

# Combustion time of the oxygenated and non-oxygenated fuels in an Otto cycle engine

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**Abstract** Speed flame propagation in Otto cycle engines is one of the principal characteristics of fuel and is fundamental in defining the ignition advance. The greater the propagation speed the less the negative work required to compress the mixture before the piston reaches the top dead center and the higher the cycle's efficiency. This paper presents experimental results of time measurements of the fuel's ignition and the maximum pressure rating in the combustion chamber of a Cooperative Fuel Research engine specially instrumented. The combustion duration measurements of oxygenated and non-oxygenated fuels were taken as a function of the compression ratio (8:1, 10:1 and 12:1) and lambda ( $\lambda$ ). The speed flame propagation in the combustion chamber is significantly changed with the change of the lambda different compression ratios. The VNG has a maximum in the speed flame propagation in the stoichiometric region ( $\lambda = 1.0$ ) in all compression rates in this study. Similar behavior occurs with ethanol and gasohol, but only in compression ratio 12:1. Ethanol and gasohol have the higher rate of flame propagation for all

compression ratios measured as compared to the non-oxygenated (isooctane) and oxygenated fuels (MTBE and TAEE).

**Keywords** Fuels · Speed flame propagation · Otto cycle engine

## List of symbols

AKI	Antiknock Index ( $[(MON + RON)/2]$ )
Hz	Hertz frequency unit is derived from the International System (IS) for frequency, which is expressed in terms of cycles per second
Kpa	$10^3$ Pa (Pascal) unidade padrão Multiple of the standard unit of pressure and tension of the International System (IS)
$\lambda$	lambda expresses a relationship between the mass of air and fuel admitted to in a given instant by the engine, compared to the masses of air and fuel chemically balanced

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## 1 Introduction

In conventional spark-ignition engines, the fuel and air pass through intake manifold into the cylinder's, where they are mixed with the residual gas and then compressed. Under normal operating conditions, combustion is triggered by an electric discharge during compression. The flame front propagates through the air–fuel mixture until it reaches the walls of the combustion chamber, where it is extinguished. At the onset of combustion caused by the spark from the spark-plug, the speed flame propagation is small and is accompanied by a slight increase in pressure (delay time) due to combustion. The flame front increases progressively and propagates through the combustion chamber until the

pressure inside the cylinder reaches its maximum after the top dead center (TDC).

The combustion process is divided into four distinct phases: ignition, development, propagation and end of flame. The maximum combustion pressure must occur at a point (15° after TDC) that provides the maximum power or maximum torque. Combustion starts immediately after spark ignition, but, because the initial stages are quite slow, the ignition delay period is used to describe the first stage. Second Ishii et al. [1] it is difficult precisely determine the end of combustion because of burning late phenomenon [2]. For these reasons, ignition delay and end of combustion are usually defined arbitrarily. Ignition delay can, for example, be considered the time period in which the ratio of burned gas reaches a value of 5 [3] or 10 % [4]; [5] and end of combustion as the time period in which this ratio reaches the value of 85 [4, 6] or 90 % [3]. If the onset of combustion advances progressively before the TDC, the negative work of compression increases. If ignition is delayed, the pressure peak occurs later and its value is reduced. The appropriate time for the onset of the combustion process depends on the speed flame propagation and on the geometric configuration of the combustion chamber. These depend on the engine design, the operating conditions and the properties of the fuel–air mixture [7–9, 4].

Complete combustion is described as that in which the flame front moves through the combustion chamber until the air–fuel mixture is completely consumed and transformed into carbon dioxide and water. During the combustion process, carbon monoxide, unburned hydrocarbons and nitrogen oxide are present due to kinetic effects. Factors such as the fuel's composition, air–fuel ratio, geometric details, temperature, pressure, compression ratio and deposits in the chamber determine the physical and chemical characteristics of the combustion process [10–13]. The reactions that occur during the combustion process are determined by serial and parallel mechanisms whose stages initiate in response to the formation of radicals and reactions due to equilibrium displacement. In many cases, due to the high release of energy and the resulting increase in temperature, the speed of the reaction increases exponentially, leading to possible detonation or knocking [4]. Knocking depends on the temperature, the system's pressure, and the fuel's physicochemical characteristics [9]. At very low pressures, the system is outside the knock region and the mixture reacts mildly [4, 14–16].

In engines with optimal design, the turbulence inside the combustion chamber is proportional to the engine's revolution and increases the speed flame propagation [4, 17]. High revolution requires a great advance in the ignition, because the spark time for a given mass of fuel and air inside the cylinder is set to give the maximum brake torque

for an operating condition [4]. The engine's geometric parameters affect the turbulence and surface area of the flame front. Aerodynamic details of the intake system contribute to the creation of an oriented gas flow, changing the flame's surface area and path. The shape of the combustion chamber also affects other criteria such as volumetric efficiency, knocking trend and pollutant emissions. Combustion speed measurements in a laminar system using pure hydrocarbons indicated that, for the same air–fuel ratio, the speed flame varies as a function of the fuel's chemical structure [9, 18, 19].

Due to the constant increase in crude oil-derived liquid fuel prices and the growing restrictions with respect to environmental contamination, interest has focused increasingly on alternative fuels [20–27]. These fuels can be classified as synthetic gasoline, gasoline with oxygenated compound additives such as methyl *tert*-butyl ether (MTBE), *tert*-amyl-ethyl-ether (TAEE) [28, 29]. Studies of the speed flame propagation of new oxygenated fuels such as TAEE are important to determine the best ignition advance to reach the maximum brake torque. This paper analyzes the variation in burning speed of a Brazilian commercial gasoline (gasohol), isooctane, MTBE, TAEE, ethanol and vehicular natural gas (VNG) at different compression ratios and air–fuel ratios. Compression ratios used in this study were 8:1, 10:1, and 12:1 for the liquid fuels and 14:1 for VNG. The air–fuel ratios used for each compression ratio were 0.8 and 0.9 (rich mixture), 1.0 (stoichiometric), and 1.1 and 1.2 (lean conditions).

## 2 Materials and methods

Six different fuels were used in this study: a Brazilian commercial gasoline (gasohol), isooctane 99 %, MTBE 99 %, TAEE 99 %, ethanol 99.3 % and VNG. The VNG contained 89 % methane, 7 % ethane and 2 % propane. Table 1 lists the physicochemical characteristics of the liquid fuels used here.

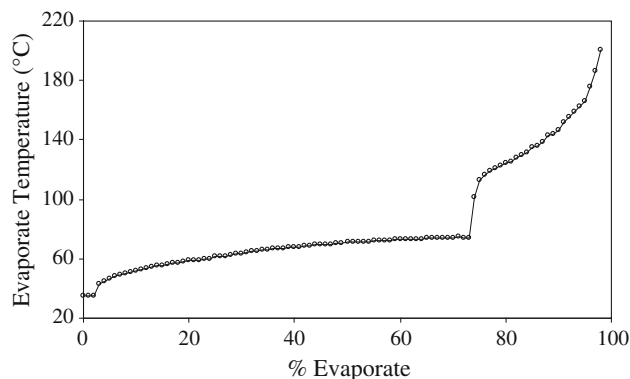
According to table, ethanol shows a higher antiknock index (AKI) than the ramified ethers (TAEE and MTBE) and isooctane. The antiknocking strength is an extremely important property of fuel for Otto cycle engines, since it should not auto ignition by compression, but by sparking. Combustibles having a high octane index produce a milder and more effective combustion. The octane number is simply a numerical description of its resistance to auto ignition without entering into a knocking process (spontaneous burning of the mixture). The average of the Motor Octane Number (MON) (ASTM D 2700-97) [30] and Research Octane Number (RON) (ASTM D 2699-97) [31] is known as the Antiknock,  $AKI = [(MON + RON)/2]$ .

**Table 1** Physicochemical characteristics of the liquid fuels used in this study {Fonte: [10, 29]}

Physicochemical characteristics	Gasohol	Isooctane	Ethanol	TAAE*	MTBE
Reid vapor pressure (kPa)	69	11	16	13	54
Distillation range (°C)	35–201	99	78	102	55
Antiknock index (AKI)	87	100	115	108	109
Density (g cm <sup>-3</sup> ) 20/20 °C	0.740	0.692	0.792	0.767	0.742

Volatility can be understood as the ease with which fuel evaporates. This property is extremely important since the combustion process takes in a gaseous environment, i.e., inside the engine, where the fuel must first vaporize so that combustion can occur. The volatility is determined based on the properties of Reid vapor pressure and distillation curve.

Fuels with high vapor pressure present high rate emission of volatile organic compounds (VOCs). However, there is a minimum required for starting at low temperatures. Fuels with low vapor pressures and high enthalpy of vaporization have trouble in completing vaporization in the intake manifold, making the combustion process and reducing engine performance. The higher the enthalpy of vaporization the smaller is the resulting temperature in the intake manifold, increasing the volumetric efficiency of the plunger. There is currently a strong trend in reducing the volatility of parameters, among these the vapor pressure in gasoline [32–35]. The behavior of the vehicle with respect to the volatility of gasoline at low temperatures can cause incomplete fuel vaporization and in an uncontrolled air/fuel ratio. Considerable progress has occurred in this area, such as replacing the carburetor injection system multipoint [9]. The volatility parameters that have an effect on performance at low temperature have been determined by testing vehicles at high speeds. At high temperatures, the most common phenomena affecting fuel systems in vehicles are mainly steam “lock” and percolation. Vapor “lock” is the occurrence of a stock vaporized fuel from the fuel tank and the carburetor or fuel injection system. Percolation is the result of an uncontrolled vaporization of fuel, which can happen when the vehicle stops after a long period at high operating temperatures. The enthalpy of vaporization has a strong influence on engine power because the higher the value, the greater the amount of heat removed from the intake manifold and thereby lower the temperature of the mixture allowed. This lowering of the temperature in the combustion chamber increases the density of the air/fuel mixture and with it, the mass of fuel to be oxidized is increased, favoring an increase of engine power. In general, the vapor pressure of a volatile liquid is the pressure

**Fig. 1** Distillation profile of gasohol

exerted by the vapors of the same confined space (the force per unit area exerted on the liquid surface). For pure hydrocarbon, steam pressure depends on the temperature and enthalpy of vaporization. For gasoline, which are mixtures of hydrocarbons, depends on the relative concentrations and the interaction of the constituents of the mixture [35].

Figure 1 shows the profile gasohol. The distillation curve gives an idea of the volatility of gasoline throughout the range of distillation. The distillation curve can, in simple terms, be represented by three points: T10, T50 and T90, which represent the temperatures at which 10, 50 and 90 % vaporization of de gasoline’s initial volume occurs (ASTM D-86) [36–37]. These temperatures characterize the volatility of the fuel’s light, medium and heavy fractions. These fractions, in turn, affect the engine’s different operating regimes. It should also be remembered that the control of one or more properties of gasoline results in proper combustion in car engines, and emissions of nitrogen oxides (NO<sub>x</sub>), carbon monoxide (CO) and unburned hydrocarbons (HCs) may be reduced. The ideal fuel for reducing contaminants must have a Reid vapor pressure not greater than 51.68 kPa, essentially zero olefins in its composition and temperature for the 50 % distillation point (ASTM D-86) [36] greater than 82 °C but <96.1 °C. The profile abnormal distillation gasohol is due to azeotrope formed by ethanol and hydrocarbons, wherein the

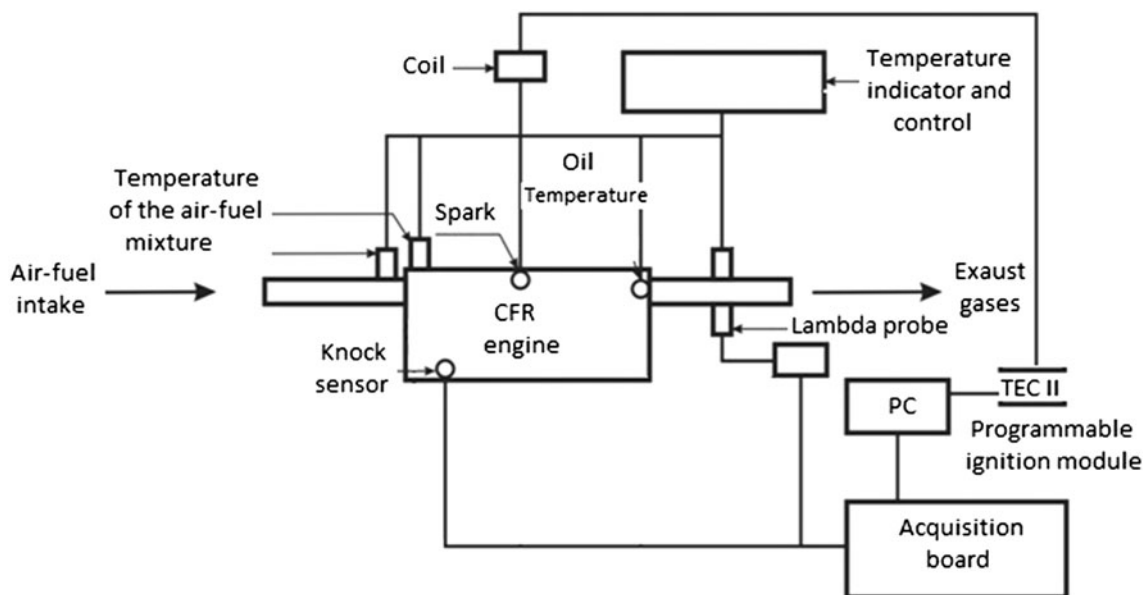
composition volumetric gasohol consisted of 13 % normal paraffins, 23 % isoparaffins, 13 % naphthenes, 11 % olefins, 9 % aromatics and ethanol 25 %.

When considering the flow of the fuel through a carburetor jet, the coefficient of discharge for normal hydrocarbon fuels is virtually constant above a critical value of the Reynolds number. Under these conditions, the mass flow is a function of the density of the fuel so that increases mass flow and lowers the air–fuel ratio of the mixture. This is partly offset by the lower level in the float chamber that occurs with heavier fuels. There is such compensating factor with injection vehicles where the mass of fuel injected will depend directly on the density of the fuel. Variations in density between commercial fuels will therefore influence the air/fuel ratio unless there is an engine management system to hold it at the optimum level. Most gasolines have a density between about 0.72 and 0.78. The density will depend on the types of components used in blending the gasoline, aromatics compounds have the highest density, with the olefins being intermediate and the paraffins having the lowest density, when compounds having the same number of carbon atoms are compared. Of the pure compounds used in this study, ethanol has the highest density value. The tests to measure the speed flame propagation were performed with a CFR engine (a Cooperative Fuel Research engine (year 1951) manufactured by the Waukesha Engine Division) modified with an Otto cycle cylinder head being specially instrumented with sensors and temperature indicators, D-1 Knock sensor, Lambda probe, acquisition board, as shown in Fig. 2 a schematic diagram of the system with the instrumentation used in the tests. The VNG was stored in a high-pressure

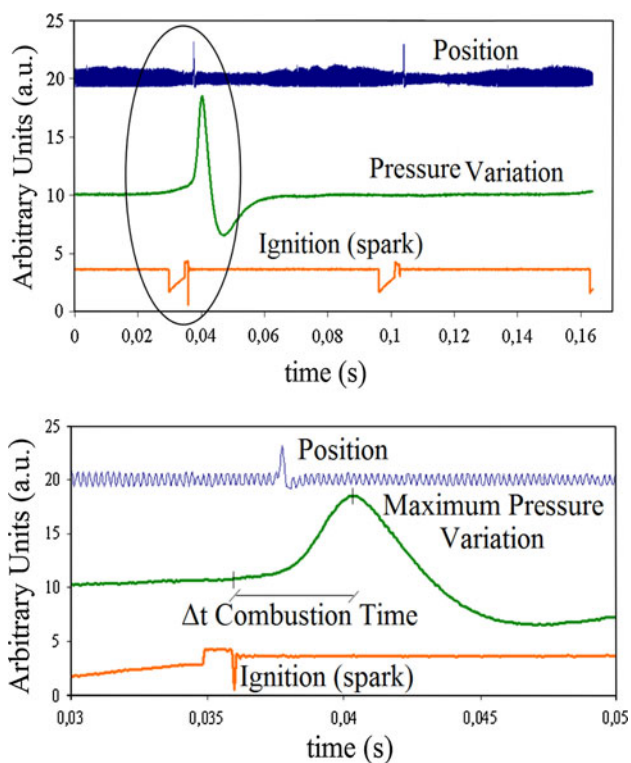
cylinder equipped with a pressure control valve and the dosing was done with a mixer mounted on the intake manifold of the CFR engine. The engine's original system, which was used for the other fuels, consisted of a triple barrel carburetor with adjustable height for adjustment of the air–fuel ratio.

The speed flame was obtained using a D-1 knock sensor (a standard component of the CFR engine). The D-1 sensor has a stainless steel diaphragm, which induces a voltage in the sensor through the action of pressure. The induced voltage signal is proportional to the pressure rate in the combustion chamber. In this work, speed flame propagation was defined as the angle between the spark ignition and the maximum pressure rate inside the combustion chamber divided by time. According Ishii et al. [1], the angle considered in flame propagation includes the ignition delay [4] and the maximum pressure rate. The ignition advance was set at  $10^\circ$  to compare the speed flame propagation of the fuels under study. The ignition was determined from the signal voltage of the electronic ignition control system. The piston's position was determined by a 600 mm diameter toothed wheel with 304 straight teeth and one slanted in the position relative to the top dead center with the aid of an inductive sensor.

The parameter ( $\lambda$ ) lambda expresses a relationship between the mass of air and fuel admitted to in a given instant by the engine, compared to the masses of air and fuel chemically balanced. It is called “rich mixture” ( $\lambda < 1$ ) when there is a lack of oxygen for the complete oxidation of all the reagent available in the combustion chamber and is called “lean” ( $\lambda > 1$ ) when there is excess oxygen to the oxidation of all the available reagent into the



**Fig. 2** Schematic diagram of the instrumentation system used in the tests



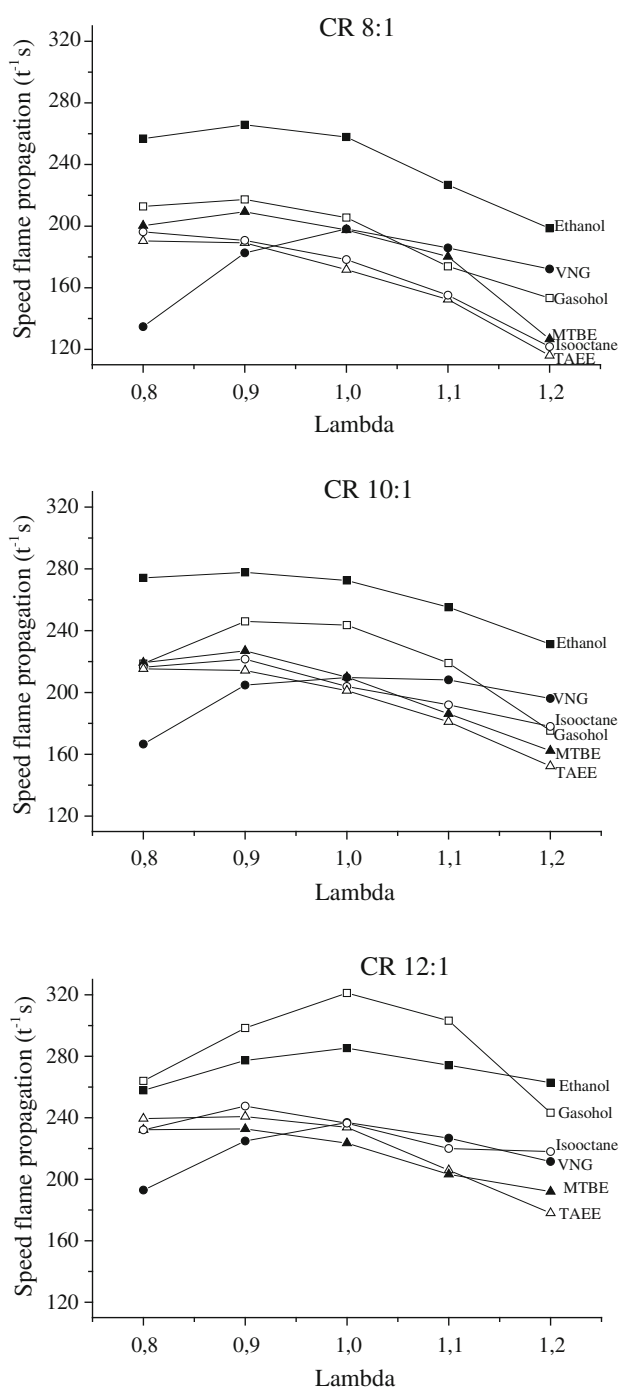
**Fig. 3** Characteristic profiles of the signals from the position, pressure variation and ignition sensors as a function of acquisition time

combustion chamber. The air–fuel ratio was adjusted using a Bosch model LSU4 broadband lambda probe with a MoTec PLM converter. The recorded pressure rate, position and ignition sensor signals were obtained using a CIO EXP GP and CIO DAS 16/330 data acquisition system at a sampling rate of 55 kHz.

The burning time in the combustion chamber was defined as the interval between ignition and the maximum pressure variation inside the combustion chamber. The dimension of combustion duration was expressed as  $s^{-1}$ , and the tests were performed at a fixed revolution of 600 rpm (10 Hz). The “burn time” value of each fuel under the different conditions analyzed here is the mean value of 45 consecutive cycles after complete stabilization of the operational conditions. After all the operational conditions in the tests were stabilized, a comparison was made of the results. Figure 3 depicts the characteristic profiles of the pressure rate, position and ignition sensors recorded for the engine test, using gasohol as fuel and a compression ratio of 8:1.

### 3 Results and discussion

The combustion speed flame is an important factor to adjust the best ignition advance. The cylinder’s dimension



**Fig. 4** Speed measurement flame propagation tests under turbulent conditions for fuel this study as a function of the air–fuel ratio ( $\lambda$ ) and compression ratio 8:1, 10:1 and 12:1

and the compression ratio can also be adjusted to achieve maximum torque as a function of revolution and air–fuel ratio.

The results presented in Table 1 indicate that the oxygenated compounds showed the highest densities and

antiknock index. The oxygenated compound TAEЕ presented the lowest vapor pressure.

Figure 4 shows the results of the speed flame propagation of fuels used in this study in function of the lambda in different compression ratios (8:1, 10:1 and 12:1).

It is observed that the speed flame propagation in the combustion chamber is significantly changed with the change of the lambda to different compression rates. The VNG has a maximum in speed flame propagation in the region stoichiometric ( $\lambda = 1.0$ ) in all compression rates in this study. Similar behavior occurs with ethanol and gasohol, but only in compression ratio 12:1.

Evaluating rich region ( $\lambda = 0.8$  and  $0.9$ ) shows that the fuel reached the lowest flame propagation rate is VNG, while the lean region ( $\lambda = 1.1$  and  $1.2$ ) the TAEЕ begins to show a slower flame propagation in all compression ratios studied. MTBE presents a behavior similar to TAEЕ to the lean region can be explained of the oxygenated fuel TAEЕ, whose molecular structure is similar to that of MTBE but with a higher molecular weight.

Ethanol and gasohol have the higher flame propagation rate for all compression ratios measured as compared to the non-oxygenated (Isooctane) and oxygenated fuels (MTBE and TAEЕ). Ethanol has a vapor pressure of about 16 kPa, according to Table 1, however, when mixed with gasoline, form azeotropes with the hydrocarbons of minimum, which have a boiling point lower than that of two separate components. This effect results in an increase of the vapor pressure of the mixture.

Gasohol is a complex mixture of hydrocarbons and ethanol and it is impossible to attribute the increase or decrease to any specific compound. However, it is known that linear chain compounds and olefins have a higher combustion speed than branched and saturated compounds. However, oxygenated compounds such as MTBE and TAEЕ presented lower speed flame propagation than gasoline, although ethanol showed values close to that of Brazilian commercial fuel, whose composition contains about 25 % v/v of ethanol.

Higher burning speed requires smaller advancements. Fuels with low flame speed need a higher ignition advance to maximize the torque. For VNG and gasohol, when used compression ratio 12, the highest combustion speed occurs in the stoichiometric mixture condition, while the burning speed of oxygenated fuels is higher when the mixture is rich ( $\lambda = 0.8$ – $0.9$ ) and diminishes when the mixture is lean ( $\lambda = 1.1$ – $1.2$ ), showing the profile for all compression ratio used in this study. For air–fuel ratios above the stoichiometric ratio (lean mixture), the combustion process becomes unstable, leading to failures in the combustion cycle.

It should be noted that, as the compression ratio increases, so does the speed flame propagation, due to the

higher diffusion of mass and energy when density is high [38–39].

Note that among the rich mixtures ( $\lambda = 0.8$ ), fuels such as ethers (MTBE and TAEЕ), gasohol and isooctane present similar speed flame propagation. Among the oxygenated compounds, ethanol presents the highest speed flame propagation and among the hydrocarbons, VNG shows the lowest flame speed with rich mixtures, which increases with lean mixtures. Methane has high activation energy in the oxidation process, which may explain the lower speed flame propagation observed at air–fuel ratios below the stoichiometric condition when the combustion temperature is low. The best ignition advance for VNG should be higher than that of gasohol and oxygenated fuels to achieve maximum torque. Ethanol has higher speed flame propagation, while the high flame speed of gasohol can be attributed of the ethanol in its formulation. As mentioned by Lawes et al. [40], the speed flame propagation of ethanol can be explained by the formation of intermediate species such as acetaldehyde and formaldehyde, whose high reactivity favors the oxidation process [41–42].

The speed flame propagation of all the fuels analyzed, except for VNG, here increases when the air–fuel ratio is rich and close to the stoichiometric ratio. This behavior can be explained by the fact that the combustion temperature increases when it approaches the stoichiometric ratio. When the air–fuel ratio is above the stoichiometric ratio (lean mixture), the speed flame propagation diminishes in response to the decrease in combustion temperature caused by excess air.

#### 4 Conclusions

The proper determination of the compression ratio and ignition advance for the optimal performance of engines is directly related with the type of fuel employed. Fuels with branched-chain and ether compounds show low speed flame propagation. The speed flame propagation of ethanol is higher than that of branched-chain and ether compounds. The use of compression ratio 12:1 increases the combustion speed of isooctane, VNG, TAEЕ, ethanol and gasohol as compared with compression ratios 8:1 and 10:1. The higher the compression ratio the greater is the speed flame propagation. To achieve high thermodynamic and mechanical performance, internal combustion engines with spark ignition should operate with high compression rates, and to minimize the negative work of compression, the ignition advance should be reduced to obtain the maximum brake torque. The combustion speed diminishes at air–fuel ratios above the stoichiometric ratio. The VNG has a maximum in the speed flame propagation in the

stoichiometric region ( $\lambda = 1.0$ ) in all compression rates in this study. Similar behavior occurs with ethanol and gasohol, but only in compression ratio 12:1. Ethanol and gasohol have the higher rate of flame propagation for all compression ratios measured as compared to the non-oxygenated fuels (isooctane) and oxygenated (MTBE, and TAAE) fuels with high octane ratings can be used in engines with high compression ratios.

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