

Low Cost Wet Ethanol for Spark-Ignited Engines: Further Investigations

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ABSTRACT

This study evaluates the performance of an ethanol fueled spark ignited engine running with high levels of hydration. Ethanol is a renewable fuel and has been considered a promising alternative to counteract global warming and to reduce pollutant emissions. Its use is well established in ICE as the main fuel or blended with gasoline. However, due to its lower calorific value, it shows increased fuel consumption when compared to gasoline, rendering its use sometimes less attractive. The energy demand to produce ethanol, especially at the distillation phase, increases exponentially as the concentration of ethanol-in-water goes from 80% onwards. Thus, mixtures with less than 80% of ethanol-in-water would reduce the energy consumption during production, yielding a less expensive fuel. In previous studies, to evaluate the feasibility of wet ethanol as a fuel for spark-ignited engines, results have shown that it was possible to use mixtures of up to 40% of water-in-ethanol. This further experimental results aiming at optimizing combustion phasing for a fixed 50% mass fraction burnt point. The engine used in this research was a 4-stroke single cylinder 0.668L, originally Diesel fueled but converted to ethanol with a PFI system, an electronic controlled throttle body and a spark plug inside the swirl chamber. Tests have shown stable combustion despite the large water content. Emissions results have shown reduced NO_x and higher CO for the higher water content ethanol.

CITATION: Martins, M., LanzaNova, T., and Sari, R., "Low Cost Wet Ethanol for Spark-Ignited Engines: Further Investigations," *SAE Int. J. Fuels Lubr.* 8(2):2015, doi:10.4271/2015-01-0954.

INTRODUCTION

According to IPCC (Intergovernmental Panel on Climate Change) there was recent 70% increase in greenhouse gases emissions, attributed, mainly, to fossil fuel combustion [1]. WWF (World Wildlife Fund) says that climate changes carried by this high level of emissions can lead to global impact such as floods, storms and heat waves [2]. In this global scenario, the production and use of renewable energy sources are pre-requisites for economic development [3,4]. In this context, the use of biofuels is pointed out as one of main solution to decrease emissions and at the same time bring, economic benefits to the rural sector. Comparative researches indicates the wide experience in sugar/ethanol production and natural resources as factors that place Brazil ahead in relation to other countries with temperate and tropical climate in biofuels production. USA and Brazil are together responsible for 90% of world ethanol production [5]. Brazil is the main ethanol producer from sugarcane and the pioneer in the development of engine technology adapted to this fuel [6].

Through the Brazilian PROALCOOL program (National Program of alcohol) instituted in 1975, a reduction of 11% in CO₂ emissions was achieved [7]. Together with the law that demands the use of biodiesel, in effect since 2005, the National Plan for Bioenergy 2006-2011 [8] and other governmental incentive programs the use of ethanol in Brazil is then inevitable, also due to its well established distribution network its good technical and environmental performance[9].

Ethanol presents high knock resistance, allowing higher compression ratios in comparison to gasoline. Despite of its 37% lower calorific value [10], its specific fuel consumption can be only 20% higher than gasoline [11]. If made from sugarcane, it offers great advantages regarding greenhouse gas emissions since its combustion-generated CO₂ emissions are recovered by photosynthesis during crop production. When used in internal combustion engines, its higher latent heat of vaporization cools the intake air, increasing the air density, leading to higher volumetric efficiencies. During combustion, ethanol presents higher laminar burning velocities and inflammability limits, enabling the operation with lean and heavily diluted mixtures.

Despite of all advantages mentioned above, it is still necessary to verify its feasibility regarding pollutant emissions and fuel economy. In regards to fuel prices, there is an important opportunity for reduction should more water content be allowed in ethanol. The production of the current 95.6%azeotropic hydrous mixture is highly energy demanding. The energy necessary for production increases linearly until 80% of ethanol-in-water (v/v) and exponentially until 95.6% (azeotropic point). Thus, it is easy to realize that if more water content could be allowed the fuel costs would reduce [12,13,14,15].

Several studies report the use of ethanol in advanced combustion concepts, like the one carried out in a 1.9L, four-cylinder turbocharged engine with direct injection and working in HCCI (Homogeneous Charge Compression Ignition) mode, fuelled with wet

ethanol. As results, it was achieved stable operation for mixtures containing up to 40% of water. Also, as expected in HCCI engines, NO_x levels were extremely low. Unburned hydrocarbons and carbon monoxide increased according to higher water percentage [13]. Musin et al studied the performance and combustion of a 0.398L engines, with compression ratio of 10.2:1 and electronic controlled injection system when operating with water-in- ethanol percentages up to 40% [16]. It was verified an increase in specific fuel consumption and lower indicated efficiency for higher water percentage. In agreement with Mack et al. work, it was also verified reduction in NO_x emissions and higher HC and CO levels [17].

Cristensen and Johansson [18] evaluated the influence of water injection in the intake manifold in a 4-cylinder 1.6 L engine, with compression ratio of 18:1, naturally aspirated and in turbocharged mode, running in HCCI mode. The auto ignition event was controlled through intake air pre-heating. Water addition allowed extending the operation range without knock occurrence due to lower in-cylinder temperatures. Unburned HC and CO emissions showed increased values attributed to lower combustion efficiencies and flame quenching.

A different research was carried out by Idaho University in partnership with RAI (Automotive Resources, Inc.), in which catalytic igniters were used to burn high water content ethanol. Tests were carried out in a 3 cylinders engine, 0.998 liters, originally running with compression ignition. A mixture containing 30 v/v of water in ethanol was chosen. To enable ethanol usage a new PFI injection system was developed for wet ethanol SI operation mode. The combustion followed the concept of flame torch ignition. Using catalytic igniters the combustion took place in lower temperatures due to lower energy required to burn the fuel. Reduction in CO and NO_x emissions were found due to lower temperatures attributed to the use of catalytic igniters and increase in water content when comparing to the original diesel operation.

In a recent work Saxena et al. [17], investigated the optimum conditions for power generation using hydrous ethanol and HCCI combustion concept. Engine speed was fixed in 1800 RPM and the mixtures ranged from 70% to 100% of ethanol percentage in volumetric content. Each operation point was defined by ignition phasing, boost pressure and equivalence ratio. Optimum operation point was found for boost pressure of 2 bar, equivalence ratio of 0.55 and combustion timing (50% of mass fraction burned) close to 8° ATDC for mixtures containing water percentages of 30% of water in ethanol in volumetric content.

A preliminary study in a single-cylinder engine carried out by the authors of this paper, have shown that at fixed ignition timing, it was found reduced NO_x emissions, and stable combustion for increased amounts of water.

Hence, this paper aims to deepen the previous studies and provide further investigations about the influence of water on combustion of water-ethanol fuel blends on a SI single-cylinder experimental engine. Concentrations ranging from 5% to 40% by volume of water

in ethanol were tested with ignition timing set for a fixed 50% mass fraction burned crank angle. A simulation software was used to process and analyze combustion data.

METHODOLOGY

Engine

A 0.668 L, single cylinder engine was used. It was originally diesel fuelled, converted to spark ignition. Table 1 presents the engine characteristics. Before modifications, diesel was injected in a swirl chamber, starting a robust combustion propagating to the main chamber. This feature was explored in this engine to run with high levels of hydration, following the concept of flame torch ignition, widely used to burn diluted charges.

To enable ethanol usage, some engine modifications were necessary. Compression ratio was decreased through spacer rings to 12:1. The original diesel injector was replaced by a spark plug to start combustion in the swirl-chamber, as illustrated in Figure 1. A new intake runner was designed containing throttle body and PFI (Port Fuel Injection) injection system. In the exhaust runner it was installed an oxygen sensor to provide air-fuel ratio. All these devices were controlled by an electronic control unit, Fueltech® FT 400.

Table 1. Engine characteristics

Parameter	Description
Number of cylinders	1
Bore x stroke (mm)	90.0 x 105.0
Displaced volume (cm ³)	668
Compression ratio	12
Intake valve diameter (mm)	41.0
Exhaust valve diameter (mm)	35.0
Intake valve lift (mm)	9.0
Exhaust valve lift (mm)	9.5
Intake valve open	36° BTDC
Intake valve close	184° ATDC
Exhaust valve open	204° BTDC
Exhaust valve close	64° ATDC
Idle speed (rpm)	1100-1200

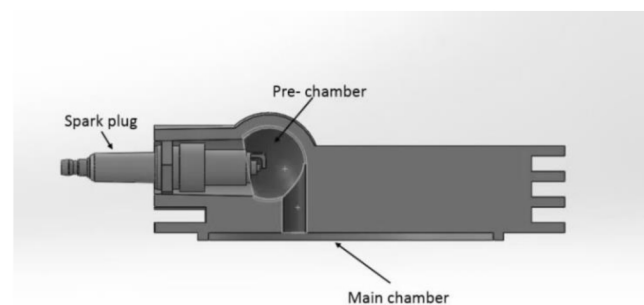


Figure 1. Spark plug location

Test Bench Setup

An eddy current dynamometer was used to load the engine as illustrated in [Figure 2](#). A Labview® routine was developed to control the dynamometer, aiming at constant speed engine operation. For this, it was employed a National Instruments data acquisition board model 6009.

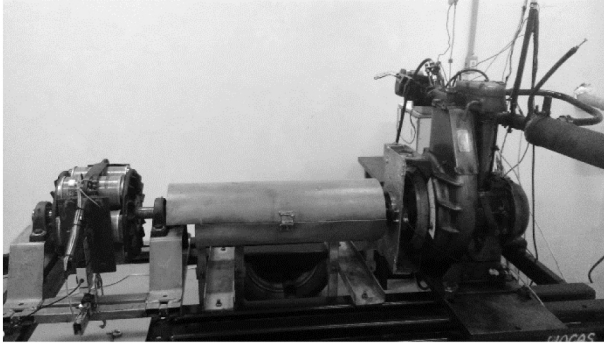


Figure 2. Dynamometer setup

Instrumentation and Data Acquisition

All the required sensors were installed so as to perform data acquisition for heat release analysis. Pressure sensors (MPX 4250 AP) for instantaneous manifold pressure monitoring were installed. For instantaneous in-cylinder pressure measurement it was chosen an AVL® solution constituted by an AVL GH14D® sensor e a signal conditioner AVL FlexIFEM®. This sensor presents high reliability and reduced size due to its composition (gallium orthophosphate) leading to no required water cooling. This sensor was installed in the cylinder head, as illustrate by [Figure 3](#), in the opposite side of the pre-chamber, where combustion starts. Thermocouples were installed to monitor mean temperatures of intake and exhaust runners. Torque was measured using a commercial load cell. Fuel consumption was measured through a burette with 0.2 resolution. All sensors were connected to a NI 6250 board. These signals were used as inputs to an online combustion processing routine, allowing real time combustion data visualization.

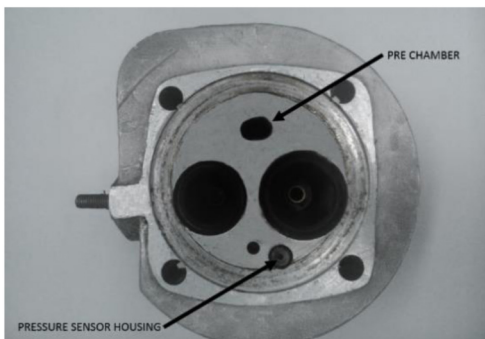


Figure 3. In-cylinder pressure sensor location

In this research, emissions were also evaluated by means of a commercial automotive gas analyzer Alfatest® Discovery G4®. The experimental setup scheme is shown in [Figure 4](#). All the sensors accuracies are presented in [table 2](#).

Table 2. sensors accuracy

Device	Accuracy
In-cylinder pressure transducer	0.3 % of Full Scale
Intake pressure transducer	1.5% of Full scale
Exhaust pressure transducer	1.5% of Full scale
Lambda meter	1.7% of measurements
Fuel flow	1.8% of measurements
Air Flow (calculated through lambda and fuel flow)	4.0 % of measurements
Load cell	0.03 of Output voltage
K type thermocouples	0.75%of measurements
Gas analyzer	
NO _x Probe	4% of full scale
CO Probe	3% of Full scale

Experimental procedure

The cylinder head was tested on a flow bench to acquire discharge coefficient of valves and ports, which were used as inputs for calculating residual fraction and to perform heat release analysis. Valve lift and characteristic engine dimensions were also directly measured.

During the engine testes, the behavior of all blends was analyzed with ignition timing adjusted for a fixed center of combustion (50% MFB at 10° CA after TDC). This method was chosen so as to obtain better comparison than methods aiming at MBT or just with a fixed spark timing such as used in previous works. Load was kept in 37 Nm, corresponding to a BMEP of 7 bar, so as to keep a safe margin before maximum engine load. All tests were carried out using stoichiometric lambda and ignition timing was modified to adjust for 50% CA of mass fraction burned. The nomenclature utilized in this research was EXXWXX, where “E” stands for ethanol and “W” for water. “XX” represents the percentage of each substance in the mixture. Tests were performed according to the following steps:

1. Engine warm up at 1400 RPM;
2. Engine speed setup;
3. Load application;
4. Stabilization at the desired operation point, adjusting ignition timing and air-fuel ratio;
5. Fuel consumption measurement and data acquisition;
6. Fuel change;
7. Repeat the steps above for all the mixtures.