

Characterization of Flame Development with Hydrous and Anhydrous Ethanol Fuels in a Spark-Ignition Engine with Direct Injection and Port Injection Systems

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Abstract

This paper presents a study of the combustion mechanism of hydrous and anhydrous ethanol in comparison to *iso*-octane and gasoline fuels in a single-cylinder spark-ignition research engine operated at 1000 rpm with 0.5 bar intake plenum pressure. The engine was equipped with optical access and tests were conducted with both Port Fuel Injection (PFI) and Direct Injection (DI) mixture preparation methods; all tests were conducted at stoichiometric conditions. The results showed that all alcohol fuels, both hydrous and anhydrous, burned faster than *iso*-octane and gasoline for both PFI and DI operation. The rate of combustion and peak cylinder pressure decreased with water content in ethanol for both modes of mixture preparation. Flame growth data were obtained by high-speed chemiluminescence imaging. These showed similar trends to the mass fraction burned curves obtained by in-cylinder heat release analysis for PFI operation; however, the trend with DI was not as consistent as with PFI. OH planar Laser induced fluorescence images were also acquired for identification of the local flame front structure of all tested fuels.

Introduction

Over the last century, internal combustion (IC) engines have been the prime source of power in the transport sector. It is believed that IC engines will be central to transportation for several decades to come due to their low cost, high performance, high reliability and the potential to operate on various fuels. Hydrocarbon fuels as at present still dominate the automobile sector as for a long time it has been relatively cheap to obtain the finished fuel product from crude oil. Liquid hydrocarbon fuels are also attractive due to their high energy content per unit volume and mass. However, IC engines are faced with increasingly intense international obligations to reduce CO₂ emissions [1, 2], a major culprit to the global warming issue. IC engines of motor vehicles account for more than 70% of global carbon monoxide (CO) emissions and 19% of global CO₂ emissions [3]. Fuel supply security is also a top priority amongst countries due to political instabilities in some major crude oil exporting areas. The aforementioned scenarios have precipitated requirements for cars to operate greener [4] and more efficiently through implementation of various mixture preparation strategies in the combustion cylinder and the use of sustainable reduced-carbon fuels, such as alcohols, provided that they do not compete with food chains. Alcohols

have received increased research attention over the past decade due to their ability to replace common hydrocarbon fuel stock in the automotive transportation industry [5]. Various governments around the world have legislations towards increase in alcohol blend in gasoline in the near future. One example is the Directive 2009/28/EC of the European Parliament and of the Council on the promotion of the use of energy from renewable sources. Each Member State needs to ensure that the share of energy from renewable sources in all forms of transport by 2020 is at least 10% of the final consumption of energy in transport in that Member State [6]. Furthermore, the United States Renewable Fuel Standard has a running program to increase the production of ethanol and advanced biofuels to 36 billion gallons by 2022 [5]. Of these alcohols, ethanol is most commonly used as automobile fuel in spark-ignition engines. Its relatively high octane rating and greater heat of evaporation compared to gasoline [7], makes it suitable for advanced IC engines. The high octane value permits an increase in CR [8] with associated thermal conversion efficiency benefits, whilst the high latent heat of evaporation cools down the incoming air and promotes volumetric efficiency. Ethanol also has marginally higher laminar flame speed than typical fuels like *iso*-octane at engine-like conditions [1].

When blended with gasoline, ethanol fuel specifications worldwide traditionally dictate use of anhydrous ethanol (less than 1% water) for gasoline blending [9]. However, hydrous ethanol is the most concentrated grade of ethanol that can be produced by simple distillation, without further dehydration steps to produce anhydrous ethanol [9]. At ethanol purity greater than 95.57% (E95.5/W4.5), hydrous ethanol is an azeotropic mixture [10], hence the production of anhydrous ethanol requires an additional and costly processing step which is less advantageous with regard to Life Cycle Inventory [11]. This processing cost has also been a challenge in fuel price competitiveness and economic gains can be achieved from reduced distillation costs if the final ethanol concentration is below the azeotropic limit [10]. Considering the above economics, the use of hydrous ethanol as automobile fuel may become a preferred option for high percentage ethanol blends, hence future engine technologies designed for ethanol may need to accommodate either form [9].

The increase in water content in ethanol has been shown to cause a reduction in NO_x [9, 12, 13] and this has been

attributed to lower peak temperatures and slower combustion rate [2]. Hydrous ethanol addition to gasoline has also been shown to reduced CO and HC, but increased CO₂ [14, 15]. However Musin et al [13] discovered that addition of water from 20% to 40% by volume to anhydrous ethanol resulted in incomplete combustion which led to increased CO and HC emissions. Limited publications have reported the effect of hydrous ethanol on unregulated emissions such as formaldehyde (CH₂O). This includes a research conducted by Melo et al [15] where they investigated the combustion and emission performance of a an engine working with different hydrous ethanol blends at different operating conditions. Gasoline with 25% vol/vol of anhydrous ethanol (E25) was set as the reference fuel and identified as H0 and 100% vol/vol hydrous ethanol as H100. H0 (E25) was blended by volume with 30%, 50% and 80% of H100. Their results showed a steep increase in both acetaldehyde and formaldehyde as hydrous ethanol percentage in volume increases for all engine operating points. The CH₂O trend reported by these authors shows a sharp difference when compared to those reported by Wallner et al [16] and Broustail et al [17] who used anhydrous ethanol and recorded only marginal or no increase in CH₂O with ethanol increase. Another interesting study was published by Munsin et al [13]. They focused on the effects of the use of hydrous ethanol with high water contents of up to 40% on the performance and emissions of a small SI engine for a power generator. Ethanol blends with water at levels of 20–40% per volume were investigated. The authors observed that increasing the water content resulted in increased CH₂O emissions both before and after the catalytic converter, though it was lower for the latter. This they attributed to incomplete combustion with increased water content.

Present contribution

Previous work on hydrous ethanol combustion in engines has focused mainly on engine performance and engine exhaust out emissions. No publications have presented detailed results on the in-cylinder combustion mechanism of hydrous and anhydrous ethanol fuels and with various mixture preparation methods. The first objective of the current work was to conduct heat release analysis on in-cylinder pressure records from a single-cylinder spark-ignition research engine when fuelled with anhydrous and hydrous ethanol with varying water content. Both PFI and DI mixture preparation methods were employed. The pressure records were obtained simultaneously with high-speed flame chemiluminescence imaging. The flame images were processed to characterize the flame's behavior. Tests were also conducted with gasoline and *iso*-octane fuelling for direct comparison with the hydrous and anhydrous ethanol data. Additionally, Planar Laser Induced Fluorescence (PLIF) images of OH were acquired for comparison of the local flame front shape and structure of all fuels.

Experimental apparatus and procedure

Research engine

The research engine used for the current work was a single-cylinder optical engine designed and built at UCL, with a capacity of about 0.5 l. The engine's geometric properties and specifications have been summarized in Table 1. For

completeness, this table also summarises the operating conditions used for this work, as will be detailed later. In-cylinder optical access was achieved by using a quartz piston crown of 66 mm diameter and a hollow 'Bowditch' piston. Oil smearing of the 45° mirror and piston crown was prevented by using a vacuum pump connected to the crankcase. The engine design also accommodated a pentroof window which allowed side optical access to the combustion chamber. Engine control was achieved by using a shaft encoder, with a resolution of 1800 pulses per revolution, fitted to the engine's camshafts, as well as an AVL 427 Engine Timing Unit (ETU). The encoder also fed a Top Dead Centre (TDC) reference to the ETU. The engine's head and block were heated via an independent water circulation system and heat exchanger. Further details of the engine and ancillary equipment can be found in previous publications [18, 19].

Table 1. Research engine specifications and conditions used.

Engine Head	4-Valve Pentroof
Piston Crown	FLAT
Bore/Stroke [mm]	89/79
Engine Speed [RPM]	1000
Inlet Plenum Pressure [bar]	0.5
Spark Advance	30° CA
Injection Systems	PFI, Single-hole Injector Side DI, Multi-hole Injector
Injection Timing	PFI: TDC Firing DI: 300° CA BTDC Firing

Fuels

Hydrous ethanol with 6% water per volume (E94W6) and 10% water per volume (E90W10), as well as pure ethanol (E100) were used in this research. The hydrous ethanol fuels were prepared by adding distilled water to pure ethanol (99.99% pure chemical grade ethanol). Tests were also conducted with *iso*-octane and gasoline fuels for comparison. The distillation curve of the gasoline used has been presented elsewhere [1]. The distillation curves of the hydrous ethanol fuels were measured. Considering that the boiling point of ethanol at 1 bar is 78.1 °C, the hydrous ethanol blends with low water content exhibited almost vertical distillation curves very close to that single temperature point. E90W10 showed a deviation from this past 90% evaporation; its distillation curve is shown in Fig. 1. Various fuel properties are also presented in Table 2.

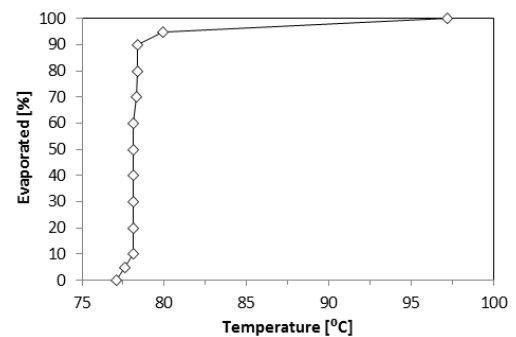


Figure 1. Distillation curve of E90W10.

Table 2. Fuel properties

Fuel	C %w/w	H %w/w	O %w/w	Density (20°C) [g/cm ³]	Kinematic Viscosity (20°C) [cSt]	Surface Tension (20°C) [dyn/cm]	Calorific Value [MJ/kg]	Injection Duration, PFI [ms]	Injection Duration, DI [ms]
<i>iso</i> -Octane	84.1	15.9	0	0.69	0.72	18.30	45	1.37	1.73
Gasoline	86.1	13.9	0	0.72	~0.5–0.6	~20	43	1.41	1.81
E100W0	52.1	13.1	34.7	0.79	1.52	24.05	27	1.93	2.61
E90W10	45.4	12.7	41.5	0.82	2.11	26.10	–	2.08	2.79

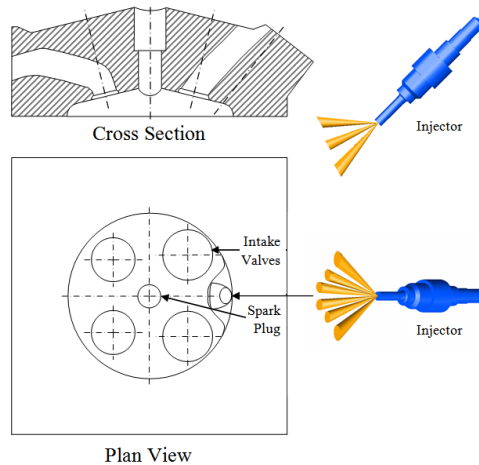
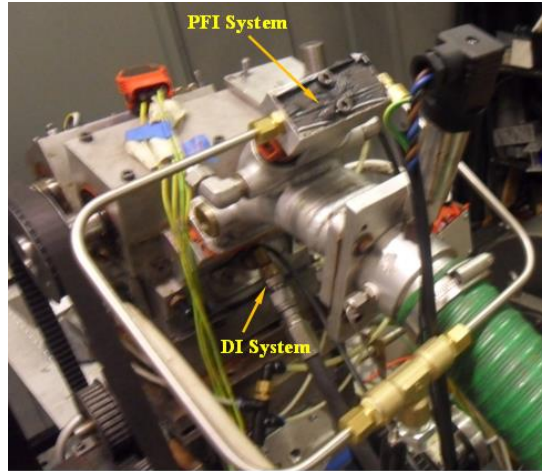


Figure 2. Schematic of combustion chamber with side multi-hole DI and picture of engine head showing the injection systems

Injectors

The optical engine had a fully flexible fuelling system, capable of both PFI and DI operation. For PFI operation two liquid injectors (Bosch Fuel Injector 92TF-AA) were placed directly upstream of the left and right intake port, respectively. The fuel pressure for the PFI injectors was fixed at 4 bar. The DI injector was a six-hole injector with asymmetric pattern, located at 45° inclination below the intake valves and the fuel pressure was fixed at 100 bar. Trigger signals were supplied to the DI and PFI injectors from a Bosch injector driver unit. The fuel supply system comprised of a fuel ram. The Start of injection (SOI) for PFI was fixed at Top Dead Centre (TDC) firing to ensure that the sprayed fuel hit the back of the hot injection valves, i.e. closed valve injection to promote fuel evaporation before air and fuel were drawn into the combustion cylinder of the next intake stroke event. It is noted that throughout this paper TDC refers to firing TDC. For determination of SOI with DI operation, the engine was fired with SOI varied from 320 to 220 Crank Angle degrees (° CA) Before Top Dead Centre (BTDC) with a fixed spark advance of 30° CA BTDC. A SOI of 300 CA BTDC firing (60° CA into the intake stroke) proved optimum with regards to engine combustion stability for 'homogenous DI' operation as a balance between time available for fuel evaporation and degree of fuel impingement on the piston crown. A schematic representation of the DI injector and its location on the engine and a picture of the



engine head showing the injectors, spark coil and inlet manifold pressure sensor are presented in Fig. 2.

Engine operating conditions

All engine tests were conducted at 1000 RPM engine speed with intake plenum pressure set at 0.5 bar by the use of a throttle upstream the plenum chamber. The engine coolant was heated to 85 °C to simulate typical warm running engine conditions. The temperature was allowed to stabilize over a period of 45 min before acquiring any measurements. Only stoichiometric combustion events are presented in the current study. Stoichiometry was monitored by the use of a wide range Air-Fuel Ratio (AFR) oxygen sensor and an ECM AFR Analyser 1200. The humidity of the environment and each fuel's O/C and H/C ratios were entered into the analyser. Ignition sweeps were conducted with *iso*-octane and anhydrous ethanol for both PFI and DI operating conditions with spark advance varied from 50° to 10° CA in 5° CA degrees steps to identify the area of Minimum spark advance for Best Torque (MBT). For the SOI timings used in this work, TDC firing for PFI and 300° CA BTDC firing for DI, ethanol required an MBT of 25° CA and 30° CA for PFI and DI respectively, whilst *iso*-octane needed a spark advance of 30° CA for both PFI and DI. However, in general, the IMEP curves versus spark advance were fairly flat and it was decided to use a fixed spark advance of 30° CA for all fuels for both PFI and

DI. This was to ensure that, nominally, the same flow field conditions existed on average at ignition timing for all tests.

In-cylinder pressure measurements

In-cylinder pressure measurements were conducted with a water-cooled piezoelectric Kistler 6041A pressure transducer. The transducer was connected to a data acquisition system via a charge amplifier and the pressure was determined using the calibrated sensitivities of the transducer and amplifier (pC/bar, mv/pC). The pressure signals were digitized at a rate of 15 KHz by a 12-bit analogue-to-digital converter (National Instrument PCI-MIO-16E-4). The digitized pressure traces and other signals were displayed on a computer monitor screen using a LABVIEW program. The pressure traces were post-processed to calculate heat release using methods published extensively (e.g. see [1]).

Flame imaging and processing

Flame chemiluminescence images were acquired with a Photron APX-RS high-speed camera. Since the engine was run at 1000 RPM, a frame rate of 6 kHz was used with 640×480 pixels image size, corresponding to 1 image/° CA at 1000 RPM engine speed. The camera had on board memory of 2.14 GB which allowed continuous acquisition of 6826 8-bit greyscale images at that pixel resolution. A 60 mm f/2.8 Nikon lens was used on the camera. 60 images were obtained from each cycle right from ignition (spark) timing to 60° CA After Ignition Timing (AIT); 110 cycles were recorded for every test point. Ignition timing is used to denote 'spark timing' throughout this paper. It is noted that the number of cycles was selected by plotting the Coefficient of Variation (COV) of the Indicated Mean Effective Pressure (IMEP) against different numbers of cycle. It was found that this relationship turned asymptotic between 100 to 130 cycles. Considering the limitations of the optical engine and the camera storage capacity (6800 images at the resolution used), 110 cycles (at 60 frames per cycle) were averaged for every test point in the end. The camera was triggered by a TTL signal from the AVL ETU. To obtain the rate of flame growth for all tested fuels, enflamed areas were obtained on a cycle by cycle basis via thresholding and binarisation. The image resolution was ~0.142 mm per pixel. A sensitivity study was carried out using threshold values in the region of 1–3% of the maximum greyscale. It was found that the results were not sensitive to the threshold level past about 10–12° CA AIT and by optimization, in order to capture the flame growth earlier in the cycle than that, an overall threshold of 1.5% was finally selected and used throughout the full set of flame data for PFI and DI. Other details on the image processing methodology have been discussed elsewhere and are not reiterated here for brevity [1].

OH planar Laser induced fluorescence

OH PLIF investigation was conducted by means of a pump and dye Laser assembly. The pump Laser was a Continuum Surelite Nd:YAG Laser matched to its SSP-1 separator box. In this particular application the second harmonic generator doubler unit was installed and produced nominal pump energies of 350 mJ at 532 nm. A set of 532 nm dichroics were employed in the SSP-1 separator unit. The beam was then steered into a Sirah pulsed dye Laser with two 532 nm mirrors.

The resonator and amplifiers of the dye Laser were operated with Rhodamine 6G dissolved in methanol. The dye Laser wavelength of 566 nm was then doubled with a BBO crystal. First and second harmonics were separated by four Pellin-Broca prisms. The output Laser pulse energy obtained near 283 nm was 18 mJ. The Laser beam was later stirred to the optical engine using a periscope with 283 nm mirrors and formed into a Laser sheet of 50 mm width and 0.5 mm thickness using cylindrical lenses.

PLIF imaging was initially conducted on a propane/butane flame using a broad range of wavelength sweep in order to find the rotational line that produced the highest fluorescent yield for OH radical excitation. This fine incremental stepping of wavelength was done via the Sirah Control 2.5 software, a Labview based application which allowed a large array of Laser parameters to be controlled via a serial connection to the PC. The wavelength was adjusted along with the actual grating positions for maximum output at any chosen wavelength. Eventually, excitation at 282.9496 nm produced the best fluorescence signal on the burner and it was then used for the OH PLIF investigation in the engine.

PLIF images were capture by a gated Intensified Charge Couple Device (ICCD) camera from Princeton Instruments (P-MAX III). The camera had a CCD array size of 512×512 pixels, with digitalisation of up to 5 MHz, and a capturing area of 12.14×12.4 mm. Two UV lenses, a Pentax 78 mm f/3.8 UV lens and a Nikon 105 mm f/4.5 UV lens, were used for fluorescence signal collection in this study. Depending on the engine plane investigated and the desired magnification, one of those lenses was mounted on the camera and the light was collected via the UV-enhanced 45° mirror located in the hollow Bowditch piston extension. The optical parts were realised in UV polished fused silica for both piston and side pentroof window. In order to resourcefully isolate the fluorescence signal from broadband chemiluminescence and block scattered light, a combination of Schott UG11 and Schott WG305 filters were placed in front of the UV lens allowing the transmission of about 56% of the incident radiation to the camera between 305–320 nm. The Laser and camera were triggered by a Stanford signal generator DG535 which received a reference input from the AVL ETU. For all PLIF work, the ICCD camera gate width was set at 50 ns. Fig. 3 shows a picture of the PLIF optical arrangement. One image per cycle was captured over a series of 60 consecutive cycles. Images were acquired in the range 10°–30° CA AIT in steps of 5° CA.

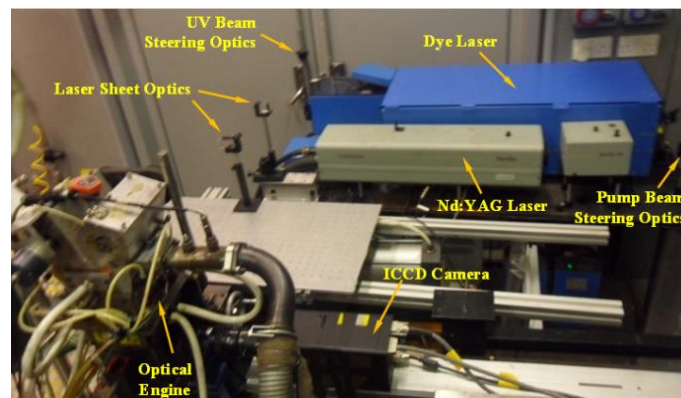


Figure 3. OH PLIF measurement setup

RESULTS

Heat release

The mean in-cylinder pressure traces are presented in Fig. 4 for all fuels tested under both PFI and DI operation. It can be observed that peak in-cylinder pressures are higher for PFI operations compared to DI for all fuels tested. This is in agreement with the trends reported by Daniel et al [20]. These authors reported the effect of PFI and DI on peak cylinder pressure when running an engine on gasoline in homogenous mode. Their tests were performed on a single-cylinder SI engine with a 4-valve cylinder head having spray-guided DI and PFI fuel delivery systems and with engine speed of 1500 RPM. The SOI used for DI operation was 280° CA BTDC with injection pressures of 150 bar and 3 bar for DI and PFI, respectively. They observed that at IMEP values less than 7 bar, PFI operations produced higher cylinder peak pressures, greater efficiencies (higher indicated thermal efficiency and lower fuel consumption) and lower combustion durations. The authors attributed this to a reduction in cooling effect on the intake temperature for PFI as a result of hot manifold walls and valves leading to higher combustion temperature and reduction in combustion duration. The IMEP produced by the current engine at the conditions tested was of the order 2 bar. Analysis of the engine's cylinder pressure traces and polytropic indices for all fuels revealed a pressure difference of less than 0.1 bar for all fuels at ignition timing under PFI operation. Similar analysis for DI showed pressure differences close to 0.2 bar at ignition timing amongst all fuels which corresponded to about 20 K temperature difference. Previous work done by our group at UCL has also shown higher peak pressures for PFI than DI in an optical engine with side DI running at 1500 RPM, 0.5 bar intake pressure using hydrocarbons and anhydrous ethanol blends [21].

In the present work, the least peak pressure was achieved by *iso*-octane for both PFI and DI fuel delivery method followed by gasoline. Ethanol attained the highest peak in-cylinder pressure of all fuels tested with 13.6 bar and 14.4 bar for DI and PFI respectively. This is consistent with the findings reported by Aleiferis et al [1] though they presented results obtained from only a DI fuelled engine similar to that used by Daniel et al [20] with a 6 hole multihole injector and engine speed of 1500 RPM. Peak cylinder pressure can also be seen to reduce with increase in water content in ethanol for both PFI and DI with E90W10 attaining the least of the alcohols for DI and PFI respectively. This trend may be as a result of the superior latent heat of vaporisation for water compared to ethanol hence an increase in water content leads to greater cooling effect on the pre-ignition charge. There are similarities in peak pressures between gasoline and E90W10 with the former attaining values of 11.9 bar and 12.8 bar for DI and PFI respectively. The timings of peak cylinder pressure for all fuels also show that the alcohols attained peak pressures earlier than the hydrocarbons for both DI and PFI, with ethanol the

fastest. This timing was found to be delayed with increase in water content in ethanol for both PFI and DI. However, the change was marginal amongst the alcohols with the largest difference of 1.5° CA existing between E96W4 and E90W10 for DI.

The mass fraction burned (MFB) curves are shown in Fig. 5. For PFI operation, anhydrous ethanol was generally the fastest attaining 50% MFB at 34.4° CA AIT, closely followed by E96W4 which attained 50% MFB at 34.8° CA. Both the former and the latter approximately overlapped till about 27° CA AIT. The difference between both fuels at a much earlier CA AIT is clearer in the flame radius curves presented later in this work. E90W10 was significant slower than the other two ethanol fuels, attaining 50% MFB at 36.4° CA AIT. This trend is generally in agreement with the findings of Schiffer et al [9] who observed that addition of water to ethanol-gasoline blends slowed down the combustion process. Their hydrous ethanol fuel had a fixed water content of 4% per volume and they observed that as the hydrous ethanol content in gasoline was increased, the combustion process was delayed. However, in the present study, E90W10 was found to be significantly faster than gasoline and *iso*-octane. The latter two fuels attained 50% MBF at 38.4° CA AIT and 40.8° CA AIT, respectively.

The MFB curves show similar trends for DI operation in comparison to PFI. However, the disparity between the fastest fuel (anhydrous ethanol) and the slowest (*iso*-octane) is much lower for DI than for PFI. Anhydrous ethanol attained 50% MFB at 37.6° CA AIT while *iso*-octane attained the same value at 42° CA AIT, i.e. exhibiting a difference of 4.4° CA compared to the difference of 6.4° CA recorded for PFI. Also, the overlapping between E100W0 and E96W4 continued until 70% MFB at 40.8° CA AIT. This can be interesting because of the azeotropic nature of ethanol around this concentration and the reduced production cost [13]. It is also clear here that the combustion was faster for PFI than DI for all fuels tested. Differences in pressure and temperature at ignition timing between PFI and DI can be related to this. A higher temperature increases the laminar flame speed, whilst a higher pressure decreases it but the effect of temperature is more prominent (e.g. see [22]). Extended analysis on such effects has been presented in [1] and references therein.

Engine IMEP values and COV of IMEP are presented in Fig. 6. IMEP values were larger under DI operation when compared to PFI for all fuels tested. The IMEP also dropped when the fuel was changed from anhydrous ethanol to E96W4 for PFI operation, but it increased when the same fuel change was applied under DI operation. Furthermore, the COV of IMEP was larger with PFI than with DI for all fuels except for anhydrous ethanol which showed similar values for both mixture preparation methods. The hydrous ethanol generally recorded higher COV in IMEP under PFI.

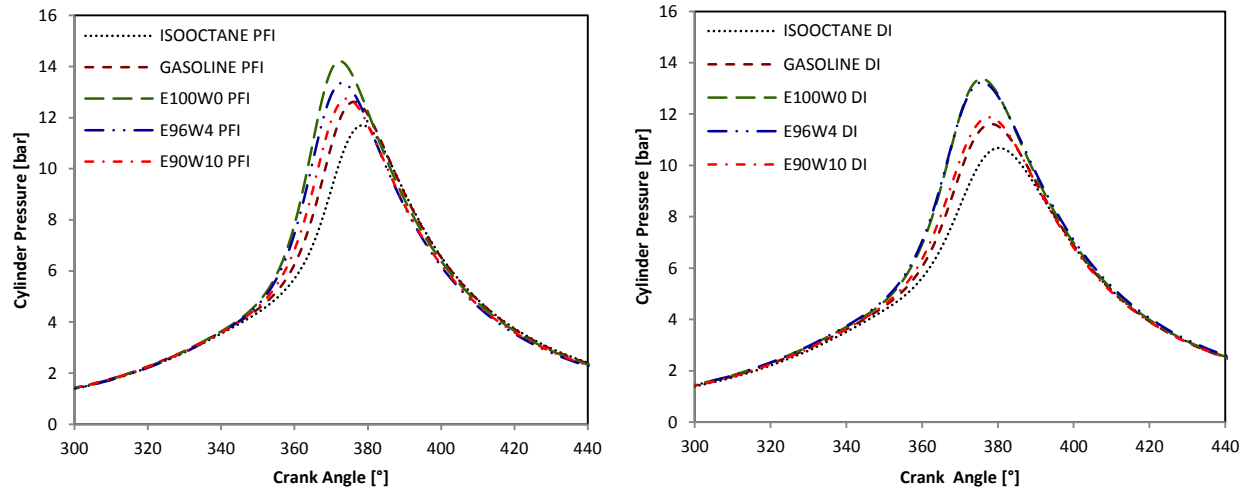


Figure 4. Mean in-cylinder pressure traces for PFI and DI operation

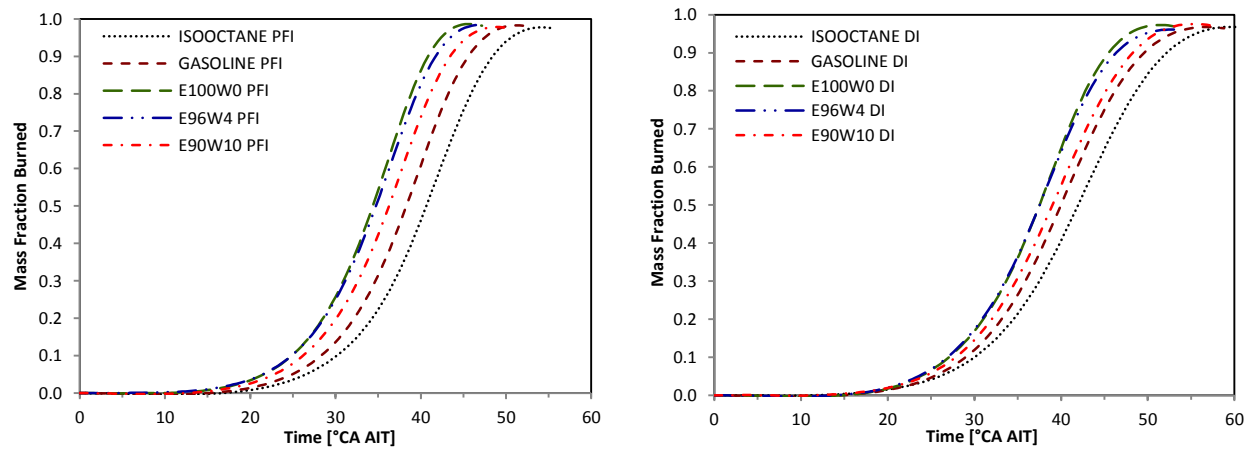


Figure 5. Mass fraction burned for PFI and DI operation

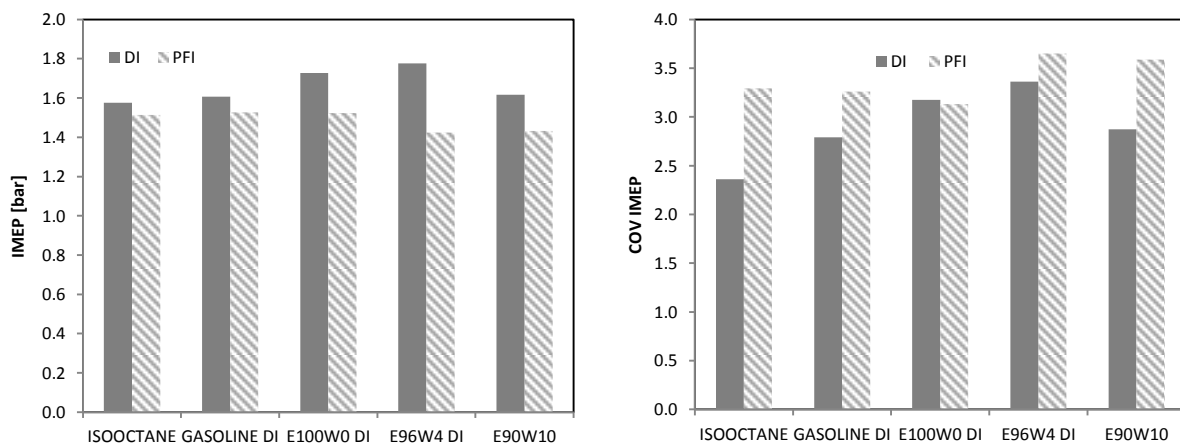


Figure 6. IMEP and COV of IMEP for PFI and DI operation

Characteristics of flame growth

General observations

Figure 7 and Fig. 8 present typical flame images at different crank angle timings following ignition for PFI and DI operation, respectively. For PFI, gasoline flames appeared more luminous compared to the other fuels within the early stage as can be seen particularly from 20° to 28° CA AIT. After this period, the ethanol flames generally showed higher luminosity compared to the hydrocarbons. *Iso*-octane appeared the least luminous of all flames with a more ‘coarse’ texture compared to the rest. *Iso*-octane flame also showed more variation in flame luminosity within the flame’s central region compared to the other fuels. Furthermore, *iso*-octane flames contained a smaller amount of bright spots (which could be diffusion burning of fuel droplets), followed by gasoline. The flame edges were generally more distinguishable from the dark engine background for the hydrocarbon fuels; the ethanol fuels appear cloudier at the edges. The water content was also seen to affect the general flame structure and the flame’s edges. Anhydrous ethanol was less cloudy and had a clearer edge structure during early flame development (20°–32° CA AIT) compared to E96W4 and E90W10. Under DI operation, similar observations to those of PFI were made. However, the flames were generally smaller for all fuels for DI when compared to PFI at the same timings. Also, the bright spots within the flames were significantly more with DI flames but, again, with *iso*-octane exhibiting the least amount of spots, followed by

gasoline. This is in agreement with previous work on hydrocarbon and anhydrous ethanol fuels [21]. The bright spots tended to increase with water content in ethanol.

Flame radius

Equivalent flame radii were calculated at different crank angles following ignition timing. These were based on methods described previously, e.g. [1, 23]. Specifically, the equivalent flame radius was obtained from the square root of the measured enflamed area when divided by π . Results are presented in Fig. 9 (a close-up section is also shown). For PFI operation, anhydrous ethanol was fast from the very early stages and continued as the fastest throughout the imaged combustion process. E96W4 was essentially as fast as anhydrous ethanol very early on but started exhibiting slightly slower growth from about 8–10° CA, yet clearly the second fastest fuel throughout the rest of the imaged combustion process. E90W10 showed clearly faster growth than gasoline and *iso*-octane from about 12° CA AIT. Those flame radius trends are generally in agreement with the MFB curves shown earlier. Anhydrous ethanol and E96W4 were clearly the fastest attaining a radius of 23 mm and 21 mm respectively at 25° CA AIT. They were followed by E90W10 which recorded a flame radius of ~19 mm at the same timing. *Iso*-octane which was the slowest attained a flame radius of 15 mm at 25° CA AIT, whilst gasoline recorded a radius of about 16 mm at the same timing.

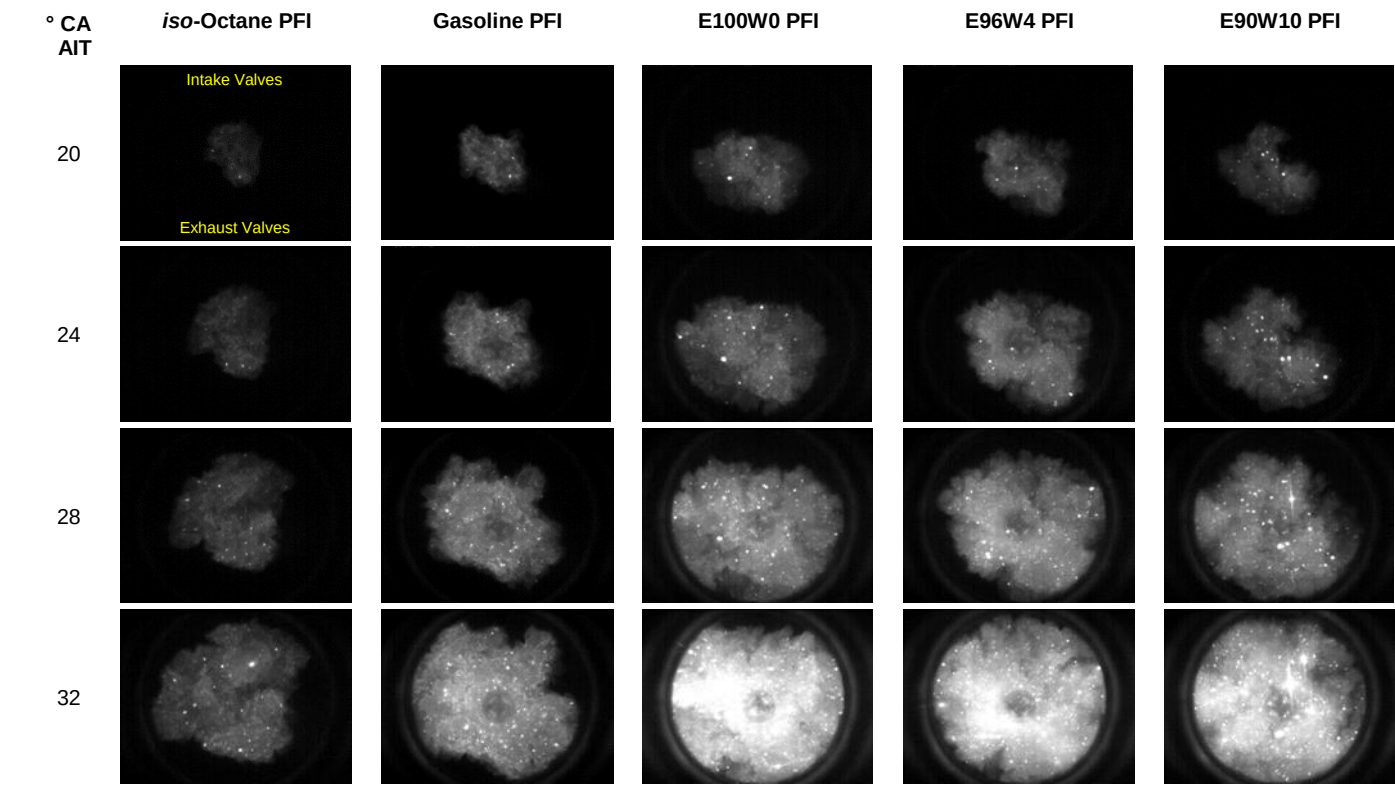


Figure 7. Flame development for PFI operation

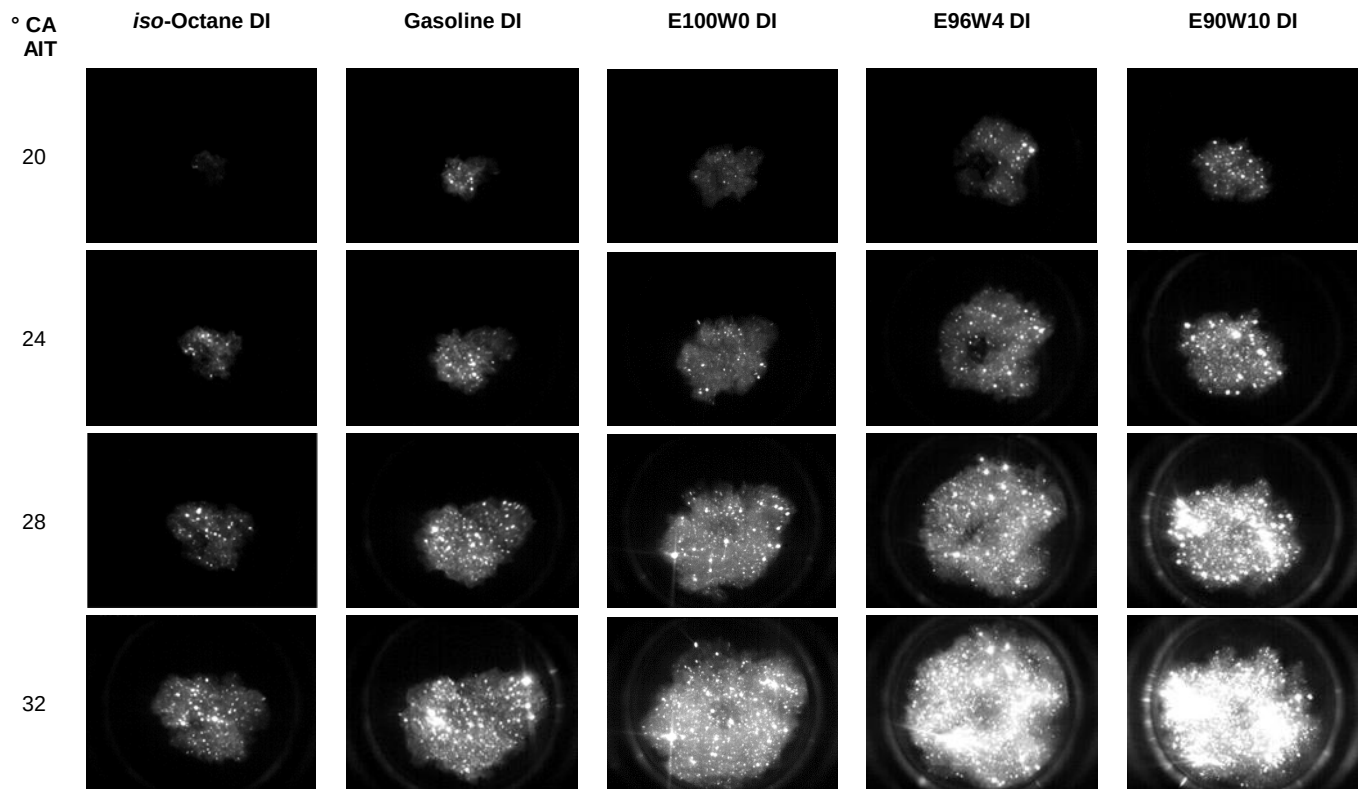


Figure 8. Flame development for DI operation

For DI operation, all fuels had similar flame radii from spark timing until about 8° CA AIT. Gasoline was fastest from the start and up to ~13° CA AIT where it was surpassed by E96W4 and anhydrous ethanol. The flame radii curves generally showed *iso*-octane as the slowest of all fuels, attaining a flame radius of ~13 mm at 25° CA AIT, followed by gasoline which achieved a value of ~15 mm at the same timing. Of the ethanol fuels, E90W10 was the slowest with a flame radius of about 17

mm at 25° CA AIT. Again, E96W4 and anhydrous ethanol were very close just as observed in their MFB curves. Both radii almost overlapped except for the earlier flame development stage. Within this period, E96W4 showed as marginally faster than anhydrous ethanol until about 25° CA AIT. E96W4 and anhydrous ethanol were clearly the fastest for DI operation, attaining a flame radius of about 19 mm at 25° CA AIT. This is in agreement with the trends shown previously for MFB.

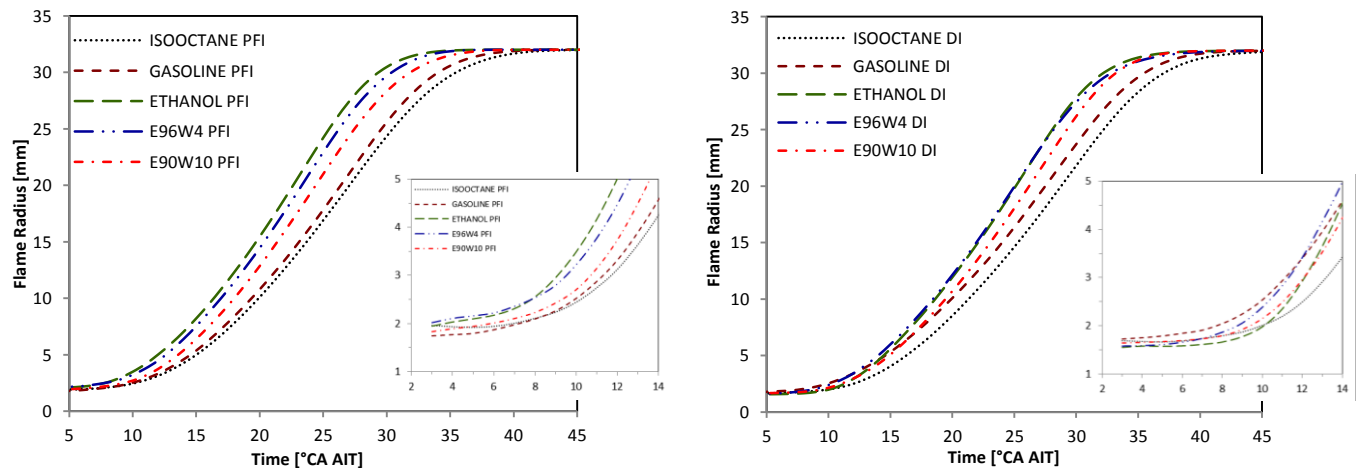


Figure 9. Flame radius evolution for PFI and DI operation

Flame growth speed

The flame growth speed curves are presented in Fig. 10 for PFI and DI; only the first 24° CA of combustion are shown in that figure. This is because from about 22° CA AIT the fastest flames started to get 'clipped' by the boundary of the optical crown, therefore they appeared to decelerate. This was not quantitatively realistic as the in-cylinder flame growth speed continues to increase in practice almost up to the end of combustion event (with a gradient that depends on the timing of piston crown impingement, heat losses, etc.). For PFI, The ethanol fuels accelerated clearly faster than the hydrocarbons, with anhydrous ethanol being the fastest attaining a peak average flame growth speed of about ~11 m/s within 20–22° CA AIT). E96W4 is close to anhydrous ethanol and shows a greater gradient after 10° CA AIT, tending towards the anhydrous ethanol curve. The gasoline curve suggest that on average gasoline flames grew faster than E90W10, largely within the spark overlap period, before surpassed by the latter whose curve accelerated towards the ethanol blends. *iso*-octane flames grew the slowest of all fuels under PFI operation, attaining a maximum flame growth speed of ~9.5 m/s. For DI operation, *iso*-octane again grew the slowest, exhibiting a peak speed of ~9 m/s, followed by gasoline. However, gasoline flames grew faster than all the ethanol fuels from the start to about 8° CA AIT except for E96W4 which

largely overlaps with gasoline within this period. Combustion studies with central DI fuel delivery in [1] have also shown that gasoline flames grew faster than ethanol's within this period. Of the alcohols, anhydrous ethanol was the slowest from start to about 9° CA AIT, before accelerating ahead of all the other fuels and attaining a peak flame growth speed of ~10.5 m/s. E90W10 was the slowest of the alcohols attaining a peak flame growth speed of ~10 m/s. Overall, PFI had higher values of flame growth speed for all fuels compared to DI.

Plots of flame growth speed against flame radius are presented in Fig. 11. The upper limit of the radius scale has been set to 16 mm as past this the flames have been affected by the boundaries of optical access. For PFI, there are two distinguishable groups, the alcohols and hydrocarbons. In both groups, the fuels essentially fall over each other from 5–16 mm flame radius. The alcohols form the fastest group of the two. For DI operation, although the two distinct grouping still persist, anhydrous ethanol appears fastest at the same radius throughout, followed by E96W4 and E90W10. *iso*-octane typically recorded higher values of flame growth speed than gasoline at the same radius. The values of flame growth speed shown for *iso*-octane gasoline and anhydrous ethanol are very similar quantitatively and in trends to what has been shown in a recent publication on a DI engine with centrally mounted multi-hole injector and injection strategy in the intake stroke as used in the current study.

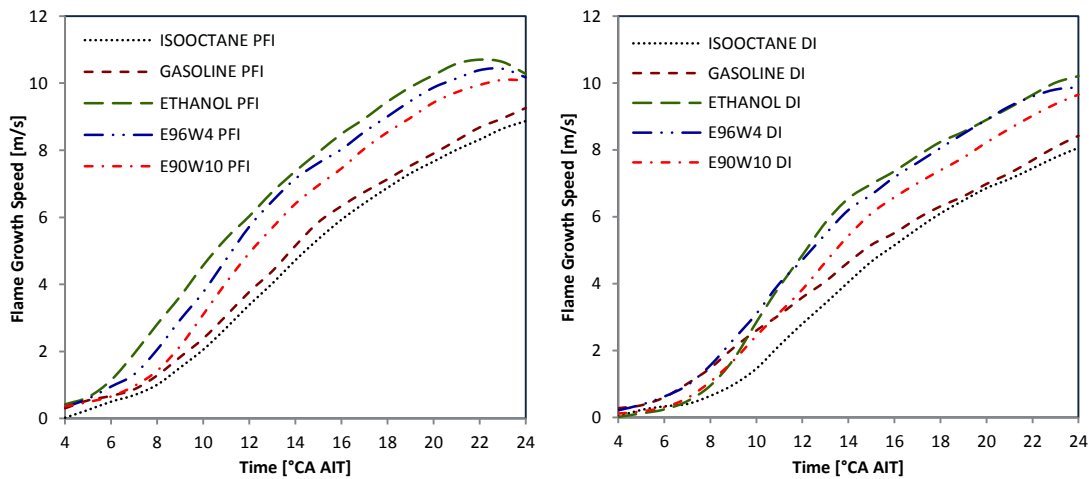


Figure 10. Flame growth speed for PFI and DI operation

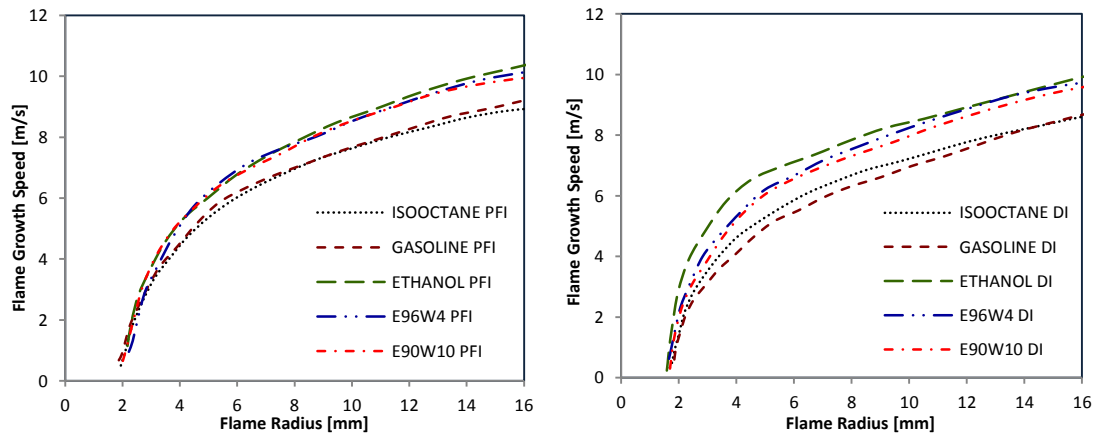


Figure 11. Flame growth speed relative to flame radius for PFI and DI operation

Planar Laser induced fluorescence

Typical OH PLIF images obtained from PFI operation for *iso*-octane and anhydrous ethanol are presented in Fig. 12. The same figure also contains images of flame chemiluminescence taken with the same ICCD camera used for PLIF for direct comparison. The intake valves are at the top and exhaust valves at the bottom, as shown earlier in Fig. 7. The PLIF images were acquired on a horizontal plane 5 mm below the tip of the spark-plug's ground electrode with the 78 mm f/3.8 UV lens. Four sets of images of different cycles are shown here to illustrate the typical degree of cyclic variability. It is noted that the Laser sheet enters the chamber from the right hand side on these images. It is also worth mentioning here that these images are discussed here by qualitative analysis based on observations. A more detailed statistical analysis is currently under study and quantitative comparisons will be presented in a follow up publication. For *iso*-octane, at 10° CA AIT, the part of the flame which has entered the Laser sheet shows greater spatial variations in the OH distribution than at later timings. Some OH images are more convoluted than others, whilst others exhibit separate fragmented entities (*i.e.*

islands that may be connected to the flame on other planes). The PLIF images appear smaller in area when compared to the chemiluminescence flame images as expected [24]. At 10° CA AIT, the early kernels all appear approximately centrally located around the spark plug with the edges showing greater OH signal intensity. At 20° CA AIT, large portions of flame have emerged into the plane of the Laser sheet and are gradually showing a bias towards the exhaust-valve side of the combustion chamber. Though already greatly wrinkled and convoluted at this timing, the OH images generally exhibit a single entity. At 30° CA AIT, the PLIF images appear majorly on the exhaust-valve half of the combustion chamber and are generally brighter on the left hand side. Although the flame chemiluminescence images show greater intensity at 30° CA AIT compared to earlier timings, the PLIF signals generally show less intensity around the spark plug location. This is because most of the flame has already propagated through the Laser sheet, leaving behind a large portion of burnt gases. The OH produced in the flame front is slowly consumed but remains present in the burnt gas region at equilibrium concentrations [25]. Flame fragmentation can be observed to reoccur at 30° CA AIT.

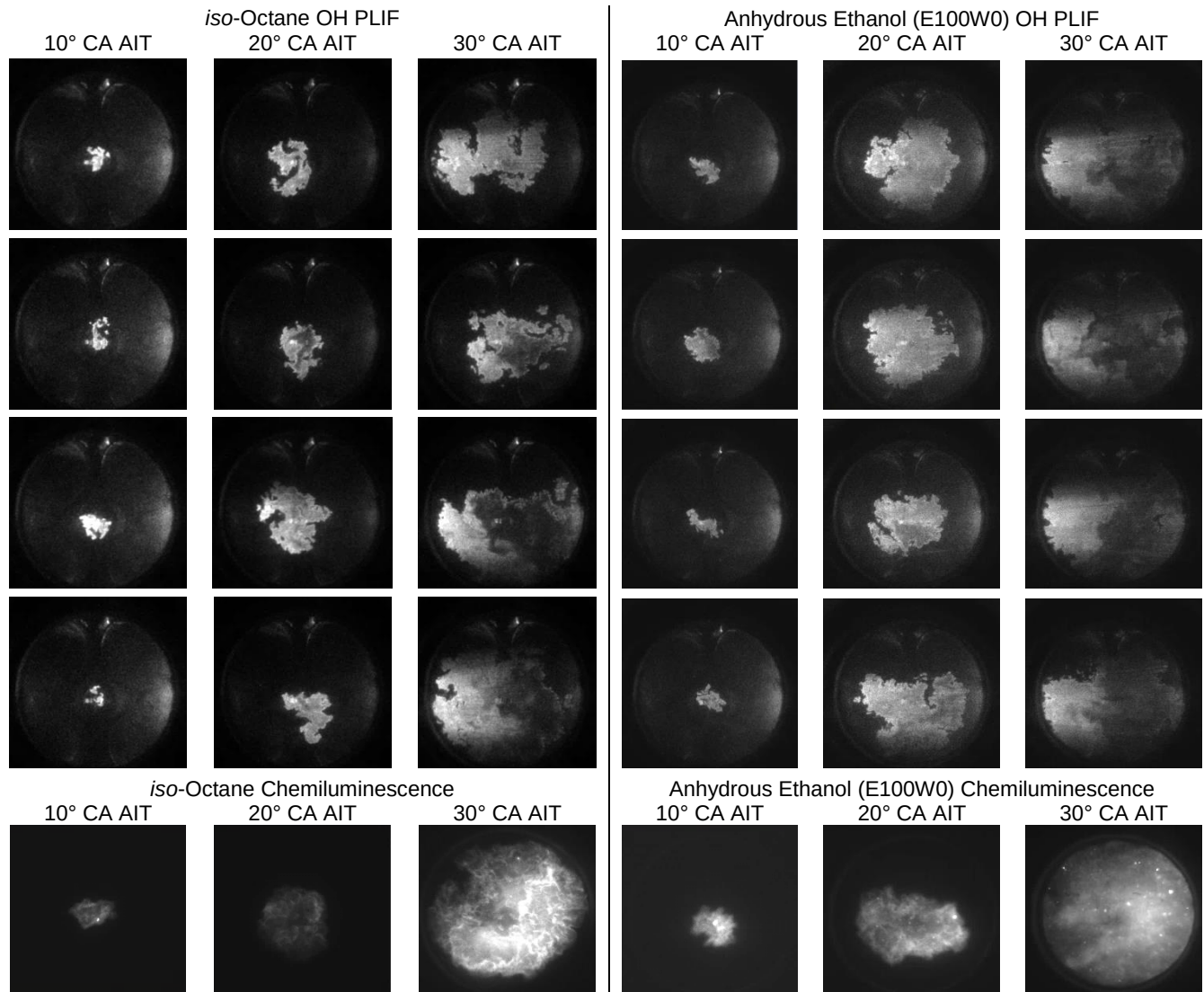


Figure 12. OH PLIF and flame chemiluminescence images of *iso*-Octane and E100W0 at various crank angles AIT for PFI operation

Another interesting observation is the characteristics exhibited in the flame chemiluminescence images. Those images show highly luminous structures around the right-hand-side exhaust valve. This behaviour was typical amongst all the flame images obtained at 30° CA AIT and is related to the light integration along the line of sight, *i.e.* along the size of the flame. The OH PLIF images do not replicate this characteristic; they simply reveal the burnt gases region as it lies behind the tip of flame 'sphere' with the highest integrated combustion luminosity.

The OH PLIF images of anhydrous ethanol also exhibit variability from cycle-to-cycle, especially for 10° CA AIT. Is it clear that even at 10° CA, the OH areas are relatively larger than those obtained with *iso*-octane, albeit the fact that both exhibit similarly wrinkled shape. At 20° CA AIT, ethanol's OH imaged areas are significantly larger than *iso*-octane's. This may be attributed to the marginally higher laminar burning velocity of ethanol at engine-like conditions which when coupled to turbulence leads to much faster burning. Ethanol's images also show 'more round' OH shapes on a macroscale when compared to *iso*-octane. They also remain largely centralized around the spark plug compared to *iso*-octane. The latter, by 20°CA AIT, exhibits OH patterns that have been displaced slightly towards the exhaust valve side. At 30° CA, ethanol's OH images have completely occupied the field of view on the Laser plane and still show clearer roundness than *iso*-octane's. In contrast, *iso*-octane's images at 30°CA AIT exhibit already large OH areas filled with burnt gas on the Laser's plane. However, *iso*-octane shows higher intensity compared to ethanol, especially at 10 CA, with both fuels

consistently having higher intensities on the left side of the images.

Further OH PLIF images are presented in Fig. 13 and Fig. 14. The imaged plane in these figures was set to coincide with the tip of the spark-plug's electrodes to investigate difference between fuels around this vicinity. The distinct consistent bright four orthogonal spots in those images are light reflections from the tip of the four ground electrodes of the spark plug. The 105 mm F/4.5 UV lens was used for this PLIF study to increase magnification. Although the gain for the intensified camera was fixed for the all the images in Figs 12–14, the contrast of the images in Figs 13 and 14 has been adjusted to increase brightness and compensate for loss in captured light from the change in lens aperture. Hence a direct comparison of the OH intensities in Fig. 13 and Figs 14 and 15 should be done with some caution.

It is clear that *iso*-octane's images are also smaller at the spark plug location and look more distorted and fragmented when compared to anhydrous ethanol; the latter fuel's OH images appear larger and less distorted on a macro scale. OH intensity is typically higher for *iso*-octane in comparison to ethanol. A comparison between the anhydrous ethanol and the E90W10 blend reveals that the former is associated with generally larger area than the latter - which also demonstrated greater cycle-to-cycle variability in size. The OH images of anhydrous ethanol show clearer and better defined edges than E90W10. Anhydrous ethanol also showed greater local flame distortion and corrugation than E90W10.

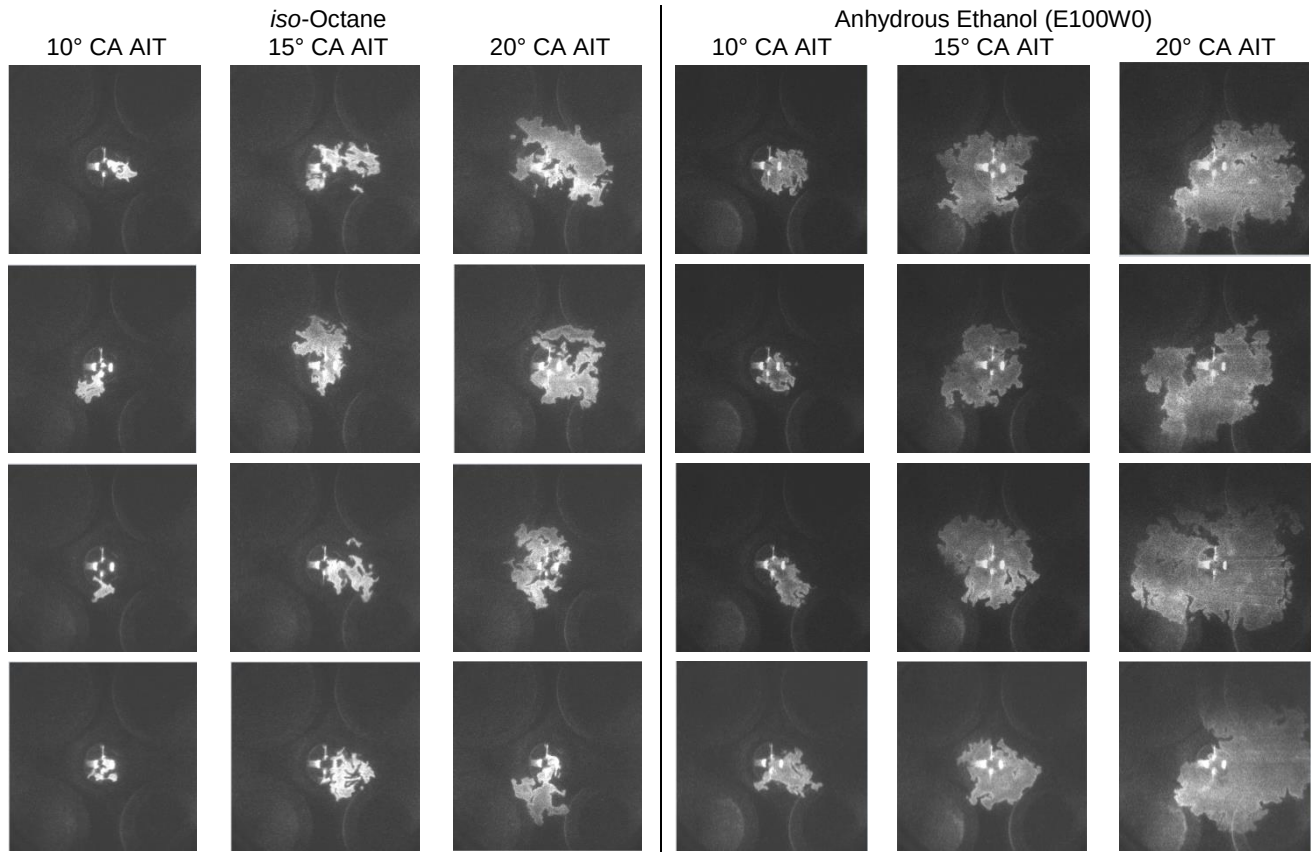


Figure 13. OH PLIF images of *iso*-Octane and E100W0 at spark plug location at various crank angles AIT for PFI operation

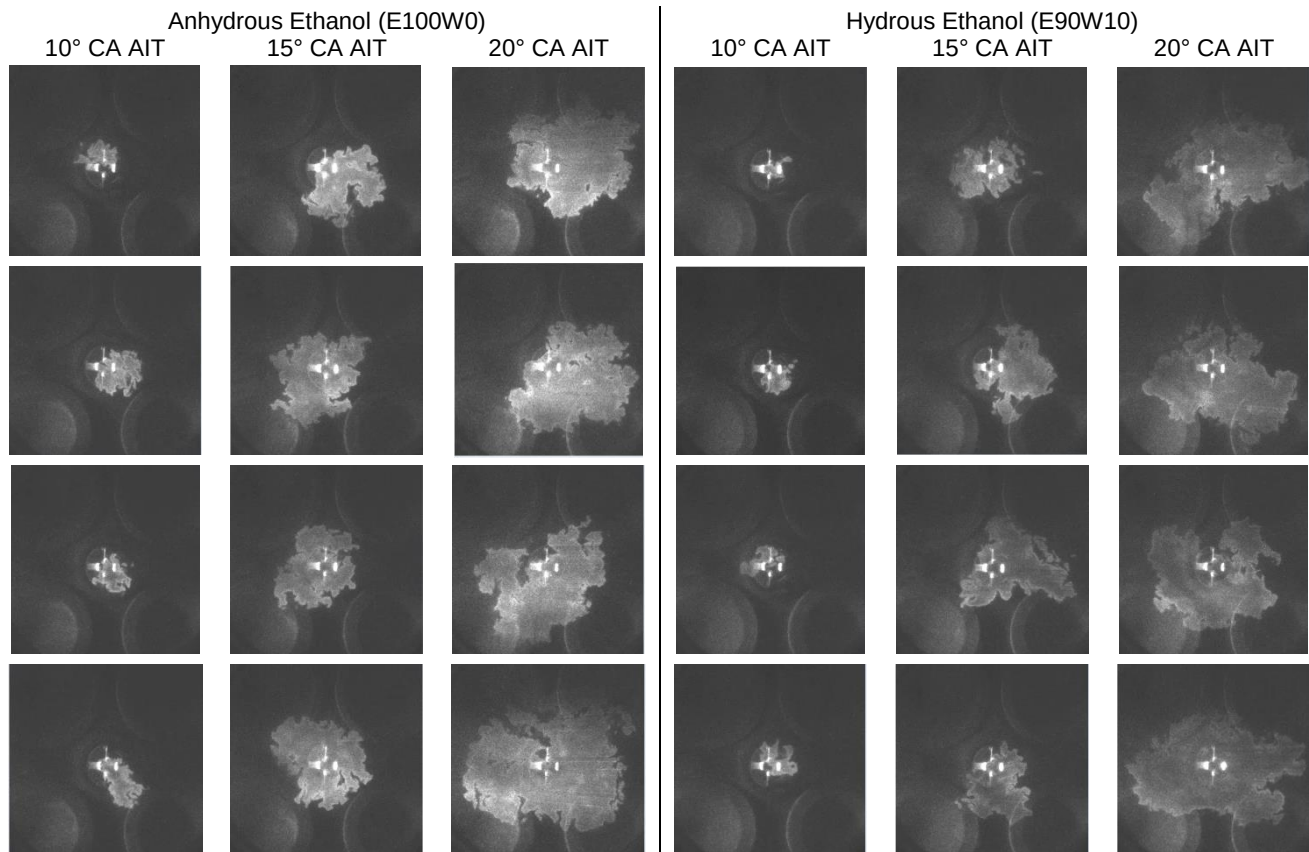


Figure 14. OH PLIF images of E100W0 and E90W10 at spark plug location at various crank angles AIT for PFI operation

Conclusions

The present work focused on characterising the combustion behavior of hydrous and anhydrous ethanol fuels in comparison to *iso*-octane and gasoline in a single-cylinder spark-ignition research engine operated at 1000 rpm with 0.5 bar intake plenum pressure. The engine was equipped with optical access and tests were conducted with both PFI and DI mixture preparation methods at stoichiometric conditions. Heat release analysis was conducted simultaneously with high-speed chemiluminescence image processing. OH PLIF images were also acquired to study the local flame front shape and structure. The main conclusions of this work can be summarized as follows:

Heat release

- Peak in-cylinder pressure reduced as water content in hydrous ethanol was increased for both PFI and DI operations and peak in-cylinder pressures were higher for PFI operations compared to DI for all fuels tested. The alcohol fuels also recorded higher peak cylinder pressure in comparison to the hydrocarbon. However, E90W10 had close peak pressure values to gasoline for both DI and PFI.

- The MFB analysis showed increasing water in hydrous ethanol slowed combustion rate for both PFI and DI fuel delivery method. Combustion rate was also faster when engine was operated with PFI strategy compared to DI operation irrespective of the fuel been burned. Of all fuels tested, anhydrous ethanol burned the fastest for both operations though very closely followed by E96W4 with both fuels overlapping over a large period AIT for both PFI and DI, reaching the point of 70% MFB for DI.
- The alcohol fuels burned faster than the hydrocarbons for both PFI and DI with *iso*-octane recording the lowest combustion rates in both operations. However, the range in combustion rate between fuels i.e. the fastest and the slowest based on 50% MFB was larger for PFI compared to DI. The latter was 4.4° CA while the latter was 6.4° CA.

Flame analysis

- PFI flames were generally larger than DI flames for all fuels. The DI flames also contained larger quantity of high intensity spots within the flames compared to PFI and the quantity increased with water content in ethanol for DI operation.
- For PFI, the flame radius curve trend was generally in agreement with the MFB curve with anhydrous ethanol

and E96W4 clearly the fastest followed by E90W10. *Iso*-octane was the slowest followed by gasoline.

- For DI operation, all fuels had similar flame radius from ignition timing until about 8° CA AIT. Gasoline was the fastest up to about 13° CA AIT before surpassed by E96W4 and anhydrous ethanol. E96W4 was marginally faster than anhydrous ethanol until about 25° CA AIT with both curves overlapping afterwards.
- Plots of flame growth speed against time showed the alcohols accelerated more than the hydrocarbons with anhydrous ethanol attaining a maximum flame growth speed of ~11 m/s for PFI. E90W10 was the slowest of the alcohols for both PFI and DI, attaining a maximum flame growth speed of ~10 m/s for DI operation. Overall, PFI recorded peak values of flame growth speed higher than DI for all fuels compared to DI.
- Flame growth speed against flame radius showed that for PFI, the alcohols all had similar magnitudes and were distinguished from the hydrocarbons. However, for DI, though the two groups persisted, there was a clear difference amongst fuels especially within flame radius of 2 to 10 mm.

OH PLIF

- OH PLIF images showed variability from cycle to cycle especially for 10° CA AIT for both *iso*-octane and ethanol with the former exhibiting a more wrinkled shape. However, for 15° CA onwards, ethanol showed a distinctive roundness in shape compared to *iso*-octane which still exhibited wrinkled shapes at 20° CA AIT similar to those recorded at 10 CA AIT.
- OH PLIF also showed there tend to be larger burned gases regions on planes behind the plane of the flame sphere showing the greatest flame luminosity.
- OH intensities were generally higher for *iso*-octane compared to anhydrous ethanol especially within 10° to 20° CA AIT.
- *Iso*-octane's images were also smaller at the spark plug location and looked more distorted and fragmented when compared to anhydrous ethanol; the latter's OH images appeared larger and less distorted on a macro scale.
- OH fluorescence intensities and areas were generally higher for anhydrous ethanol compared to hydrous ethanol with the latter showing more cycle to cycle variability.
- The OH images of anhydrous ethanol show clearer and better defined edges than E90W10. Anhydrous ethanol also showed greater local flame distortion and corrugation than E90W10.

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Definitions/Abbreviations

AIT	After Ignition Timing
ATDC	After Top Dead Center
BTDC	Before Top Dead Center
COV	Coefficient Of Variation
CR	Compression Ratio
DI	Direct Injection
DISI	Direct Injection Spark Ignition
E100W0	Anhydrous ethanol
E96W4	Blend of 96% anhydrous ethanol, 4% water (volume).
E90W10	Blend of 90% anhydrous ethanol, 10% water (volume)
ICCD	Intensified Charge Couple Device
IMEP	Indicated Mean Effective Pressure
IVC	Intake Valve Closure
PFI	Port Fuel Injection
PLIF	Planar Laser Induced Fluorescence
RPM	Revolution Per Minute
SI	Spark Ignition
SOI	Start of Injection
TDC	Top Dead Center