

UV-visible Optical Characterization of the Early Combustion Stage in a DISI Engine Fuelled with Butanol-Gasoline Blend

Simona Silvia Merola, Luca Marchitto, Cinzia Tornatore, and Gerardo Valentino
Istituto Motori CNR

Adrian Irimescu
Politehnica University of Timisoara

ABSTRACT

Detailed experimental information on the early stages of spark ignition process represent a substantial part for guiding the development of engines with higher efficiencies and reduced pollutant emissions. Flame kernel formation influences strongly combustion development inside the cylinder, especially for a direct injection spark ignition engine. This study presents the analysis of the evolution of spark-ignited flame kernels with detailed view upon cycle-to-cycle variations. Experiments are performed in a SI optical engine equipped with the cylinder head and injection system of a commercial turbocharged engine. Blend of commercial gasoline and butanol (40% by volume) is tested at stoichiometric and lean mixture conditions.

Experiments are carried out at 2000 rpm through conventional tests (based on in-cylinder pressure measurements and exhaust emission analysis) and through optical diagnostics. In particular, UV-visible digital imaging and natural emission spectroscopy are applied. A post-detection procedure is used to evaluate flame kernel areas evolution and its correlation to the MFB. UV-visible natural emission spectroscopy is applied to investigate the formation and the evolution of the main compounds characterizing the spark ignition and combustion process.

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INTRODUCTION

Butanol features several advantages over alcohols with lower molecular weight such as methanol or ethanol, given its increased heating value and higher compatibility with existing fuel system components intended for gasoline use. Several studies of butanol and its mixtures with gasoline used in SI engines have been performed mostly on port injection power units. A somewhat comprehensive and recent literature review on butanol production advances and its use as an energy source can be found in [1]. As a general conclusion, only the use of blends containing high concentrations of the four carbon atoms alcohol was found to have a significant impact on engine performance and efficiency if the air-fuel ratio is to be maintained within the same range as for gasoline operation.

A study featuring stoichiometric operation of a port injection SI engine found an increase of HC and CO

emissions, along with a reduction of NO_x and soot formation at wide open throttle [2]. Other studies have found similar results and the literature features numerous articles on butanol use and its effects on emissions variation in port injection SI engines; direct injection however, was studied to a lesser degree and for low concentrations of alcohol mixed with gasoline [3, 4].

For fundamental understanding of in-cylinder processes related to the use of butanol fuels in new generation DISI engines, originally designed to operate on hydrocarbons, it is necessary the fully integration of experimental methodologies based on conventional (as in-cylinder pressure and pollutant exhaust measurements) and unconventional diagnostics as the optical techniques.

Recent works [5, 6] investigate in-cylinder spray combustion process due to alcohols and hydrocarbons fuel injection in the early intake stroke. Also in his work, the

homogeneous engine operation was considered; on the other hand the pressure related data and exhaust measurements were correlated with UV-visible chemiluminescence imaging and natural emission spectroscopy data. Moreover, commercial gasoline fuel neat and blended with n-butanol (40% in volume) were tested. The final target is to realize a database useful for a better knowledge of the basic combustion mechanisms under realistic engine conditions.

EXPERIMENTAL SET-UP

The engine used for the experiments was an optically accessible single cylinder DISI engine. It was equipped with the cylinder four-valve head of a commercial SI turbocharged engine with the same geometrical specifications (bore, stroke, compression ratio). Also the injection system was the same as the reference real engine with a commercial 6 hole solenoid injector (flow rate Exxsol D40 at 100 bar $Q_{stat}=9.00$ g/s). The valve overlap on this engine resulted in levels of internal Exhaust Gas Residuals (EGR) of 15-20%. Further details on the engine are reported in [Table 1](#).

All measurements were performed at 2000 rpm, WOT and 100 bar fuel injection pressure. Intake air temperature was in the range 300-310 K. Cooling and lubricant fluid temperature values were maintained within 320-340 K by a thermal monitoring and conditioning system. Engine was fuelled by commercial gasoline (BU00) and a blend (BU40) of gasoline and n-butanol (40% volume).

Table 1. Engine specifications

Displaced volume	399 cc
Stroke	81.3 mm
Bore	79 mm
Connecting Rod	143 mm
Compression ratio	10:1
Number of Valves	4
Exhaust Valve Open	153° ATDC
Exhaust Valve Close	360° ATDC
Inlet Valve Open	357° ATDC
Inlet Valve Close	144° BTDC

Table 2. Operating points with relative air-fuel ratio and spark timing settings

Test label		λ	Spark advance
SA7_ST	Stoichiometric	1	7 deg BTDC
SA15_ST			15 deg BTDC
SA15_L	Lean	1.15	15 deg BTDC

Injection start was set at 300 deg BTDC for all conditions, as this point was found to ensure maximum performance, probably due to the fuel jet-piston interaction. Pulse duration for opening the injector was 39.5 deg during gasoline fueling and 42.5 deg for BU40 (40% volumetric concentration of butanol mixed with gasoline) with stoichiometric, and 33 deg for the first, with 35.5 deg for the latter fuel type when using

lean mixtures. Relative air-fuel ratio was measured using a wide band exhaust gas oxygen sensor, with an accuracy of $\pm 1\%$. Relative air-fuel ratios and spark timing settings used in the experiments are shown in [table 2](#).

For each engine operative conditions and for both the fuels, tests were performed according to the following procedure: after 1 minute of warm-up in motored conditions, the engine works in fired conditions for 200 consecutive cycles. Optical data detected in the last 160 cycles were considered.

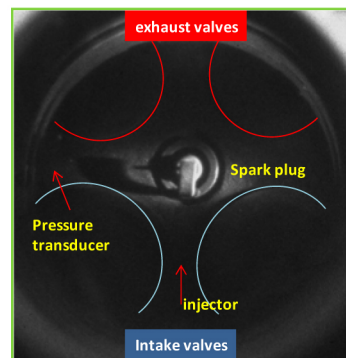
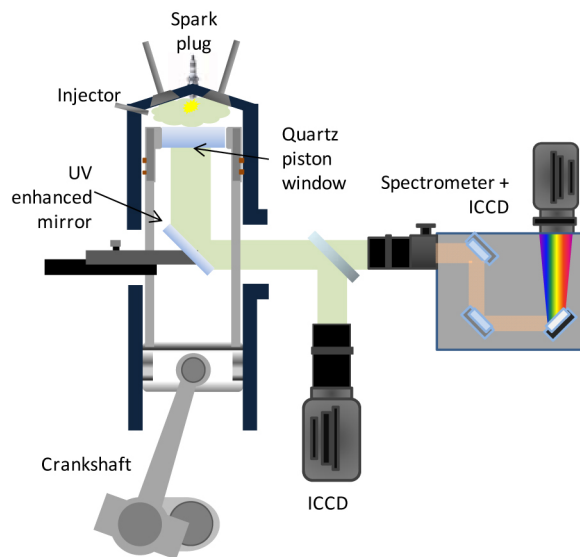


Figure 1. Experimental setup for optical investigations and bottom field of view of the combustion chamber.

Sketch of the experimental apparatus is shown in [Figure 1](#). A quartz pressure transducer was flush-installed in the region between intake-exhaust valves at the side of the spark plug. The transducer allowed to perform in-cylinder pressure measurements in real-time with an accuracy of $\pm 0.5\%$. Pressure values were related to piston movement by triggering the data acquisition system based on an optical crank angle encoder with a resolution of 0.1 deg. Spark timing is given throughout the article as advance angle before TDC at the end of compression. Unless otherwise specified, such as giving the crank angle interval elapsed after spark

(AS), all other crank angle references are made in relation to the same point (i.e. TDC at the start of expansion).

For the current work, a conventional elongated hollow Bowditch piston was used. An optical crown, accommodating a 20mm-thick fused-silica window (57mm diameter), was screwed onto the piston. The combustion chamber was visible through an UV enhanced 45-degree mirror, mounted within the hollow piston. Slotted graphite piston rings were used to provide oil-less lubrication with uninterrupted bronze-teflon rings used for sealing.

Spark ignition and flame growth were detected by an intensified CCD camera (array size of 1024×1024 pixels with a pixel size of $13 \times 13 \mu\text{m}$, 16-bit pixel digitization and 1MHz sustained repetition rate). ICCD was equipped with a 78 mm focal length, f/3.8 UV Nikon objective. This optical set-up allowed to reach a spatial resolution of $53 \mu\text{m}/\text{pixel}$.

In this work, the ICCD was used both in “repetitive mode” and “sequential mode”. The first mode allowed to acquire images at the same CAD for consecutive engine cycles. The sequential gating mode was used to study the temporal evolution of the flame propagation: it allowed to collect one frame for each cycle keeping the same gate width but variable delay with respect to the trigger. For both the acquisition modes the exposure time was fixed at $42 \mu\text{s}$ (~ 0.5 CAD @ 2000 rpm). Series of 160 images were recorded for each operating condition. Repetitive measurements were performed with a delay of $600 \mu\text{s}$ relative to the spark (~ 7.2 CAD @ 2000 rpm), in order to evaluate cycle-to-cycle variations. Images were also recorded in “sequential mode”; the dwell time between two consecutive acquisitions (performed on two consecutive engine cycles) was set at $83 \mu\text{s}$ (~ 1 CAD @ 2000 rpm). All the ICCD acquisitions started on the rising edge. The ICCD trigger was activated on the falling edge of the spark triggering pulse (end of the coil charge).

In order to realize the image processing, a custom numerical procedure was developed by Vision Assistant and LabView of National Instruments (NI). For quantitative analysis of the flame front characteristics were obtained via thresholding/binarization of each flame image. In particular the maximum entropy method was applied on the luminous intensity plane [7]. Before the image processing, an appropriate circular region of interest on the flame images was fixed in order to cut the spurious light from reflections at the boundaries of the optical access. Mean flame growth curves were plotted for all fuels in terms of circle equivalent area. To compare flame area with burned volume cross-section, the values were normalized with respect to the cylinder bore.

The flame area were estimated by the average from the areas calculated from external and internal flame boundaries of binarized image. The effect thresholding method was also evaluated. In particular the effect of 6-bit manual thresholding was considered. Figure 2 shows a comparison between the external and internal flame boundaries obtained by entropy method and manual thresholding. For each

binarized image the area and Waddel disk, that is the diameter of a disk with the same area as the particle, were listed. For both the quantities, relative uncertainties resulted lower than 10% (Table 3). Similar results were obtained for all the test conditions. It should be noted that all references made to flame area are values given by the shadow flame area. Moreover, it was possible to follow the flame growth until the optical limit (at around 70% of the area described by the cylinder bore $\pi B^2/4$).

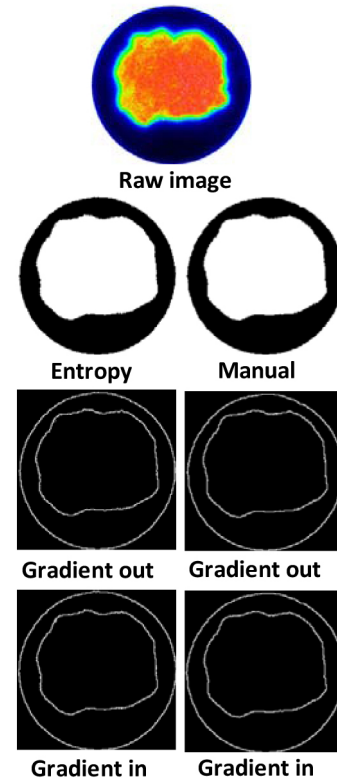


Figure 2. Effect of threshold method on the binarized flame image.

Table 3. Effect of threshold method on the equivalent diameter and area of the flame.

Threshold method	Boundary	Waddel Diameter [pixel]	Area [pixel]
Entropy	out	28.635	644
Entropy	in	28.524	639
Manual	out	28.479	637
Manual	in	28.255	627

For the UV-visible spectroscopy experiments, the radiative emission from the combustion chamber was focused by a 78 mm focal length, f/3.8 UV Nikon objective onto the micrometer controlled entrance slit of a spectrometer with 150 mm focal length and 600 groove/mm grating (central wavelength 350 nm). From the grating the radiations were detected by the ICCD camera previously described. The exposure time was fixed at $42 \mu\text{s}$ and the dwell time between two consecutive acquisitions was set at $83 \mu\text{s}$. Spectroscopic

investigations were carried out in the central region of the combustion chamber. The emission spectra were corrected for the wavelength dependent sensitivity of the optical devices. The wavelength calibration was performed using a mercury lamp.

Spectroscopic data post-processing allowed to evaluate the time evolutions of chemical species featuring the combustion process in all the operative conditions. For each individual- cycle spectrum, the heights of the CH (431 nm), CN (388 nm) and OH (309 nm) emission peaks were evaluated relative to the local background, which was estimated by linear interpolation between the nearest intensity minima on either side of the band containing the feature of interest. The local background was principally due to HCO, H₂CO* and CO₂*. Soot signal was evaluated at 510 nm.

Exhaust gas concentrations of unburned hydrocarbons (HC) and nitrogen oxides (NO_x) were measured using a 5-gas analyzer with resolutions of 0.01% for the first component and 1 ppm for the other one. Smoke opacity determination was performed with a smoke-meter featuring a 0.01% resolution.

RESULTS AND DISCUSSION

Given that injection rates were increased by ~7% when switching from BU00 to BU40, air-fuel ratio was maintained within roughly the same limits (variability has been estimated to be lower than 1% in λ) at $\lambda \sim 1$ during stoichiometric, and $\lambda \sim 1.15$ for lean operation. Therefore, any significant difference in working parameters would be the result of fuel properties.

With regard to ignition timing, two points that were roughly at the same crank angle interval away from MBT setting (found to be at 10 deg BTDC for both fuels with $\lambda \sim 1$) were chosen, while the more advanced spark setting was kept when changing from stoichiometric to lean operation. It should be mentioned that the torque curve was relatively flat within this spark advance region, and allowed to operate the engine without any abnormal combustion event, avoiding even light knock. Therefore, the entire discussion is organized around the variation of these two parameters, namely relative air-fuel ratio and spark timing.

A brief comparison of the pressure traces recorded with gasoline and BU40 (Figure 3), reveals that similar values were obtained for both fuel types at the same spark timing and air-fuel ratio. Differences of peak pressure were as low as ~0.2 bar for stoichiometric operation at 15 deg BTDC spark timing and up to ~1 bar with the 7 deg advance setting. The difference of maximum pressure for lean mixtures was between the two values previously mentioned, namely ~0.6 bar. In all cases, the higher values were obtained for BU40 fuel. Considering that at 50 bar the accuracy of $\pm 0.5\%$ means an absolute error of 0.25 bar, it can be concluded that indeed the presence of alcohol in the fuel blend did have an effect; however, this effect is reduced and resulted in no penalty or significant improvement of performance. Therefore, mixing

gasoline with butanol up to 40% volumetric concentration can be performed in DISI engines without any negative effects and most likely without the need for changing spark advance settings as compared to gasoline operation. These observations are of course valid for the conditions investigated, namely naturally aspirated WOT.

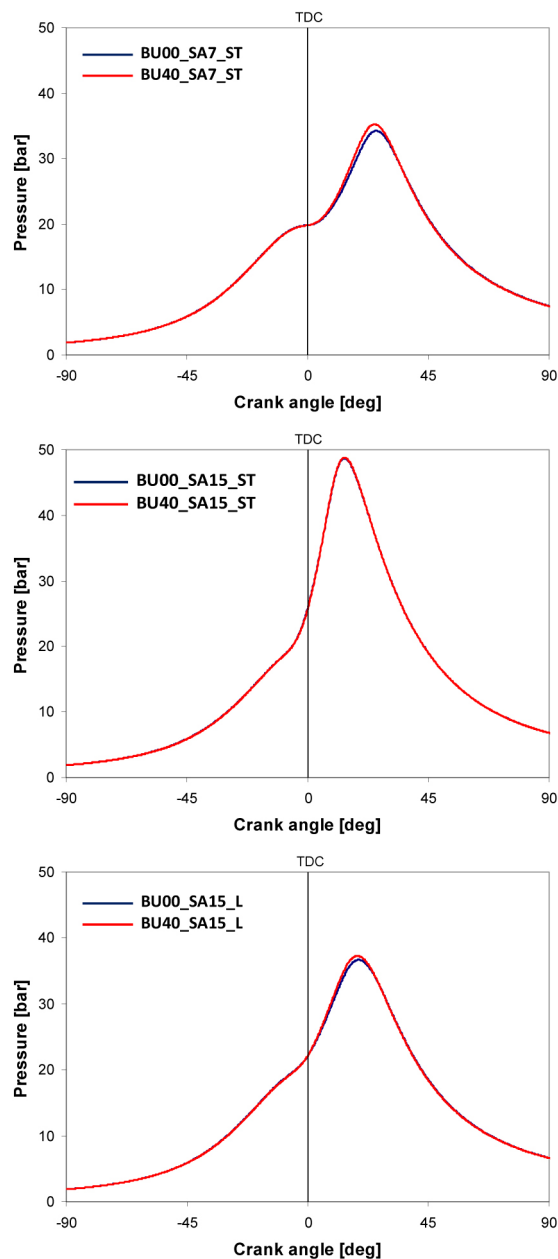


Figure 3. Pressure traces averaged on 200 consecutive engine cycles and measured for the two fuel types at different spark timings and relative air-fuel ratios.

Increased peak pressure and advanced location of this point is to be expected, given that n-butanol features faster laminar flame development compared to gasoline. The curves for the alcohol blend consistently featured slightly higher peak pressure (Figure 4), most likely due to this fuel property.

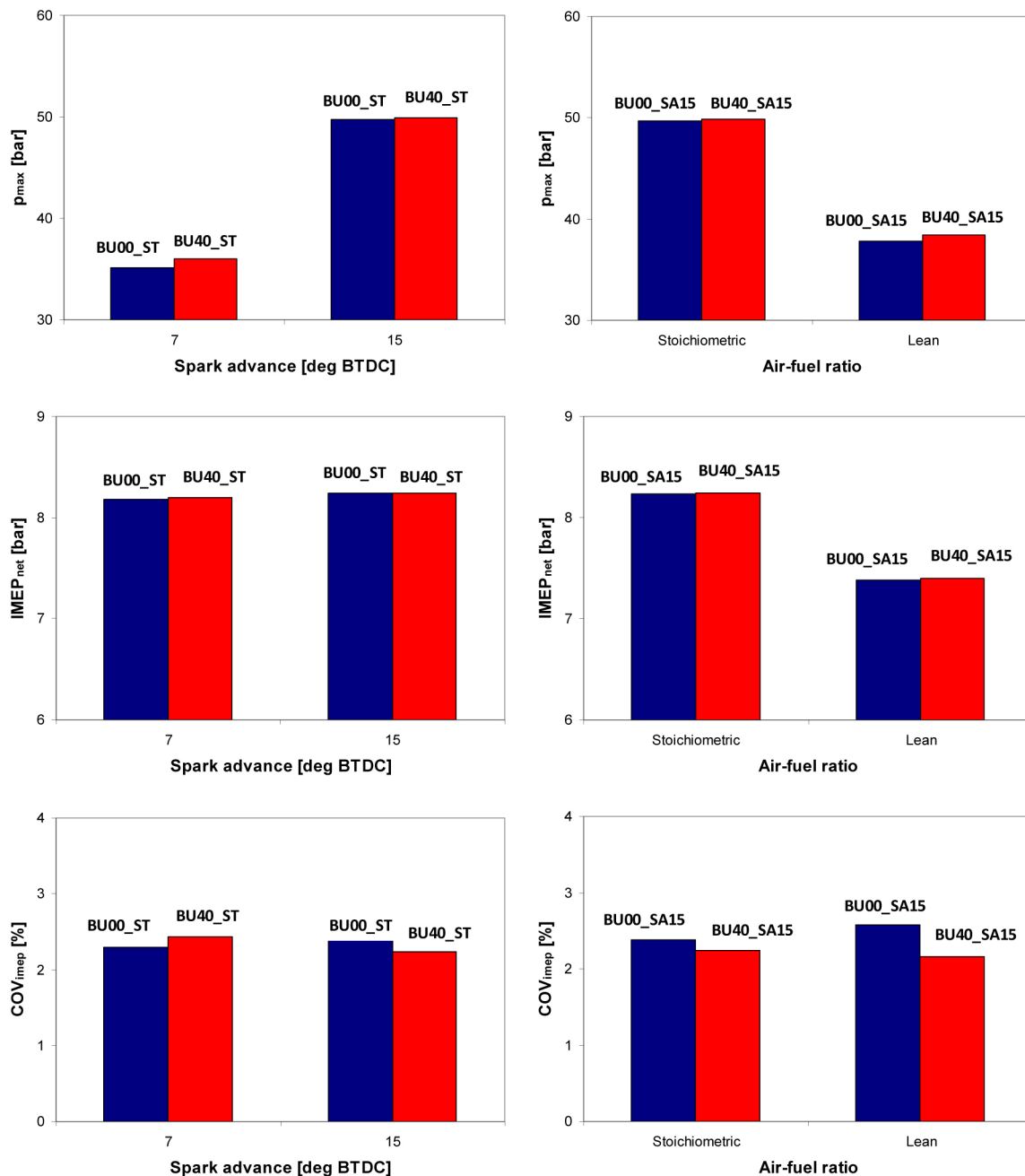


Figure 4. Mean operating parameters for the two fuel types at different spark timing (left) and relative air-fuel ratios (right).

Faster flame development also seems to be supported by the location of peak pressure when switching from gasoline to BU40, with an advance of 0.5 deg and 0.7 deg on average for the lean operation points and stoichiometric with retarded timing respectively. Given that the observed advancement is higher than the resolution of the crank angle encoder, this effect too can be attributed to the change in fuel properties. Again, the difference is not that significant so as to induce changes in performance or require spark timing adaptations.

As a direct result of the similar pressure traces recorded for the two fuel types, indicated power and cycle-to-cycle variability were also very close at the same operating point.

Evidently, leaning out the mixture from stoichiometric to a relative air-fuel ratio of ~ 1.15 lead to a reduction of $\sim 10\%$ in indicated mean effective pressure. Moreover while the cycle-to-cycle variation remained roughly the same for BU40 a little decrease in the stability for BU00 was observed.

This effect can be due to the higher laminar flame speed of BU40. These findings are in line with the ones obtained in previous studies [8, 9], even though mixture formation was performed completely inside the cylinder. It is exactly this difference that is likely to contribute to the slight improvement in combustion stability for the alcohol containing blend, since the mixture is stratified in a more

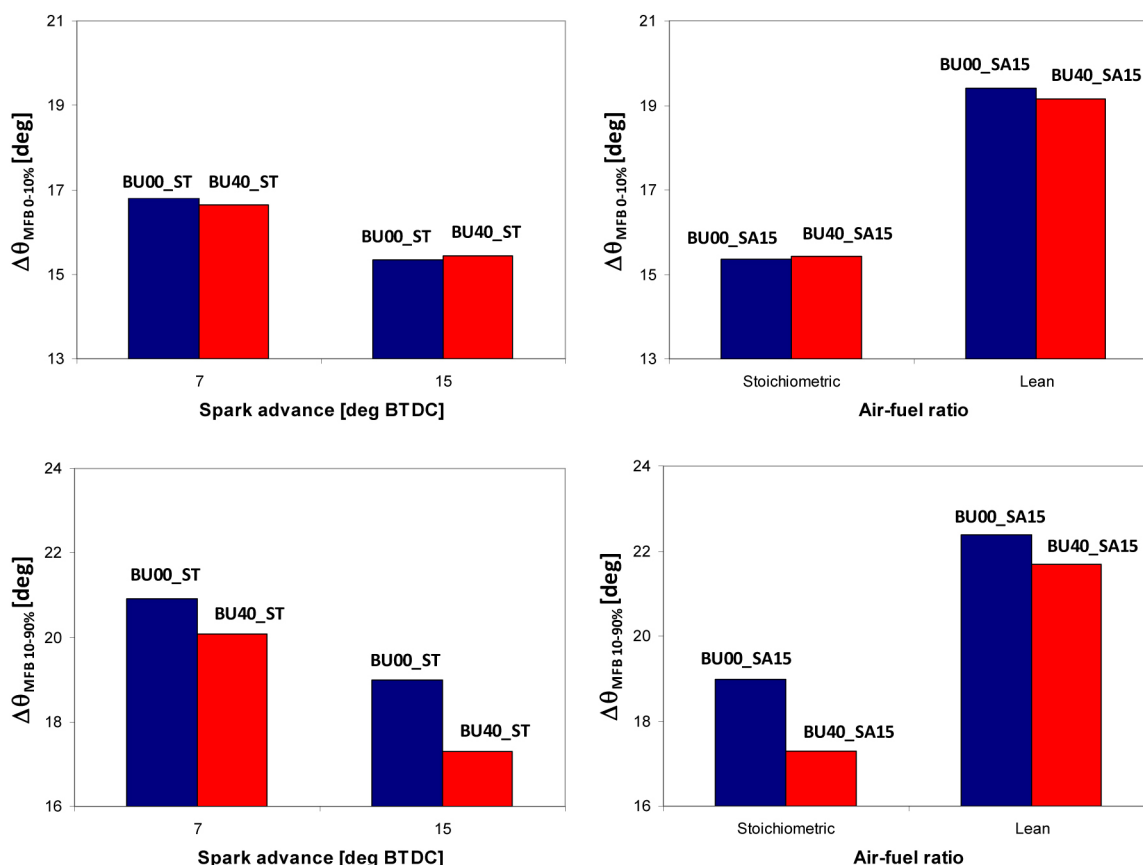


Figure 5. Duration of initial and main combustion phases.

controlled manner in DI engines by the process of fuel impingement on the piston crown [10]. Further investigations into the issue of fuel jet break-up and vaporization should provide more insight, but were beyond the scope of this research. The aforementioned effect of higher laminar flame speed, along with a different fuel stratification (only a minor stratification is foreseeable, given that injection was performed well before ignition) can be considered as the mechanisms that resulted in faster combustion development and slightly improved stability.

MFB values and duration of different combustion phases were calculated using the Rassweiler and Withrow technique [11]. This simpler method was preferred to applying a heat transfer correlation and calculating MFB values based on heat release analysis, in order to ensure consistency when comparing results (i.e. reducing possible variations due to model complexity); also, fuel evaporation is very likely to occur even after intake valve closure, thus requiring a separate heat and mass transfer model to account for this effect.

A reduction in combustion duration was recorded for BU40, except for the 10% MFB point with a spark advance of 15 deg BTDC and using stoichiometric air-fuel mixtures (Figure 5). As expected, the duration of first and main combustion phases was higher for lean as compared to stoichiometric operation. The effect of spark timing was

particularly interesting, even though the two advance settings were chosen at roughly the same span related to the point of MBT located at ~ 10 deg BTDC for both fuel types. Interestingly enough, the first combustion phase, from 0 to 10% MFB, was very close duration wise for both fuels, even though it would have been expected that laminar flame speed would have a greater influence in this period [12]. A more consistent difference was observed during the main combustion phase, from 10 to 90% MFB, in line with previous results. This seems to indicate that the air-fuel mixture was stratified to a certain degree (again, even though injection was performed during the intake stroke) and combustion evolved in a similar manner for both fuel types.

Given that no significant information could be obtained based on pressure trace analysis in the initial combustion phase, a more detailed study was performed using optical investigations. The early stage of combustion plays a key role for engine stability and efficiency because in this period the main exothermic reactions influencing the combustion process take place. The UV-visible imaging allowed to characterize both the flame propagation and the spatial-temporal evolution of the main excited species participating to the chemical reactions occurring during the combustion.

Combustion imaging allows a more detailed analysis of phenomena occurring within the combustion chamber, including the study of different processes characteristics at