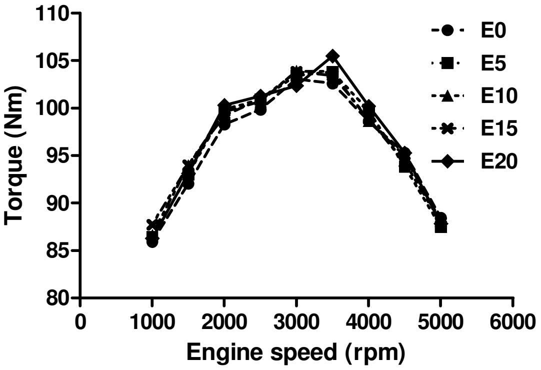

Figure 3. Experimental results of Torque, at different fuel blends and engine speeds

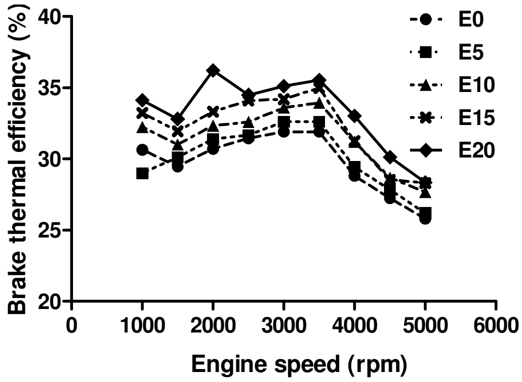

Figure 4. Experimental results of brake thermal efficiency, at different fuel blends and engine speeds

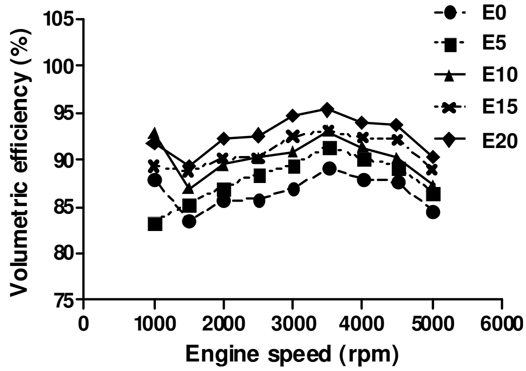

Figure 5. Experimental results of volumetric efficiency, at different fuel blends and engine speeds

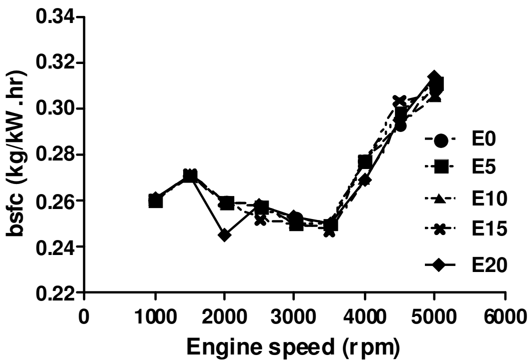

Figure 6. Experimental results of brake specific fuel consumption at different fuel blends and engine speeds

C. Engine emission studies

When the ethanol percentage increased, the CO concentrations decreased, indicating more complete combustion with ethanol blend. Fig. 7 shows the concentrations of CO emission for different engine speeds. It can be seen from this figure that when ethanol percentage increases, the CO concentration decreases which means the combustion is tuned to be completed. The CO concentration in the exhaust gas emission at 3000 rpm for gasoline fuel was 4.69 (%V), while the CO concentration of E5, E10, E15 and E20 at 3000 rpm was 4.05 (%V), 3.55 (%V), 3.38 (%V) and

2.56 (%V) respectively. The CO concentrations at 3000 rpm using E5, E10, E15 and E20 was decreased by 13.7%, 24.31%, 27.93% and 45.42% respectively in comparison to gasoline. This decrease in CO concentration was due to the fact that ethanol has less carbon than gasoline. Also, given the same fuel dispersion pattern as for gasoline, the oxygen content of the blended fuels would help to increase the oxygen-to-fuel ratio in the fuel-rich regions consequently combustion becomes more complete. It was found that the CO2 concentration increased as the ethanol percentage increased. CO2 emissions depend on relative air-fuel ratio and CO emission concentration (Fig. 8). The CO2 concentration in the exhaust gas emission at 3000 rpm for gasoline fuel was 12.4 (%V), while the CO2 concentration of E5, E10, E15 and E20 at 3000 rpm was 12.9 (%V), 13.2 (%V), 13.3 (%V) and 13.8 (%V) respectively. The CO2 concentrations at 3000 rpm using E5, E10, E15 and E20 was increased by 3.87%, 6.06%, 6.76% and 10.14% respectively in comparison to gasoline. HC emissions for different speeds are illustrated in Fig. 9. The HC concentration in the exhaust gas emission at 3000 rpm for gasoline fuel was 183 (ppm), while the HC concentration of E5, E10, E15 and E20 at 3000 rpm was 152 (ppm), 139 (ppm), 137 (ppm) and 125 (ppm) respectively. The HC concentration at 3000 rpm using E5, E10, E15 and E20 was decreased by 16.94%, 24.04%, 25.14% and 31.69% at 3000 rpm, respectively in comparison to gasoline. This result indicates that ethanol can significantly reduce HC emissions. The concentration of HC emission 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.. Considering the NOx emission, Fig. 10 shows that the NOx concentration is higher when ethanol percentage increases. It shows that as the percentage of ethanol in the blends increased, NOx emission was increased. The NOx concentration in the exhaust gas emission at 3000 rpm for gasoline fuel was 876 (ppm), while the NOx concentration of E5, E10, E15 and E20 at 3000 rpm was 1002 (ppm), 1326 (ppm), 1319 (ppm) and 1609 (ppm) respectively. The NOx concentrations at 3000 rpm using E5, E10, E15 and E20 was increased by 12.57%, 33.94%, 33.6% and 45.55% respectively in comparison to gasoline.

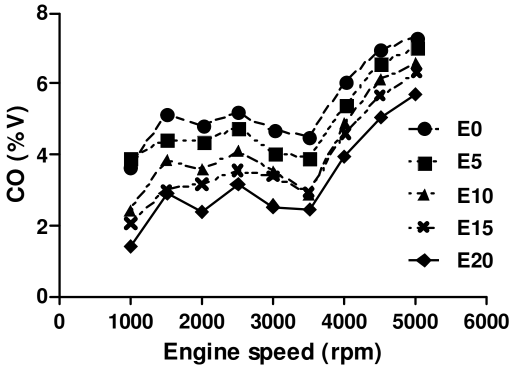

Figure 7. Experimental results of CO at different gasoline-ethanol blends and engine speeds

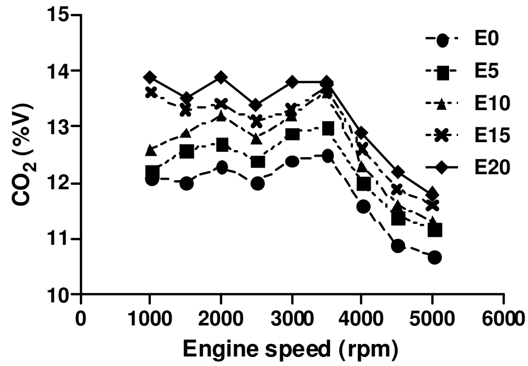

Figure 8. Experimental results of CO_2 at different gasoline-ethanol blends and engine speeds

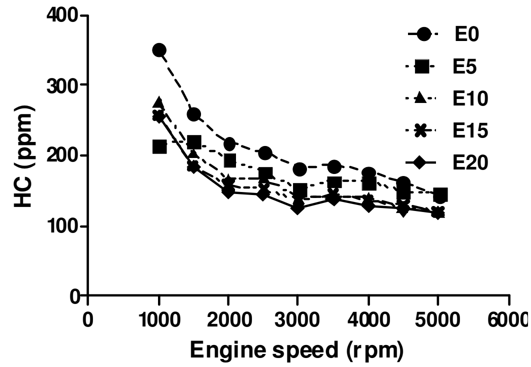

Figure 9. Experimental results of HC at different gasoline-ethanol blends and engine speeds

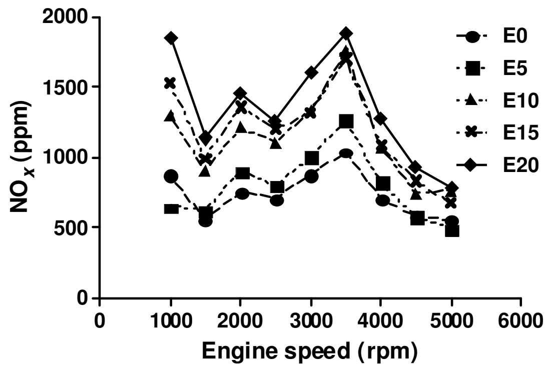

Figure 10. Experimental results of NO_x at different gasoline-ethanol blends and engine speeds

III. THEORETICAL STUDY

The predictive capability of a thermodynamic simulation is currently being tested using the experimental bioethanol-gasoline results of the previous section. From the first law of thermodynamics, the internal energy of the engine cylinder system can be defined as:

$$\Delta U = Q - W \tag{1}$$

Taking the derivative of equation (1) with respect to crank angle, the energy equation can be written:

$$m\frac{du}{d\theta}+u\frac{dm}{d\theta}=\frac{dQ}{d\theta}-P\frac{dV}{d\theta}-\frac{\dot{m}_1 h_1}{\omega} \quad (2)$$

Where m is the mass of gas in the zone under consideration, m1 is cylinder leakage mass flow rate, h1 is enthalpy of the blowby mass, u is internal energy, Q is heat transfer, p is pressure, V is volume, θ is crank angle and ω is engine speed. We can use a similar equation in the burned and unburned zones of the engine and the subscripts b and u are used to represent the burned and unburned gas zones. Relevant derivatives can be expressed as functions of crank angle, pressure, unburned gas temperature and burned gas temperature. Solving these equations with appropriate input data enables the determination of the indicated work, enthalpy and heat loss throughout the system since indicated work; enthalpy and heat loss can be expressed as a function of pressure and temperature as well.

$$\frac{dP}{d\theta}=f_1(\theta,P,T_b,T_u),\ \frac{dT_b}{d\theta}=f_2(\theta,P,T_b,T_u),\ \frac{dT_u}{d\theta}=f_3(\theta,P,T_b,T_u) \quad (3)$$

$$\frac{dW}{d\theta}=f_4(\theta,P),\ \frac{dQ_l}{d\theta}=f_5(\theta,P,T_b,T_u),\ \frac{dH_l}{d\theta}=f_6(\theta,P,T_b,T_u) \quad (4)$$

Properties of pure gasoline fuel and gasoline-ethanol blended fuels are determined at the beginning of the cycle. Certain properties of ethanol and gasoline are given in Table 4. If the properties of pure ethanol and gasoline are known, properties of the blended fuels are calculated as follows (assumes 'blend ratio' is available and is the liquid volume fraction of ethanol in the ethanol-gasoline mixture) [5]:

$$\rho=\sum_{i=1}^{2}(X_V)_i\times\rho_i \quad (5)$$

$$(F/A)_{sbl}=\frac{\sum X_{Vi}\times\rho i(F/A)si}{\sum X_{Vi}\times\rho_i} \quad (6)$$

Which:

$i=1,2$ $1=Gasoline; C_7H_{17}$ $2=Ethanol; C_2H_5O$

ρ =density of blended fuel (g/Cm^3)

ρ_i=density of given component in fuel blend (gasoline and ethanol)

$(X_v)_i$=volume fraction of given component in fuel blend , (vol.%)

$(F/A)_{sbl}$ = Stoichiometric Fuel Air ratio of blended fuel

$(F/A)si$ = Stoichiometric Fuel Air ratio of given component in fuel blend (gasoline and ethanol)

Table 4. GASOLINE AND ETHANOL FUELS PROPERTIES
Table 5.

Property	Gasoline	Ethanol
Density (kg/m^3)	800	700
Molecular formula	C_7H_{17}	C_2H_6O
Molecular weight (kg/kmol)	101	46
Stoichiometric fuel/air ratio	0.07	0.11

IV. RESULTS AND COMPARISONS

The simulations were performed using Matlab, and results were obtained to examine the SI engine performance using different blend ratios. Experimental and theoretical results have been compared at the same operating conditions and results obtained for gasoline-ethanol blended fuels and gasoline. Results are compared graphically in the following figures. Ethanol addition to gasoline leads to leaner operation and causes to increase decrease the fuel-air equivalence ratio (Fig. 11).

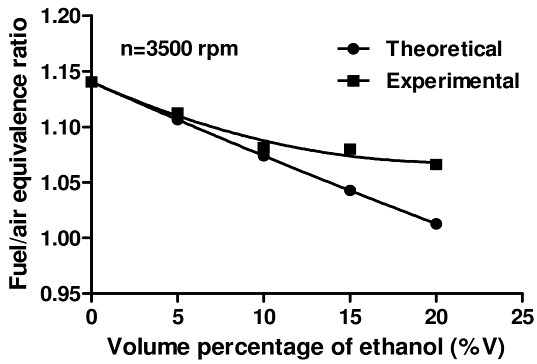

Figure 11. Fuel/air equivalence ratio with volume percentage of Ethanol at WOT condition

Ethanol addition to gasoline causes leaner operation as shown in figure 11, and completion of combustion therefore, flame temperature and cylinder pressure rise to their maximum values [5]. From this figure it can be seen that the increase of the ethanol lead to reduce the fuel percentage in the mixture which means lean combustion.

The effects of ethanol addition to gasoline on cylinder pressure and temperature have been investigated theoretically. The theoretical maximum cylinder pressure at 3500rpm for gasoline fuel was 43.9 bar, while the theoretical maximum cylinder pressure of E5, E10, E15 and E20 at 3500 rpm was 44.2 bar, 44.5 bar, 45.5 bar and 48 bar respectively. The theoretical maximum cylinder pressure at 3500 rpm using E5, E10, E15 and E20 was increased by 0.68%, 1.35%, 3.52% and 8.5% respectively in comparison to gasoline. The theoretical maximum cylinder temperature at 3500rpm for gasoline fuel was 2664 (°K), while the theoretical maximum cylinder temperature of E5, E10, E15 and E20 at 3500 rpm was 2666 (°K), 2667 (°K), 2672 (°K), and 2674 (°K) respectively. This showed that there is a slight increase of the temperature which means higher power but may lead to NOx concentration increased. While this model testing and running, some parameter was maintained to be identical to the experimental ones. These parameters are the ignition temperature, ignition duration, masses per cylinder per cycle, initial pressure and temperatures as well the heat transfer techniques that commonly used for the heat lost calculations.

As shown in figure, the brake mean effective pressure of the engine using bioethanol were generally increased due to the increase of the in cylinder pressure (Fig. 12), therefore, engine brake power and brake was found to be increased with increasing ethanol volume percentage in the blended fuel (Fig. 13). Figure 14 describes the effect of the blend ratios on the brake thermal efficiency BTE. The BTE noticed to be experimentally increased when the blend ratio increased, this nicely match with the theoretical model. But the gap between the real and the theoretical results seems to be increase when blend ratio above 15%. From this figure, the BTE is dropped when 20% ethanol added; this was due to the Ethanol low heating value of 42 MJ/Kg in comparison to the gasoline heating value which is 48 MJ/kg. Therefore more fuel is needed to obtain same power as gasoline [5]. For this reason, brake specific fuel consumption for ethanol/gasoline blends of 15% and 20% ethanol is higher than gasoline fuel (Fig. 15).

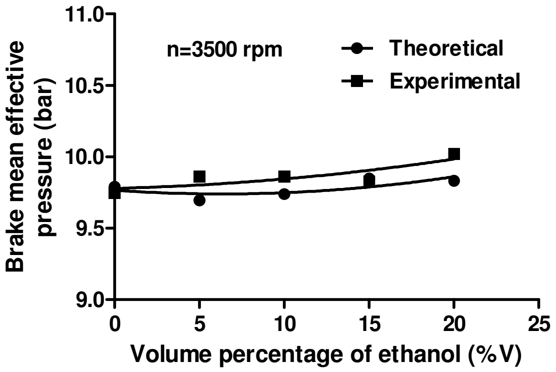

Figure 12. brake mean effective pressure with volume percentage of ethanol at WOT condition

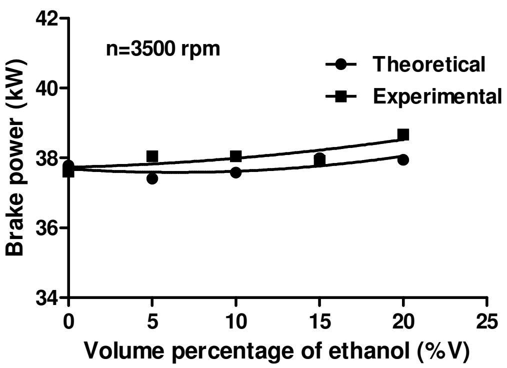

Figure 13. brake power with volume percentage of ethanol at WOT condition

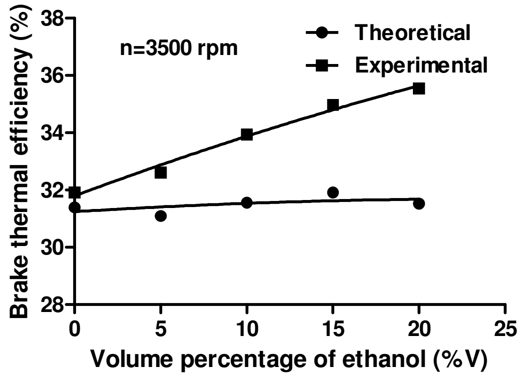

Figure 14. brake thermal efficiency with volume percentage of ethanol at WOT condition

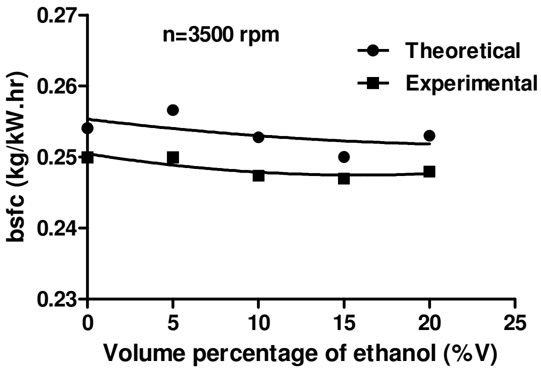

Figure 15. brake specific fuel consumption with volume percentageof ethanol at WOT condition

V. CONCLUSION

Adding ethanol to gasoline will lead to a leaner better combustion. It was experimentally demonstrated that adding 5-15% ethanol to the blends led to an increase in the engine brake power, torque and brake thermal efficiency, volumetric efficiency and decreases the brake specific fuel consumption. The lean combustion improves the completeness of combustion and therefore the CO emission is expected to be decreased. The oxygen enrichment generated from ethanol increased the oxygen ratio in the charge and lead to lean combustion. The CO2 emission increased because of the improvement of the combustion and the chemical properties of Ethanol. Unburned HC is a product of incomplete combustion which is related to A/F ratio. It can be concluded that that adding ethanol to the blends will reduces the HC emission because of oxygen enhancement. When the combustion process is closer to stoichiometric, flame temperature increases, therefore, NOx formation is expected to be increased. The results obtained with presented theoretical model are in acceptable agreement with those experimental ones. An agreement of 11% was determined between experimental and theoretical results.

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