

# Split Injection in a DISI Engine Fuelled with Butanol and Gasoline Analyzed through Integrated Methodologies

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## ABSTRACT

In this study, experiments were carried out in an optical single-cylinder Direct Injection Spark Ignition engine fuelled with n-butanol and gasoline, alternatively. The engine is equipped with the head of a commercial turbocharged engine with similar geometrical specifications (bore, stroke, compression ratio). The head has four valves and a centrally located spark device with surface charge ignition. A conventional elongated hollow Bowditch piston is used and an optical crown, accommodating fused-silica window, is screwed onto it. The injector is side mounted and features 6 holes oriented to guide the jets towards the piston crown.

During the experimental activity, the injection pressure was maintained at 100 bar for all conditions; the injection timing and the number of injections were adjusted to investigate their influence on combustion and emissions. Different optical diagnostics were applied in order to follow the combustion process, from ignition to the completion of flame front propagation: cycle resolved visualization, chemiluminescence and natural emission flame spectroscopy in the UV-Vis range.

The optical data was correlated with conventional measurements of thermodynamic analysis and exhaust emissions. The effect of split injection (i.e. two injections per cycle) was found to be relatively negligible for gasoline, while for the alternative fuel significant differences were recorded when switching from single to double injection. This points to a different mixture formation process, mainly related to the evaporative properties of n-butanol.

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## INTRODUCTION

The internal combustion engine represents a well-known and proven technology that is still the power unit of choice in the automotive sector. Even if it features inherently lower fuel conversion efficiency compared to its compression ignition counterpart, the spark ignition (SI) engine still holds the majority in automotive applications, mainly due to higher adaptability to varying fuel properties and less complex exhaust gas after-treatment systems [1]. The reduction in pollutants emissions has become an issue as pressing as the need to improve efficiency; it makes the choice of injection, ignition and other secondary strategies (such as exhaust gas recirculation) a complex control problem that takes into account an ever increasing number of monitored parameters and output variables [2, 3]. Alternative fuels make the problem even more complex, given that certain characteristics exert multi-level influences that require special control strategies. One example is the use of alcohols as gasoline replacement or blended with the classical fuel, in the drive towards reduced environmental impact and increased energy security, as well as to capitalize on their higher laminar flame speed, which results in improved performance [4]. A higher flame speed enables higher thermal efficiency due to a shorter period of heat release and it has a positive influence on the tolerance for charge dilution with excess air and inert gases, as well [5, 6, 7, 8, 9].

Direct injection (DI) only recently has been adopted on a wide scale by manufacturers, mainly due to prohibitive costs, but also due to the problem of higher particulate emissions compared to the extensively used port injection strategy [10]. Therefore, the benefits of improved fuel conversion efficiency are to a large degree offset by the penalty of higher rates of particulate matter formation during stratified charge operation. When considering DISI control, there are basically two routes, represented by homogenous and stratified air-fuel mixture inside the combustion chamber; the first choice is more readily available and easier to apply given that injection pressure is lower and thus implementation costs can be significantly reduced. Stratified operation is quite difficult to achieve with wall guided systems and requires special air motion control [11]; spray guided systems are more robust, but feature higher injection pressure and often multiple injections per cycle are needed to obtain proper ignition [12]. This last control strategy is receiving increased attention in wall guided injection systems, as it may provide several advantages when applied in certain conditions. The main drive behind the application of split- or double-injection is to reduce jet impingement on the piston crown and cylinder liner, in order to avoid the formation of fuel film on these surfaces [13]. Improved air-fuel charge stratification (much less prominent compared to spray guided injection) can also result in improved efficiency and engine stability [14]. Another example is the

use of split injection during the intake stroke in order to reduce the tendency towards the occurrence of super-knock [15]; this result can be regarded as an important aspect of full load operation, given that such extreme events cannot be controlled by using existing knock mitigation systems [16].

Within this context, the present study investigated the effect of injection splitting in an optically accessible DISI engine. N-butanol fuelling was compared to gasoline operation. Improved understanding of global and local phenomena was provided through integrated methodologies employing thermodynamic and optical measurements.

Flame visualization and spectroscopy were applied in an optically accessible DISI engine featuring wall guided air-fuel mixture formation with a side mounted injector. These two high spatial resolution techniques were combined with standard in-cylinder pressure trace and exhaust gas emissions recording. These latter measurements were also coupled with extinction spectroscopy at the exhaust in order to provide a clearer picture on particulate emissions, given that this is one of the main issues in DISI engines.

## EXPERIMENTAL APPARATUS

Combined thermodynamic and optical measurements were performed in a single cylinder SI engine. Optical accessibility was ensured through a 20 mm-thick fused-silica window fixed on the piston crown featuring a Bowditch design (Figure 1), with a 45 degree UV-enhanced mirror fitted within the hollow piston. Slotted graphite piston rings were used to provide oil-less lubrication with uninterrupted bronze-Teflon rings used for sealing.

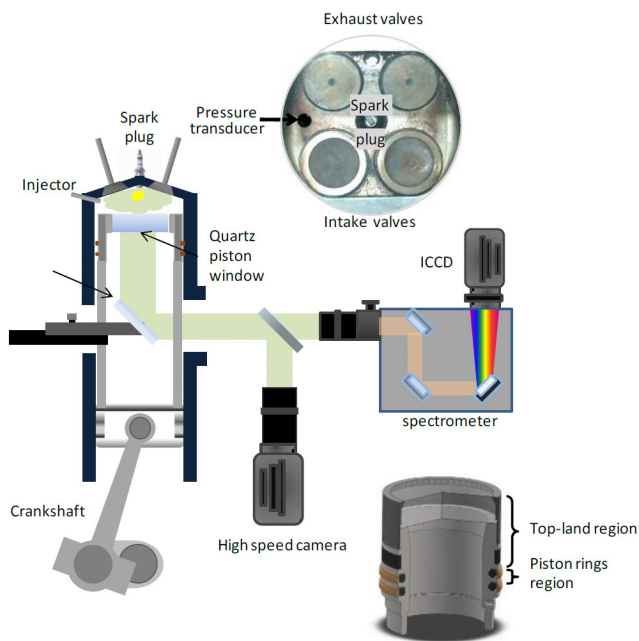


Figure 1. Experimental setup.

The cylinder head was taken from a 16-valve, 4 cylinder commercial power unit featuring a central spark plug and wall guided direct injection at 100 bar. Fuel delivery was ensured by a low pressure circuit combined with a high pressure mechanical pump which

delivered the fuel to a 6 hole solenoid injector that directed the spray towards the piston crown; pressure was maintained constant in the injector rail using an electronic unit with feed-back control. Further details on engine characteristics are listed in Table 1.

The fuels used for the experimental activity were commercial gasoline and n-butanol (referred simply as butanol or BU100 hereafter). Intake air temperature was around 300 K and ambient pressure 1 atm; coolant and lubricant temperature was maintained at 330 K using a thermal conditioning unit. A fixed value for engine speed was set at 2000 rpm, chosen as representative for mid-road load automotive use. All cases that were investigated featured wide open throttle (WOT) and the spark timing was set at 15 CAD BTDC, in these conditions knocking event never takes place. Fuel delivery was performed in a single event (start of injection SOI300 CAD BTDC) and split into two separate injections (divided 50-50% with respect to fuel quantity); the first injection was fixed at 300 CAD BTDC, while the second one was performed at two different instances during the cycle, namely 120 and 140 CAD BTDC, referred to as SOI300\_120 and SOI300\_140 respectively. Injector opening time was increased when switching from gasoline to butanol by a ratio of 1.21 in order to compensate for the combined effect of lower stoichiometric air-fuel ratio and higher density of butanol (Table 2). Two relative air-fuel ratios were chosen: close to stoichiometric (i.e.  $\lambda \approx 1$ ) and lean (i.e.  $\lambda \approx 1.15$ ) operation. This parameter was monitored with an accuracy of  $\pm 1$  % using a wide band oxygen sensor fitted at the exhaust.

Table 1. Single cylinder SI engine characteristics.

|                    |          |
|--------------------|----------|
| Displaced volume   | 399 cc   |
| Stroke             | 81.3 mm  |
| Bore               | 79.0 mm  |
| Connecting Rod     | 143 mm   |
| Compression ratio  | 10:1     |
| Number of Valves   | 4        |
| IVO                | 3° BTDC  |
| IVC                | 36° ABDC |
| EVO                | 27° BBDC |
| EVC                | 0° ATDC  |
| Injection pressure | 100 bar  |

Exhaust gas concentration was determined through a gas analyzer using the non-dispersive infrared measurement principle for carbon monoxide (CO) and unburned hydrocarbons (HC), and the electrochemical method for nitrogen oxides ( $\text{NO}_x$ ) emissions. Accuracy was within  $\pm 3$  % and the resolution of readings for the first component was 0.01% and 1 ppm for the other two species. Soot and particulate were detected by a spectroscopic technique in order to overcome difficulties related to the use of the smoke-meter and opacimeter in the exhaust gas stream of an optical engine in lean combustion conditions. Optical diagnostics allowed to perform a real time gaseous and solid emission detection avoiding the trouble of ex-situ sampling or exhaust dilution.

Table 2. Fuel properties [17, 18].

| Fuel                                          | Gasoline | n-Butanol |
|-----------------------------------------------|----------|-----------|
| Test label                                    | G100     | BU100     |
| Chemical formula                              | C4-C12   | C4H9OH    |
| Lower heating value (MJ/kg)                   | 42.9     | 33.0      |
| Density (kg/m <sup>3</sup> )                  | 736      | 810       |
| Octane number (R + M)/2                       | 97       | 89        |
| Boiling temperature (°C)                      | 25–215   | 118       |
| Latent heat of vaporization (kJ/kg)           | 380–500  | 716       |
| Self-ignition temperature (°C)                | ~300     | 343       |
| Stoichiometric air/fuel ratio                 | 14.7     | 11.2      |
| Laminar flame speed (cm/s) <sup>a,b [9]</sup> | 51       | 58.5      |
| Adiabatic flame temperature (K)               | 2370     | 2340      |

(a) T = 393 K and P = 100 kPa, at stoichiometric mixture. (b) The gasoline was represented by iso-octane here.

### Optical Set-Up for Exhaust Measurements

Multi-wavelength extinction spectroscopy technique [19, 20] was applied at the undiluted engine exhaust to identify the contribution to the extinction of different chemical compounds with specific interest in soot and particulate emissions. The extinction coefficient due to an unpolarized source, can be written as

$$K_{ext}(\lambda) = \sum_{i=0}^N N_i C_{i,ext}(\lambda)$$

where  $\lambda$  is the wavelength of the incident beam,  $N_i$  and  $C_{i,ext}$  are the concentration number and the extinction cross-section respectively of the  $i$ th compound [21].

The most of the molecule extinction (or absorption) cross-sections in the UV-visible-near IR range are available in literature, as well as their dependence on temperature at ambient pressure. On the other hand, the extinction cross-section of particles is a complex function of the chemical and physical properties and can be interpreted in the framework of electromagnetic theory of light by

$$K_{ext}(\lambda) = N C_{ext}(\lambda, D_p, n, k)$$

where  $N$  is the number density of the particle,  $D_p$  is the particle diameter,  $n$  and  $k$  are the real and the imaginary part respectively of the complex refractive index  $m = n - ik$ , and  $C_{ext}$  is the extinction cross-section [21].

Assuming that the particles are spherical and constitute a polydispersed system, the extinction coefficients depend on the particle diameter and can be written according to the Mie theory as

$$K_{ext}(\lambda) = \int_{D_{min}}^{D_{max}} C_{ext}(m, \alpha_D) n(D_p) dD_p$$

where  $\alpha_D = \frac{\pi D_p}{\lambda}$  is the dimensionless size parameter and  $n(D_p)$  is the number distribution.

The retrieving of optical data related to exhaust particles induces strong hypothesis on refractive indices and/or size distribution [21, 22]. To avoid this influence, the opacity instead of extinction coefficient was considered. Opacity is calculated according to the relation

$$N = 100 - \tau$$

where  $N$  is the opacity in percentage and  $\tau$  is the ratio of transmitted light in percent.  $\tau$  is related to the extinction coefficient through the Beer-Lambert law.

$$K_{ext} = \frac{1}{L} \ln \left( \frac{\tau}{100} \right)$$

where  $K_{ext}$  is the extinction coefficient and  $L$  is the effective optical path length.

In this work the opacity was evaluated at 700 nm to avoid the overestimation that can occur at lower wavelength due to the overlapping of absorption species such as heavy HC and NO<sub>2</sub> [23, 24, 25].

To apply multi-wavelength extinction spectroscopy, the exhaust line was customized for the insertion of two optical accesses. A sketch of the optical set-up is shown in Figure 2. The path-length for the light in the system was basically the diameter of the exhaust duct: approximately 5.6 cm. The distance from the exhaust manifold to the measurement volume was about 2 m. The optical assembly consisted of a pulsed Xenon lamp (Ocean Optics, Model PX-2) and a spectrometer (Ocean Optics, Model USB2000+). The pulsed Xenon lamp is characterized by high flash rate and short-arc lamp emitting in the spectral range 220-850 nm. It operates at speeds up to 220Hz with 5μs pulse duration and offers excellent pulse-to-pulse stability. As it produces a pulsed signal, the PX-2 is less likely to contribute to solarization in optical fiber assemblies, which can occur when fibers are illuminated with a signal with  $\lambda < 260$ nm.

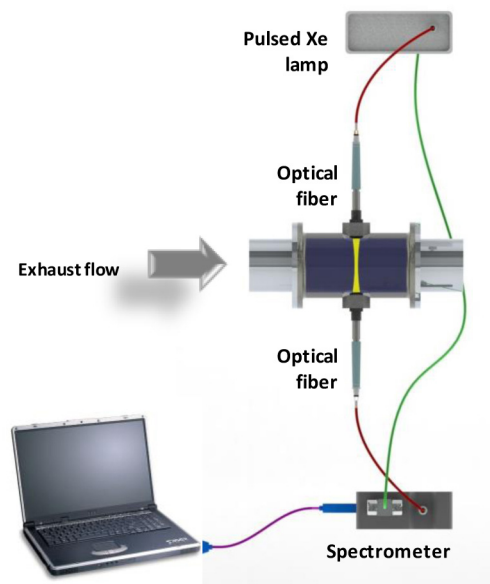


Figure 2. Sketch of the optical set-up for the extinction spectroscopy at the engine exhaust.

USB2000+ is a miniature fiber optic spectrometer specifically designed to match the spectral range of the Xenon lamp. The light was detected by an optical fiber connected to the spectrometer with a core diameter of 600  $\mu\text{m}$  and an aperture number of 0.22. An aspheric fused silica lens collected the light from the optical fiber and collimated the light for transmission through a fused silica window on the exhaust system. The same optical arrangement on the opposite side of the exhaust collected and transmitted the light back to the spectrometer. The optical design of the spectrometer is the crossed Czerny-Turner configuration with a 600-lines/mm grating which disperses collected light across a 2048-element linear CCD array. The spectral range of the spectrometer was 650 nm (leading to a dispersion of 0.32 nm/pixel). An external TTL positive pulse was used to trigger and synchronize the operation of the light source and spectrometer. Exposure time was fixed at 256 ms and each spectrum was referred to the average on 6 consecutive scans.

In the exhaust line, the extinction of the UV-visible-near IR radiation is due to soot extinction, fuel vapor, hydrocarbons and nitrogen oxides absorption. The nitrogen monoxide, NO, is characterized by isolated absorption tight bands below 250 nm. The nitrogen dioxide is featured by two wide absorption bands centered at 220 and 400 nm. The NO<sub>2</sub> band, in the visible, is extended from 250 nm to 550 nm [25]. Finally, aromatic hydrocarbons show a high intensity band below 280 nm due to light aromatics and weaker visible band systems typical of polycyclic aromatic hydrocarbons around 500 nm [23, 24]. Quantitative analysis of HCs is not possible because the contribution of different compounds cannot be resolved. The condensed (particulate) matter extinction appears as a continuous background that decreased with wavelength according to  $\lambda^{-1}$  law typical of soot particles in Rayleigh regime [21, 22]. As a consequence, above 700 nm the extinction coefficient can be only due to the soot extinction.

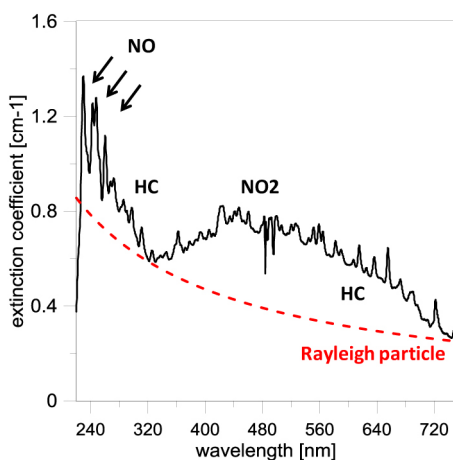


Figure 3. Typical extinction spectrum detected at the engine exhaust

Figure 3 reports a typical extinction spectrum detected for G100 single injection in stoichiometric condition from UV (220 nm) to near-IR (750 nm). All the spectra are featured in different proportions by the band systems of NOx, hydrocarbons ( $\lambda < 260$  nm and  $500\text{nm} < \lambda < 600\text{nm}$ ). These extinctions were overlying on the continuous background of the soot particle. The  $\lambda^{-1}$  curve typical of soot particle in the Rayleigh regime is reported as a dotted line in Figure 3.

### Optical Set-Up for In-Cylinder Measurements

Cycle-resolved imaging and UV-visible chemiluminescence were applied to visualize the flame inception and growth in the combustion chamber. Synchronisation of various control triggers for ignition, injection and cameras was achieved using the optical encoder mounted on the crankshaft as an external clock connected to the AVL Engine Timing Unit.

UV-visible natural emission spectroscopy allowed to follow the evolution of radiative chemical species involved in the combustion process from the spark ignition until the late combustion phase. During the spectroscopic investigations, the radiative emission from the combustion chamber was focused by a 78 mm focal length, f/3.8 UV Nikon objective onto the 250  $\mu\text{m}$  micrometer controlled entrance slit of a spectrometer with 150 mm focal length and 300 groove/mm grating (central wavelength 350 nm). From the grating, the radiations were detected by an intensified CCD camera (array size of  $1024 \times 1024$  pixels with a pixel size of  $13 \times 13 \mu\text{m}$ , 16-bit pixel digitization and 1MHz sustained repetition rate). During the UV-visible chemiluminescence measurements, the objective was directly coupled with the ICCD. No filters were used in order to capture broadband light emission. The intensified camera was employed in full-chip configuration. The set-up allowed a resolution of 90  $\mu\text{m}$  per pixel.

For UV-visible spectroscopy and visualization, ICCD sequential gating mode was used: it allowed to collect one spectrum and/or frame per cycle with fixed gate width but variable delay with respect to the trigger (i.e. falling edge of the ignition coil charge signal). Optical data were recorded for 100 consecutive cycles, at a step of 1 CAD crank angle (83  $\mu\text{s}$ ), with an exposure time of 42  $\mu\text{s}$ .

Spectroscopic investigations were carried out in the central region of the combustion chamber. The binning of 920 spectra was considered. The emission spectra were corrected for the wavelength dependent sensitivity of the optical devices. Wavelength calibration was performed using a mercury lamp. Spectroscopic data post-processing allowed to evaluate the time evolutions of chemical species featured by the combustion process in all the operative conditions. It should be noted that all the detected species through emission spectroscopy are in excited electronic states; for this reason the notation “\*” that identifies the excited species will be omitted. In this work, for each individual-cycle spectrum, OH emission was evaluated at 309 nm 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. Soot precursors evolution was evaluated at 490nm considering the tail of blackbody carbon emission.

A high-speed CMOS camera (Optronis CamRecord5000, 8-bit,  $16 \times 16 \mu\text{m}$  pixel size) equipped with a 50 mm Nikon objective was used for cycle-resolved imaging. Specifically, the camera was employed at a frame rate of 5 kHz, corresponding respectively to 2.4 CAD between frames at 2000 rpm, for image sizes of  $512 \times 512$  pixels. The set-up gave a resolution of 175  $\mu\text{m}$  per pixel.

The two digital imaging methodologies were applied to obtain a more complete characterization of the flame front propagation. The ICCD ensured a higher signal to noise ratio than the CMOS camera even for

flames with reduced luminosity (such as was the case for lean operation). This occurred as a consequence of the high quantum efficiency of the intensified detector in the UV spectral range (from 250 nm to 400 nm). Since the exothermal reactions typical of the first stages of the combustion process and of flame front outline are characterized by radiative emission in that UV wavelength range, the ICCD camera is particularly well suited for flame kernel optical investigations. Moreover the ICCD flame area resulted larger than CMOS cycle resolved one as shown in Fig. 4.

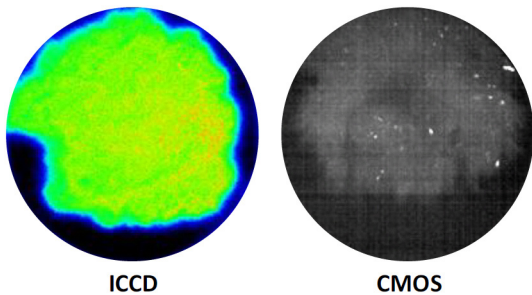


Figure 4. Comparison between ICCD and CMOS acquisition of flame at 22 CAD ASOS with the engine operated at 2000 rpm, WOT, stoichiometric air-fuel gasoline mixture and 15 CAD BTDC spark timing.

### Image Processing

In order to obtain quantitative information from the combustion visualization, macroscopic parameters related to flame morphology were evaluated via thresholding/binarisation of each flame image [26, 27]. All image sequences detected by CMOS as well as ICCD were downloaded as movie .avi file. Ad hoc image processing methodologies were performed for each visualization technique by Vision Assistant 2011 of National Instruments. The procedures allowed to operate with the movie files, thus reducing calculation time for each 100 frame acquisition to less than 10 seconds.

For cycle resolved imaging, as a first step, an appropriate circular mask was fixed in order to cut the spurious light from reflections at the boundaries of the circular window of the piston crown. Then the contrast and brightness of the images were adjusted fixing the brightness at 180, contrast at 64 and the gamma value at 1.0. After these preliminary steps, a look-up table (LUT) transformation was applied for each pixel through the power function

$$f(\text{pixel}) = \text{pixel}^{1/1.5}$$

Then thresholding was performed by fixing the minimum image intensity at 64 on 256. Thresholding segmented the image into two regions, foreground and background respectively, obtaining a binarized image. After thresholding, morphological transformations were applied to fill holes and remove small border objects. For each frame, the program calculated the flame areas as the “projected” in-cylinder enflamed area by summing up the pixels of binarized images. Other parameters such as the Waddel diameter and centroid coordinates were also evaluated. The Waddel disk diameter is the diameter of a disk with the same area as the particle. The geometric centroid of area for each flame was identified in x and y image coordinates and calculated for the duration of flame kernel growth.

ICCD image processing was similar to the procedure applied for CMOS cycle resolved visualization. ICCD sequences were converted in 8-bit grey-level by extracting the intensity plane. After applying the circular mask, contrast and brightness were adjusted by fixing the brightness at 180, contrast at 64 and the gamma value at 1.01. After this step, as in the previous methodology, a LUT power function  $1/x$  with  $x=1.5$  was applied to each pixel.

Contrary to the procedure applied for analyzing the CMOS data, automatic thresholding was performed. Compared to manual thresholding, this methodology did not require setting minimum and maximum image intensities by the user. In this work the “metric” method was used [27, 28]. For each threshold, a value determined by the surfaces representing the initial gray scale is calculated. The optimal threshold corresponds to the smallest value. In details, the threshold value is the pixel value  $k$  at which the following expression is minimized:

$$\sum_{i=0}^k h(i)|i - \mu_1| + \sum_{i=k+1}^{N-1} h(i)|i - \mu_2|$$

where

$\mu_1$  is the mean of all pixel values in the image that lie between 0 and  $k$ ,  $\mu_2$  is the mean of all pixel values in the image that lie between  $k+1$  and  $N=255$ . After the thresholding, morphological transformations were applied to fill holes and remove small border objects. For each binarised frame, the enflamed area and Heywood Circularity Factor were evaluated. The last parameter refers to the perimeter divided by the circumference of a circle with the same area. The closer the shape of a particle is to a disk, the closer the Heywood circularity factor is to 1.

### In-cylinder Pressure Data Processing

Overall performance was evaluated by calculating indicated mean effective pressure (IMEP) and its coefficient of variation (COV), as well as considering peak pressure and the location of 50% mass fraction burned (MFB); all four parameters were calculated based on data averaged over 200 consecutive fired cycles, chosen by considering the COV and the fact that quasi steady-state was achieved during measurements [29].

In order to provide a general view of thermodynamic processes that were taking place inside the cylinder for the investigated conditions, pressure traces were analysed employing a simplified heat release approach [30]. The rate of heat release (ROHR) was calculated using the following equation,

$$dQ = \frac{\gamma}{\gamma - 1} \cdot p \cdot dV + \frac{1}{\gamma - 1} \cdot V \cdot dp$$

where  $Q$  is the net heat released through fuel oxidation measured in J,  $\gamma$  the ratio of specific heats,  $p$  in-cylinder pressure in Pa and  $V$  cylinder volume in  $\text{m}^3$ . A constant value of 1.35 was set for  $\gamma$ ; even though this simplification introduces additional uncertainty in the evaluation of different phenomena [31]. For the scope of this research the influence on results interpretation should be minimal, given the

similarities between the properties of the working fluid (even when switching from gasoline to butanol), also considering the fact that only qualitative comparisons were sought after. For a closer correlation of the evolution of thermodynamic parameters during combustion and flame development, volume fraction burned (VFB) was also calculated for each condition using the following equation,

$$VFB = \frac{1}{1 + \frac{1 - MFB}{MFB \cdot \frac{\rho_u}{\rho_b}}}$$

where MFB was calculated as  $MFB = (Q - Q_{spark}) / (Q_{EVO} - Q_{spark})$ , with exhaust valves opening (EVO) considered as the point of combustion completion and the ratio of unburned-burned gas density  $\rho_u / \rho_b$  set as 4 [30].

## RESULTS

### Thermodynamic Analysis

As stated in the 'Introduction' section, one of the main features of alcohol fuels is their higher laminar flame speed compared to gasoline, a property which results in improved performance of SI engines. Latent heat of evaporation, on the other hand, can become an issue, given the increased heat flux required for proper air-fuel mixture preparation. At low load, no shortcomings were identified with pure butanol fuelling for the engine used in this study [32], by using the same injection strategy, namely single injection initiated at 300 CAD BTDC; this SOI setting is the one employed by the commercial power unit from which the cylinder head was taken and it ensures the best compromise between air velocity during injection and fuel jet-piston crown impact. However, when using the same strategy during WOT operation, a slight drop in IMEP was noted when switching from gasoline to butanol (Figure 5). This effect was even more pronounced for lean fuelling. Given that the major difference between the two fuel types (besides the stoichiometric air-fuel ratio which requires increased injection duration for the alcohol) is the evaporative behaviour (i.e. latent heat of vaporization and saturation or dew point), a change in injection strategy was investigated, so as to identify a way to overcome the aforementioned shortcoming.

Therefore, split injection (by dividing it in two separate events with equal amounts of fuel delivery) was employed for each fuel in order to provide improved insight into the effect of such changes; the equivalent spray-piston crown interaction as for single injection starting at 300 CAD BTDC would have been obtained for the second injection ending at 60 CAD BTDC. This strategy was found to feature numerous slow burn cycles and even misfires, most likely due to the fact that the fuel was not properly vaporized before ignition.

Therefore, the results of CFD simulations [14] were used for choosing two SOI points for the second injection, with the first one fixed at 300 CAD BTDC; these points were SOI300\_120 and 300\_140 CAD BTDC. The last one is the optimal setting with lean

operation at a relative air-fuel ratio of 1.15. The stoichiometric points were found to feature reduced sensitivity (i.e. variations in engine output, with differences lower than  $\pm 2\%$ ) to the injection strategy; nonetheless, a slight improvement in IMEP was noted for butanol when splitting the injection in two separate events. Reduced differences were recorded for lean fuelling as well when using gasoline; for butanol, on the other hand, an improvement of around 10% for both SOI300\_120 and SOI300\_140 cases was obtained (as displayed in figure 5).

While the effect of injection splitting can be attributed to both fuel evaporation and fuel distribution inside the combustion chamber, the fact that the significant improvement was recorded for lean fuelling seems to point to the second parameter, which can have an important effect and emphasize cycle-to-cycle variations, especially in the initial phase of combustion [33]. This latter conclusion is also suggested by the COV values recorded for all cases; injection splitting consistently reduced cyclic variability, with a more pronounced effect during lean operation. In fact, the modified strategy resulted in more stable operation with the alcohol compared to gasoline at  $\lambda \approx 1.15$ . Peak pressure was found to be comparable for all stoichiometric cases, and again, for lean fuelling, the double injection strategy resulted in higher values especially for butanol.

With regard to combustion phasing, the crank angle of MFB50% showed relatively reduced variation at changes in injection strategy; a consistent reduction of this parameter was obtained only for the higher relative air-fuel ratio. Judging from the analysis of the overall combustion indicators, it can be concluded that injection splitting can bring benefits in performance and stability, more so with lean fuelling and for the alternative fuel.

A more detailed analysis, based on in-cylinder pressure traces, reveals that changes in injection strategy result in the variation of combustion phasing (Figs. 6 and 7). Slightly lower pressure was recorded during compression, before ignition, when employing double injection as compared to the case for a single event. This can be attributed to a shift in fuel evaporation timing, since half of the injected quantity entered the cylinder later during the cycle. Both split injections sped up the initial phase of combustion; the effect was more pronounced for SOI300\_140, which resulted in higher peak pressure compared to the SOI300\_120 case for gasoline as well as butanol. The SOI300\_120 strategy featured lower heat release rates than SOI300\_140, but in the final stages of combustion this parameter increased for both fuel types. Butanol combustion was found to be more sensitive (i.e. larger variations in the rate of heat release) during lean operation for which the SOI300\_140 setting resulted in a significant shift of heat release, advancing the peak by more than 7 CAD as compared to the single injection case. This seems to indicate that injection splitting changes the charge stratification, providing more fuel in the central region of the combustion chamber, thus ensuring increased heat release during flame development rather than the final stages of the process.

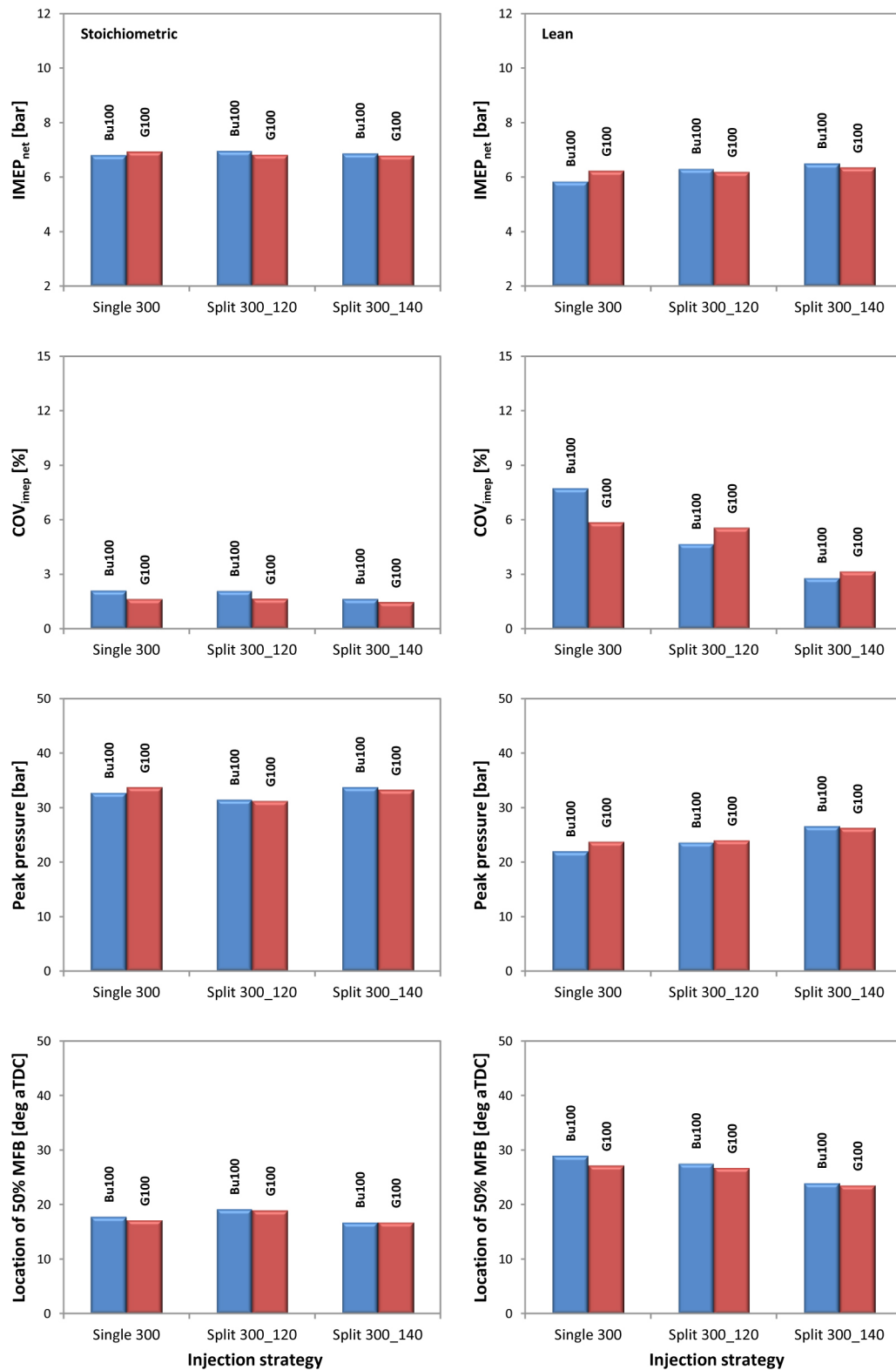


Figure 5. Average thermodynamic parameters for stoichiometric (left) and lean (right) conditions, with gasoline and butanol fueling.