

BIOETHANOL OBTAINED FROM WOODEN BIOMASS.  
AN APPROPRIATE ALTERNATIVE FUEL FOR SPARK IGNITION  
ENGINES

CORNELIU CINCU, LAURENTIU FARA, ALEXANDRU RACOVITZĂ and  
LUCIAN LOBONT\*

*Polytechnic University of Bucharest, 313 Splaiul Independentei, Bucharest, Romania*  
*\*"Lucian Blaga" University of Sibiu, 2-4 Emil Cioran Str., Sibiu, Romania*

*Received January 29, 2010*

Among other biofuels, ethanol proves significant qualities that recommend it as an alternative fuel for classic thermal engines, thus offering opportunities of saving conventional petroleum fuels and diminishing the levels of the main exhaust emissions, especially CO, CO<sub>2</sub> and hydrocarbons (HC). Ethanol could be generously obtained by applying technologies that transform biomass into sugar compounds and alcohol. Vegetal species like poplars, with a rapid growing cycle, offer the main source for obtaining wooden biomass, thus permitting ethanol production by industrial methods, such as enzymatic fermentation of the sugar compounds. The resulting product is tested as to the requested properties, for its further utilization as a fuel for Spark Ignition (SI) Engines. Therefore, regarding its octane number, caloric heat, lubrication capacity, etc., bioethanol shows good features to be used as a single fuel or in mixture with gasoline to supply SI Engines.

**Keywords:** bioethanol, wooden biomass, enzymatic fermentation, ethanol-gasoline mixture, gas emission control, engine efficiency improvement

## INTRODUCTION

Ethanol represents an adequate fuel for SI engines due to its properties – which permits its mixing with gasoline in a stabile and miscible fuel blend –, due to its very high octane number (over 100) and comparatively low toxicity. This type of alcohol fuel represents a renewable fuel due to the great opportunity it provides – to be produced from vegetal resources, crops and wooden remains. Fast-growing poplars are used for their rapid regenerative cycle, thus ensuring the necessary biomass. Generally speaking, UE is encouraging<sup>1</sup> the substitution of fossil fuels up to 5.75% volumetric fractions by 2010, and up to 20% by 2020. Ethanol could be easily produced from biomass by two well-known procedures: hydrolysis and fermentation of the sugar compounds existing in the biomass composition. The biomass resulted from vegetal species contains a complex mixture of carbohydrate polymers, known as cellulose, hemicellulose and lignin. To

obtain sugar components from biomass, it has to be treated with acids or enzymes. Thus, such polymers lead to a sucrose process and, subsequently, to alcohol production. Three methods to extract sugar from biomass<sup>2</sup> are now available:

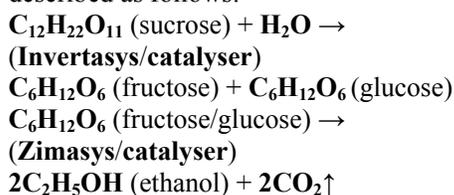
A) *Hydrolysis with concentrated acids* (the Arkhanol method). The biomass is treated with sulphuric acid (70-77% concentration) after being dried to 10% humidity. One part of biomass corresponds to 1.25 parts of acid, at 50 °C. Water is added to dilute the acid to 25-30%, then the mixture is heated up to 100 °C for 1 h. The obtained gelatin is pressed, to obtain only the sugar-acid mixture, their separation being performed with a chromatographic column;

B) *Hydrolysis with diluted acids* – one of the simplest and the most efficient methods to obtain ethanol. The diluted acid is used to extract sucrose from the biomass. In the first step, sulphuric acid (0.7% concentration) is

used at 190 °C for hemicellulose hydrolysis. The second stage consists in cellulose hydrolysis with sulphuric acid at 215 °C and 0.4% concentration. The liquid resulted from hydrolysis is then neutralized and reused in the process.

C) *Hydrolysis of biomass with enzyme fermentation.* It is a new and revolutionary process being developed with high costs and investments.<sup>4</sup>

The reactions through which, finally, ethanol could be obtained by enzymatic fermentation of the sugar compounds are described as follows:



The ethanol obtained by applying the fermentation reactions also contains a significant water amount. Water is supposed to be eliminated through fractioned

distillation. Due to the fact that, at normal atmospheric pressure, ethanol has a lower vaporization temperature (78.5 °C) than water (100 °C), it can be vaporized before water and, subsequently, condensed and separated from it. As an alcohol fuel, bioethanol could form a new type of fuel in mixture with gasoline.<sup>3</sup> These mixtures have already been standardized as E class fuels, depending on the percentage of alcohol used (*e.g.* E85, E15 in which the value is given by the volumetric ethanol ratio). Gasoline-ethanol mixtures are most likely to be used instead of ethanol-only fuel, due to the benefits offered by both fuels. Gasoline is characterized by high caloric heat and good lubricating properties, to be taken into account as to the components of the supplying systems, while ethanol has a higher octane number, which confers the air-fuel mixture better knock resistance.

Table 1 provides a comparison between the main physical properties of ethanol and ethanol-gasoline fuel mixtures.

Table 1  
Physical properties of gasoline and ethanol-gasoline fuel mixtures

Properties	Bioethanol	E85	E15
Equivalent molar weight (kg/kmole)	46.1	-	-
Carbon (% mass)	52.2	56-58	-
Hydrogen (% mass)	13.1	13-14	-
Oxygen (% mass)	34.8	29-30	5
Water (% mass)	<6.2	2.9-6.6	-
Phosphorus (ppm)	-	<0.5	-
Sulphur (ppm)	-	<30	0.01%
Density (15 °C/1 bar) (g/cm <sup>3</sup> )	0.8-0.82	0.78-0.80	0.815
Boiling temperature (°C/1 bar)	78	49-80	-
Melting temperature (°C)	-114	-	-
Autoignition temperature (°C)	423	>257	-
Inferior caloric heat (MJ/kg)	25-27	27-29	-
Octane number	108.6	107	-
Vaporization specific heat (kJ/kg)	923	836	-
Specific heat (kJ/kgK)	2.4	2.3	-

## THEORETICAL INVESTIGATION

The paper presents some results, obtained by computer simulation, on the combustion of a bioethanol-gasoline mixture in the cylinder of a spark ignition engine. A computer model has been proposed for describing the development of combustion and its heat release inside the engine, when gasoline-ethanol mixtures are supplied to a Dacia-Renault commercial four-cylinder SI engine. The engine has the following features: 45 kW maximum power, 4 in-line cylinders, 9:1 compression ratio,

5000 rpm maximum speed, 0.95 relative air-fuel ratio at maximum power. Certain usual hypotheses have been considered when writing the equations of the mathematical model, as follows:

- ethanol-gasoline mixtures are homogeneous blends, considered as new types of fuels with modified values for carbon, hydrogen and oxygen mass participations (with respect to  $x_{\text{eth}}$  – the ethanol substitution rate for gasoline);
- the combustion model is a zero-dimensional one, with no influence on the

combustion geometry volume. The motor fluid is divided into burned and unburned zones, under the same thermal state conditions;

- the air-fuel mixture is preformed and homogeneous before combustion starts. The delay between the spark ignition point per cycle and the flame kernel occurrence is neglected;

- the combustion model operates with the well-known Wiebe formalization equation of the in-cylinder heat release. Thus, a fraction of fresh unburned fluid leads to an equal heat release fraction related to the whole amount of released heat per cycle;

- the equations taken into consideration are the following: energy conservation, mass conservation and thermal state:<sup>5</sup>

$$m_{fuel} \cdot \frac{\delta Q_i}{\delta \alpha} + \frac{dH_{fuel}}{d\alpha} = p \cdot \frac{dV}{d\alpha} + \frac{dU}{d\alpha} + \frac{\delta Q_w}{\delta \alpha} \quad [\text{KJ}/^\circ\text{CA}] \quad (1)$$

where  $m_{fuel}$  is the burned cycle fuel quantity,  $Q_i$  is the caloric heat of the fuel,  $H_{fuel}$  is the enthalpy of the in-cylinder injected fuel,  $p$  is the in-cylinder pressure,  $V$  is the cylinder volume,  $U$  is the internal energy of the motor fluid,  $Q_w$  is the heat transferred to the combustion chamber walls and  $\alpha$  is the crankshaft rotation angle ( $^\circ\text{CA}$  – Crankshaft Angle degree):

$$\frac{dm_{fc}}{d\alpha} = -\frac{dm_{bg}}{d\alpha} \quad [\text{kg}/^\circ\text{CA}] \quad (2)$$

where  $m_{fc}$  is the mass of the fresh charge (unburned), and  $m_{bg}$  is the mass of the burned gas.

$$p \cdot V = (v_{fc} + v_{bg}) \cdot R_M \cdot T \quad [\text{kJ}] \quad (3)$$

where  $v_{fc}$  and  $v_{bg}$  are the molar quantities of the fresh charge and burned gas inside the cylinder, and  $R_M$  is the universal gas constant (8314 J/moleK).

The purpose of this thermodynamic analysis on the engine cycle is that of plotting the pressure diagram corresponding to the combustion process. Thus, the indicated mean effective pressure ( $p_i$ -IMEP) of the cycle would be calculated.

$$p_i = \frac{1}{720} \cdot \int_0^{720} p(\alpha) \cdot d\alpha \quad [\text{MPa}] \quad (4)$$

This operation leads to the calculation of the indicated engine power:

$$P_i = \frac{p_i [\text{MPa}] \cdot V_t [L] \cdot n [\text{rpm}]}{120} \quad [\text{kW}] \quad (5)$$

where  $V_t$  is the total engine displacement and  $n$  is the engine revolution speed.

The indicated specific consumption could be also determined:

$$c_i = \frac{10^3 \cdot C_h^{fuel}}{P_i} \quad [\text{g/kWh}] \quad (6)$$

where  $C_h^{fuel}$  is fuel consumption per hour.

The indicated efficiency of the engine is the main energetically variable of interest in this study.

$$\eta_i = \frac{3.6 \cdot 10^8}{c_i \cdot Q_i} \quad [\%] \quad (7)$$

The last problem to be solved regarding the simulation model is to evaluate the total period of combustion per cycle. The period of combustion is connected to the value of flame propagation velocity through homogeneous air-fuel mixtures. Preliminary tests have been conducted on a single-cylinder CFR (Committee for Fuels Researches) engine, in order to approximate the velocity of flame propagation through gasoline-air and ethanol-air fuel mixtures.<sup>6</sup> Measurements were taken at a constant engine speed of 900 rpm, full load and a compression ratio set to 7.25. The results are shown in Figure 1, in which the ratio between the ethanol-air flame velocity and the gasoline-air flame velocity is plotted *versus* various relative air-fuel ratios  $\lambda$  (within the 0.8-1.2 range). Flame propagation velocity could be determined by measuring the time of flame propagation between two ionization plugs mounted inside the cylinder, at a certain distance from one another.<sup>7</sup>

The trend of the obtained curve could be explained by the peculiarities of both combustion types. In rich air-fuel mixtures, the flame velocity of gasoline-air mixtures is low, due to the lack of oxygen; on the contrary, the ethanol molecule contains a high oxygen amount, favoring reactants diffusion and flame propagation. With lean air-fuel mixtures, flame velocity through the gasoline-air mixtures decreases, because of the lack of mixture flammability, which permits the conclusion that using dual gasoline-ethanol mixtures as SI fueling agent might increase the flame propagation velocity inside the cylinder, for decreasing the global combustion duration and, subsequently, for increasing the engine thermal efficiency. Further tests are planned to be performed on a SI four-cylinder engine with emission measurements, to study the influence of combined combustion types on the main exhaust emission levels.

## RESULTS

For applying the above model to analyze the combustion process and to plot the pressure diagram vs. the crankshaft rotation angle, the following engine operating parameters were used: maximum speed (5000 rpm), various load coefficients (30, 50, 70 and 100%), stoichiometric air-fuel ratios for air-gasoline only mixtures (the reference) and ethanol (20%)-gasoline-air fuel mixtures; another value of 1.1 was chosen for the air-fuel ratio of the ethanol (20%)-gasoline mixture. Figure 2 shows the results of the simulation concerning the values of the indicated efficiency vs. engine load, when only gasoline (reference case) and 20% ethanol-gasoline fuels mixture were used at different air-fuel ratios. The

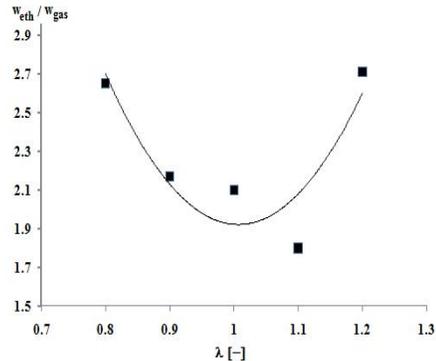


Figure 1: Ratio of flame propagation velocities through ethanol and gasoline air-fuel mixtures vs. relative air-fuel ratio

highest efficiency values were obtained when using ethanol at a stoichiometric dosage value, which is conclusive as to the fact that combustion in SI engines is well centered on the top dead center point, at an optimum spark advance value (about 25 degrees of crankshaft rotation angle).

It is obvious that the combustion of another type of oxygenated fuel is much different from the standard one with classic fuel. Combustion with a high oxygen reserve, present in the alcohol molecule, is rapid, generating high temperature values of the motor fluid. Thus, the results of the theoretical investigation should be correlated with the experimental measurements performed on the engine test bed, adequately equipped.

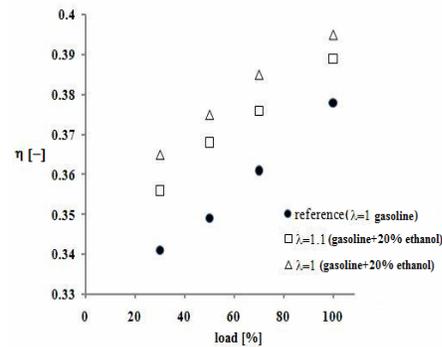


Figure 2: Indicated efficiency vs. engine load at maximum speed and 20% ethanol mass substitution rate

## CONCLUSIONS

In the near future, the production of bioethanol will mark a significant development, due to the new technologies applied, such as enzymatic fermentation of the sugar compounds from wooden biomass.

Theoretically, the simulation of combustion and of the heat release from the in-cylinder volume could reveal an appreciable increase in the indicated efficiency of the engine, when substituting a 20% mass fraction of the classic fuel.

This thermodynamic analysis of the motor cycle obtained by simulation has to be confirmed by further tests carried on the engine mounted on a special test bed. It is also imperative to carry out such tests measuring the values of engine emissions, in order to correlate them with the engine regime performance.

Globally, the entire array of engine regimes should be investigated, especially the most frequently used ones (urban regimes at medium loads and speeds), when supplying the engine with different ethanol-gasoline ratios and also under different relative air-fuel ratios.

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