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Performance and Emission Comparison and Investigation of Alternative Fuels in SI Engines

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ABSTRACT

Alternative fuels are of great interest since they can be refined from renewable feedstocks, and their emission levels can be lower than those of conventional fueled engines. Despite the fact that alternative fuels are not currently widely used in vehicular applications, using these kinds of fuels is definitely inevitable in the future. In this paper, a computer code is developed in Matlab environment and then its results are validated with experimental data. This simulated engine model could be used as an appropriate mean to investigate the performance and emission of a given SI engine fueled by alternative fuels including hydrogen, propane, methane, ethanol and methanol. Also, the superior of alternative fuels is shown by comparing the performance and emissions of alternative fueled engines to those in conventional fueled engines.

INTRODUCTION

The strict regulation of environmental laws, the price of oil and its restricted resources, has made the engine manufacturers use other energy resources instead of oil and its products. Alternative fuels are derived from resources other than petroleum. The benefit of these fuels is that they emit less air pollutant compare to gasoline and most of them are more economically beneficial compared to oil and they are renewable [1-6]. Alternative fuels such as methane, propane, ethanol, methanol and hydrogen have been used since 1930 [1]. During the 1970s the price of crude petroleum rose rapidly. Its cost (in real terms) in 1970 and concern built up regarding the longer-term availability of petroleum. Pressures for substantial improvements in internal combustion engine (ICE) efficiency (in all its many applications) have become very substantial indeed. Much work is being done on the use of alternative fuels instead of gasoline and diesel [1]. Of the non-petroleum-based fuels, natural gas, methanol and ethanol (methyl and ethyl alcohols) are receiving the greatest attention, while synthetic gasoline and diesel made from shale oil or coal, and hydrogen could be longer-term possibilities [1]. The most common fuels that are used as alternative fuels are natural gas, propane, ethanol, methanol and hydrogen.

Methane, the main content of natural gas (up to 96 percent), is the most common alternative fuel and is one of the cleanest burning fuels [2]. It can be used in the form of compressed natural gas (CNG) or liquefied natural gas (LNG) to fuel vehicles. Dedicated natural gas vehicles are designed to run on natural gas only. Dual-fuel or bi-fuel vehicles are capable of operating on either gasoline or natural gas. That allows alternative fuel usage which is more economical without sacrificing vehicle range and mobility with wide-spread availability of gasoline or diesel [2].

Ethanol and methanol are alcohol-based fuels made by fermenting and distilling starch crops, such as corn. Both ethanol and methanol produce less emission than gasoline [3]. In Brazil, ethanol is well known as a clean, economic and available fuel for vehicles. But engines work on alcoholic fuels will experience a decrement in brake torque and power compared to gasoline [3].

Propane or liquefied petroleum gas (LPG) is a clean-burning fossil fuel that can be used to power ICEs. LPG-fueled vehicles produce fewer toxic and smog-forming air pollutants. LPG is usually less expensive than gasoline [4].

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Hydrogen (H2) is an attractive alternative energy carrier. Hydrogen is being widely explored as a fuel for passenger vehicles. It can be used in fuel cells to power electric motors or burned in ICEs. Hydrogen could make harmful emissions, global warming and the insecurity concerning oil supply a thing of the past since it produces no air pollutants or greenhouse gases when used in fuel cells and it produces only NOx when burned in ICEs. [5, 6]

In this paper a full detailed comparison between some alternative fuels has been made. For approaching this aim, the experiment study is done in term of engine performance and exhaust emissions for gasoline, methane and methanol in a wide range of operating conditions. In order to examine other fuels performance and comparisons between them a thermodynamic model of an SI engine in Matlab environment is developed. Combustion of each one of these fuels differs from others so the code has been modified for each of them. For estimating the turbulent speed of flame, different methods have been used. The code has the ability to evaluate performance and emission characteristic of an engine such as brake power, brake torque, brake mean effective pressure, volumetric and thermodynamic efficiency, NOx and CO concentrations.

EXPERIMENTAL SETUP

Experiments were carried out on a four cylinder Mazda B2000i engine. The engine and dynamometer specifications are listed in table 1 and 2.

The test engine was coupled to an eddy-current dynamometer for engine speed and load measurement. In-cylinder pressure data was taken by a Kistler 6117B piezoelectric high-pressure transducer. The crank shaft position was measured by a Kistler 2613B with a resolution of 1 degree of CA. Emission data were taken using a Pierburg HGA 400 exhaust gas analyzer. Air/fuel ratio was monitored by a HORIBA lambda analyzer. Emerson micro motion elite sensor and AVL 753 fuel mass flow meter were used to measure the mass and temperature of injected fuel. Tests have been done for gasoline, methane and methanol under engine steady state conditions. For engine running on methane, a CNG kit was installed. The CNG kit used in the tests was PRINS (VSI). Also a CNG storage tank was used. Fuels properties that have been used in presented research are listed in appendix I.

MODELING

Most of the predictive engine models generally drop within three types: zero-dimensional, Quasi-dimensional and multi-dimensional ones. Zero-dimensional model is

Table 1 Engine specifications

Engine type	Four stroke, spark ignition
Induction	Naturally aspirated
Number of cylinders	Four in line
Bore (mm)	86
Stroke (mm)	86
Connecting rod length (mm)	153
Compression ratio	8.6
Max. power	70 kw @5000 (rpm)
Max. torque	151 $N.m$ @2500 (rpm)
Number of valves per cylinder	3
Intake valve opening (CA)	10 BTDC
Intake valve closing (CA)	49 ABDC
Exhaust valve opening (CA)	55 BBDC
Exhaust valve closing (CA)	12 ATDC

Table 2 Dynamometer properties

Dynamometer Type	Ricardo FE 760-S
Max. torque ($N.m$)	610
Max speed (rpm)	12000
Max power (kw)	191.17
Inertia (kg/m^2)	0.176
Tensional spring $(N.m/rad)*1000$	239
Weight (kg)	474

a thermodynamic model based on mass and energy conservation only. Zero-dimensional models totally neglect all the geometries and treats combustion simply as a heat addition process. The rate of heat addition is usually acquired by empirical formulas, such as Wiebe functions, which approximate the shape of experimental observed burn rate curve by calibration [1]. Disadvantage of this kind of models is that the model cannot predict burn rate because it cannot simulate the flame propagation. Rousseau et al. in their work did a great number of experiments to find the correlation between the three constants in Wiebe functions (which are required for the simulation of burn rate) with engine operating variables [7]. Multi-dimensional model is the most powerful model for predicting engine performance and emission characteristics, it is also the most complicated one [1]. Basically, it solves combustion chemistry and species transport equations with the 3D computational fluid dynamics (CFD) model so this model runs very slow [1]. In presented work, a quasi-dimensional model is used which, has the ability to simulate burn rate (by using the turbulent combustion model) and, compared to the multi-dimensional model, is quite efficient and faster.

The engine model is a quasi-dimensional two-zone model which solves the basic differential equations for the intake, compression, power and exhaust strokes. In this model, the combustion chamber is divided into two zones, which one should be considered as being divided by flame front, zone 1 contains unburned mixture and zone 2 contains burned mixture. This can be observed in Fig.1. Thermal NOx formation also takes place in burned zone, which is described by the extended Zeldovich mechanism [1]. The flame front is assumed to travel by a speed called turbulent flame speed which is a function of laminar flame speed. The engine model uses Woschni correlation [8] to estimate engine heat transfer. It is assumed that the flame travels in a sphere like shape.

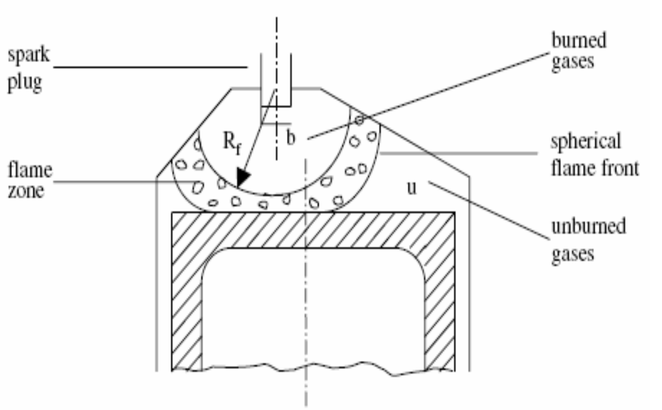

Figure 1. Schematic representation of cylinder contents during combustion period [4].

The engine model also includes a friction model to predict brake mean effective pressure. The frictional processes in an internal combustion engine can be categorized into three main components:

1. The mechanical friction
2. The pumping work
3. The accessory work

For this work a method, which calculates the total friction work accurately, was used [9]. The applied friction model spontaneously predicts all the categories above. The composition of the reaction products is calculated from the chemical equilibrium at a given pressure and temperature of the 12 species N2 , NO, N, CO2, CO, OH, H,O2, O, H2O, H2, Ar. An optimization calculation procedure is used to calculate the mole fraction of each species and the total mole fraction [10].

FUELS LAMINAR FLAME SPEED

Since the presented model is predictive, it must evaluate the burn rate of the in-cylinder mixture. The burn rate is completely affected by flame speed of fuel so flame speed is a very important parameter in engine performance and emission characteristics. First of all laminar flame speed of mixture must be calculated then by using laminar flame speed, turbulent flame speed can be calculated.

Various laminar flame speed measurement methods for different fuels have been presented in literatures [11, 12, 13, 14, 15 and 16]. The applied methods in this paper will be discussed as following.

HYDROGEN - Various kinds of relations for laminar burning velocity of Hydrogen/Air mixture have been proposed in previous works. But in this paper Iljima and Takeno's formulation [11] has been used because in their correlation, flame speed is a function of equivalence ratio, temperature and pressure. But other formulas are not a function of all of the parameters above and one of the parameters does not exist in other formulas. According to Iljima and Takeno's formula, laminar burning velocity of hydrogen/air mixture is:

$$u_l = u_{l0} \left(\frac{T_u}{T_0}\right)^{\alpha_T} \left[1 + \beta_p Log \frac{P}{P_0}\right] \qquad (1)$$

Where; P is the pressure, T_u is the unburned temperature, $T_0 = 358(K), P_0 = 0.1(MPa)$. α_T and β_p are as follows:

$$\alpha_T = 1.54 + 0.026(\phi - 1) \qquad (2)$$

$$\beta_p = 0.43 + 0.003(\phi - 1) \qquad (3)$$

u_{l0}, which is the laminar burning velocity of hydrogen at 291 K and 1 (atm) , in (m/s), given by:

$$u_{l0} = 2.98 - (\phi - 1)^2 + 0.32(\phi - 1.70)^3 \quad (4)$$

and ϕ is the equivalence ratio[11].

ETHANOL - Burning velocity of Ethanol/Air mixture is calculated based on a recent work [12]. According to that research, laminar burning velocity is given by the formulation below:

$$u_l = u_{l0} (\frac{T_u}{T_0})^{\alpha_T} \left[1 + \beta_p Log \ \frac{P}{P_0} \right] \quad (5)$$

Where; $T_0 = 358(K)$ and $P_0 = 0.1(MPa)$. α_T and β_p are as follows:

$$\alpha_T = 1.783 - 0.375 \ (\phi - 1) \quad (6)$$

$$\beta_p = \begin{cases} -0.17\sqrt{\phi} & \phi \geq 1.0 \\ -0.17 / \sqrt{\phi} & \phi < 1.0 \end{cases} \quad (7)$$

u_{l0}, which is the laminar burning velocity of ethanol at 358 K and 1 (atm), in (m/s) given by:

$$u_{l0} = -2.0707 \ \phi^2 + 4.501 \ \phi - 1.8971 \quad (8)$$

METHANOL, PROPANE, GASOLINE - Burning velocity of methanol, propane and gasoline can be calculated from Metghalchi and Keck formulation [1]. Their correlation is defined as following.

$$u_L = u_{L,0} (\frac{T_u}{T_0})^{\alpha} (\frac{P}{P_0})^{\beta} (1 - 2.06 x_b^{\ 0.77}) \quad (9)$$

Where; $T_0 = 298 \ K$ and $P_0 = 1(atm)$ are the reference temperature and pressure, and $u_{l,0}$, α and β are functions of equivalence ratio for a given fuel, and x_b is unburned gas diluent fraction. For propane, isooctane and methanol, these constants can be represented as follows:

$$\alpha = 2.18 - 0.8(\phi - 1) \quad (10)$$

$$\beta = -0.16 + 0.22(\phi - 1) \quad (11)$$

$$u_{l,0} = B_m + B_{\phi}(\phi - \phi_m)^2 \quad (12)$$

Where ϕ_m is the equivalence ratio at which $u_{l,0}$ is maximum with respect to value B_m.Values of ϕ_m, B_m and B_{ϕ} are listed in table 3.

Table 3. Values of ϕ_m , B_m and B_{ϕ}

Fuel	ϕ_m	$B_m (m/s)$	$B_{\phi} (m/s)$
Methanol	1.11	0.369	-1.405
Propane	1.08	0.342	-1.387
Gasoline	1.21	0.305	-0.549

METHANE - Flame speed of methane/air mixture is calculated according to X. J. Gu et al. formulation [13]. In their literature, a simple correlation of burning velocities through the empirical expression below was used [14]:

$$u_l = u_{l,0} \left(\frac{T_u}{T_0} \right)^{\alpha_T} \left(\frac{P_u}{P_0} \right)^{\beta_P} \quad (13)$$

The parameters α_T and β_P which depend upon ϕ, are optimized over the full experimental data. The X. J. Gu et al. formulation is given by:

$$u_l = 0.259 \left(\frac{T_u}{T_0} \right)^{2.105} \left(\frac{P_u}{P_0} \right)^{-0.504} \quad \text{for } \phi = 0.8 \quad (14)$$

$$u_l = 0.360 \left(\frac{T_u}{T_0} \right)^{1.162} \left(\frac{P_u}{P_0} \right)^{-0.374} \quad \text{for } \phi = 1.0 \quad (15)$$

$$u_l = 0.314 \left(\frac{T_u}{T_0} \right)^{2.000} \left(\frac{P_u}{P_0} \right)^{-0.438} \quad \text{for } \phi = 1.2 \quad (16)$$

In their literature standard deviations of the difference between experimental and simulation were 0.008, 0.011, and 0.014 (m/s) for ϕ=0.8, 1.0, and 1.2 respectively.

TURBULENT FLAME SPEED

Many methods for describing and calculating the turbulent flame speed have been developed by many researches. In this paper “Damkohler and derivatives” method is used [15, 16]. According to this model turbulent flame speed is as follows:

$$u_t = u' + u_l \quad (17)$$

Where:

$$u' = u'_{TDC}(1 - 0.5\frac{\theta - 360}{45}) \quad (18)$$

$$u'_{TDC} = 0.75\overline{U}_P \quad (19)$$

θ is the crank angle (360 at TDC of compression).

MODEL VALIDATION

In order to validate the above predictive model for all desired fuels, several experiments were conducted. And several diagrams for in-cylinder pressure (P) versus crank shaft position (crank angle) and NOx concentration were achieved for each one of the fuels in a wide range of engine operating conditions. Recorded data from experimental tests and the results that achieved from developed codes were compared. In Figs.2 to 7 the in-cylinder pressure versus crank shaft position is shown for gasoline, methane and methanol. It can be seen that the predicted and experimental data match quite well.

Propane, ethanol and hydrogen were not tested in presented experiment. For validating model for ethanol, hydrogen and propane, some experimental works studied and the model calibrated in order to match them [3, 4 and 17]. Figs. 8, 9 and 10 are comparisons between experimental data and model predictions. As it can be seen, a very small deviation between simulation results and experimental data exist .The validity of applied relations, mechanisms and assumptions in model are proven.

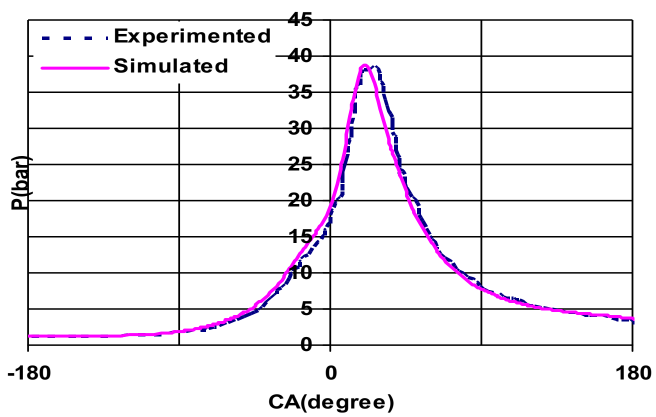

Figure 2 Comparison between experimental and simulated data. Fuel: Gasoline ,Engine speed: 5000rpm, Fuel/Air equivalence ratio: 1, Spark advance: 30 BTDC

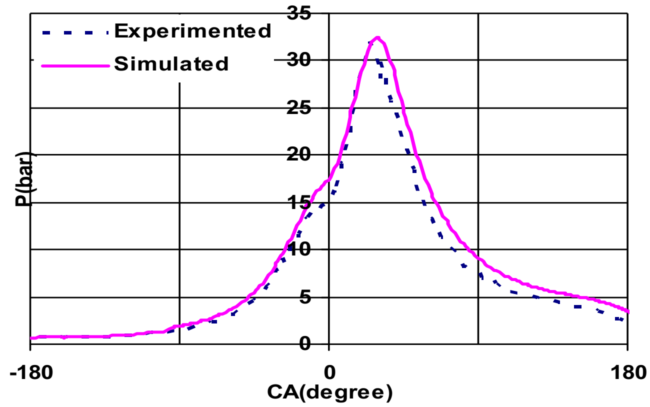

Figure 3 Comparison between experimental and simulated data. Fuel: Gasoline, Engine speed: 2000rpm, Fuel/Air equivalence ratio: 0.856, Spark advance: 12 BTDC

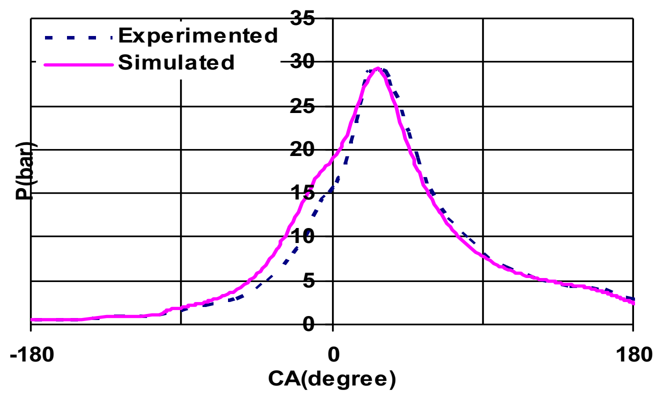

Figure 4 Comparison between experimental and simulated data. Fuel: Methane, Engine speed: 2000rpm, Fuel/Air equivalence ratio: 1, Spark advance: 27 BTDC

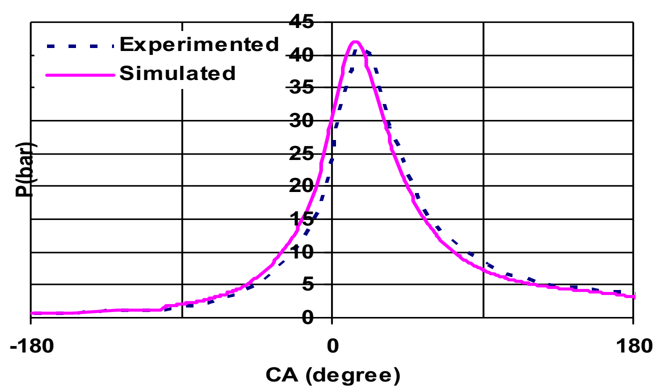

Figure 5 Comparison between experimental and simulated data. Fuel: Methane, Engine speed: 5000rpm, Fuel/Air equivalence ratio: 1, Spark advance: 53 BTDC

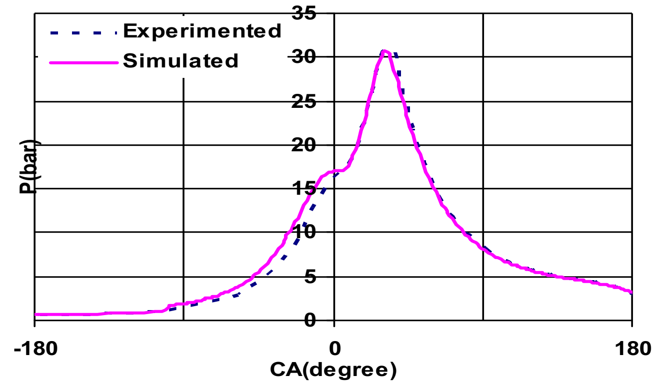

Figure 6 Comparison between experimental and simulated data. Fuel: Methanol, Engine speed: 2000rpm, Fuel/Air equivalence ratio:1, Spark advance: 10 BTDC

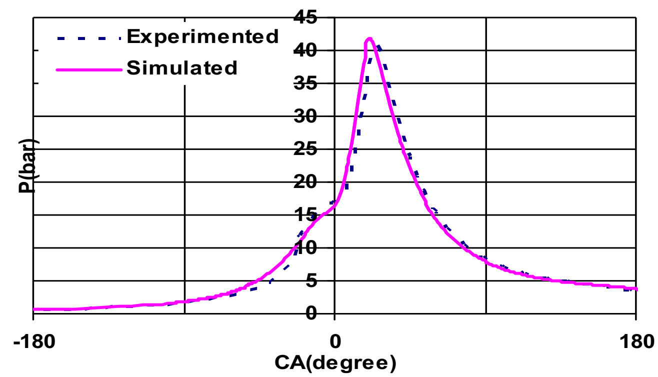

Figure 7 Comparison between experimental and simulated data. Fuel: Methanol, Engine speed: 5000rpm, Fuel/Air equivalence ratio: 1, Spark advance: 16 BTDC

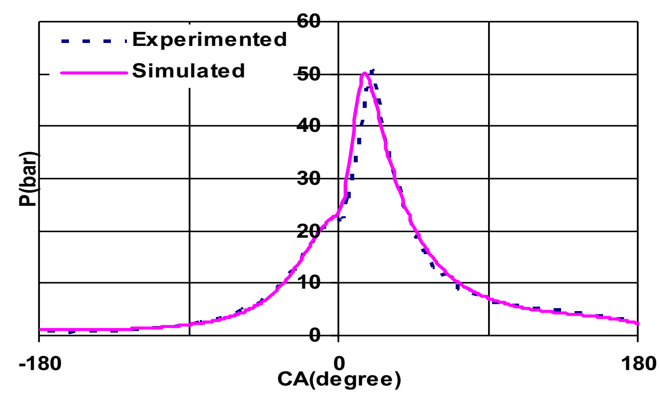

Figure 8 Comparison between experimental and simulated data. Fuel: Hydrogen, Engine speed: 2830rpm, Fuel/Air equivalence ratio: 1.06, Spark advance: 4 BTDC

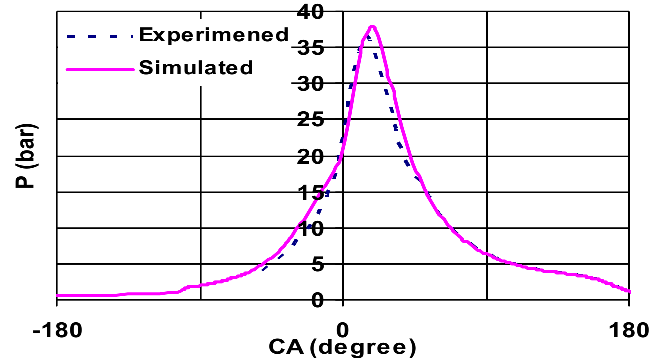

Figure 9 Comparison between experimental and simulated data. Fuel: Propane, Engine speed: 1500rpm, Fuel/Air equivalence ratio: 0.869 , spark advance: 16 BTDC

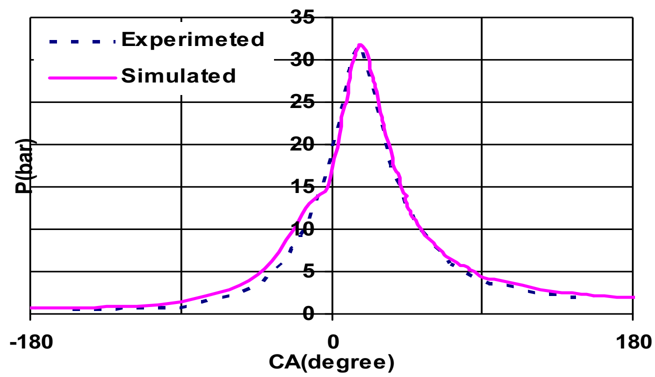

Figure 10 Comparison between experimental and simulated data. Fuel: Ethanol, Engine speed: 2500rpm, Fuel/Air equivalence ratio:1 , Spark advance: 28 BTDC

ENGINE PERFORMANCE AND EMISSION

Type of fuel directly affects all performance and emission characteristics of an engine. In this work, the practical engine performance and emission parameters of interest are power, torque, brake mean effective pressure, brake specific fuel consumption, brake specific NOx and produced CO.

Power and torque depend on an engine's in-cylinder mixture conditions. Therefore volumetric efficiency plays one of the most important roles among the other engine parameters. In Fig.11, engine volumetric efficiency can be observed for the fuels that have been tested on engine. As known liquid fuels have latent heat of vaporization, as a result they produce a cooling effect on intake charge while vaporizing. Therefore, the density of intake mixture is increased and engine volumetric efficiency can rise. Gaseous fuels are vapor in ambient temperature so the cooling effect would not occur for them. On the other hand gaseous fuels have another effect on decreasing volumetric efficiency, this decrease is due to larger volume of fuel in inlet mixture. These are the main reasons of decrease in volumetric efficiency when engine is gaseous fueled.

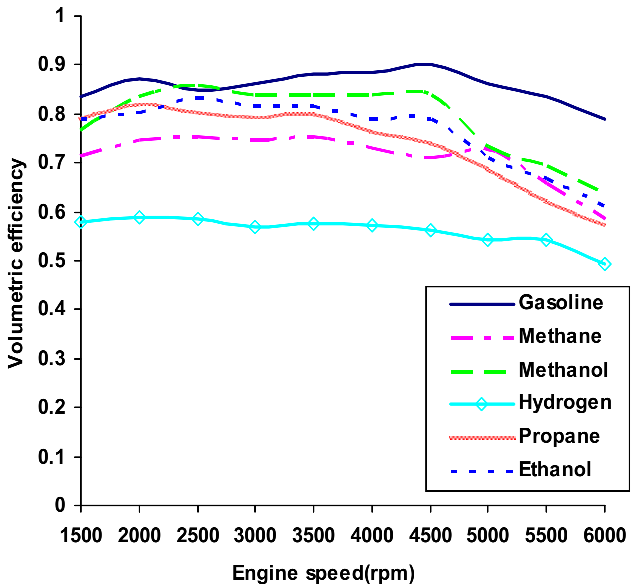

Figure 11. Volumetric efficiency with different fuels versus engine speed

As expected, it can be observed in Fig.11 that engine has the minimum volumetric efficiency when fueled with hydrogen and methane. The other fuels are liquid at ambient temperature. In this case the fuel which has the largest latent heat of vaporization will be more cooled than the others.

Fig.12 compares engine power for operation on gasoline and alternative fuels.

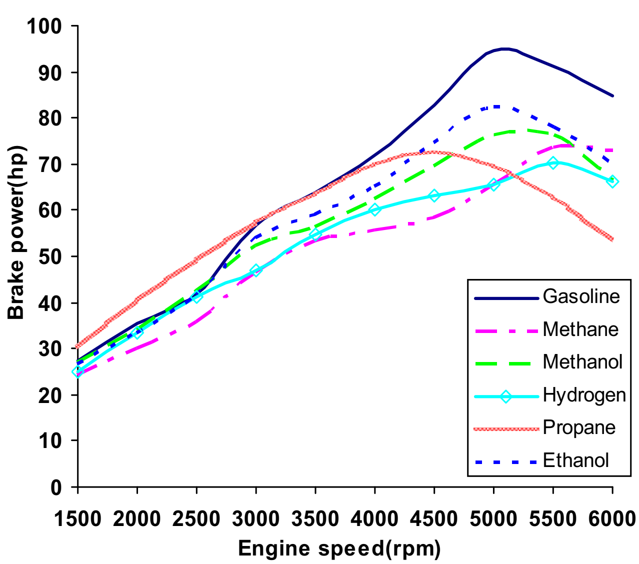

Figure 12. Variation of brake power versus engine speed for different fuels

It can be seen that engine produce less power when operating on methane and hydrogen. As mentioned before, this is because of lower volumetric efficiency of methane and hydrogen fueled engine. In order to regain that lost power, two methods are used. Turbo-charging and raising the compression ratio under naturally aspirated operation. Engine maximum power for all of the fuels happens between 4500 and 5500 (rpm).

Engine power when operating on gasoline is higher than other fuels because the engine is designed for gasoline. All the other fuels have an octane number, higher than that of gasoline, so engine compression ratio could be higher if the engine was dedicated to those fuels, and therefore engine performance could be improved.

Fig.13 shows comparison of brake mean effective pressure (BMEP) of all fuels. For naturally aspirated spark ignition engines, maximum values are in the range 850 to 1050 (kPa) at the engine speed where maximum torque is obtained. At the speed where maximum power occurred, BMEP values are 10 to 15 percent lower. This fact has been mentioned by Heywood too [1].

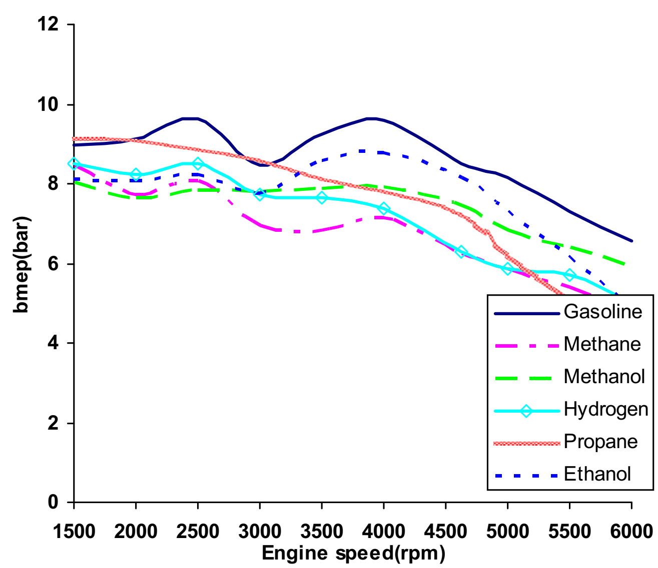

Figure 13. Variation of brake mean effective pressure versus engine speed for different fuels

The reduction in BMEP with methane operation is seen through out the speed range. Part of this BMEP loss is due to longer ignition delay and lower flame speed of methane. As combustion starts earlier with respect to TDC, there is a greater amount of negative work done on the piston before TDC compared to gasoline. The remainder of the BMEP loss is due to the displacement of air by gaseous fuels.

Brake specific fuel consumptions (BSFC) for different fuels are compared in Fig.14. BSFC of methanol is the highest followed by ethanol. Because methanol and ethanol heating values are the lowest between the others and their stoichiometric air/fuel ratios are the smallest. That means for specific air/fuel equivalence, more fuel is needed. BSFC of methane has been measured lower than that of gasoline. The reason is that methane heating value is higher than gasoline. Therefore specified amount of heat can be released with less amount of fuel. As expected because of highest heating value and highest stoichiometric air/fuel ratio, the engine has the lowest BSFC when fueled by hydrogen.

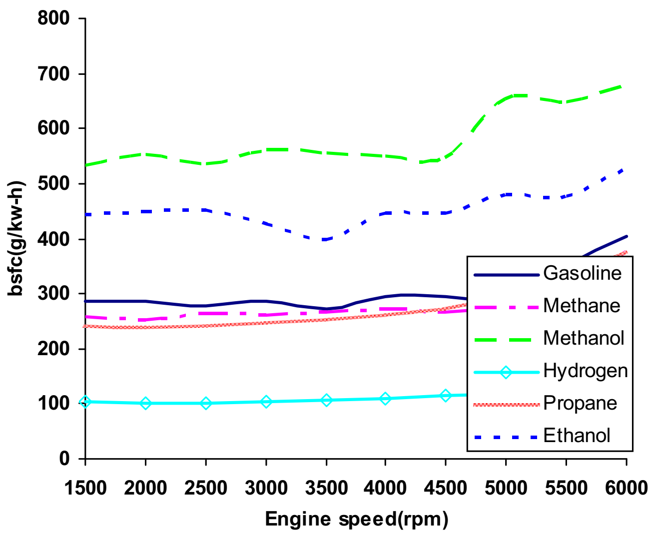

Figure. 14 Variation of brake specific fuel consumption versus engine speed for different fuels

In Fig.15 brake specific NOx (BSNOx) for each fuel is compared. According to measured and predicted data hydrogen and methane cause more BSNOx. It must be remembered that NOx formation take place at high temperatures and the increase of BSNOx is caused by higher combustion temperature of hydrogen and methane.

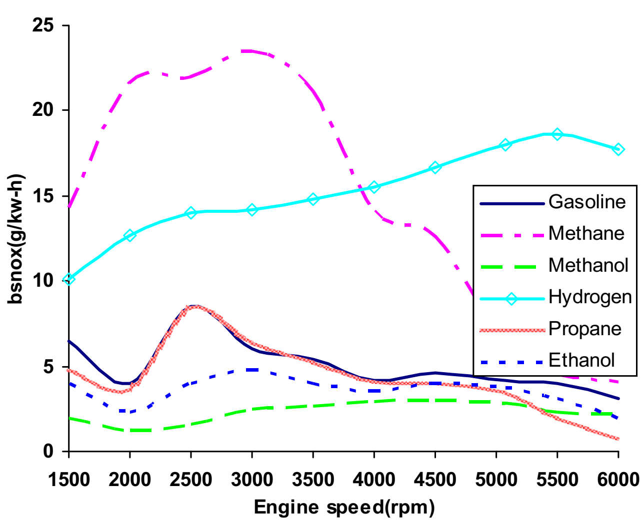

Figure. 15 Variation of brake specific NOx versus engine speed for different fuels

There are two main reasons for this increase in temperature. Firstly, as noted, for gaseous fuels there is no cooling effect which will cause a higher initial temperature for in-cylinder charge. And secondly which is occurred for methane, is that more spark advance is needed because of the low flame speed of methane which rises peak of combustion temperature and pressure.

Methanol and ethanol have the lowest heating value. Methanol flame speed is the highest after than that of hydrogen, consequently lower spark advance is used and combustion temperature is lowered. So BSNOx produced by methanol is less than other fuels. It must be mentioned that hydrogen fueled engine is tested at stoichiometric mixture condition and the value of NOx in that situation is dramatically high. It is clear that hydrogen can perform on much lower equivalence ratio which in that condition NOx value would be really lower in comparison to stoichiometric condition.

In Fig.16, CO mass fraction is shown. CO concentration in exhaust mostly depends on air/fuel ratio. A rich mixture causes more CO in exhaust gases [1, 9]. The closer to stoichiometric point the less amount of CO. Carbon to hydrogen ratio (C/H ratio) of fuel is another parameter which formation of CO is affected by [1]. In lean mixtures there is another additional source of CO caused by the flame interaction with the walls, the oil films and the deposits [9]. It is obvious that hydrogen produces no CO because there are no carbon atoms in hydrogen molecules but in fact as noted, there is a lubricant oil film on cylinder walls that interact with the flame and produces a little amount of CO even with hydrogen fueled engine.

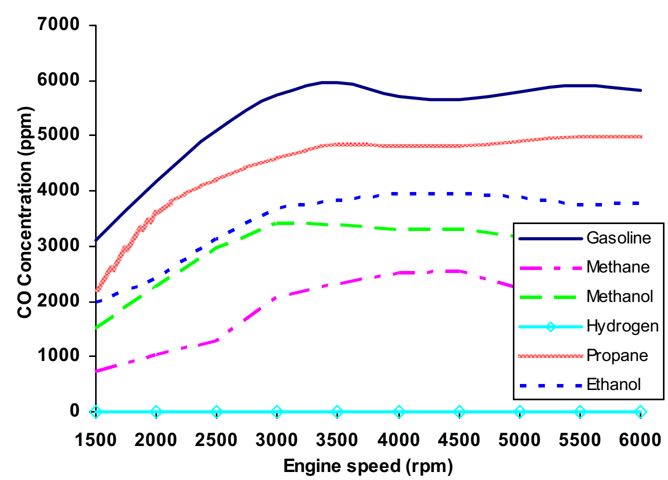

Figure. 16 Variation of CO concentration versus engine speed for different fuels

CONCLUSIONS

Performance and NOx emission of an engine was experimentally measured for gasoline, methane and methanol. All tests have been done in practical engine conditions in a wide range of engine speeds.

A Matlab oriented code was developed to simulate the SI engine performance and emission characteristics fueled by methane, gasoline, methanol, propane, ethanol and hydrogen. Code results were validated by presented experiments and previous literatures. Measured and predicted performance and emission characteristic of engine were compared in detail.

Gaseous fuels which are practiced in this paper have common properties that provide them some advantages and disadvantages relative to liquid fuels. They provide a better air fuel mixture formation and can be used with higher compression ratios. Gaseous fuels decrease volumetric efficiency and increase combustion temperature which results in increase of BSNOx.

Gaseous operation in an engine which is designed for gasoline generally suffers from reduced power, due to low volumetric efficiency, to obviate this disadvantage, engine compression ratio could rises or engine could be turbo-charged.

Liquid fuels tested in presented paper, produce more power rather than gaseous fuels and they produce less NOx. Brake specific fuel consumption of engine, operating on hydrogen, propane and methane is less than that of gasoline while BSFC of methanol is nearly two times of gasoline.

Engine produces more CO when performing with gasoline due to its rich running at high speeds and smaller H/C ratio while CO production of hydrogen fueled engine is very low.

Fuels with low flame speed need more spark advance which causes reduction in BMEP. BMEP of engine fueled with gasoline is the best. That shows, the major role of engine design which is for gasoline.

For gaining high performance and low emission, engine should dedicatedly design for each one of the fuels. In a dual-fuel or flexible-fuel engine some performance properties might be sacrificed.

In this paper, the major benefits of alternative fuels to conventional ones and also the disadvantages of alternative fuels are discussed. Decreasing the disadvantages can be an interesting topic of research for future studies.

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DEFINITIONS, ACRONYMS, ABBREVIATIONS

A/F: Air to Fuel

ABDC: After BDC

ATDC: After TDC

atm : Atmosphere

BBDC: Before BDC

BDC: Bottom Dead Centre

BMEP: Brake Mean Effective Pressure

BSFC: Brake Specific Fuel Consumption

BSNOx: Brake Specific NOx

BTDC: Before TDC

CA, θ : Crank Angle

CFD: Computational Fluid Dynamics

CNG: Compressed Natural Gas

EVO: Exhaust Valve Opening

hp: Horse power

ICE: Internal Combustion Engine

IVO: Intake Valve Opening

LNG: Liquefied Natural Gas

LPG: Liquefied Petroleum Gas

NOx: Oxides of nitrogen

P : Pressure

PPM: Particle Per Million

RON: Research Octane Number

rpm : Revolution per minute

SA: Spark advance

SI: Spark Ignition

T : Temperature

TDC: Top Dead Centre

u : Burning velocity

u' : Root mean square turbulent velocity

$\overline{U_P}$: Mean piston speed

α_T : Temperature exponent

β_P : Pressure exponent

ϕ : Fuel to Air equivalence ratio

SUBSCRIPTS

0: Reference condition

l : Laminar

t : Turbulent

u : Unburned

APPENDIX I

Fuels chemical properties [1]

Fuel Type	RON	Formula	Molecular Weight	Density (kg/m^3)	Heat of vaporization @298K (kJ/kg)	Lower Heating Value (MJ/kg)	Stoichiometric Air/Fuel Ratio
Gasoline	95.8	$C_{7.56}H_{15.5}$	106.22	750	305	44.0	14.6
Methane	120	CH_4	16.04	720	-	50.0	17.23
Methanol	106	CH_4O	32.04	792	1103	20.0	6.47
Ethanol	107	C_2H_6O	46.07	785	840	26.9	9.00
Propane	112	C_3H_8	44.10	545	426	46.4	15.67
Hydrogen	106	H_2	2.015	90	-	120.0	34.3