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Experimental analysis and modeling of internal combustion engine operating with wet ethanol

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ABSTRACT

The use of ethanol as fuel has encouraged many countries to improve their production processes. However, costs related to its current production model have been occasionally mentioned as a disadvantage to the use of this biofuel. One of the main costs is the energy expenditure for an effective distillation of the fermented ethanol wort, to result a final product called hydrous ethanol (about 95% by volume of ethanol in water). A promising alternative is the use of ethanol fuel with high fractions of water (above 5% by volume), the so-called wet ethanol, which would reduce the energy cost of production during the distillation. Thus, this study proposes the development of a mathematical model that, along with experimental data, is able to predict the effect that the use of wet ethanol has on the performance of internal combustion engines. In order to do so, along with commercial hydrous ethanol, blends were prepared with the following volume fractions of water: 10% (E90W10), 20% (E80W20), 30% (E70W30) and 40% (E60W40). Tests were performed considering two engine operating conditions: (a) fixed ignition timing and (b) adjusted ignition timing for maximum engine torque. The model was able to successfully simulate the gradients of pressure and temperature in the cylinder and it showed good ability to predict engine performance based on the variations of power, torque, conversion efficiency, and specific fuel consumption. Except for the specific fuel consumption, where the estimated error for variations was greater than 20%, the estimates for the other performance parameter presented relative errors lower than 7%. For all tests, the relative error was lower than 13%. The gradual increase of specific fuel consumption was associated with the increasing water content. Among the fuels tested, E70W30 showed the best performance, followed by the E80W20 blend; both were more efficient than the commercial ethanol.

1. Introduction

The search for energy sources that can reduce economic and environmental impacts without a need for severe changes in modern lifestyle has encouraged the research and development for alternative fuels. In Brazil, ethanol has been considered the main alternative, in the form of anhydrous ethanol fuel (AEF), with minimum alcohol content of 99.6% v/v, to be used in the formulation of gasoline type C, and as hydrous ethanol fuel (HEF) for sale at gas stations, containing levels from 95.1% to 96% v/v [1].

Compared to gasoline, ethanol's highest octane rating provides excellent anti-knock properties. Likewise, due to a considerably larger latent heat of vaporization the charge cooling effect can be used to increases the engine volumetric efficiency [2]. On the other hand, its calorific value is considerably lower than that of gasoline and diesel, and its high flash point makes engine cold start more difficult at low temperatures, requiring a cold-start system when below 13 °C [3].

The use of ethanol fuel at high volume fractions of water, referred to in this study as *wet ethanol* (for mixtures containing more than 5% volume of water) appears as a resource that would lower the cost of fractionation during the production of ethanol, making its energy balance much more attractive. The ethanol blend is miscible with water in all proportions, enabling combustion reactions to occur in both liquid phase and in gas phase [4]. The use of wet ethanol in HCCI engines is well known due to its capacity to run on low-grade fuels as long as the fuel can be heated to the point of ignition. As for SI and diesel engines, the calculation of the maximum water concentration that allows efficient engine

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operation is a difficult task, but the use of a medium to high grade (30–10% water content) wet ethanol could save considerable distillation and dehydration energy [5].

The phase change of water requires 2.26 MJ/kg at 100 °C [6], against only 0.839 MJ/kg for ethanol. Thus, when high wet ethanol is used as fuel, the mean in-cylinder combustion temperature tends to be lower. On the other hand, based on the ideal gas law, it is possible to understand that, for a given temperature and pressure, the specific volume of water in the gaseous state is greater than that of ethanol.

Additional water injection for internal cooling and knock reduction have been proposed during the second world war and were further studied by Melton et al. [7] and Greeves et al. [8], showing potential efficiency increase in old diesel engines. Lately, the direct water injection have been proposed as a possibility to decrease the in-cylinder temperature in highly boosted downsized engines, with possibility of injection during, after or before the power stroke [9]. It has been advised that water injection should be used only for high loads and when exhaust temperature could cause problem to the turbine.

Among the existing simulation models for internal combustion engines, the models known as thermodynamic stand out for their reasonable simplicity and quick response when compared to multidimensional models. Chiodi [10] considers the thermodynamic models most proper for bench test analysis. Among these, two-zone models, with each zone representing the burned and unburned gas, are interesting simulation tools. The success of simulations with this approach depends on the accuracy of the heat transfer models [11].

Accordingly, this work aims to develop a mathematical model to assess the operation and performance of a spark ignited swirl chamber internal combustion engine using different fuel blends: hydrous ethanol fuel (HEF) and mixtures of ethanol and water in the proportions of 90% (E90W10) 80% (E80W20), 70% (E70W30) and 60% (E60W40) by volume of ethanol. The MatLab® software was used in the modeling process due to its integrated modeling, simulation and optimization interface. Thus, the development of an accessible model, with an open code, operating in an widespread programming software, such as MatLab®, becomes a useful tool to be used in future studies.

2. Description of the model

For the simulations, a two-zone model was adopted with a burned and an unburned zone, each spatially homogeneous and separated by an interface of infinitesimal thickness, surrounded by a control volume comprising the interior of a cylinder in an engine operating on Otto cycle. The governing equations of the model include the energy conservation equation, the mass conservation equation, the equation of ideal gas and sub models comprising combustion, heat transfer, variation of the cylinder volume, and composition of air, fuel, and residual gas mixture. The equations describing these models are grouped into several publications and works that deal with internal combustion engines [11–14].

The model simulates the interval of the cycle where the intake and exhaust valves are closed, so it is assumed that the initial condition is one in which the compression cycle begins (PMI, $\theta = -180°$) and the final one in which the expansion cycle ends (PMS, $\theta = +180°$). All in-cylinder gas is considered as a homogeneous mixture of ideal gases. The air–fuel mixture is assumed to be completely vaporized at the time of intake, even under high levels of ethanol hydration. This implies that the heat loss caused by the possible effects of vaporization of the water inside the cylinder shall not be computed as an explicit term in the heat transfer model. On the other hand, the effect of fuel hydration is assumed to focus solely on the heat exchange coefficient of each of the zones. The model does not consider any other effects due to fuel hydration.

Based on the first law of thermodynamics, an open system internal energy is equal to the difference between heat and work generated by the system:

$$\Delta U = Q - W \tag{1}$$

The purpose of the model is to determine the change in thermodynamic state during the cycle as a function of the crank angle, θ. Thus, taking the derivative of Eq. (1), yields:

$$m\frac{du}{d\theta} + u\frac{dm}{d\theta} = \frac{dQ}{d\theta} - P\frac{dV}{d\theta} + \frac{d(\dot{m}_l\dot{h}_l)}{d\theta} \tag{2}$$

where $\dot{m}_l$ and $\dot{h}_l$ are the mass and enthalpy flows in the system, u is the internal energy, Q the transferred heat, P is pressure, and V is volume. The model assumes an engine in which the valves are closed, the mass variation in the cylinder due only to losses from leaks in the rings (blow-by), thus:

$$\frac{d(\dot{m}_l\dot{h}_l)}{d\theta} = -\frac{\dot{m}_b\dot{h}_b}{\omega} \tag{3}$$

where ω is the rotation speed of the engine. Substituting Eq. (3) in (2) we obtain the general form of the energy equation for a control volume that accommodates the contents of the cylinder:

$$m\frac{du}{d\theta} + u\frac{dm}{d\theta} = \frac{dQ}{d\theta} - P\frac{dV}{d\theta} - \frac{\dot{m}_b\dot{h}_b}{\omega} \tag{4}$$

2.1. Thermodynamic properties

Eq. (4) describes how the properties of interest will vary with the crank angle, but not as they vary as a function of temperature and pressure. This dependence is expressed by the following relations:

$$u = f(T,P) \rightarrow u = U/m = xu_b + (1-x)u_u \tag{5}$$

$$v = f(T,P) \rightarrow v = V/m = xv_b + (1-x)v_u \tag{6}$$

$$h = f(T,P) \rightarrow \left(\frac{\partial h}{\partial T}\right)_p = C_p \tag{7}$$

$$s_u = f(T,P) \tag{8}$$

Subscripts b and u represent the burned and unburned zones, respectively, and x represent the fraction of the blend between them. Entropy is measured only in the unburned gas zone, as this is considered as an open system losing mass due to the combustion process. Thus, for the burned zone:

$$\frac{dv_b}{d\theta} = \frac{v_b}{T_b}\frac{\partial \ln v_b}{\partial \ln T_b}\frac{dT_b}{d\theta} + \frac{v_b}{P}\frac{\partial \ln v_b}{\partial \ln P}\frac{dP}{d\theta} \tag{9}$$

$$\frac{du_b}{d\theta} = \left(C_{pb} - \frac{Pv_b}{T_b}\frac{\partial \ln v_b}{\partial \ln T_b}\right)\frac{dT_b}{d\theta} - v_b\left(\frac{\partial \ln v_b}{\partial \ln T_b} + \frac{\partial \ln v_b}{\partial \ln P}\right)\frac{dP}{d\theta} \tag{10}$$

And for the unburned zone:

$$\frac{dv_u}{d\theta} = \frac{v_u}{T_u}\frac{\partial \ln v_u}{\partial \ln T_u}\frac{dT_u}{d\theta} + \frac{v_u}{P}\frac{\partial \ln v_u}{\partial \ln P}\frac{dP}{d\theta} \tag{11}$$

$$\frac{du_u}{d\theta} = \left(C_{pu} - \frac{Pv_u}{T_u}\frac{\partial \ln v_u}{\partial \ln T_u}\right)\frac{dT_u}{d\theta} - v_u\left(\frac{\partial \ln v_u}{\partial \ln T_u} + \frac{\partial \ln v_u}{\partial \ln P}\right)\frac{dP}{d\theta} \tag{12}$$

$$\frac{ds_u}{d\theta} = \frac{C_{pu}}{T_u}\frac{dT_u}{d\theta} - \frac{v_u}{T_u}\frac{\partial \ln v_u}{\partial \ln T_u}\frac{dP}{d\theta} \tag{13}$$