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MODELING AND OPTIMIZATION OF AN OTTO CYCLE USING THE ETHANOL-GASOLINE BLEND

MODELADO Y OPTIMIZACIÓN DE UN CICLO OTTO UTILIZANDO LA MEZCLA ETANOL-GASOLINA

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Abstract

The use of bio-fuels, like bio-ethanol mixed with gasoline, is growing interest in the automotive industry, to reduce the fossil fuels dependence. However, the air/fuel ratio for any blend is not found in the literature. It is found only for specific values such as: E10, E50 and E85 for example. For this reason, in the present work a mathematical model is presented to determine the stoichiometric air/fuel ratio of the ethanol-gasoline blend in the range between 0% to 100% of ethanol molar percentage in the blend. The model is based on combustion chemical analysis for any composition ethanol-gasoline. The optimum compression ratio for maximum network is also obtained, considering isentropic processes during compression and expansion. The analysis of the Otto cycle with the air/fuel model and optimum compression ratio is developed. Results show that the stoichiometric air/fuel ratio of the blend ethanol-gasoline is not linear. The maximum difference between air/fuel ratios predicted and experimentally reported is 7% in the whole range of the blend. The analysis of the Otto cycle, using the equation derived shows that the power and the torque decrease when the ethanol mole fraction grows. The equations obtained in this work can be used to predict the performance of internal combustion engines using the ethanol-gasoline blend in the continuous range of ethanol molar percentage between 0% and 100%.

Keywords: modeling, Otto cycle, ethanol-gasoline blend.

Resumen

La relación aire/combustible para cualquier mezcla no está reportada en la literatura, sólo para valores específicos. En este trabajo, se presenta un modelo matemático para determinar la relación estequiométrica aire/combustible de la mezcla etanol-gasolina en el intervalo de 0% a 100% del porcentaje molar de etanol en la mezcla. El modelo se basa en el análisis químico de la combustión para cualquier composición etanol-gasolina. También se obtiene la relación de compresión óptima para el trabajo neto máximo, considerando procesos isentrópicos durante la compresión y la expansión. Se realiza el análisis del ciclo Otto con el modelo aire/combustible y la relación de compresión óptima. Los resultados muestran que la relación estequiométrica aire/combustible de la mezcla etanol-gasolina no es lineal. La diferencia máxima reportada entre las relaciones aire/combustible que se predicen y experimentales es de 7% en todo el intervalo de la mezcla. El análisis del ciclo Otto, utilizando la ecuación derivada muestra que la potencia y el torque decrecen cuando la fracción mol de etanol aumenta. Este trabajo se puede utilizar para predecir el rendimiento de los motores de combustión interna utilizando la mezcla etanol-gasolina en el intervalo continuo entre 0% y 100%.

Palabras clave: modelado, ciclo Otto, mezcla etanol-gasolina.

1 Introduction

Deployment of fossil fuels has increased the interest in using new alternatives energy sources. Otto and Diesel engine models with cyclic variability were performed by Rocha-Martínez *et al.* (2002), while the desires properties for a rocket fuel were studied by Miranda (2003). The synthesis and characterization of perovskites for fuel cells as an alternative energy source was analyzed by Chávez-Guerrero *et al.* (2003), while Carbon nanotubes produced from

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hexane and ethanol has been used by Mendoza *et al.* (2006). The use of bio-fuels, like biodiesel studied by Gonca and Dobrucali (2016) o bio-methanol and bio-ethanol mixed with gasoline, is getting growing interest in the automotive industry, to reduce the fossil fuels dependence Murali *et al.* (2012). The use of ethanol blend gasoline is demonstrated also by Yücesu *et al.* (2006), who give results of the ethanol use to reduce the gasoline consumption, the emissions during the cold start up of a flex fuel engine, the experimental determination of ethanol physical properties, and the heat flux characteristics of spray wall impingement with ethanol, respectively. The economy and emissions of light vehicles, and the dynamic analysis of the supply chain feasibility for the ethanol-gasoline mixture in Mexico, are shown in Hernández *et al.* (2014). The use of this fuels blendin motorcycles was reported by Yao *et al.* (2013), where the authors show the good results obtained.

The air/fuel ratio is of significant importance for the analysis and measurement of combustion, as shown by Clements and Smy (1976) for the measurement of the ionization density. Someone papers have been found on this issue. Polymeropoulos and Sernas (1977) determined the droplet size and the fuel-air ratio for a spray, Deadmore *et al.* (1979) studied the effect of fuel-to-air ratio on burner rig hot corrosion, Desoky and Rabie (1983) showed the fuel economy benefits of using alcohols gasoline blends when experimental investigations were carried out to judge the performance of small spark ignition engines running on alcohols, gasoline and alcohol-gasoline blends. Bardaie and Janius (1984) during their experimental investigation found the power loss of 3-4% when using ethanol in SI engine with a modified carburetor. Barwan (1985) studied blended fuel ranging from E10 to E70 and concluded that the highest antiknock capability was obtained with E50. Palmer (1986) in his experimental investigation reported the engine power improved by 5% when 10% in gasoline was used as a fuel additive. Hamdam and Jubran (1986) during their investigations using 5% ethanol in gasoline under partial load found the thermal efficiency improved by 4-12%. Ohsuga and Ohyama (1986) studied the oxygen-biased wide range air-fuel ratio sensor for rich and lean air-fuel ratios. Additionally, according to Rigatos *et al.* (2014) the control of the air/fuel ratio is very important in the operation of spark-ignition engines.To study the performance of internal combustion engines using this mixture, the air/fuel ratio is required. This ratio has been determined in different ways for the ethanol-gasoline blend; one of them is taking into account the volume percentage considering that for gasoline is 14.6 and for ethanol 9.0, according to Pulkrabek (2003). On the other hand, as it can be observed in Chen *et al.* (2012), the vaporization enthalpy respect to the fraction of ethanol in gasoline is linear when it is based on the volume. On the contrary, its tendency is quadratic if it is based on the molar mass. Further, the air/fuel ratio for the ethanol-gasoline blend has been compiled in Kasseris (2011), based on different bibliographic data. There, the ratio is presented for the mixtures E0, E10, E20, E50, E85 and E100 (the number represents the percentage of ethanol in gasoline). Further, the air/fuel ratio, AF, can be obtained by different ways as discussed briefly in the following:

1. By means of chemical analysis, considering: flame velocity, problem geometry, mass flow rate, time, species concentration, etc. This analysis is complex, due to partial differential equations are coupled and they require numerical solution. This process may increase the cost of the research, mainly for the computing time that can be excessive. This makes difficult the determination of the air/fuel ratio in the entire range of ethanol mole fractions. This method is useful to find experimentally the air/fuel ratio.

2. Experimentally, by two different ways: a) using the Brettschneider's equation Brettschneider (1979, 1997), Schifter *et al.* (2011), Zhang *et al.* (2013) that shows how the fuel is burned by means of the combustion exhaust gases analysis. Results obtained from this equation, using experimental data of exhaust gases, show if the combustion is clean or not. Experiments need to be performed as many times as the blend composition of ethanol-gasoline is changed. b) using the following equation, Cengel and Boles (2012):

$$AF = \frac{m_a}{m_f} \tag{1}$$

where: m_a: air mass, measured using a flowmeter.

m_f: fuel mass, calculated from the heat supplied during combustion, Q_s.

The heat supplied, Q_s, is calculated as:

$$Q_s = m_f LHV \tag{2}$$

where:

   LHV: lower heating value, which should be obtained for each ethanol-gasoline blend.

   In the same way, the experiments should be repeated for each one of the blend compositions tested.

3. Analytically, performing a chemical analysis of combustion for each ethanol-gasoline composition. One way to obtain the air/fuel ratio of this mixture is shown in Cordeiro *et al*. (2012), where the authors give and equation for this ratio proposed by Heywood Heywood (1988). However, in order to use this equation the ratios H/C and O/C should be known, making the analysis complex.

The chemical analysis of combustion can be developed supposing that combustion is instantaneous and 100% efficient (ideal combustion), i.e., the reactants burn totally generating a clean combustion. Further, products are considered to be: water vapor, carbon dioxide and gaseous nitrogen. Taking into account these assumptions, a chemical equation can be written including in the reactants the percentages of ethanol and gasoline, reacting with atmospheric air. This analysis can be performed for the whole range of the ethanol-gasoline blend, from 0% to 100% of ethanol (E0 to E100, respectively).

This method simplifies considerably the determination of the air/fuel ratio, because using a single equation it is possible obtaining the ratio for the continuous range between E0 and E100. Additionally, this equation can be modified for more specific cases in a simple manner, for example if excess of air or a third fuel is present, etc.

This last method is used in the present work to derive a general equation for the air/fuel ratio of the ethanol-gasoline blend for any composition between 0% and 100% of ethanol. This equation is a function only of the mole fraction of ethanol in the blend.

This work is organized in four sections. The model development section presents: i) the chemical analysis for the whole range of the ethanol-gasoline blend, ii) the analysis developed to the ideal Otto cycle for the ethanol-gasoline blend, iii) a compression ratio equation that maximizes the net work of the cycle, for each ethanol fraction in the blend. Finally, iv) the results and discussion section presents the principal findings of this research.

2 Model development

2.1 Air/fuel ratio equation

The procedure used in this section, is described in a detailed way in Gómez (2014), it is used to obtain the equation for the air/fuel ratio as a function of the ethanol mole fraction.

As previously commented, combustion products only include CO_2, H_2O and N_2, the excess air for tempering is not considered. Other products like CO and nitrogen oxides are neglected, because an ideal analysis is performed. Also, nitrogen and other gases of atmospheric air are considered to be chemically neutral, as in Pulkrabek (2003). However, nitrogen affects the temperature and pressure during the combustion process. The maximum heat generated by combustion depend on the LHV and on the ethanol concentration in the blend, it can be estimated from equations 11 and 12 presented later. The higher temperature reached correspond to E0 (only gasoline) in the entire range, the higher temperature generated by the stoichiometric reaction correspond to E0 or lower for any blend. The mechanical elements of combustion chamber are designed for this operation conditions, the used of ethanol/gasoline blends do not need any modification in design or in mechanical elements to be implemented.

The ideal combustion equation for the blend ethanol-gasoline can be written as follows, considering that x is the mole fraction of ethanol and $1-x$ the mole fraction of gasoline:

$$xC_2H_5OH + (1-x)C_8H_15 + a(O_2 + 3.76N_2) \rightarrow bCO_2 + cH_2O + dN_2 \quad (3)$$

Taking into account the law of mass conservation, algebraic equations can be written for each chemical element, obtaining the following results for the coefficients of Eq. (3):

$$a = \frac{47-35x}{4} \quad (4)$$

$$b = 8-6x \quad (5)$$

$$c = \frac{15-9x}{2} \quad (6)$$

$$d = 3.76\left(\frac{47-35x}{4}\right) \quad (7)$$

The number of moles of each chemical element in the reactants and combustion products can be seen in Table 1.

Table 1. Number of atoms in the reactants and combustion products.

Number of atoms	Element
$8-6x$	C
$15-9x$	H
$\frac{47-35x}{2}$	O
$3.76\left(\frac{47-35x}{2}\right)$	N

Table 2. Molecular weights of the reactants and combustion products.

Compound	Molecular weight (g/mole)
C_2H_5OH (Ethanol)	46
C_8H_{15} (Gasoline)	111
O_2	32
N_2	28
CO_2	46
H_2O	18

Taking into account the molecular weights of reactants and combustion products (Table 2), the mass of each individual compound can be obtained using the number of moles of each compound calculated from Eqs. (4) to (7).

Further, the total mass of fuel is:

$$m_f = m_e + m_g \tag{8}$$

where:

m_e: mass of ethanol

m_g: mass of gasoline

In this way, the following equation is obtained for the air/fuel ratio of the ethanol-gasoline blend, using the definition given in Eq. (1):

$$AF = 34.32\left(\frac{47-35x}{111-65x}\right) \tag{9}$$

Eq. (9) is only a function of the mole fraction of ethanol in the blend. It expresses the ratio between the mass of atmospheric air and blend required to obtain an ideal combustion process.

2.2 Analysis of the Otto-cycle

The analysis developed in this section applies to the ideal air-standard conditions, Po= 101.325 kPa, To=25°C of Otto cycle, represented in Fig. 1. The procedure followed is similar to that described by Pulkrabek (2003). The engine develops an isentropic compression from state 1 to 2, heat is supplied at constant volume from state 2 to 3, the useful stroke is developed during the isentropic expansion from state 3 to 4 and, finally, heat is released to the cold reservoir following a constant volume process from state 4 to 1, closing the cycle. All the processes are developed in a closed system by ideal gases with constant specific heats.

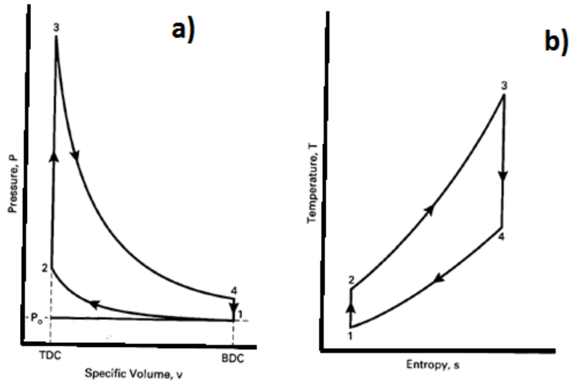

Fig. 1. Otto Cycle on a a) pressure-specific volume diagram, b) temperature-specific entropy diagram.

Heat supplied during process 2-3 can be calculated as:

$$Q_{in} = m_f LHV_f \eta_c = (m_a + m_f)C_v(T_3 - T_2) \tag{10}$$

LHV_f can be obtained from:

$$LHV_f = \omega LHV_e + (1-\omega)LHV_g \tag{11}$$

The ethanol mass fraction in the blend, ω, is obtained from Eq. (12) considering the molecular weight, M, of ethanol and gasoline:

$$\omega = \frac{xM_e}{xM_e + (1-x)M_g} = \frac{m_e}{m_t} \tag{12}$$

Net work of the cycle is the difference between the work produced during expansion and the one consumed during compression of the air-fuel mixture:

$$W_n = W_{34} - W_{12} \tag{13}$$

where:

$$W_{34} = (m_a + m_f)C_v(T_3 - T_4) \tag{14}$$

$$W_{12} = (m_a + m_f)C_v(T_2 - T_1) \tag{15}$$

Thermodynamic efficiency of the Otto cycle is defined as:

$$\eta = \frac{W_n}{Q_{in}} \tag{16}$$

Power produced by the engine, as a function of the regime is (W_n in kJ):

$$N_t = \frac{W_n \cdot n \cdot Ncyl}{89.52} \quad \text{(in HP)} \tag{17}$$

Torque can be calculated from:

$$\tau = \frac{60 W_n}{2\pi n} \tag{18}$$

2.3 *Optimum compression ratio for maximum net work*

With the aim of searching for an optimal fraction of ethanol in the blend, in this section an equation is obtained that allows calculating the compression ratio that maximizes the net work of the cycle, for each ethanol fraction in the blend. In this way, the blends having optimum compression ratios achievable in practice can be identified.

The specific net work of the Otto cycles is:

$$w_n = \frac{W_n}{(m_a + m_f)} \tag{19}$$

Considering the isentropic processes 1-2 and 3-4:

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma-1} = \frac{T_3}{T_4} = r^{\gamma-1} \tag{20}$$

Combining Eqs. (13) to (15), (19) and (20), the following equation is obtained for the specific net work of the cycle:

$$w_n = C_v T_3 \left(1 - \frac{1}{r^{\gamma-1}}\right) - C_v T_1 \left(r^{\gamma-1} - 1\right) \tag{21}$$

Deriving Eq. (21) respect to the compression ratio, r, and making the result equal to zero:

$$\frac{\partial w_n}{\partial r} = -C_v T_3 (1-\gamma) r^{-\gamma} - C_v T_1 (\gamma - 1) r^{\gamma-2} = 0 \tag{22}$$

then:

$$\frac{T_3}{T_1} = r^{2(\gamma-1)} \tag{23}$$

From Eq. (23), the optimum compression ratio that maximizes the net work of the cycle is obtained:

$$r_{opt} = \left(\frac{T_3}{T_1}\right)^{\frac{1}{2(\gamma-1)}} \tag{24}$$

Assuming that combustion efficiency is 100%, the maximum temperature of the cycle can be obtained from Eqs. (10) and (20):

$$T_3 = \frac{m_f LHV_f)}{(m_a + m_f) C_v} + T_1 r^{\gamma-1} \tag{25}$$

Substituting Eq. (25) for T_3 in Eq. (24), the optimum compression ratio is obtained:

$$r_{opt} = \left(\frac{m_f LHV_f}{(m_a + m_f) C_v T_1}\right)^{\frac{1}{\gamma-1}} = \left(\frac{LHV_f}{(AF+1) C_v T_1}\right)^{\frac{1}{\gamma-1}} \tag{26}$$

In Eq. (26), LHV_f, AF, C_v and γ are a function of x. In this model, C_p and C_v are calculated as follows:

$$C_p = \frac{AF}{AF+1} C_{p_a} + \frac{1}{AF+1} C_{p_f} \tag{27}$$

$$C_v = \frac{AF}{AF+1} C_{v_a} + \frac{1}{AF+1} C_{v_f} \tag{28}$$

Finally, the efficiency of the cycle corresponding to the optimized compression ratio is:

$$\eta_{opt} = 1 - \frac{1}{r_{opt}^{\gamma-1}} = 1 - \frac{(AF+1) C_v T_1}{LHV_f} \tag{29}$$

3 Results and discussion

3.1 *Air/fuel ratio*

Variation of the air/fuel ratio, as a function of the ethanol mole fraction in the mixture, is calculated using Eq. (9) and represented in Fig. 2. As it is observed, this ratio is not a linear function of the ethanol mole fraction. A similar tendency was shown by Yao *et al.* (2013) for the vaporization enthalpy of the blend. Results found in this paper can be used to simulate the engine performance in the whole range of concentrations. As commented before, air/fuel ratios can be found in the open literature for discrete values of ethanol mole fraction.

Results found using the model developed in this work has been compared with experimental and theoretical values found in the open literature. Fig. 2 shows the comparison with theoretical results obtained by Kasseris (2011), Orbital (2002) and Mantilla (2010). It is shown that, for ethanol mole fractions lower than 20% and higher than 95%, results are very similar to the ones obtained in the present work. However, the results in the intermediate range defer more significantly. As observed in Fig. 2, results published in the open literature can be fitted using a linear function, similar to a linear interpolation between air/fuel ratios of pure components.

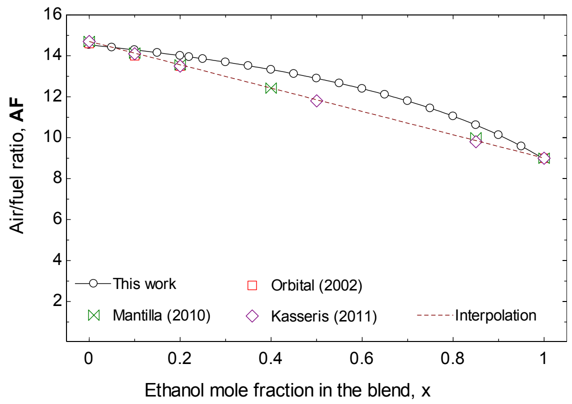

Fig. 2. Comparison between air/fuel ratios obtained in the present work and theoretical data found in the open literature, as a function of the ethanol mole fraction in the blend.

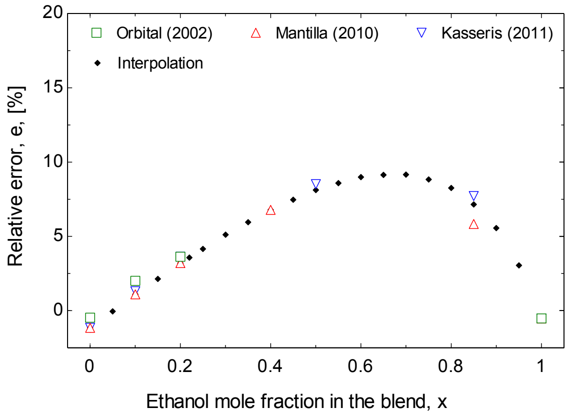

Fig. 3 Relative error between air/fuel ratios calculated in the present work and theoretical data found in the open literature, as a function of the ethanol mole fraction in the blend.

Relative errors between results predicted in this work and theoretical values found in the open literature can be observed in Fig. 3. As commented before, major differences exist in the range between 20% and 95% of ethanol in the blend. For lower and higher values, respectively, the differences are as small as 0.7% in the blend E5.

A comparison of Eq. 9 with experimental results reported in the literature was done to verify the predictive capacity. Results calculated by the present model and experimental values found in the open literature Costa and Sodré (2010), Szybist (2010) and Camarillo (2011) can be observed in Fig. 4. The last work Camarillo (2011) gives results of the air/fuel ratio for blends of ethanol-gasoline and hydrated ethanol-gasoline.

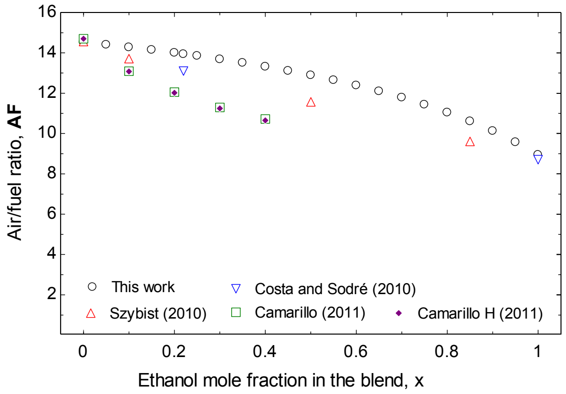

Fig. 4 Comparison between air/fuel ratios obtained in the present work and experimental data found in the open literature, as a function of the ethanol mole fraction in the blend.

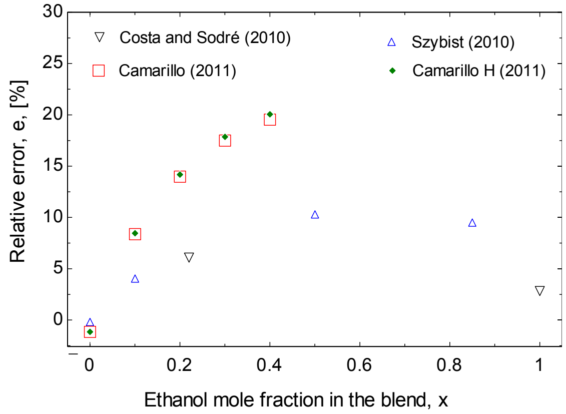

Fig. 5 Relative error between air/fuel ratios calculated in the present work and experimental data found in the open literature, as a function of the ethanol mole fraction in the blend.

Similarly, to theoretical studies, experimental air/fuel ratios found in the open literature are very scarce and do not cover the whole range of ethanol-gasoline blends. As observed, results predicted are similar to those of Costa and Sodré (2010) and Szybist (2010), but highly differ from values given by Camarillo (2011). This difference can be because of sensors errors in the experimentation or probable not experimental stoichiometric conditions. The difference between experimental data and values calculated using Eq. (9) is shown in Fig. 5. Minimum difference in the whole range of ethanol concentration is about 0.7% respect to results of Costa and Sodré (2010) and Szybist (2010).

Table 3. Characteristics of the 4T Otto cycle analyzed.

Parameter	Value
Displacement volume	2400 cm^3
Number of cylinders	4
Bore	72 mm
Stroke	74 mm
Compression ratio	8.6:1

Table 4. Intake conditions in the 4T Otto cycle analyzed.

Parameter	Value
Intake temperature, T1	25 °C
Intake pressure, P1	101.325 kPa
Volume percentage of ethanol in the blend	0% to 100%
Rotational speed	1500, 3000, 4500, 6000 rpm

Table 5 Thermodynamic properties, taken from Clements and Smy (1976).

Parameter	Air	Bio-ethanol	Gasoline
Specific heat, Cp, kJ/kg K	1.005	2.3	2.22
Specific heat, Cv, kJ/kg K	0.718	2.3	2.22
Lower heating value, kJ/kg	-	26,900	44,300

The results here shown indicate that Eq. (9) could be utilized to calculate the air/fuel ratio for the ethanol-gasoline blends with satisfactory results in the interval of 0% to 100% of ethanol mole fraction.

3.2 *Analysis of Otto-cycle*

Previous studies found in the open literature about Otto cycles using mixtures of fuels can be found in Pulkrabek (2003), Kasseris (2011) and Szybist (2010). In the present work, an analysis of a 4 times Otto cycle using different concentrations of the ethanol-gasoline blend is performed. The characteristics of the engine used are shown in Table 3, similar to Pulkrabek (2003). Intake conditions considered are shown in Table 4, corresponding to temperature and pressure at sea level.

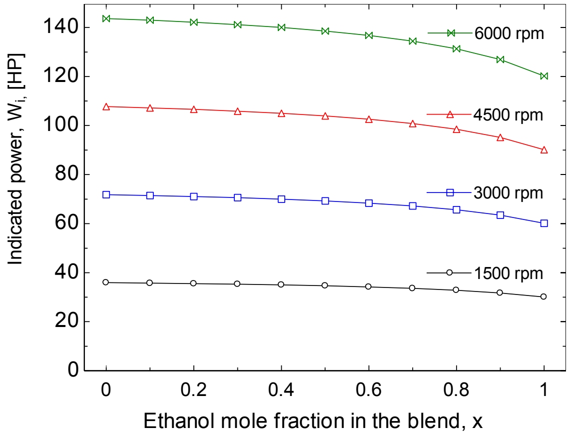

Fig. 6. Indicated engine power as a function of the ethanol mole fraction in the blend for different regimes.

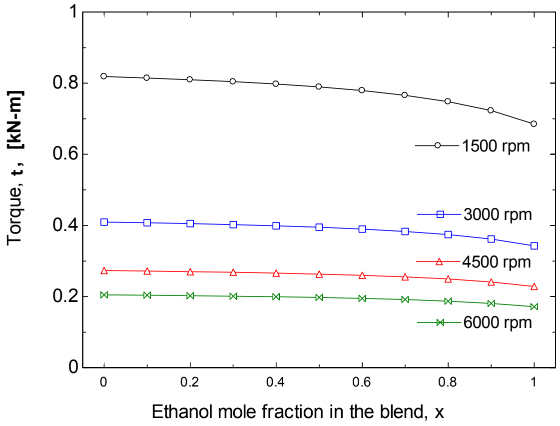

Fig. 7. Torque of the engine as a function of the ethanol mole fraction in the blend.

Thermodynamic properties are shown in Table 5. The model previously described was implemented in a computer code developed by the authors using Engineering Equation Solver software, EES^{TM} Klein (2015).

Fig. 6 shows the indicated power of the engine, as a function of the ethanol mole fraction, for different rotational speeds of the engine. It can be observed that the indicated power decreases slightly when the ethanol mole fraction augments and a more pronounced reduction takes place for compositions of ethanol higher than 80%. Moreover, the effect of concentration increases as the engine speed rises.

In a similar way to the power, the torque decreases slightly when the ethanol mole fraction augments (Fig. 7). In this case, the effect of concentration increases as the engine speed decreases.

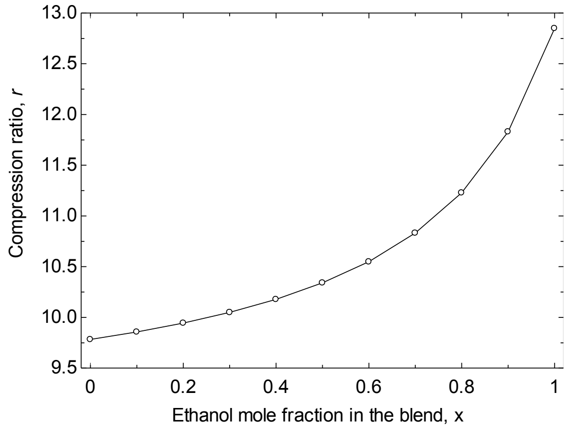

Fig. 8. Compression ratio that maximizes the net work of the Otto cycle as a function of the ethanol mole fraction in the blend.

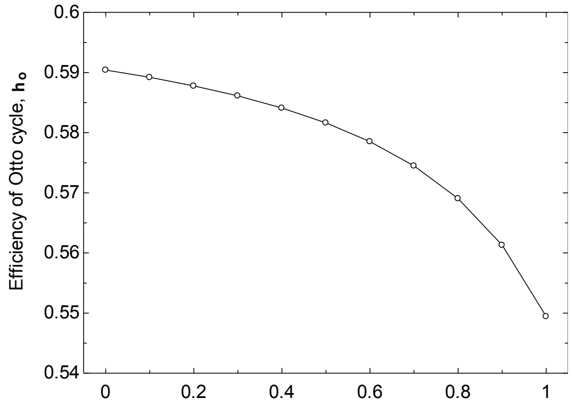

Fig. 9. Thermodynamic efficiency of the Otto cycle as a function of ethanol mole fraction in the blend.

The torque corresponding to pure gasoline E0 is slightly higher than that of pure ethanol E100. Maximum difference reaches 16.3% at 1500 rpm.

3.3 *Optimum compression ratio*

The optimum compression ratio of the cycle, corresponding to the maximum specific net work, at T_3=1340 K, is shown in Fig. 8 as a function of the blend composition. It can be observed that the optimum compression ratio rises as the ethanol mole fraction in the blend increases. As real compression ratios used in gasoline engines are lower than 15, all of the cycles with ethanol-gasoline blend can operate at their optimum compression ratio.

According to Fig. 9, the efficiency of the cycle decreases when the ethanol mole fraction grows. In this figure, the cycle operates at the optimum compression ratio of each blend represented in Fig. 8. As it is observed, a more pronounced decrease of the efficiency occurs for ethanol mole fractions higher than 80% approximately. For this reason, it is recommended to use ethanol concentrations lower than this value.

Conclusions

In this paper, an equation for the stoichiometric air/fuel ratio of the ethanol-gasoline blend is derived and the performance of a 4 times Otto cycle using this mixture is evaluated. The following conclusions have been derived:

- The equation derived for the air/fuel ratio of the ethanol-gasoline mixture can be used straightaway and without computational cost. It depends only on the ethanol mole fraction in the blend and is valid for the whole range of ethanol concentrations between 0% and 100%. The equation obtained offers exact results for the stoichiometric air/fuel ratio of the blend. A comparison with experimental data reported in literature shows that results are very similar for low and high ethanol concentrations.

- Optimization of the compression ratio to maximize the specific net work of the cycle, at T_3 = 1340 K, provides ratios achievable in real gasoline engines.

- The ethanol-gasoline blend offers suitable results of power and torque. They reduce as maximum 6.4 and 6.44% (8.56 and 8.59% with E80) respectively, respect to pure gasoline, when the blend E70 is used and the rotational speed of the engine is 6000 rpm. The lower heating value of ethanol (40% lower), and the fact that the fuel is oxygenated, are the reasons of the performance decrease respect to pure gasoline.

- The efficiency of the cycle operating at the optimum compression ratio is evaluated. A more pronounced decrease in the efficiency is obtained for blends with ethanol mole fraction higher than 80% approximately. The use of ethanol decreases the efficiency because, as commented before, fuel is oxygenated and it has a lower heating value.

This work can be improved considering a better kinetic scheme and secondary products, resulting from an inefficient combustion.

Nomenclature

AF	air/fuel ratio
C_p	specific heat at constant pressure, kJ/kg K
C_v	specific heat at constant volume, kJ/kg K
LHV	lower heating value, kJ/kg
m	mass, kg
M	molecular weight, g/mole
n	rotational speed, rpm
N_t	theoretical power, W
N_{cyl}	cylinder number
Q	heat supplied, J
r	compression ratio
R	ideal gas constant, kJ/kg K
T	temperature, K
w	specific work, J/kg
W	work, J
x	mole fraction of ethanol in the flexible fuel
Greek symbols	
γ	ratio between C_p and C_v
η	efficiency
τ	torque, N m
ω	mass fraction of ethanol in the flexible fuel
Subscripts:	
$1-4$	thermodynamic states of the cycle
a	air
c	combustion
e	ethanol
f	fuel
g	gasoline
n	net
opt	optimum
t	total

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