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The Effect of Ethanol on Combustion Characteristics of Spray Guided Direct Injection Spark Ignition Engine

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Abstract

The objective of this study was to investigate experimentally the effect of adding ethanol at different proportions (0 to 85%) on the combustion behaviour of a SI direct injection engine. Although the effect of ethanol on combustion behaviour was studied by several researchers, the variation in the results among those researches illustrated a need for a better understanding of ethanol effects.

Combustion behaviour has a significant effect on engine's emission and performance. There is an agreement among researchers that faster burn rate is a favourable characteristic. Shorter burn duration produce more robust and repeatable combustion pattern.

In addition to changing fuel content, experimental data was obtained for different running conditions including different speeds, loads, spark timing, equivalence ratio and exhaust gas recirculation. The Rassweiler and Withrow method was used to estimate the burn duration in the engine. Furthermore, the effect of increasing ethanol concentration on combustion stability and EGR tolerance was evaluated. Results show that combustion duration decreases significantly at high ethanol content (85% by volume). However, there is not a linear relationship between increasing ethanol and combustion duration, comparable durations were obtained for gasoline and low ethanol contents. This advantage of decrease burn duration manifests itself through increase in tolerance to dilution and EGR

Keywords Internal Combustion Engine, Ethanol, Alcohol Fuel, Alternative Fuel, Combustion Characteristics.

تأثير استخدام كحول الإيثانول مخلوطاً بالبنزين بنسب مختلفة في محرك احتراق داخلي على خصائص الاحتراق داخل المحرك

ملخص

يهدف البحث إلى دراسة تأثير الكحول الإثيلي مختلطاً بالبنزين بنسب متفاوتة على الاحتراق داخل محرك البنزين. وعملية الاحتراق لها تأثير كبير على العوادم التي تخرج من السيارات وأداء المحرك بشكل عام. تأثير الإيثانول على المحركات دُرس بواسطة كثير من الباحثين ولكن تضارب النتائج بين الباحثين كان الدافع الرئيسي لهذه الدراسة. فمعدل الاحتراق وطول مدة الاحتراق هما العوامل الرئيسية في فهم الاحتراق داخل المحرك. وهناك اتفاق بين الباحثين على أن كلما قلت الفترة الزمنية للاحتراق كلما كان احتراقاً كاملاً. وإن تقليل المدة الزمنية للاحتراق سيكون له تأثير إيجابي في تقليل نسبة استهلاك الوقود خصوصاً في السرع والعزوم المتوسطة والقليلة.

بالإضافة إلى تغيير نسبة الإيثانول بالنسبة إلى البنزين سنقوم بتغيير عدة عوامل داخل المحرك مثل السرعة والعزم ووقت الاحتراق ونسبة العوادم المرجعة إلى الاسطوانة. ستستخدم طريقة رازويلر وويثرو لتقييم مدة الاحتراق داخل المحرك. بالإضافة إلى السابق تم دراسة تأثير الإيثانول على ثبات الاحتراق والنسبة القصوى المسموح بها من العوادم المرجعة إلى المحرك. وأوضحت النتائج على أن استخدام نسبة عالية في الوقود أدى إلى نقصان ملحوظ في المدة الزمنية للاحتراق. لكن لا يوجد علاقة طردية بين زيادة نسبة الإيثانول ومدة الاحتراق (مدة الاحتراق متقاربة بين استخدام البنزين واستخدام الإيثانول بنسبة صغيرة). ميزة تقليل مدة الاحتراق ظهرت جلياً في زيادة النسبة المسموحة بها من العوادم المرجعة إلى المحرك.

كلمات مفتاحية: آلات احتراق داخلي، إيثانول، وقود بديل، وقود كحولي، معدل احتراق.

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1. Introduction:

The topics investigated in this paper are related to the use of ethanol mixed with gasoline at different proportions in SI engines. The use of ethanol in SI engines can be traced back to the end of the Nineteenth Century, when Henry Ford designed a car that used ethanol as fuel. Gasoline later gained prominence as fuel refined for SI engines due to the availability and cheap supply of crude oil [3]. In the last few years, however, ethanol has again attracted attention as an automotive fuel. This renewed interest in ethanol and alternative fuels in general is driven by several factors including the rise in fuel prices, the increase in demand
for fuel the increase in demand for fuel especially in the developing world, the growing awareness that fossil fuel reserves are finite, and finally concerns over rising levels of greenhouse. Biofuels, particularly ethanol, are today the only direct substitute for fossil fuels in transport that are available on a significant scale [3]. Ethanol can be used today in ordinary vehicle engines without major modification (unmodified for low blends or with cheap modifications to accept high blends).

The most obvious difference between gasoline and ethanol is that the latter is a single species that might be viewed as partially oxidized hydrocarbon [4], the former is complex and composed of variable mixtures of hydrocarbons [5]. The presence of oxygen in ethanol, coupled with its lower molecular weight and H/C ratio, will cause substantial differences physiochemical and combustion properties for ethanol compared to gasoline.

Several researchers studied the effect of ethanol on engine combustion behaviour. Malcolm et al. [5] examined the combustion behaviour of blends of gasoline, isooctane and a variety of alcohols under part-load engine operation at 1500 rpm, with port fuel injection. The tested fuels were gasoline, E85 and isooctane, with ethanol content levels at 25% and 85%, as well as a blend with 25% butanol content. The tests were carried out in an optical SI engine and the combustion duration was tested using high-speed crank-angle resolved natural light imaging in conjunction with in-cylinder pressure analysis over batches of 100 ycles. It was found that E85 shows a faster mass fraction burned traces and faster flame radius growth than the rest of the fuel for most test cases, irrespective of the change in spark timing. The same results were also obtained by Yeliana et al.[6], who studied the effects on combustion duration of blending ethanol with gasoline at different proportions (up to 85% ethanol content, in 20% gradual increments). One-dimensional single zone and two zone analyses have been conducted to calculate the mass fraction burned using the cylinder pressure and volume data. In both analyses, E85 showed a decrease in the combustion duration compared to that for all other fuel blends. The decrease was clear at both Flame Development Angle, FDA, and Rapid burn angle, RBA. For the other fuel mixtures, with low and medium ethanol content FDA showed a linear decrease as ethanol ratio increased. RBA on the other hand, show very little difference between the various fuel blends.

The same FDA results were also obtained by Cairns et al. [7]. However, RBA showed comparable results between different fuel blends, including E85.

The decrease in RBA as ethanol introduced in the fuel was also shown by Zhang et al [8] when 10% ethanol-gaoline mixture was use in a direct injection SI engine.

Other researchers (Varde et al. [9], Yoon et al. [10] and Wallner et al. [11]) found different results where ethanol, whether at high or low content levels, exhibited no effects on either FDA or RBA.

Detailed understanding of combustion duration is essential. Modern engines are required to meet the stringent emissions regulations in the world. In addition, Customers' demands high level of refinement, performance and reliability of their vehicles. Meeting regulations and customer expectations leaves little room for unknown effects of fuel quality and it is this area which the author work has focused on. Combustion characteristics influence engine performance in terms of specific power, engine knock behaviour and emissions.

2. Experimental facilities and Fuel blends characteristics:

The experimental studies were carried out on a prototype, four cylinders inline, compression ratio of 11.51:1, 1.6L Spray Guided Direct Injection, SGDI, gasoline engine manufactured by Ford motor company. The injection timing was fixed at 300 degree Before Top Dead Centre, °BTDC (homogeneous charge). Several transducers and thermocouples were installed at different locations inside the engine as seen in Figure 1. In-cylinder pressure is of a particular importance for this study since it is an indicative of the burning duration inside the engine, see following sections for more details. In-cylinder pressure was measured over 100 cycles by the piezoelectric sensors. The piezoelectric sensors are differential sensors and need to be referenced to a known pressure at a given point in order to obtain an absolute pressure. Therefore, in-cylinder pressure at BDC during the intake stroke was sensibly assumed to be equal to the pressure in the intake manifold.

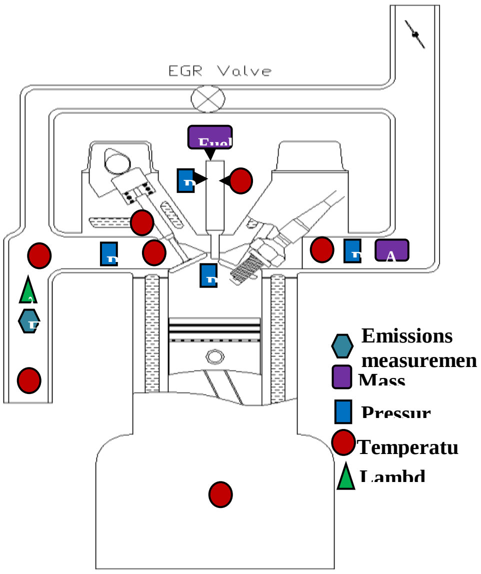


Figure 1 *Schematic diagram of the engine instrumentation*

The comparison between different fuels mixtures were carried out at a constant running condition. That includes constant speed, load, spark timing and equivalence ratio. This is to ensure a direct comparison between the different fuel blends, with change in ethanol content in the fuel as the only variable. Experimental data for each fuel blend was obtained for different running conditions including different speeds, loads, spark timing (ST), equivalence ratio and exhaust gas recirculation (EGR) as shown in Table 1. Those ranges were chosen to investigate not only the effect of the different fuel blends over a wide range of running condition but also the consistency and the repeatability of the results.

Table 1 *Engine running condition*

Variable	Range
Speed	1500-4000 rpm
BMEP	1.57-8 bar
Equivalence ratio, φ	0.8-1.25
EGR	0-20%

Ethanol and gasoline were mixed on a volume basis. The blends were mixed just before carrying out the test in order to avoid any absorption of moisture from atmosphere, which can cause phase separation. Phase separation can occur because ethanol is miscible with water while gasoline is not. Four blends were tested including 10% (E10), 20% (E20), 50% (E50) and 85 % (E85) ethanol ratio. The volume fractions were chosen for several reasons. E10 was of interest as it is already of use in US markets and is being considered for the EU market [3]. E20 and E50 were selected to provide fuels with moderate content which are already used by countries such as Brazil [3]. Finally, E85 has already emerged in some markets, such as USA and Sweden [3], and was required to provide information about the effect fuel with high ethanol content on the engine performance. In addition, a wide range of ethanol ratios were used to aid characterisation of the physical and the chemical properties that might not be linear. Table 2 shows the properties of the different fuel blend used in this study. These properties were calculated by the author based on the properties of pure ethanol and gasoline.

Table 2 *Properties of tested fuel blends*

Properties	E0	E10	E20	E50	E85	E100
Molar mass (kg/kmol)	111.2	107.8	100.9	80.3	56.3	46.1
AFR_{stoich} [kg_{air}/kg_{fuel}]	14.54	13.95	13.08	11.66	9.75	9.00
O_2 contents [wt%]	0.00	3.2	9.9	24.1	34.7	34.7
H/C ratio	1.88	1.95	2.03	2.31	2.76	3.00
Q_{LHV} [MJ/kg]	43.66	42.98	42.21	39.10	32.60	27.74
h_{fg} [24].[kJ/kg]	300	379	457	517	564	948
T_{add} [K]	2438	2431	2424	2400	2364	2345

Despite the fact that ethanol has a lower Q_{LHV} than that of gasoline, its lower AFRs enables an equivalent or even greater amount of energy to be released for a given amount of air, as illustrated in the equations below;

Gasoline :

$$C_{8.26}H_{15.5} + 12.135(O_2 + 3.78N_2) \rightarrow 8.26CO_2 + 7.75H_2O + 45.63N_2 + 4.86MJ \quad (1)$$

Ethanol :

$$4.05C_2H_5OH + 12.135(O_2 + 3.78N_2) \rightarrow 8.05CO_2 + 12.135H_2O + 45.63N_2 + 4.99MJ \quad (2)$$

This will be unfortunately at the expense of Brake Specific Fuel Consumption, BSFC, due to lower heat content.

3. Methodology:

3.1 Combustion Process characterization:

A Mass fraction burned (x_b) profile as a function of crank angle provides a convenient basis for defining various stages of combustion processes by their crank angle duration. x_b can be defined as the percentage of the cylinder charge that has been burned at a certain instant after spark discharge.

At the initial part of the combustion process, immediately after the spark discharge, the air fuel mixture burns at a low rate. The charge burn rate starts to increase until it reaches its maximum about half way through the burning process, and then decreases to zero as the burning process ends. The previous stages of combustion process and energy release can be characterized in two main definitions [1]:

Flame development angle, FDA is the crank angle duration that starts immediately after spark discharge until a small but significant fraction of the cylinder charge has been burned or fuel energy has been released. This fraction is usually 10% [1].

Rapid burn angle, RBA is the crank angle interval where the bulk of the cylinder charge is burned. It starts after the FDA stage and continues until the end of the flame propagation process. Heywood [1] defines the RBA as crank angle interval that covers 10% to 90% of the x_b. The Heywood definition is adopted in this paper. 90% x_b limit was chosen to avoid errors associated with locating the end of combustion since the final stage of combustion is hard to identify [12].

3.2 Rassweiler and Withrow Method:

In this study, the method developed by Rassweiler and Withrow [2] was used to calculate x_b from experimental pressure and volume variation data. This approach is based on two main observations from a constant volume bomb experiment; firstly, it was noticed that the mass fraction burned is approximately equal to the fraction pressure rise.

$$x_b = \frac{m_b}{m_{tot}} \approx \frac{P_b}{P_{tot}} \qquad \textbf{(3)}$$

where m_b is the mass of charge burned at certain instance, m_{tot} is the total charged burnt, P_b refer to pressure due to combustion at a certain instance and P_{tot} is total pressure in the system. Secondly, they observed that for a given amount of energy release, combustion pressure rise is inversely proportional to the volume. In order to apply this equation to SI engine conditions, the change in total pressure, P_{tot} across a discrete crank angle interval is considered to be the sum of pressure changes due to volume, P_V, and combustion, P_C.

$$\Delta P_{tot} = \Delta P_C + \Delta P_V \qquad \textbf{(4)}$$

The pressure rise from change in volume can be calculated at small crank angle intervals assuming polytropic process.

$$\Delta P_V = \Delta P_\theta \left[\left(\frac{V_\theta}{V_{\theta+1}} \right)^n - 1 \right] \qquad \textbf{(5)}$$

In order to compensate for the change in the volume of the combustion chamber compared to a constant volume of the bomb used by Rassweiler and Withrow, the combustion pressure $\Delta P_C'$ has to be related to reference volume, V_{ref}.

$$\Delta P_C' = \Delta P_C \times \frac{V_\theta}{V_{ref}} \qquad \textbf{(6)}$$

V_{ref} was assumed to be equal to the combustion chamber volume at TDC. The x_b at a particular crank angle θ is therefore,

$$x_b = \frac{m_b}{m_{tot}} \approx \frac{\sum_{\theta_0}^{\theta} \Delta P_C'}{\sum_{\theta_0}^{EOC} \Delta P_C'} \qquad \textbf{(7)}$$

where EOC refers to End of Combustion.

The major difficulty with using the Rassweiler and Withrow method is selecting appropriate polytropic index values to calculate P_V. The sensitivity of pressure to the polytropic index increases with increasing in-cylinder pressure. For this reason the sensitivity of the x_b profile to the compression index, n_{comp},is relatively low. The expansion index, n_{exp}, is more important since the pressure reaches its maximum after Top Dead Centre, TDC. x_b profile sensitivity to the change in n_{exp} is shown in Figure 2. Prior to spark ignition, during the compression stroke, the process was assumed to be polytropic that starts from input valve closed, IVC. The polytropic index was calculated from slope of (log P, log V) diagram over 30 degrees before ST as shown in Figure 3. However, during the expansion stroke, the value of n_{exp} varied due to heat transfer, work exchange and turbulent intensity. The use of the correct n_{exp} will keep the burn rate at 100% once the combustion is over until Exhaust Valve Open, EVO. This will satisfy the zero combustion pressure conditions [13]. Several techniques have been developed to calculate n_{exp}. The iterative method is the most common. This starts from a value of n_{exp} = 1.3, changing the value of n_{exp} and EOC location accordingly until a reasonable x_b S-shape profile is reached, as show in Figure 2, EOC, at which n_{exp} was chosen, was determined from the calculated combustion pressure, P_C.

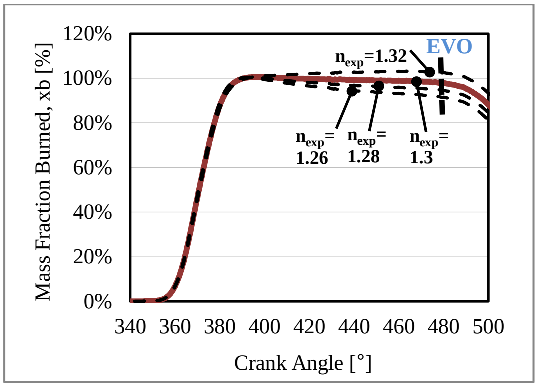


Figure 2 *Changing expansion index and x_b profile.*

In this study, the iterative method was used to determine n_{exp}, with EOC was allocated using 'negligible P_C fraction index'. EOC was determined when P_C becomes a negligible fraction of the total pressure ($P_c \leq 0.02\ P_{tot}$) at three consecutive steps. This method was seen to be more robust since it reduced the influence of signal noise and it was easier to use to define EOC.

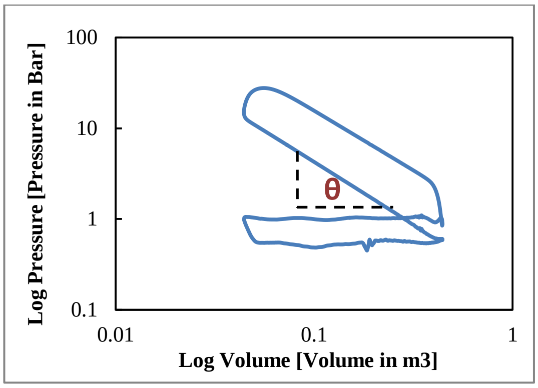


Figure 3 *Log P-V diagram used to calculate the compression index*

3.3 Estimating the error in Rassweiler and Withrow Method:

The main sources of error in Rassweiler and Withrow method could be error in pressure reading, pressure volume phasing, and polytropic index estimation. Polytropic indexes for compression and expansion are linked to pressure data and pressure volume phasing, so any error with compression index is associated with those two factors. Sensitivity analysis for both pressure error and pressure-volume phasing error was carried out. The inaccuracy in pressure reading was assumed to be ±0.098 bar. Figure 4 shows the effect of ±0.098 bar pressure error on the calculated x_b for different speeds, loads and fuel blends. X_b's error is at its maximum at FDA, and then reduces to negligible values at RBA. The results also illustrate that x_b is more sensitive to pressure error at low load than at high load. Finally x_b does not appear to be sensitive to change in Fuel blends. Figure 5 shows the error in burn rate duration as result of changing pressure, the maximum difference at FDA is around ±0.35°, while for RBA the maximum difference was around ±0.25°. An accurate allocation of TDC is hard. In this study, extreme care was taken in allocating TDC for volume pressure phasing. However there ought to be some error in TDC allocation, an error of ±0.25° was found to be a reasonable assumption. The effect of changing volume-pressure phasing on x_b can be shown from Figure 6.

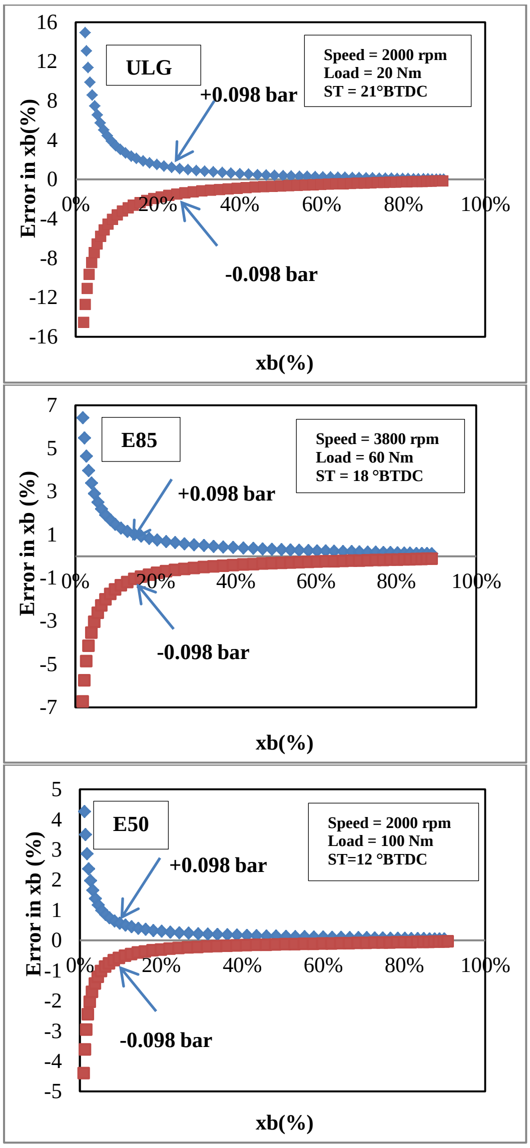
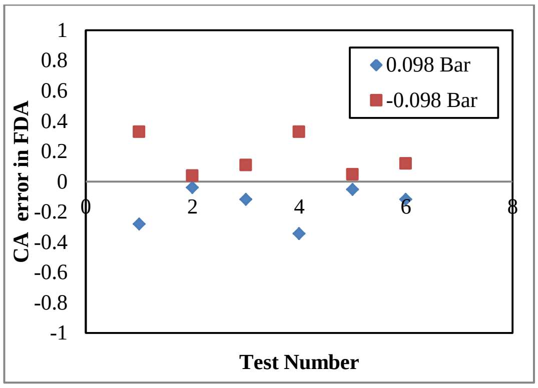


Figure 4 *Effect of changing pressure value by ±0.098 bar on the calculated x_b error for different speeds, loads and fuel blends*



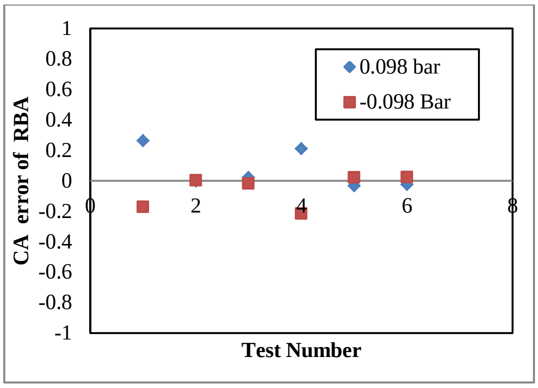
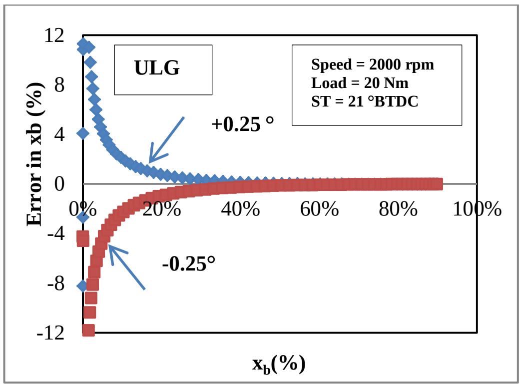
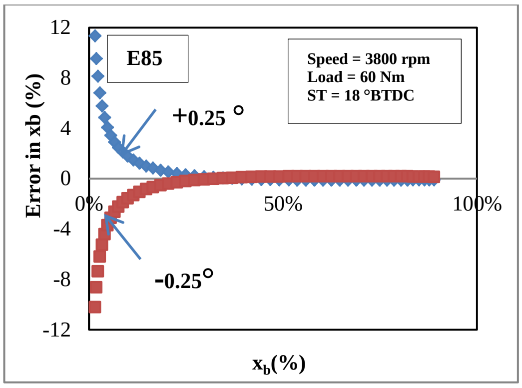


Figure 5 *Percentage difference in RBA and FDA as a result of pressure value by ±0.098 bar*





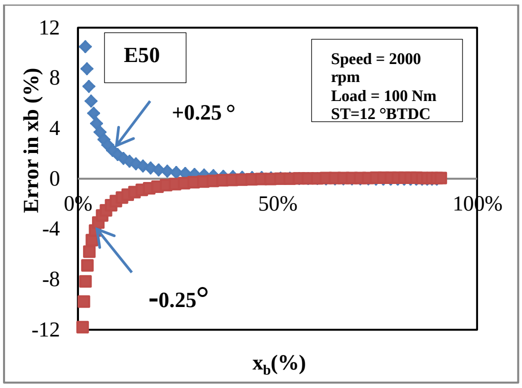
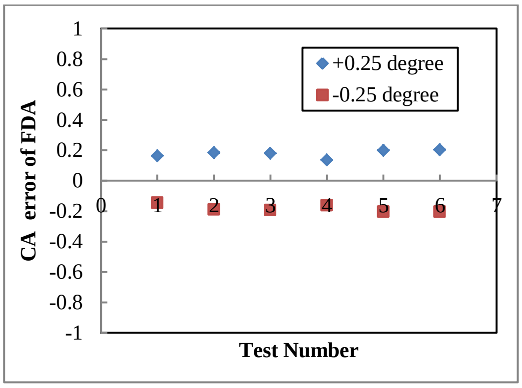


Figure 6 *Effect of changing of volume pressure phasing by ±0.25° on the calculated x_b error for different speeds, loads and fuel blends*



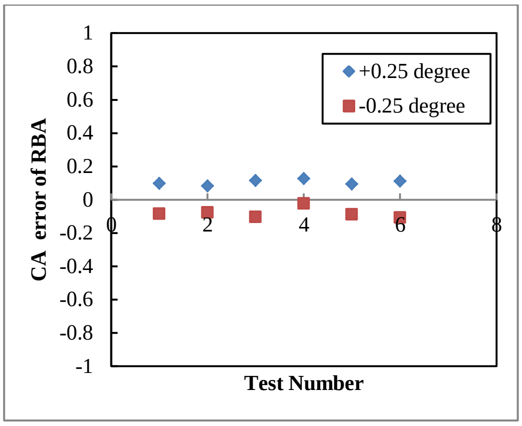


Figure 7 *Percentage difference in RBA and FDA as a result of changing of volume pressure phasing by ±0.25°*

The error was not affected by load, speed or fuel content. It is also illustrated that FDA is more sensitive to any change in pressure-volume phasing than RBA. Figure 7 illustrates that the maximum error for FDA is ±0.25°, and for RBA is around ±0.15°.

In conclusion, Rassweiller and Withrow appears to be a robust method to calculate combustion duration and it is not very sensitive to pressure error (due to thermal shock or pressure referencing error) or to pressure-volume phasing. In the worst case scenario the error occurs simultaneously, then the total error can be evaluated by adding the two source of error. Hence, the maximum error is 0.6° for FDA and 0.4° for RBA.

4. Laminar flame speed of ethanol and gasoline:

In order to understand the pure combustion behaviour of ethanol and gasoline without the extra factors that might be included in a complex system such as IC engine, a comparison between laminar flame speed of ethanol and gasoline was carried out.

Although burning rate is often expressed in terms of a turbulent burning velocity. Turbulent flames can be treated as an array of laminar flamelets with no turbulence structure residing within them [18]. Therefore, understanding of laminar combustion is important to understand flame turbulent combustion. Laminar flame speed can be defined as the rate of propagation of a flame through a gaseous fuel-and-oxidizer mixture relative to a fixed reference point [1].

Laminar burning velocity, S_L, of gasoline and ethanol has been measured using a spherical combustion bomb by various researchers. The gas motion of the spherical bomb can illustrate the features of the induced motion in an engine. Data at higher pressure and temperature have been fitted to a simple empirical correlation of the form [1].

$$S_L = S_{L,0}\left(\frac{T_u}{T_0}\right)^{\alpha}\left(\frac{P}{P_0}\right)^{\beta} \qquad (8)$$

where T_0= 298 K and P_0 =1 atm are the reference temperature and pressure, and $S_{L,0}$, α and β are constant for given fuel, equivalence ratio and burned gas diluents fraction. For gasoline these constants can be represented by [14]:

$$\alpha = 2.4 - 0.271\varphi^{3.51}$$
$$\beta = -0.357 + 0.14\varphi^{2.77}$$
$$S_{L,0} = B_m + B_{\varphi}(\varphi - \varphi_m)^2$$

where Φ_m= 1.21 is the equivalence ratio at which $S_{L,0}$ is a maximum with value of B_m. For gasoline B_m =30.5 cm/s and B_Φ=-54.9.

For ethanol these constants can be represented by [15, 16]:

$$\alpha = 1.783 - 0.375(\varphi - 1)$$
$$\beta = \begin{cases} -0.17\sqrt{\varphi} & \varphi \ge 1 \\ -0.17/\sqrt{\varphi} & \varphi \le 1 \end{cases}$$
$$S_{L,0} = Z\,W\,.\varphi^{\eta}\exp[-\varepsilon(\varphi - 1.075)^2]$$

where Z=1, W=0.465, η = 0.25 and ε=6.34.

Equation 6 was used to calculate laminar flame speed for ethanol and gasoline, with appropriate constants used for each fuel. Figure 8 shows a comparison between the laminar flame speed of gasoline and ethanol as a function of φ at a different initial pressures and temperatures. For all pressure conditions, the data illustrate clearly that ethanol has higher laminar flame speed than gasoline for most φ values. The maximum difference in laminar flame speed between ethanol and gasoline occurs at stoichiometric conditions. Ethanol seems to be more sensitive to the change in φ. Subsequently, the difference in laminar flame speed starts to decrease as φ moves away from stoichiometric, particularly as it becomes richer. As the charge becomes richer the difference in laminar flame speed between the two fuels decreases significantly up to point where gasoline will have a higher laminar speed than ethanol, at φ=1.2-1.3 depending on the pressure.

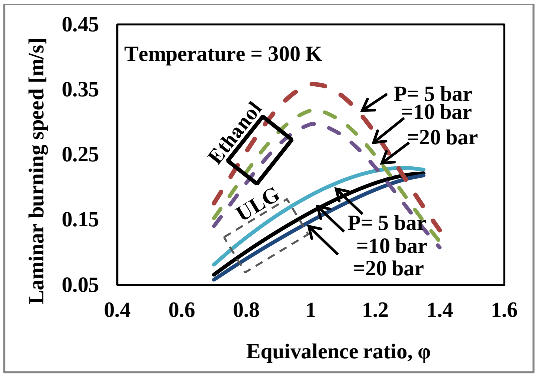


Figure 8 *Laminar flame speed of ethanol and gasoline as a function of equivalence ratio and pressure*

Laminar burning speed is influenced by several factors including molecular structure of the fuel, adiabatic flame temperature (T_{add}), pressure, upstream temperature and EGR [1]. Although T_{add} has a strong influence on laminar burning velocity, and ethanol has a lower T_{add} due its lower Q_{LHV} (see Table 2), the molecular structure of ethanol includes an oxygen molecule that will significantly increase laminar flame speed.

5. Results: Effect of ethanol blends on burning duration:

Several tests were carried out at wide range of running conditions in order to examine the repeatability and the sensitivity of the effect of ethanol on burn duration across those conditions. In addition, the effect of those running conditions on burn duration in an SGDI engine was investigated. The different running conditions are summarized in Table 1.

5.1 Different speeds, loads and spark timing:

Figure 9, Figure 10 and Figure 11 show the effect of ethanol on the FDA and the RBA for various loads, speeds and spark timing, respectively. For all engine running conditions, the results illustrate very little difference in FDA among different fuel blends. RBA results illustrate that there is not a linear relation between increasing ethanol ratio and RBA. Initially E10 showed a slight decrease in the RBA. Then, there was a very small difference or no trend in RBA between E10, E20 and E50. E85, on the other hand, clearly showed a clear faster combustion speed, shorter RBA, compared to all fuel blends and particularly gasoline. The decrease in RBA for E85 compared to gasoline ranged between 2% at low load to 6 % at high load. The lower decrease in RBA at low load compared to high load is attributed to different internal dilution among fuel mixtures. At low BMEP, for fix cam timing and power output, internal dilution increases as ethanol ratio increases as shown in Figure 12.

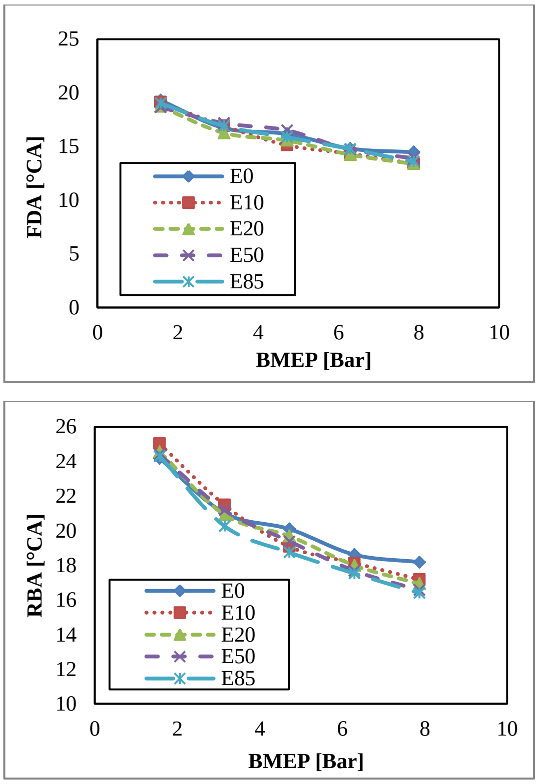


Figure 9 *RBA and FDA for different fuel blends as a function of BMEP. Engine running at 2000 rpm and fixed ST*

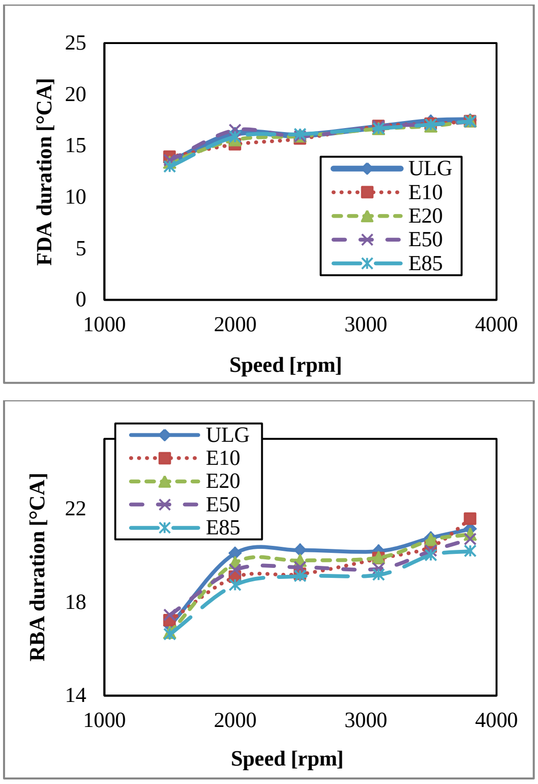


Figure 10 *RBA and FDA for fuel blends. Engine running at BMEP 4.75 bar and fixed ST*

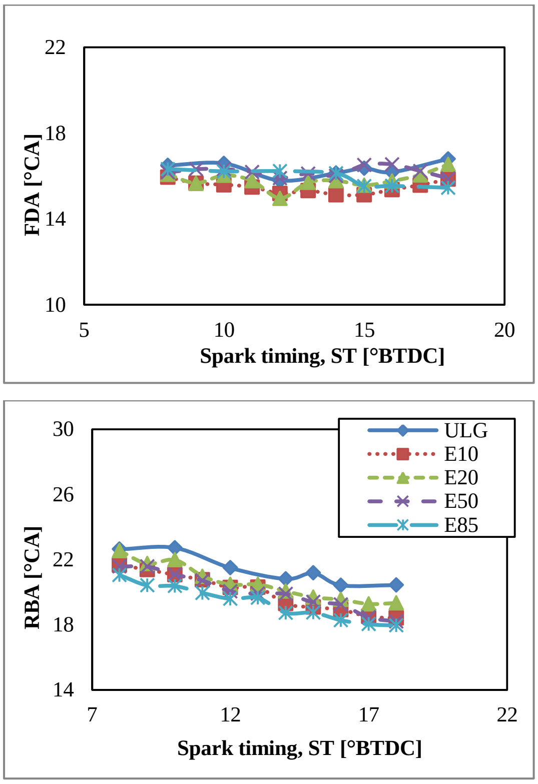


Figure 11 *FDA and RBA for fuel blends as a function of spark timing. Engine running at BMEP 4.75 bar and 2000 rpm*

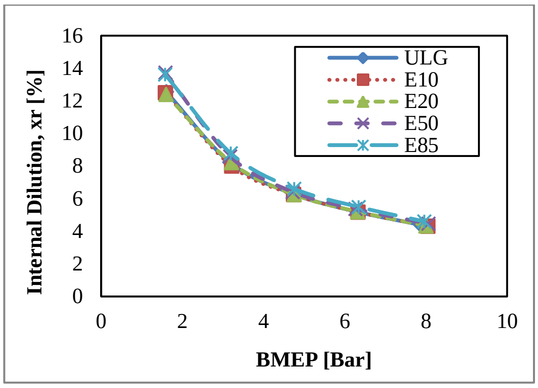


Figure 12 *Internal dilution for fuel mixtures as a function of BMEP*

The observed similarities in FDA value for the different gasoline-ethanol blends were not expected, because of the higher laminar flame speed of ethanol. This however, can probably be explained by the design of the engine under investigation. It is a high compression ratio engine (11.5:1), and consequently properties associated with high compression work, charge density and in-cylinder turbulence dominated the early stages of combustion.

The non-linear relation between increasing ethanol and the RBA is explained by the ethanol properties that influence the combustion. Ethanol has different properties, some of which may be beneficial to combustion while others have the opposite effect. The high laminar burning velocity and oxygen availability will improve combustion and reduce its duration. However, ethanol higher enthalpy of vaporisation and lower Q_{LHV} will decrease gas temperature during compression resulting in slower combustion duration. The combined influence of the two factors will affect the burn duration inside the cylinder. Consequently, the improvement in the laminar speed as a result of adding ethanol to the fuel blend will not have any apparent effect on RBA until high ethanol content.

Changing the running conditions also influences the combustion speed in the same manner for all fuel blends. Increasing load or advance ST decreases the combustion speed. This is due to the increase in pressure and temperature at the time of combustion. Furthermore, increasing load will decrease the internal dilution due to the reduction in the difference between inlet and exhaust manifold pressures. Increasing speed, on the other hand, will decrease slightly the combustion speed. The increase in piston speed will cause an increase in combustion duration in CA domain. However, increasing the speed will increase in-cylinder gas velocity and introduce swirl which will increase the turbulent intensity and subsequently increase combustion duration [1]. For that reason burn duration will only increase slowly with increasing engine speed.

5.2 Sensitivity to change charge composition (x_b & φ):

Changing the charge composition, through factors such as mass fraction burned, x_b and equivalence ratio, φ, will affect burn duration. The burned mass fraction, x_b, is defined as the sum of EGR and internal dilution, x_r. x_b was chosen instead of EGR because x_r changes for different gasoline-ethanol blends at fixed cam positions as shown in Figure 12.

Figure 13 shows the effect of x_b on RBA and FDA. For all fuel blends, both RBA and FDA increase x_b increases. Once again, FDA shows no trend between the different fuel mixtures for all x_b conditions. RBA, on the other hand, appears to be more sensitive to the change in x_b as ethanol ratio increases. At low x_b, there was an obvious reduction in RBA as ethanol ratio increases. However, the difference in RBA between the fuels blends starts to decrease as x_b increases. At high x_b level, the fuel blends show a comparable RBA.

The increase in FDA and RBA as x_b values increase is attributed to the reduction in temperature and pressure during combustion, and thus the laminar flame speed. The effect of x_b on laminar flame speed was studied by Rhodes *et al.* [1], a correlation to calculate the effect of x_b on laminar flame speed was developed as follows,

$$S_L(\tilde{x}_b) = S_L(\tilde{x}_b = 0)(1 - 2.06\tilde{x}_b^{0.77}) \qquad \textbf{(9)}$$

Equation 9 was used to calculate the effect of x_b on laminar flame speed for both ethanol and gasoline as shown in Figure 14. The plotted data demonstrate that the laminar flame speed of ethanol is more sensitive to changes in x_b than it is for gasoline. Subsequently, the difference in laminar flame speed starts to decrease as x_b value increases. This corresponds well with the data shown in Figure 13 and can explain the reduction in the difference in RBA value between the fuel mixtures as x_b increases.

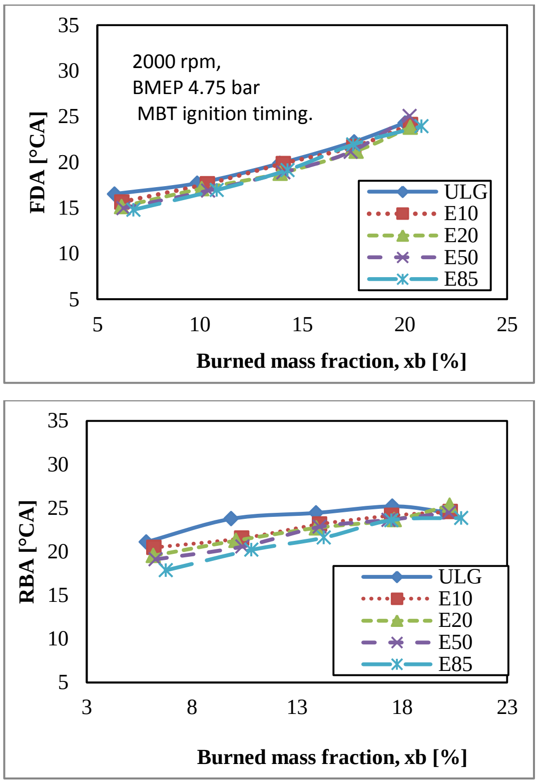


Figure 13 *RBA and FDA for fuel mixtures as a function of burned mass fraction*

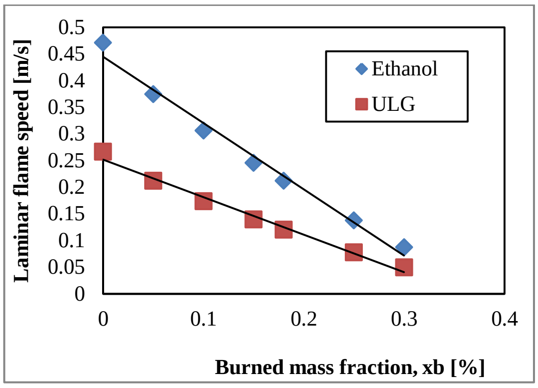


Figure 14 *Effect of x_b on laminar flame speed for gasoline and ethanol*

Equivalence ratio, φ, will also have an effect on burn duration as shown in Figure 15. For all fuel blends, FDA and RBA increases as the charge becomes leaner. The increase becomes more significant after φ=1. Comparing between the different fuel mixtures, at $\varphi = 1$ the burn duration is clearly decreasing as ethanol ratio increases. However, when the mixture becomes leaner or richer, the RBA duration difference between the different fuel mixtures slightly decreases. This corresponds well with the laminar flame speed results shown at Figure 8. The difference between gasoline and ethanol laminar flame speed starts to reduce as the charge moves away from stoichiometric. Other factors such as lower heat content and lower adiabatic flame temperature for ethanol begin to become more dominant especially when the charge is rich.

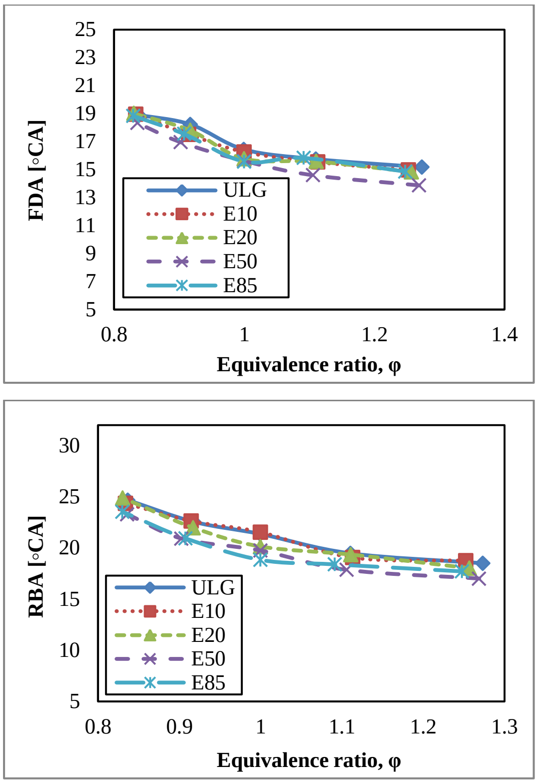


Figure 15 *RBA and FDA as a function of equivalence ratio. The engine running at 2000 rpm, constant BMEP of 4.75 bar and MBT spark timing*

6. Combustion stability and tolerance to x_b:

In order to evaluate x_b tolerance of the different fuel blends, x_b sweeps were carried out at both low and medium load and constant speed. The ST was set to maximum brake torque, MBT, for each running condition. Increasing x_b will decrease combustion speed, which makes stable combustion harder to achieve. The level of x_b that the engine can tolerate will depend on the level of the resulting decrease in combustion speed. The increase in combustion speed associated with high ethanol content in the fuel, as shown the previous sections, illustrates a potential to increase the x_b tolerance. The combustion stability is expressed as the coefficient of variation of indicated mean effective pressure, COV_{IMEP}. Experience has shown that drivability issue (combustion instability) occur when COV_{IMEP} exceed 10% [12]. Figure 16 and Figure 17 show COV_{IMEP} as a function of x_b for the different fuel blends. Both figures illustrate that for low and medium load, running at x_b less than 17% and 14%, for E85 and gasoline respectively, COV_{IMEP} remains unchanged and at a reasonable value (<5%) among the different fuel blends, which indicates excellent cyclic variability. However, as x_b level increases COV_{IMEP} starts to increase significantly and wider distribution of COV_{IMEP} between the different fuels blends starts to appear. While there is no clear trend between E10 to E50, there is a reduction in COV_{IMEP} between gasoline and E85 for high x_b levels.

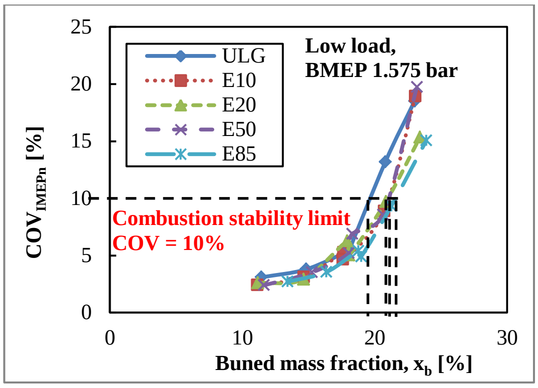


Figure 16 *Combustion stability at low load, 1.575 bar, constant speed 2000 rpm and constant ST*

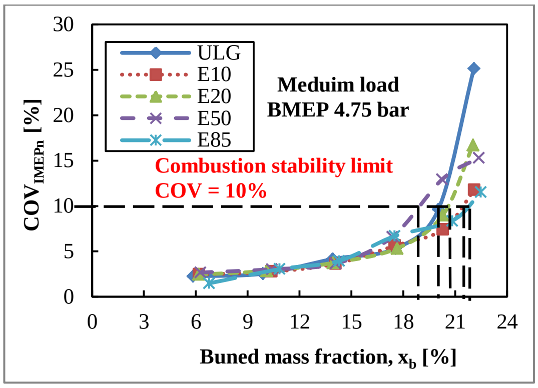


Figure 17 *Combustion stability at medium load, 4.75 bar, speed 2000 rpm and constant ST*

The plotted data was used to obtain the maximum x_b that the engine can tolerate for each fuel mixture, assuming that drivability issues occur at COV>10%, i.e combustion stability limits. Table 3 shows x_b tolerance for each fuel blend. E85 tolerance to x_b has improved.

Table 3 *Maximum EGR allowed for a stable combustion assuming COVIMEPn limit of 10%*

	Low load	Medium load
Gasoline	19.60%	20.01%
E10	21.02%	21.5%
E20	21.2%	20.5%
E50	21.2%	18.99%
E85	21.8%	21.8%

7. Discussion:

Despite the lower adiabatic flame temperature, T_{add}, for ethanol due to its lower heat content, calculated laminar flame speed for ethanol demonstrated a higher value compared to gasoline for most conditions. The increase in laminar burn speed of ethanol is attributed to the availability of oxygen in the ethanol chemical structure. Laminar flame speed for ethanol and gasoline were calculated at a different φ, pressures and temperatures. The peak difference in laminar flame speed between ethanol and gasoline occurs at stoichiometric. Ethanol is more sensitive to the change in φ than gasoline. Subsequently the difference in laminar flame speed decreases as the charge starts to move away from stoichiometric.

The increase in the laminar flame speed was not demonstrated in the FDA results obtained from the engine running at various running conditions. FDA data show comparable results between all ethanol/gasoline blends. This might be explained by the design of the engine under investigation. The engine is high compression engine (11.51:1). The effect of compression work and therefore charge density and temperature at the time of ignition becomes the dominant factor over laminar flame speed.

The RBA results show a nonlinear relation between increasing ethanol content and combustion speed. The fuel blend with highest ethanol content (E85) illustrates an increase in combustion speed compared to other fuel blends, which correspond well with the increase in laminar flame speed for ethanol. Fuel blends with low and medium ethanol content (E10, E20 and E50) showed a slight rise compared to gasoline. However, no significant difference or trend was found in RBA among those fuel blends. The difference in RBA between E85 and gasoline is between 1°C to 2.5°C which is higher than the estimated experimental error (0.4°C). This indicates that the decrease in RBA is due to addition of ethanol rather than any experimental error.

Increasing ethanol content was expected to increase laminar flame speed and hence enhance combustion and reduce duration. On the other hand, increasing ethanol content will also increase in latent heat of vaporisation, h_{fg}, and decrease Q_{LHV} which will have a negative effect on combustion. The combination of those effects will determine the combustion speed of the mixture. For that reason, the advantage of having higher laminar flame speed will not appear until high ethanol content, or E85. Those results were consistent over various engine speeds and loads.

The same effect of increasing ethanol content was also observed with changing in-cylinder composition (through x_b and φ). Once again, E85 RBA results follow similar pattern to laminar flame speed results. Increasing x_b will decrease the difference in burn speed between gasoline and E85. Laminar speed difference between ethanol and gasoline decreases as x_b increases.

Due to the change in combustion duration, increasing ethanol ratio was expected to have an effect on the engine tolerance to x_b. This tolerance is mainly influenced by combustion stability. Fuel with high ethanol ratio, E85, increased tolerance to x_b as a result of the decrease in combustion duration and subsequent increase in combustion stability.

8. Conclusion:

Despite the higher laminar flame speed of ethanol, different gasoline-ethanol blends have comparable FDA values under different running conditions. The compression work, turbulent flow and charge density dominate flame initiation in the high compression ratio engine under investigation (11.5:1).

There is no linear trend between increasing ethanol content and RBA. A small decrease is observed at E10, but no further decreases occur until E85. E85 exhibits a lower RBA compared to all other fuel blends, particularly gasoline.

Ethanol's laminar flame speed is more sensitive to changes in charge composition, such as φ and x_b, than gasoline. As a result, the difference in laminar flame speeds start to be reduced as x_b increases or the charge moves away from AFR_{stoich}. The RBA data show the same trend where the E85 data indicate a reduction in RBA compared to gasoline at AFR_{stoich}. The difference between the two fuels starts to decrease as φ or x_b changes. High ethanol ratios will slightly increase x_b tolerance as a result of shorter combustion duration and subsequently combustion stability inside the engine.

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