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ANALYSIS OF A SPARK IGNITION ENGINE FUELLED WITH HIGH WATER CONTENT ETHANOL

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Abstract. *The use of bio-fuels as a complementary energy source or even to substitute fossil fuels has been increased due to the demand to reduce carbon emissions and other greenhouse gases. In this scenario, bio-ethanol shows itself as a very important ''green energy'' source. The reduction in carbon emissions of the ethanol life cycle compared to fossil fuels can be attributed to the crop photosynthesis process in which CO2 is absorbed from the environment. The main problem in the use of bio-fuels is their high production costs when compared to fossil fuels. Knowing that in ethanol production cycle a large amount of energy is spent on the distillation phase to obtain ethanol-in-water in volumetric concentrations above 80% of ethanol, the use of ethanol-in-water fuels with high water content in internal combustion engines could highly reduce such fuel cost when compared to Brazilian commercial ethanol. This paper evaluates the use of water-in-ethanol mixtures as fuel in a port injected 0,668L single-cylinder Otto engine. Dynamometer tests were performed in order to obtain in-cylinder pressure data. Volumetric water content in ethanol is varied from 5% to 40%. One- dimensional computer simulation was made to assess the implications of using such fuel. A complete heat release analysis of the combustion process was carried. It was found that combustion stays stable and IMEP could be kept almost constant with little effect on BSFC for the test conditions evaluated.*

Keywords: *water-in-ethanol, internal combustion engines, spark ignition*

1. INTRODUCTION

The increasing economic and environmental pressure for more efficient internal combustion engines which can efficiently use renewable fuels brings out the need for research in both the internal combustion engine and the fuel fields. To achieve that, renewable fuels must be economically competitive and generate less pollutant emissions during their life cycle when compared to fossil fuels. Combustion is the most important phenomena in internal combustion engines and this is the linking field where most effort has been spent to improve both performance and emissions of renewable fuels used in internal combustion engines.

Brazil has vast experience in renewable fuels. Bio-ethanol has been extensively researched for the use in internal combustion engines since the creation of the Sugar and Alcohol Institute in 1933. Production has been greatly motivated by Brazilian government since 1970s due to the international oil crisis (Triana, 2011). This fact, which in the first instance had the intention to reduce Brazilian dependence of imported oil, also increased the country energy matrix mix (Freitas, 2011). In the environmental side, it is estimated that there was a reduction of 11% in the total CO_2 production that would be produced if only fossil fuels were used instead of ethanol (Szklo et al, 2005).

The process to produce Ethanol consists in crop plantation, mashing and cooking, fermentation, distillation and dehydration. The fermentation phase provides a mixture of ethanol-in-water at volumetric concentrations varying from 6% to 12% (Ladisch and Dyck, 1979). Through the distillation process it is possible to achieve almost 96% of ethanol-in-water, although the energy used to reach this percentage dramatically increases after 80% mixture. It is possible to decrease the amount of water in the mixture, reaching a minimum amount, defined as anhydrous ethanol. it requires techniques such as molecular sieves which, on the other hand requires high energetic power in consequence high production costs.

Some researches have been carried regarding the influence of the use of high water content fuels in spark ignition engines. Harrington and Wang (1982), studied the injection of water in both liquid and vapor states in a Waukesha CFR intake manifold with variable compression ratio. For the vaporized water charge, the combustion presented lower burn rates, there was increase of the ignition delay and combustion duration comparing to gasoline injection. The use of gasoline-liquid water leaded to higher reduction in the burn rates due to the water latent heat of vaporization. Water addition in both states enabled higher ignition advance without knock.

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Tsao and Wang (1984) investigated a water-gasoline emulsion, from 0% to 15% water mass content, in a carbureted single-cylinder engine with 375 cm³ volumetric displacement. Adding water it decreased the in cylinder temperature, improving the knock resistance. Further, NO_x, CO emissions and fuel consumption were reduced, but increase in HC emissions was related.

Christensen and Johansson (1999) evaluated the effect of water injection in a 6 cylinder 1.6 Volvo engine. It was modified to run only with one cylinder. HCCI operation was achieved using a pre-heater, and ignition timing was controlled through intake air temperature. Reducing the in-cylinder temperature due the water injection, enabled to increase the power generation limit without knock occurrence. The drawbacks were the high presence of HC and CO on the exhaust gases caused by water which decreases the combustion efficiency.

Studies carried in an original Diesel engines cycle modified to run with wet-ethanol was made in an association between the Idaho University and RAI (Automotive Resources, Inc.) (Beyerlein et al, 2001, and Cordon et al, 2002). The tests aimed to compare the efficiency and the exhaust emissions from the Diesel and the system using a catalytic igniter for water-in-ethanol mixtures with water volumetric content of 30%. The combustion process temperature was reduced due to the presence of water, decreasing the CO and NO_x emissions compared to diesel operation. HC reached high levels mainly in function of the flame quenching phenomena.

Using an air assisted direct injection turbo-charged Otto engine, Brewster et al. (2007) evaluated three levels of hydration of wet-ethanol, 6.5%, 13% and 20% of mass content. Varying the ignition time, and for a same lambda value, it was possible to see a reduction in torque when out of MBT. Water addition again enabled MBT operation without knock. At MBT, a decrease in exhaust and in-cylinder temperature was shown due to the charge higher heat capacity and higher charge latent heat of vaporization, when water was added. Lower in-cylinder temperatures improved the charge resistance to the abnormal combustion process. The possibility to develop an engine with higher compression ratio and extended turbocharger operating range was observed.

Another researcher, Mack et al. (2009) studied the behavior of a Volkswagen engine operating in HCCI using wet-ethanol from 0% to 60% water content. In order to control de ignition timing the inlet air was heated with an electric resistance as heater device, this way the water-ethanol mixture could evaporate before the intake process. The air-fuel heating was the limit to in the increase of the power output. Adding water, the rate of heat release kept relatively constant while the total heat release and the peak pressure were decreased.

Later, Saxena et al. (2012) used a heat recovery system to pre-heat the inlet air. The work had been developed as an extension of Mack's work, in the same four cylinder engine. As only one cylinder was used, turbocharger utilization had been compromised, thus, an electrical compressor was used instead the original charge mechanism. Controlling the inlet air enabled knock limit improvement. The energy used to produce ethanol with 20% of water in volumetric content is much smaller when compared to produce ethanol with 5% water. It was proved that an engine using a mixture of 80% ethanol and 20% water could bring benefits, at the same time reducing fuel costs and improving the energy balance of its lifecycle.

This paper aims to investigate the performance behavior of a single cylinder spark ignition engine using water-in-ethanol concentration from 5% to 40% of water in volumetric content trough simulation and test bench analysis.

2. EXPERIMENTAL SETUP

The engine used to perform this study was an Agrale M90 originally Diesel, modified to operate in Otto cycle. The engine's cylinder head have a swirl chamber that is usually used to achieve higher flow velocities. In addition, this fact allows charge to ignition in very diluted conditions. The figure 1 illustrates the engine. Some modifications were made to enable the engine run in Otto cycle. The decrease in the engine compression ratio could be achieved through the use of a spacer between the cylinder head and the gasket. To enable the correct amount of fuel injected each cycle a FuelTech F400 Eletronic Control Unit (ECU) was used. Modifications in the cylinder head diesel injector housing were made to install a commercial spark plug inside the swirl chamber. The fuel was inject by a PFI system (Port fuel Injector) installed in a new runner, and a commercial throttle valve with a throttle position sensor was used. An excess air meter installed in the exhaust manifold enabled the air/fuel ratio calculation. Table 1 shows the engine characteristics.

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Figure 1. Agrale M90

Table 1. Engine characteristics

Item	Characteristic	
	Original	Modified
Engine	Agrale M90	
Cylinders	1	
Strokes	4	
Ignition type	CI	SI
Fuel	Diesel	Ethanol
Fuel Injection	Swirl camber indirect injection	Port fuel injection
Refrigeration system	Forced air	
Bore (mm)	90	
Stroke (mm)	105	
Compression ratio	19:01	12:01
Displaced volume (dm^3)	0.668	
Intake Valve Open	-36° BTDC	
Intake Valve Close	184° ATDC	
Exhaust Valve Open	-204° BTDC	
Exhaust Valve Close	64° ATDC	

The engine cylinder head was submitted to flow bench tests from where data could be obtained to calculate the intake and exhaust valves discharge coefficients, in the direct and reverse flow direction. . The cylinder head was fixed in the flow bench with an adapter with a cylinder with 100 mm length and the same diameter as the piston bore. A comparator clock was attached to the device enabling the operator to reach the correct valve lift. The discharge and flow coefficients were calculate using the isentropic flow through an orifice equation described by Heywood (1988). The pressure difference used was 10 CFM kPa. Figure 2 presents the discharge coefficient related to L/D. The reference area used to calculate the discharge coefficient was the head valve area instead of the valve curtain area.

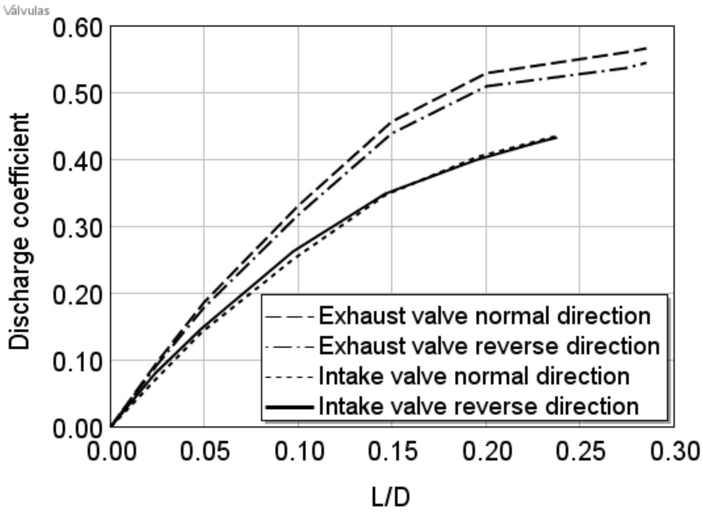

Figure 2. Discharge coefficient

To acquire the in-cylinder pressure data, an AVL GH14D was installed in the cylinder head. It was installed with a steel adaptor. A flame arrestor was used as thermo protector to reduce thermal drift. The sensor localization was chosen in a way to avoid the pressure fluctuations due the jet motion from the combustion leaving the swirl chamber, so was installed at the opposite side of the swirl chamber orifice. The AVL FLEXIFEM was used for signal amplification and conditioning, which also compensates electrical drifts.

Two piezoresistive sensors were used to obtain intake and exhaust instantaneous pressure. They were positioned in the exhaust and intake runners the nearest as possible of the respective ports. The sensors chosen to this task were two MAP (manifold absolute pressure) Freescale Semicondutors MPX4250AP piezoresistive sensors. Thermo Couples K type were installed to monitor and acquire the intake and exhaust average temperatures in the same position of the pressure sensors.

Due the engine parts arrangement it was not possible use an encoder without engine crank block modifications. Thus, a 360 teeth toothed wheel was used to relate the crankshaft position and the in-cylinder pressure traces. A variable reluctance sensor was used to generate the teeth signals enabling a 0.5 °CA resolution. The TDC was triggered by a hall sensor.

Installed sensors signals were acquired with a National Instruments® acquisition board model NI USB-6259 and Labview Signal Express® software. Post processing data was required, instead of real time processing, due to the angle indicating system . A Mathlab® routine was designed to post process all data. Pegging was developed through offset of the entire in-cylinder pressure data to equalize the instantaneous pressure at the intake phase BDC (bottom dead center) with the intake manifold pressure at the same crank angle position. Figure 3 shows the installed sensors locations at the engine.

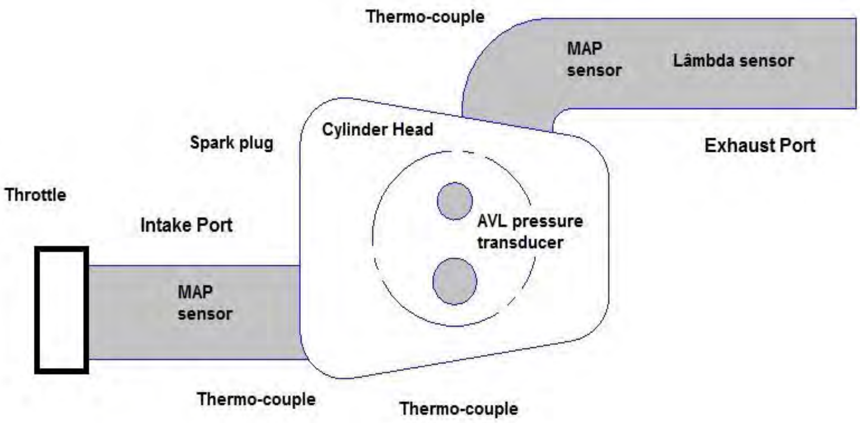

Figure 3. Engine installed sensors

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Fuel consumption was measured with a 0.2 ml resolution burette for 30 seconds engine stable operation. The ethanol-in-water mixtures were characterized in a Anton Paar DMA 4500M densimeter. Water volumetric content in the blends varied in the range from 5% to 40%. The nomenclature used to define the blends was EXXWXX, where E and W stand for ethanol and water, and the XX values are the volumetric percentage of each component. Air consumption was calculated using the fuel consumption and the lambda values. The water-in-ethanol blends used in this work was E95W5, E90W10, E80W20, E70W30 and E60W40.

For the experimental tests it was used an eddy current dynamometer and a commercial load cell for torque measurement. The tests were carried in part load operation aiming to keep the BMEP constant for different water-in-ethanol mixtures. Spark timing was modified during test until the maximum value without knock could be achieved until MBT .

The experimental tests were carried as the methodology described above:

- Using E95W05 (Brazilian commercial hydrous ethanol), the engine was warmed up;
- At 1800 rpm the brake torque of 34 Nm was applied to the engine;
- At engine stable operation, data was acquired;
- The fuel lines were drained and a new mixture was tested;

3. HEAT RELEASE ANALYSIS

For the heat release analysis the post-processed experimental data was inserted in a Three Pressure Analysis Burn Rate Calculation (TPA) in the GT-Power 7.3 software. GT-Power is an one-dimensional fluid dynamics software dedicated to simulate reciprocating internal combustion engines. During the gas exchange and compression phases the cylinder is treated as a control volume and during the combustion phase (while valves still closed) it is treated as a thermodynamic system. During combustion, for in-cylinder gases thermodynamic state calculations it is used a two-zone model, similar to the one described by Ferguson (2000) which separates the combustion chamber in a burned and an unburned zone. At the beginning of the combustion all gases are in the unburned zone. As combustion takes place, mass transfer from the unburned to the burned zone is calculated by the rate of the fuel burn, which is directly related to the heat release rate. The formulation of the two zonal model for in-cylinder fluid state calculation described by GT-Power Manual (XXXX)

- Unburned zone

$$\frac{dm_u e_u}{dt} = -p\frac{dV_u}{dt} - Q_u + \left(\frac{dm_f}{dt}h_F + \frac{dm_a}{dt}h_a\right) \tag{1}$$

- Burned zone

$$\frac{dm_b e_b}{dt} = -p\frac{dV_b}{dt} - Q_b + \left(\frac{dm_f}{dt}h_F + \frac{dm_a}{dt}h_a\right) \tag{2}$$

where the subscripts u and b define burned and unburned zones, and the subscripts f and a define fuel and air. m is the mass, e is the fluid internal energy and h is the specific enthalpy, p is the cylinder absolute pressure and V is the zone volume. Q is the term that handles in-cylinder heat transfer. As in-cylinder heat transfer depend on many different parameters, it is common in the internal combustion field to use correlation such as the Woschni correlation which is based in the Nusselt correlation.

In normal mode engine simulations (also called forward run), a given heat release curve is used to calculate the amount of mass that is transferred from one zone to the other, and consequently the energy released during combustion. The basic of the heat release analysis is to calculate how much fuel is burned each instant during combustion from the analysis of an engine cycle pressure diagram. This analysis is also called reversed run. The TPA method consists of calculating the heat release using the intake and exhaust manifold instantaneous pressures and the in-cylinder pressure. Reduced models of the intake and exhaust systems were created to calculate the gas exchange process. The intake and exhaust models consist of the respective ports and runners at which ends the instantaneous intake and exhaust pressures are used as boundary conditions. The mass flows through valves are calculated with the equation for the isentropic flow through an orifice.

Intake and exhaust systems are divided in short ducts which have similar dimensions. Each duct is discretized in smaller volumes. Figure 4 presents an example of the spatial discretization method used in GT-Power

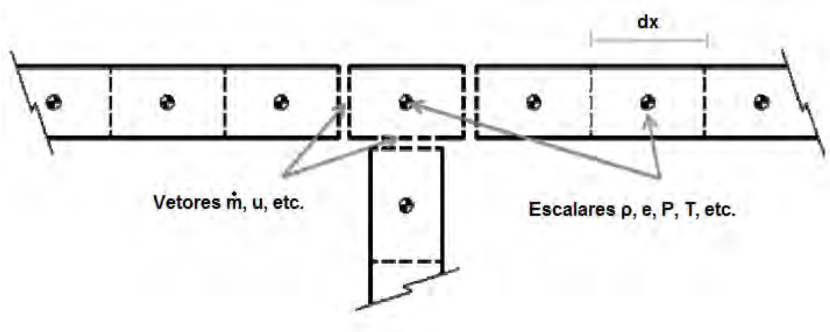

Figure 3. Discretization example of ducts used in GT-Power

As it can be seen in the figure, dx is the discretization length, and the inlet and outlet diameters are informed for each duct. Scalar quantities are calculated each volume center and the vector quantities are calculated at the volume boundaries. For the gas exchange process in the ducts the conservation equations of mass, energy and momentum are solved in the 1-D and compressible form. The compressibility is managed through the ideal gas law. The transport equations are similar to which Heywood (1988) presents in the differential form. The temporal discretization scheme used to solve the conservation equations is an explicit 1st order Euler scheme. The transport equations solved for ducts in for the explicit discretization method, as presented in GT-Power Manual (XXXX), are shown bellow

$$\frac{dm}{dt} = \dot{m}_{vc} = \sum_i \dot{m}_i - \sum_e \dot{m}_e \tag{3}$$

$$\frac{d(me)}{dt} = p\frac{dV}{dt} + \sum_i \dot{m}_i H_i - \sum_e \dot{m}_e H_e - h_c A_{sup}\left(T_f - T_w\right) \tag{4}$$

$$\frac{d\dot{m}}{dt} = \frac{dpA + \sum_i \dot{m}_i U_i - \sum_e \dot{m}_e U_e - 4C_f \frac{\rho U^2}{2}\frac{dxA}{D} - C_{Press}\left(\frac{\rho U^2}{2}\right)A}{dx} \tag{5}$$

where

$$\dot{m} = \rho A u \tag{6}$$

and m is the mass of the control volume, e is the internal energy of the control volume, p is the control volume pressure and V is its volume. H is the specific enthalpy, h_c is the convection heat transfer coefficient and T_f is the temperature of the fluid flow, while A_{sup} is the internal superficial area and T_w is the temperature of the duct wall. ρ is the density, A is the cross section area of the control volume with diameter D, U is the average velocity in the normal direction of the flow, C_f and C_{Press} are the friction and pressure loss (due to bends and shape modifications) coefficients. Both friction and pressure loss coefficients and convective heat transfer are calculated externally according to conventional formulation or table values.

The calculation of the gas exchange process through a finite volume method of the intake and exhaust systems instead of user imposed mass flow quantities is an advantage of the TPA method. In-cylinder residual fractions are also calculated with this method which is a difficult parameter to be experimentally measured.

The software automatically calculates the time step for each iteration to guarantee numerical stability through the Courant criteria. Time step can be reduced with the application of a coefficient varying from 0.1 to 1. As both spatial and time discretization can be controlled a spatial and temporal mesh study has been developed to guarantee lesser computational time spent for each analysis and which provided consistent results. It has been found that the spatial discretization of 10 mm and the time coefficient of 0.25 has been used instead the standard values of 36 mm and “1”, which did not provided consistent results due unsatisfied convergence criteria of the software.

The heat release model created for this study can be seen in Figure X. The Figure X also illustrates the geometric parts of the intake and exhaust systems. As GT-Power is an object oriented programming software, each object seen in the model represent a part or an engine system.

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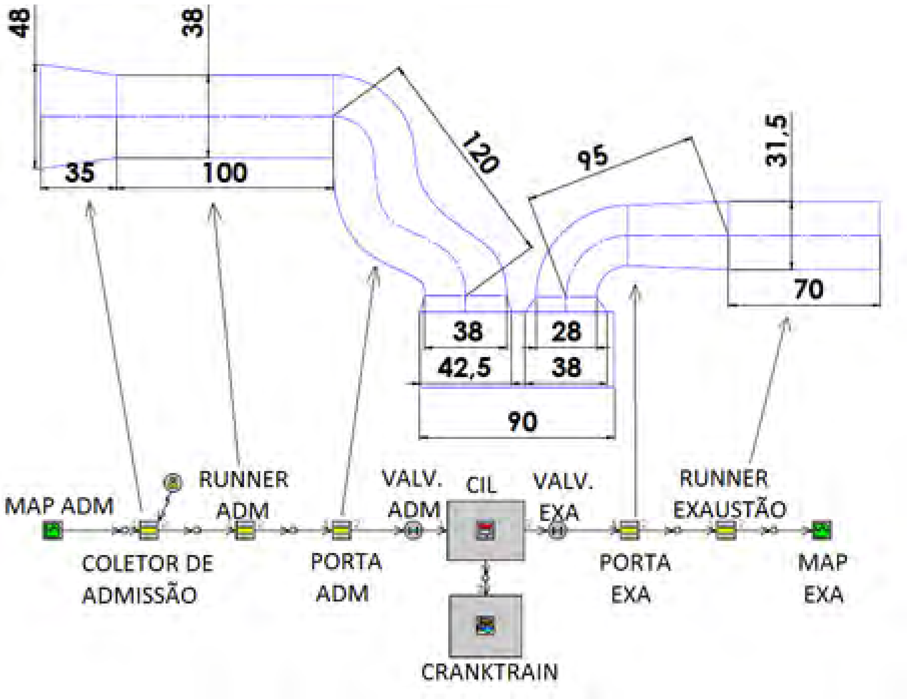

Figure 4. Heat release engine model illustration

As it can be seen in Figure 4, intake and exhaust instantaneous pressure are used as boundary conditions of the respective systems. The cylinder element contains the thermodynamic models and the cranktrain contains the mechanical models of the engine.

4. RESULTS

During experimental tests, load was kept constant around 34 Nm because it provided stable engine operation for all water-in-ethanol mixtures. Dynamometer torque curve mismatch problems occurred and lead to difficulty load support for the selected speed. This behavior resulted in BMEP variations of ±5% around the target BMEP value of 640 kPa, as shown in Figure 5, where is also shown the air mass flow per cycle for the different water-in-ethanol mixtures.

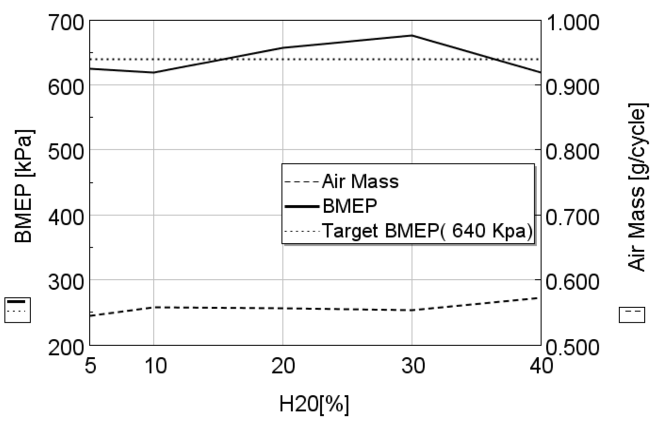

Figure 5. Experimental test BMEP and Air mass flow per cycle

As it can be seen As it can be seen , there were some variations in the target BMEP but all operation points still inside the experimental setup uncertainty. The load maintenance for different water-in-ethanol mixtures, with almost a very regular air flow for all tests, could be achieved due to spark advance reached due to water addition. Table 2 presents the spark advance values reached for all experimental tests.

Table 2. Spark advance for each fuel mixture

Fuel	E95W05	E90W10	E80W20	E70W30	E60W40
Spark Advance (°CA BTDC)	6.5	8.0	11.0	16.5	20.5

It can be seen that as the spark advance approximates the minimum spark advance for maximum brake torque (MBT), the engine energy conversion becomes more efficient. This fact is related to the geometric of the engines. The MBT is the point that will provide best energy distribution around the ideal angle to transfer the force from piston to the crankshaft. The increase in the engine brake thermal efficiency is inversely related to the brake specific fuel consumption. Figure 6 presents the engine Brake Efficiency (BEFF) and the Brake Specific Fuel Consumption (BSFC)

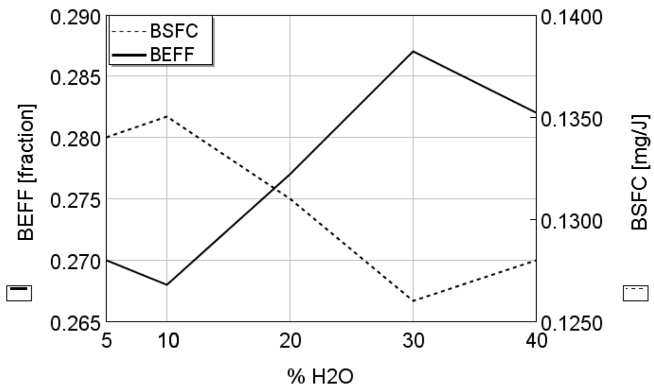

Figure 6. Engine Brake Efficiency and Brake Specific Fuel Consumption

For the heat release analysis, the experimental data was inserted in the TPA model of the engine. Figure 7 presents a comparison of the simulated and experimental data in the logP x logV graphical form. This kind of representation is commonly used in the engine field due to the fact that discrepancies are maximized.

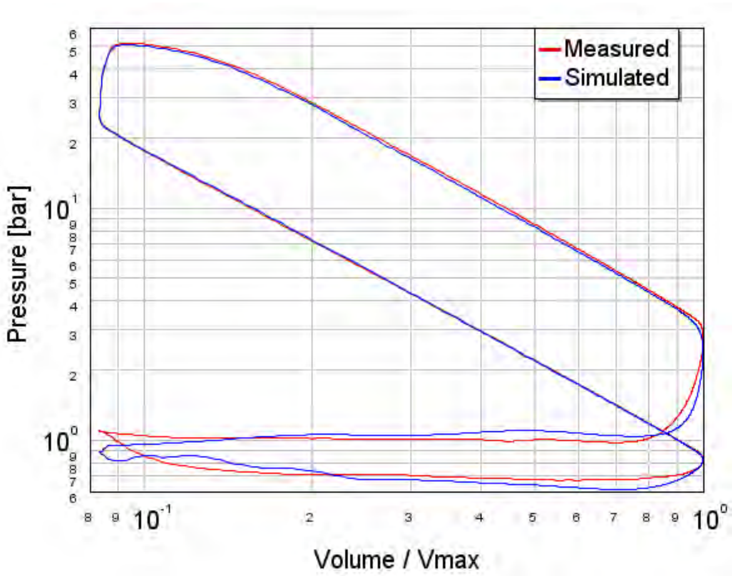

Figure 7. Experimental and simulated data comparison

The simulation proved to be very reliable and coherent with the experimental results. Ethanol mass flow and air and water mass flow are shown in the table 3. Water and air cycle mass flow are grouped because this is the way GT-Power presents these results.

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Table 3. Simulation and experimental mass flow comparison

	GT-Power (mg/cycle)	Experimental (mg/cycle)	Percent Difference (%)
	Air and water = GT Power "air"		
E95W05	530.5	549.8	3.6
E90W10	532.9	567.9	6.6
E80W20	588.6	578.4	1.8
E70W30	610.3	588.9	3.6
E60W40	621.5	625.2	0.6
	Ethanol		
E95W05	54.5	56.0	2.7
E90W10	55.5	55.9	0.7
E80W20	58.2	57.3	1.6
E70W30	58.9	57.0	3.3
E60W40	56.6	57.4	1.3

As it can be seen As it can be seen, engine volumetric efficiency and the fuel delivery are very consistent. Different in-cylinder instantaneous pressure data are shown in Figure 8. The pressure data demonstrates that as water content increases and higher spark advance is enable, more energy is released around the TDC, the point where the combustion chamber volume is minimum. This results in higher pressure during combustion. Also, water contributes to increase the molar quantity inside the combustion chamber also contributing to the pressure increase.

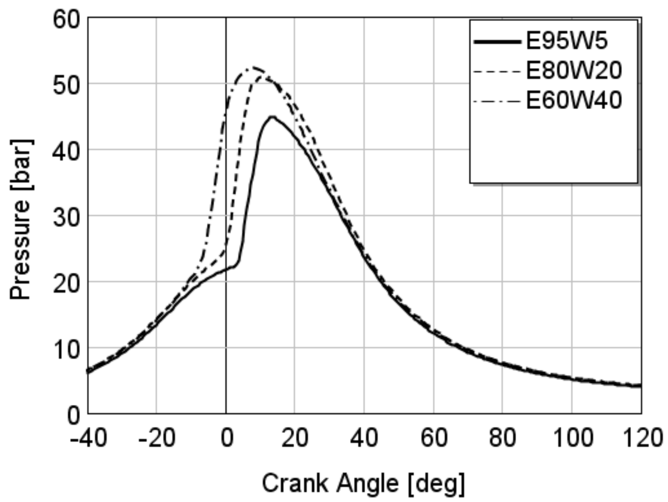

Figure 8. In-cylinder instantaneous pressure for different water-in-ethanol mixtures

The calculated burned fuel fraction is presented in Figure 9. It can be seen that as water content in the mixture increases the combustion becomes slower. Also, the rates of heat release tend to reduce - heat release rate is related to the derivative of the burned fuel fraction.

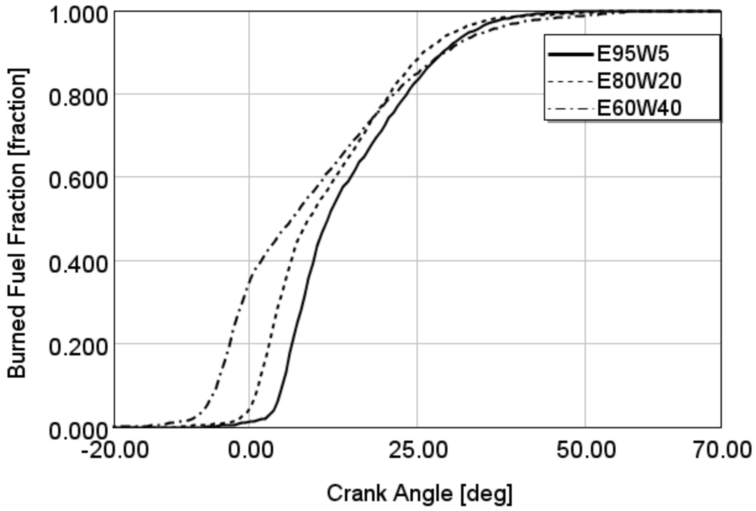

Figure 9. Burned fuel fraction of various fuel mixtures

The higher water content in the fuel mixture increases the ignition delay (0-2% mass fraction burn) due to water interaction with the initial flame development kernel. When the flame front leaves the swirl-chamber through its orifice, it finds colder charge areas, which require higher energy supply to support the combustion reactions. This reduces the heat release rate, as it is also evident in Figure 9, which shows the change in the curve's derivative around 50% MFB (mass fraction burned) – and in Figure 10, which presents the heat release rate profiles, in which two distinct zones can be seen.

The first heat release peak, shown in the heat release patterns of Figure 10, is related to the fast swirl-chamber charge combustion. As mentioned earlier, after leaving the swirl chamber the heat release rate is reduced, but as the fuel quantity is almost the same for the various fuel mixtures, the combustion duration tends to be compensated in the main chamber combustion phase. Thus, the combustion can be alternatively divided in three main events:

- Ignition delay;
- Fast charge combustion inside swirl chamber;
- Slow combustion in the main-chamber phase.

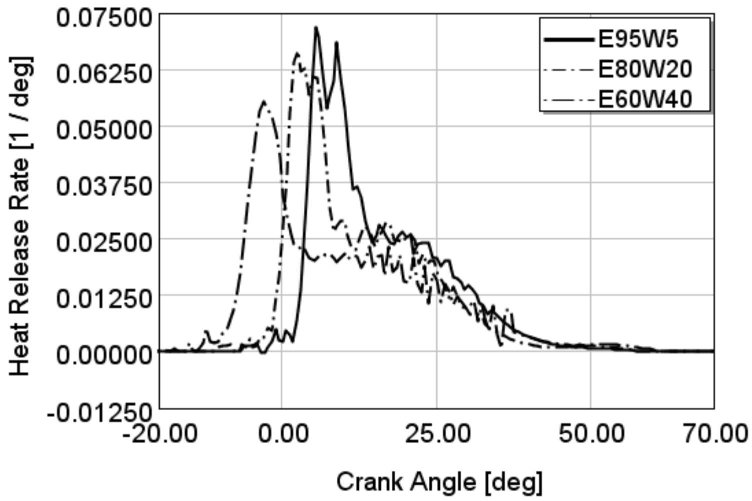

Figure 10. Heat release rates of fuel mixtures.

The increase in the water content contributes to the increase heat capacity of the cylinder charge. Water also absorbs energy from the charge during evaporation. This facts should contribute to increase combustion duration. However, due to higher combustion pressure and better combustion phasing, only achieved because of knock suppression promoted by water addition, combustion duration (10-90% MFB) is kept relatively constant as shown in Figure 11. Figure 11 also shows the ignition delay behavior as the water content in the fuel mixture is increased.

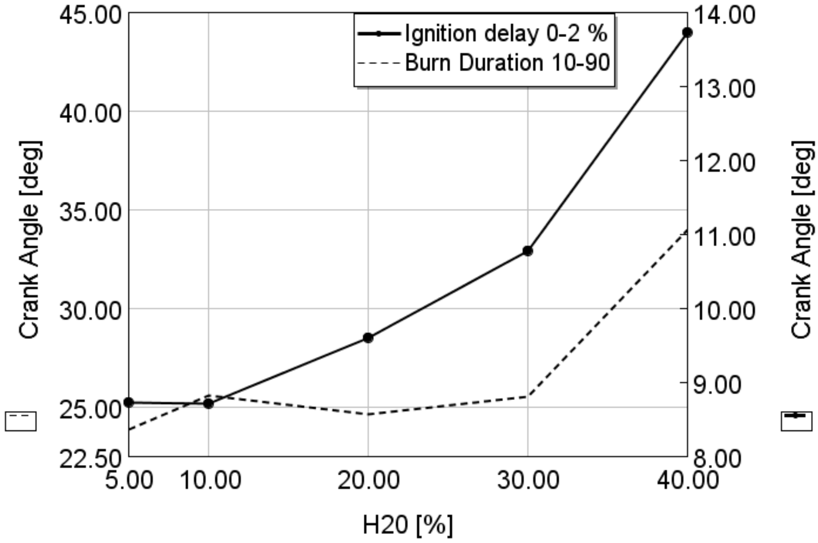

Figure 11. Burn duration 10-90% and ignition delay for different water-in-ethanol mixtures

5. CONCLUSIONS

In this study the water content in water-in-ethanol mixture was investigated through experimental tests and computer simulations. Water volumetric content varied from 5 to 40% and engine stable operation in partial load could be achieved for all fuel mixtures.

As the water content in the fuel mixture was increased, higher spark advance could be achieved without knock. This fact contributes to increased BEFF and reduced BSFC when comparing high content water-in-ethanol mixtures to commercial hydrous ethanol.

The fuel's knock suppressing characteristic due to water addition leaves room to interesting engine improvements. Higher compression ratio could be used to increase the BEFF. Also high pressure turbo-boost operation can be explored as well as an exhaust heat recovery strategy.

Finally, the use of high content water-in-ethanol mixtures in internal combustion engines instead of anhydrous ethanol can be a way to reduce this renewable fuel cost. In addition, the use of bio-ethanol instead of fossil fuels will reduce the carbon footprint and increase the matrix mix.

6. ACKNOWLEDGEMENTS

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