

Butanol-Diesel Blend Spray Combustion Investigation by UV-Visible Flame Emission in a Prototype Single Cylinder Compression Ignition Engine

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

ABSTRACT

The paper reports the results of an experimental investigation carried out in a prototype optically accessible compression ignition engine fuelled with different blends of commercial diesel and n-butanol. Thermodynamic analysis and exhaust gas measurements were supported by optical investigations performed through a wide optical access to the combustion chamber. UV-visible digital imaging and 2D chemiluminescence were applied to characterize the combustion process in terms of spatial and temporal occurrence of auto-ignition, flame propagation, soot and OH evolution.

The paper illustrates the results of the spray combustion for diesel and n-butanol-diesel blends at 20% and 40% volume fraction, exploring a single and double injection strategy (pilot+main) from a common rail multi-jet injection system. Tests were performed setting a pilot+main strategy with a fixed dwell time and different starts of injection. For the diesel case, the whole amount of injected fuel and injection pressure were set at 22 mg/str and 80 MPa corresponding to a medium load regime for an automotive light duty diesel engine. The fuel amount for the butanol-diesel blends was changed to get the same chemical energy of the delivered fuel as the reference diesel case (935J/str). The investigation was carried out at two EGR rates, 0% and 50%, corresponding to a concentration of O₂ at intake of 21 and 17%, respectively.

Taking advantages of the higher resistance to auto ignition of the butanol-diesel blends, the results showed a transition from the conventional to a partial premixed combustion for the BU40 in the single injection case. The combined effect of two-stage injection and EGR increased the ignition delay and the switch to the partial premixed combustion was enhanced. Improvement in NO_x-soot trade-off was obtained activating the double injection strategy particularly with the butanol blends.

Optical diagnostics allowed to detect and feature the spatial distributions due to the flame luminosity and OH emission at autoignition and to estimate of the lift-off length. A high concentration of OH radicals was detected for all the test cases without EGR that corresponded to an enhanced oxidation phase and a fast soot reduction. Further, the soot oxidation phase resulted more significant for the higher butanol volume fraction in the blends.

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INTRODUCTION

The improvements achieved in the last decades for Diesel engines allowed to reach high thermal efficiency and specific power output, with a drastic reduction of exhaust emissions [1][2][3][4]. In spite of the significant technology progress, the increasingly stringent regulations focus the automotive research attention to further reduce exhaust emissions, with particular care for PM and NO_x. One of the most effective solutions is the use of exhaust gas recirculation (EGR) and after-treatment devices, with a slight penalty concerning the fuel efficiency [5]. On the other hand, the growing demand for reducing the greenhouse gas emission levels imposes a reduction of consumptions for lowering CO₂. Hence, it is necessary to develop advanced combustion mechanisms such as HCCI, PPCI and LTC, for both improving fuel economy and reducing exhaust emissions [6]. Because

of the improved air-fuel mixing, these combustion modes allow to reduce the local temperature peak with a simultaneous reduction of NO_x and PM levels, without compromising the thermal efficiency.

The HCCI (Homogeneous Charge Compression Ignition) mode provides the combustion of lean homogenous air/fuel mixture, totally removing local high-temperature and/or rich-fuel zones and limiting the formation of both NO_x and PM [7][8][9]. With Partially Premixed Compression Ignition (PPCI) combustion, the mixture is not homogenous, but the injection is complete before the start of combustion. As a result, the local fuel - air equivalence ratio is lower with a consequent reduction of local maximum temperature [10][11].

Similarly to PPCI, the Low Temperature Combustion (LTC) provides more uniform air/fuel mixture with respect to conventional combustion mode. The combined use of EGR rate, injection timing and high turbulence levels allows to control the in-cylinder temperature [12][13]. The high oxygen dilution increases the combustion period, with a reduction of in-cylinder pressure peak and thus temperature peak.

The joint effect of injection advance and turbulence improves the air fuel mixing limiting local rich zones and combustion temperatures, with a reduction of both NOx and soot. Moreover, the low oxygen level lowers the adiabatic flame temperature, keeping the in-cylinder temperature below the NOx activation critical point [6][14].

The mixture temperature plays a key role for LTC and affects the combustion and exhaust emissions. In fact, the combustion temperature strongly depends on the in-cylinder temperature at the ignition start. Adjusting the mixture temperature allows to obtain a maximum combustion temperature higher than 1500 K for a complete oxidation from CO to CO₂ and, at the same time, lower than the critical temperature of NOx formation [6].

However, an accurate control of the mixture temperature is not possible since the ignition in the CI engines occurs spontaneously. Generally, the EGR can be adjusted for influencing the ignition delay. Thus, it is possible set mixture temperature at the ignition. A high level of EGR, combined with a proper injection advance, is necessary for leaving a mixing time large enough for LTC conditions. On the other hand, too high EGR levels can induce high PM formation and a penalty on the fuel consumption [15] [16].

An alternative solution is blending diesel with fuel featured by a lower cetane number to reduce CO and unburned products, improve fuel economy and combustion phasing in LTC mode [17][18]. Gasoline blended with diesel was demonstrated to increase the ignition delay and reduce HC and CO emissions; moreover soot and NOx emissions simultaneously decreased without significantly reducing in local combustion temperatures which was essentially necessary in LTC strategies [17][19].

Recently, great interest has been devoted in use of butanol as a transportation fuel. Butanol can be produced from biomass (as "biobutanol") as well as fossil fuels without difference in the chemical properties [20][21]. Butanol is a 4-carbon alcohol obtained from the same feedstock as ethanol including corn, sugar beets, and other biomass feedstock containing cellulose that could not be used as food and would instead go to waste [22][23]. The n-butanol has more advantages than ethanol, such as higher energy content, lower water adsorption and corrosive properties [24]. Moreover it is less volatile, has better solubility with diesel allowing to avoid the additive use; its viscosity is similar to diesel fuel and it has a relatively high cetane number.

The n-butanol was successfully tested as additive for diesel in c.i. engines operated in both conventional combustion and LTC mode [25][26][27]. Various studies in a high-speed DI diesel engine for passenger-car application, in which diesel fuel was mixed with butanol up to 40%vol, demonstrated that butanol-diesel blends

increased combustion pressure and burn rate while having little impact on maximum power and torque. Blended fuels slightly increased BSFC with a large increase in thermal efficiency. HC emissions increased as the butanol concentration increased, and at low loads, CO emissions increased while NOx emissions decreased with the increase of butanol in the blend ratio. However, at high loads, CO emission decreased and NOx emissions increased. Soot emissions were reduced at all conditions and less smoke was produced with higher butanol concentrations [28][29][30].

Regarding the n-butanol/diesel blends fuelling under LTC mode, trials on light-duty engine demonstrated that the combined influence of lower cetane number and higher volatility of n-butanol blends reduced pollutant exhaust emissions compared with neat diesel fuel with a slight penalty on fuel consumption [26]. The results were strongly dependent on the selected fuel injection strategies due to the great influence of fuel-air mixing.

It is consolidated that pilot injection increases the in-cylinder temperature and shorten the ignition delay of main injection, thus leading to a reduction in peak of premixed combustion and combustion noise. Further, as consequence of pilot injection, the main injection spray entrains less air because the inclusion of pilot injection products.

Even if double injection strategies have been widely investigated for diesel blended with biodiesel and ethanol, a deeper knowledge of phenomena related to fuel spray combustion is necessary, when high volume fractions of n-butanol in diesel are used [31][32][33].

To this aim, the present paper discusses the results related to the spray combustion characterization of n -butanol blending in diesel (20% and 40%) within a prototype single cylinder high swirl compression ignition engine equipped with a common rail multi-jet injection system. A double stage injection strategy with EGR of 50% was tested in order to study the attained PPCI regime. Thermodynamic analysis and exhaust pollutant measurements were performed simultaneously to UV-visible digital imaging and chemiluminescence in the optically accessible combustion chamber. 2D chemiluminescence was detected through a flame emission filtering at 310 and 690 nm to follow the OH and soot evolution.

EXPERIMENTAL SET-UP AND TEST CONDITION

The experiments were performed in a single cylinder 2-stroke compression ignition engine. The main engine specifications are reported in [Table 1](#). An external high swirl optically accessible combustion bowl (50 mm in diameter and 30 mm in depth), equipped with a high pressure common rail injection system, is connected to the main cylinder through a tangential duct. A compressed air flow is forced within the bowl as the piston approaches TDC. As result, a counter clockwise swirl flow is obtained, reproducing the fluid dynamic environment similar to a real direct injection diesel engine, with the rotation axis coincident to the symmetry axis of the chamber (cfr. [figure 1](#)).

The injector was mounted within the combustion bowl, with its axis coincident to the chamber axis. The fuel is sprayed within the swirling air motion and the combustion process mainly develops within the chamber. After TDC, as the piston moves downward, the flow reverses its motion and the hot gases flow through the tangential duct to the cylinder and finally to the exhaust ports.

Table 1. Engine specifications

Prototype Single Cylinder CI Engine	
Cylindrical Bowl (mmxmm)	50x30
Bore (mm)	150
Stroke (mm)	170
Connecting Rod (mm)	360
Compression ratio	10.1:1
Engine speed (rpm)	500
Air supply	Roots blower
Abs. intake air pressure (MPa)	0.107
Bosch Injector nozzle	7/0.136/148°

The combustion chamber provides two optical accesses: the first one is circular (50 mm diameter) and it is placed in front of the injector, used to collect images; the second one has a rectangular shape (size of 10 × 50 mm), is mounted on a side of the chamber, at 90° with respect to the bowl axis, used for the laser illumination input.

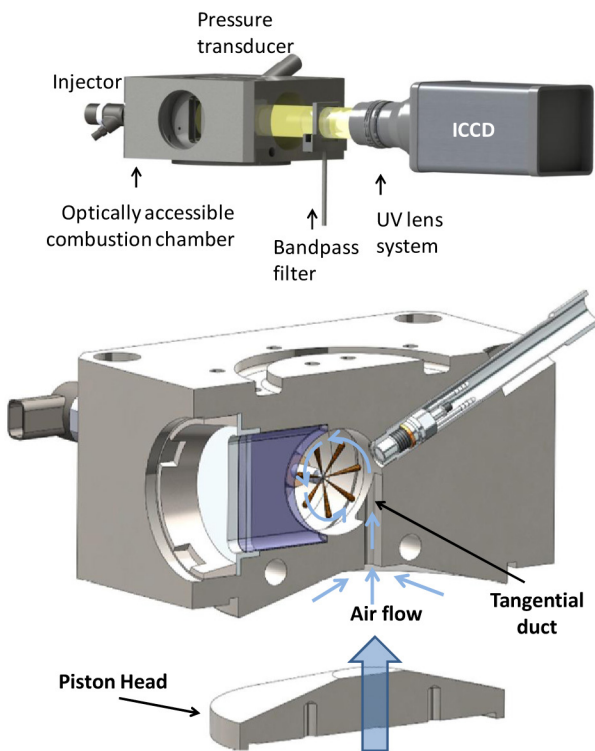


Figure 1. Sketch of the experimental apparatus for the optical investigations and cross section of the optically accessible combustion chamber.

The injection equipment includes a common rail injection system with a solenoid controlled injector located on the opposite side of the circular optical access. The nozzle is a micro-sac 7 hole, 0.136 mm diameter, 148° spray angle nozzle. An external roots blower provided an intake air pressure of 0.107 MPa with a peak pressure within the

combustion chamber of 4.3 MPa under motored conditions. The engine was warmed-up by an external heat exchanger system to keep constant the engine thermal regime during the tests.

A crank angle encoder signal synchronized the optical detector and the engine, through a delay unit. The AVL Indimodul recorded the TTL signal to monitor optical acquisitions together with the signal acquired by the pressure transducer. In this way, it was possible to determine the crank angles where optical data were detected.

Results of the in-cylinder pressure were computed averaging 300 consecutive engine cycles. Exhaust gaseous emissions were acquired by the AVL DiGas 4000 analyzer for NO_x (1 ppm resolution), smoke was measured by a part flow opacimeter (AVL Opacimeter 439). The opacity was correlated to the particulate mass concentration through an empirical formula [34].

Table 2. Fuel Properties

Fuel	BU00	BU20	BU40
Density @15° C [kg/m ³]	840	830	826
Cetane Number EN ISO4264	52	44	36
Net Heat Value [MJ/kg]	42.5	39.7	36.9
Carbon content [%]	87.0	80.2	75.2
Hydrogen content [%]	12.6	15.4	16.8
Oxygen content [%]	-	4.32	8.64
IBP [°C]	160	139	117
Distillation 50% Vol. [°C]	280	250	226
Distillation 90% Vol. [°C]	338	328	325

Tests were carried out comparing the reference commercial diesel fuel to blends of diesel and n-butanol at two different volume fractions, 20% and 40%. Two injection strategies were tested: a single and a double injection (pilot+main). A sweep of the start of injection was explored from 14 to 2 CAD BTDC with a step of 3 CAD for the single injection case. The same injection timing was set for the main injection throughout the double strategy and the pilot was advanced 7 CAD before the start of the main: the dwell time between pilot and main was kept at 7 CAD. These conditions and the amount of injected fuel were selected from the ECU data of a commercial 4-stroke light duty diesel engine under investigation to examine the effect of diesel-butanol blends on performance and emissions [26].

Table 3. Main operating conditions

Injection pressure	80 MPa
Fuel energy content (Diesel, Bu20, Bu40)	935 J/str
Motored TDC gas density	20.8 Kg/m ³
O ₂ concentration at intake	21, 17%
SOIs for single injection strategy [CAD BTDC]	14,11,8,5,2
Dwell angle between Pilot and Main starts	7 CAD

Tests were carried out at two EGR rates, 0% and 50% corresponding to a concentration of O₂ at intake of 21 and 17%, respectively. The motored TDC gas density was estimated at 20.80 kg/m³ by the in-cylinder pressure trace considering a polytropic coefficient of 1.328. The main operating conditions are reported in Table 3. In the diesel case, the whole amount of injected fuel and injection pressure

were set at 22 mg/str and 80 MPa, respectively, corresponding to a medium load regime for an automotive light duty diesel engine. The injection interval for the single injection strategy was 867 μ s, corresponding to 2.6 CAD, while an interval of 333 μ s and 833 μ s, corresponding to 1 and 2.5 CAD for pilot and main injection, respectively, was set for the double injection strategy.

The pilot duration was adjusted in order to deliver about 10% of the whole injected fuel mass. The injected fuel mass was calibrated by off-line fuel amount measurements keeping constant the temperature during the tests by a heat exchanger. Engine tests were carried out at constant chemical energy of delivered fuel, therefore the injection duration was set to achieve, for all the investigated blends, the same energy content as the reference diesel (935 J/str). The main characteristics of tested fuels are reported in [table 2](#).

OPTICAL MEASUREMENTS AND DATA PROCESSING

The spray combustion process was recorded through the quartz window using an intensified CCD camera (PiMAX 3) with a 78 mm focal length lens (Nikkor). The camera was featured by a 1024×1024 pixels sensor with 13×13 μ m pixel size, 16-bit pixel digitization and 1MHz sustained repetition. The ICCD was employed in full-chip configuration. The set-up allowed a resolution of 83 μ m per pixel. In addition to the broadband acquisition that exploits the high quantum efficiency of the ICCD in the spectral range 250-700 nm, two 50×50mm band-pass filters (Asahi Spectra Inc.) were used to selectively record flame emissions at 310 nm and 690 nm.

The high transmission (65%) and small width (10 nm FWHM) of the UV filter permitted to select the OH emission due to the A-X transition without interference of other radiative species [\[35\]\[36\]\[37\]](#).

Since the soot luminosity is characterized by a continuous broadband spectrum similar to the Planck black body curve [\[35\]\[38\]\[39\]](#), 690nm band-pass filtering allowed to separate the soot emission by the other excited species contribution that emitted in the visible wavelength range as CH and C₂. ICCD sequential and repetitive gating modes were used. Sequential acquisition collected one frame per cycle with fixed gate width but variable delay with respect to the trigger (i.e. raising edge of the injection signal). Repetitive mode detected one frame per cycle with fixed gate width and delay with respect to the trigger. For both acquisition modes, optical investigations were carried out on 100 consecutive engine cycles. In the sequential acquisitions, data were recorded at a step of 133.3 μ s (0.4 CAD @500rpm). Without spectral filtering, the camera would be overexposed due to the flame luminosity. Thus, ICCD exposure time was fixed at 16.66 μ s (0.05 CAD). Chemiluminescence investigations were performed setting the exposure time at 66.66 μ s (0.2 CAD).

In order to obtain quantitative information from the combustion visualization, “ad hoc” image processing was developed through Vision Assistant 2011 of National Instruments [\[40\]](#). [Figure 2](#) sketches the steps of the image processing for two selected images with different levels of luminosity. In detail, an appropriate circular mask was applied in order to cut the spurious light from reflections at the

boundaries of the circular window crown ([Fig. 2a](#)). At this step, for each image, with and without filtering, the spatially integrated luminosity was evaluated. Successively, false-colour 16-bit images were converted into 8-bit grey-level images and a look-up table (LUT) transformation was applied to extract the HSV value plane as shown in [Fig. 2b](#). After this preliminary procedure, a thresholding was performed. The images were segmented into two regions, flame foreground and background respectively, obtaining a binarized image ([Fig. 2c](#)). Given that a manual or an automatic thresholding can be made, the automatic methodology was chosen because it did not require setting minimum and maximum image intensities by the user.

In this work “clustering” threshold method was applied for images with and without band-pass filtering [\[41\]\[42\]\[43\]](#). Clustering method is based on selecting a threshold for separating the image into two classes so that the between-class variance is maximized. More details regarding the methodology are reported in [\[44\]](#).

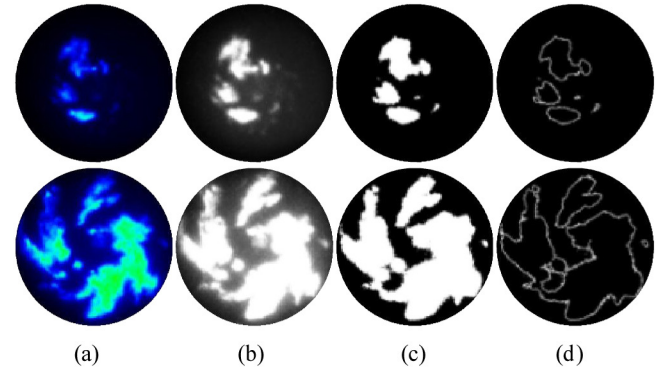


Figure 2. Sketch of the image processing steps applied to two selected images.

Assuming an image is represented in L grey levels [0,1,...L-1], the number of pixels at level i is denoted by n_i, and the total number of pixels is denoted by N. The probability p_i of gray level i is denoted by

$$p_i = \frac{n_i}{N}; p_i \geq 0; \sum_{i=0}^{L-1} p_i = 1$$

In the thresholding method, the pixels of image are divided into two classes C₁ with grey levels [0, ..., t] and C₂ with gray levels [t+1, ..., L-1] by the threshold t. The gray level probability distributions for the two classes are

$$w_1 = \Pr(C_1) = \sum_{i=0}^t p_i; \quad w_2 = \Pr(C_2) = \sum_{i=t+1}^{L-1} p_i$$

the means of class C₁ and C₂ are

$$u_1 = \sum_{i=0}^t \frac{ip_i}{w_1}; \quad u_2 = \sum_{i=t+1}^{L-1} \frac{ip_i}{w_2}$$

The total mean of gray levels is denoted by u_t

$$u_t = w_1 u_1 + w_2 u_2$$

The class variances are

$$\sigma_1^2 = \sum_{i=0}^t \frac{(i-u_1)^2 p_i}{w_1}; \quad \sigma_2^2 = \sum_{i=t+1}^{L-1} \frac{(i-u_2)^2 p_i}{w_2}$$

The within -class variance is



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$$\sigma_W^2 = \sum_k w_k \sigma_k^2$$

The between-class variance is

$$\sigma_B^2 = w_1(u_1 - u_T)^2 + w_2(u_2 - u_T)^2$$

The total variance of gray levels is

$$\sigma_T^2 = \sigma_W^2 + \sigma_B^2$$

Otsu's method chooses the optimal threshold “ t ” by maximizing the between-class variance. After thresholding, small border objects with size smaller than 2 pixels were removed through a morphological function and the luminosity region contours were extracted.

The coordinates of pixels in luminosity region contours were evaluated at the autoignition for all the conditions and images, with and without filtering. In detail, the contour with the strongest edge in the search region was selected and coordinates of pixel in extracted contour were stored. By processing the 100 images detected in the repetitive acquisition mode, the occurrence map was built. [Figure 3](#) shows the autoignition map for Diesel single injection retrieved from UV-visible flame image trial.

Autoignition maps were used to evaluate the lift-off length related to flame, OH and soot. Commonly, the lift-off was defined as the position closest to the injector where significant luminosity (due to emission or fluorescence) or light extinction was observed [45], [46], [47], [48], [49], [50], [51]. In the most of cases, the analysis was restricted to a cone around the spray axis. In this work, the autoignition luminosity occurrence allowed evaluating the probability density function (PDF) through the distribution of distances from the injection tip for each operative condition.

PDFs were represented as 50 interval histograms. Mean was considered as the average lift-off length. Moreover, variance and skewness factor (SkF) were evaluated [52][53]. The skewness quantifies the distribution shape; specifically, symmetrical distribution has a skewness of zero while an asymmetrical distribution with a long tail to the right (higher values) has a positive skew. This induces a mean value bigger than mode value, as shown in [Figure 4](#).

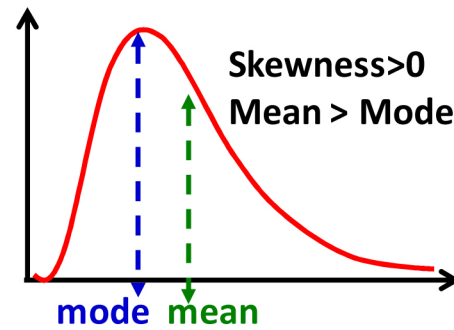


Figure 4. Sketch of central tendency and symmetry of a distribution

Histograms and related statistical values were calculated through a routine developed through Labview 2011 of National Instruments. [Figure 5](#) shows the PDFs obtained considering the UV-visible emissions due to a single injection strategy for Diesel without EGR and BU40 with EGR. The proposed methodology allowed to obtain spatially averaged lift-off values and to overcome difficulties related to high-swirl that redistributed the flame with respect to the jet axis.

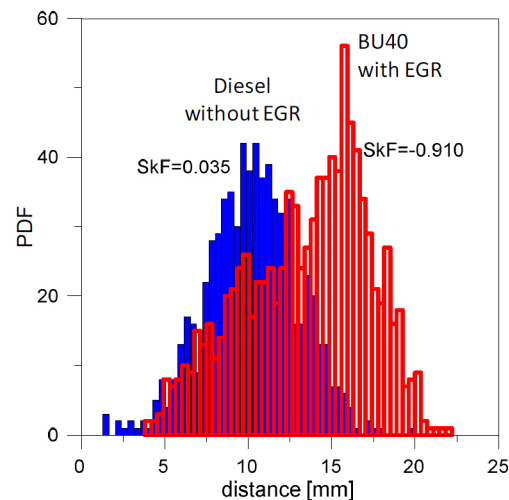


Figure 5. PDFs from the UV-visible emissions at single injection strategy for Diesel without EGR and BU40 with EGR; the skewness factors are labelled.

RESULTS AND DISCUSSIONS

Combustion and Emission Measurements

Preliminary tests were carried out to explore the effect of injection timing and fuelling on engine performance and exhaust emissions. [Figure 6](#) shows the averaged combustion pressure traces measured on 300 engine cycles for diesel fuel at different starts of injection, with 0 and 50% of EGR (EGR valve opening) corresponding to an inlet O_2 concentration of 21 and 17% in volume, for single and double injection strategy.

In order to estimate the effective ignition delay, the start of combustion was evaluated as a relative maximum of the first derivative of pressure signal. The ID was defined as the difference between the SOC occurrence and the electronic SOI. These measurements were particularly useful for investigating the pilot ignition at the most advanced SOI, even though the effective pilot combustion timing could be obtained only through optical analysis.

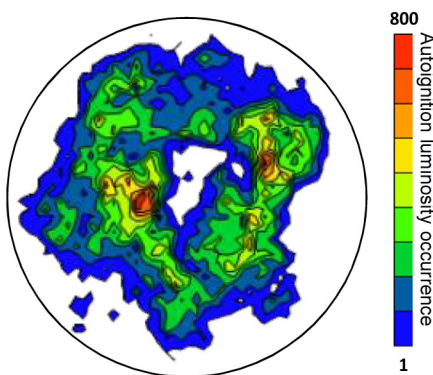


Figure 3. Autoignition occurrence map for diesel single injection evaluated from UV-visible images.