

Non-Intrusive Investigation in a Small GDI Optical Engine Fuelled with Gasoline and Ethanol

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ABSTRACT

The aim of this paper is the experimental investigation of the effect of direct fuel injection on the combustion process and pollutant formation in a spark ignition (SI) two-wheel engine. The engine is a 250cc single cylinder, four-stroke spark-ignition firstly equipped with a four-valve PFI head and then with GDI one operating with European commercial gasoline and Bio-ethanol. It is equipped with a wide sapphire window in the bottom of the chamber and quartz cylinder. In the combustion chamber, optical techniques based on 2D-digital imaging were used to follow the injection and flame propagation and spectroscopic measurements were carried out in order to evaluate the main radical species. Radical species such as OH and CH were detected and used to follow the chemical phenomena related to the fuel quality. Measurements were carried out at different engine speeds and combustion strategies based on different injection pressures. They were chosen in order to allow both optical measurements and to test the engine in real conditions.

INTRODUCTION

Two-wheel and three-wheel vehicle have been expanding rapidly over the past several years, especially in the urbanized areas of Asia and South of Europe [1]. This is due to smaller physical size of motorcycles that allows for easier navigation in heavily congested areas. Moreover, purchase and maintenance are substantially lower than the corresponding costs for even small automobiles.

Asia represented 85% of the new motorcycle market in 2004 and most of the bikes were in the 50-350 cc engine displacement category. On the other hand, the European new motorcycle market is about 2 million bikes per year. They emit substantial quantities of hydrocarbons (HCs), carbon monoxide (CO), and particulate matter (PM), which represent a limitation because they have significant adverse health effects and deteriorate environmental quality.

To address the serious pollution problems posed by two-wheel vehicles, a growing number of countries worldwide have implemented, or are in the process of implementing, motor vehicle pollution control programs. They are aimed at substantially reducing harmful emissions from spark-ignited two-wheel vehicles. An innovative technology could be the use of direct injection coupled with the use alternative/ biological fuel [2, 3, 4]. Though it is a well-established technology in automotive field [5], is not yet established in two wheels world. Direct injection (DI) would eliminate the problem of short circuit of fresh charging and would make the two wheels engines suitable to the pollutant normative reducing fuel consumption. Performances distance covered would be the same, keeping the favorable reduced report between the mass and the power of the vehicle. At the same time, the impulse to search and use Bio-ethanol fuels with respect to fossil one for these small engines must be considered. Considering these issues, the development gasoline direct injection (GDI) small engine fuelled with alcohol fuel could have simultaneously the advantages of higher thermal efficiency due to direct fuel injection and higher power output than conventional engines. In particular, GDI engines have also better transient response, more precise control of

the air-fuel ratio, an improvement of fuel economy, and a reduction of exhaust emissions thanks to globally lean stratified charge. Moreover, high compression ratios, due to the reduced possibility of knocking, lead to an improvement of the output performance using alcohol fuel [1, 3, 4]. Ethanol can be considered one of these, because it can be used as a fuel extender for petroleum-derived fuels, an oxygenate, an octane enhancer, and a locally produced pure fuel. Direct injection may therefore optimize pure ethanol-fuelled engines to a level of performance that exceeds gasoline engine efficiency, taking full advantage of ethanol's higher octane number and vaporization heat. In order to enhance these improvements, some non-intrusive measurements in the cylinder must be performed in order to assess the related chemical and physical events. Various experimental studies were carried out in optically accessible closed vessels, model combustion chambers, and rapid compression machines and only recently in optical engines in order to study the ethanol blends combustion process [6, 7, 8]. In particular, combined optical techniques were used in order to analyze the fuel spray distribution and evaporation, mixture preparation and self-ignition [6, 7]. Few works have been carried out in real engine using the ultraviolet-visible spectroscopy data, while combustion process involved in gasoline engine fuelled with mono-components fuel has been widely studied [5, 9]. The current study presents macroscopic imaging of spray and combustion and spectroscopic measurements obtained during tests carried out in a single cylinder optical engine. It was equipped with Direct Injection system and fuelled with a multi-components fuel, namely standard European commercial grade gasoline (RON 95) and Bio-ethanol. The Bio-ethanol was produced from grape pomace and has a water content of less than 1% and replies to the different specifications required by refineries in Italy and in Europe both for the direct blending and for ETBE's production. Several fuel injection pressures were tested. This results were compared with data obtained by the same engine equipped with port fuel injection system. Non-intrusive optical diagnostics were applied on both engine systems. Imaging and spectral measurements of the natural emissivity both in the visible and in the near UV were made. Simultaneous use of spectral emissivity and imaging measurements in UV-visible range have shown to be a powerful tool as well as high speed imaging of combustion process thanks to the presence in this region of the well-known "water bands" emitted by OH radical, CH, CN and C₂, HCO. Moreover, carbonaceous material and CO-O can be observed in this spectral range [10]. Finally, the in-cylinder pressure history and Heat Release (HR), as well as the flame-kernel radius growth and speed from image processing, are presented for the same engine operating conditions in order to compare the pure Bio-ethanol fuel with gasoline one.

Table 1. Chemical and physical properties for Bio-ethanol and gasoline fuel

Fuel property	Ethanol	Gasoline
Formula	C ₂ H ₅ OH	C ₄ to C ₁₂
Molecular weight[g/mol]	46,07	100–105
Carbon [mass%]	52,2	85-88
Hydrogen [mass%]	13,1	12-15
Oxygen [mass%]	34,7	2,7
Density, kg/l, 15/15 °C	0,79	0,72–0,775
Boiling point, °C	78	27–225
Vapor pres., kPa at 38 °C	15,9	48–103
Specific heat, kJkg ⁻¹ K ⁻¹	2,4	2
Viscosity, mPa s at 20 °C	1,19	0,37–0,44
Low. heating val., 10 ³ kJ/l	21,1	30–33
Autoignition temp., °C	423	257
Research octane number	108,6	98
Motor octane	92	87
(R+M)/2	100	92,5
Cetane	-	5-20
Flammability lim., Vol %	4,3/19	1,4/7,6
Water Tolerance, Vol %	Compl.miscible	Negligible
Stoichiometric air/fuel	9	14,7
Carbonyl [ppm] as C-O	567	-
Carbonyl [ppm] as acetone	1117	-
Carbonyl [ppm] as acetaldehyde	893	-
Sulphur [mg/kg]	< 0.8	10
Copper [mg/kg]	<0.1	-

EXPERIMENTAL APPARATUS

TRANSPARENT ENGINE

The engine used was an optically accessible single cylinder spark ignition (SI) engine. Details about the engine characteristics are reported in [table 2](#). It was equipped first with the cylinder head of two-wheel port fuel injection (PFI) SI engine, and then with a GDI prototype head, realized after a wide simulation study. The injector used in the PFI configuration was a commercial 3-holes. The GDI system was equipped with a prototypal 6-hole injector, located between the intake valves.

For both cases, the head had four valves and centrally located spark plug. The engine piston was flat and transparent through a sapphire window ([Figure 1a](#)). To reduce the window contaminations by lubricating oil, an elongated piston arrangement was used together with unlubricated Teflon-bronze composite piston rings in the optical section. In [figure 1c-b](#), the sketch of the bottom field of view of the combustion chambers, PFI and GDI respectively, is reported.

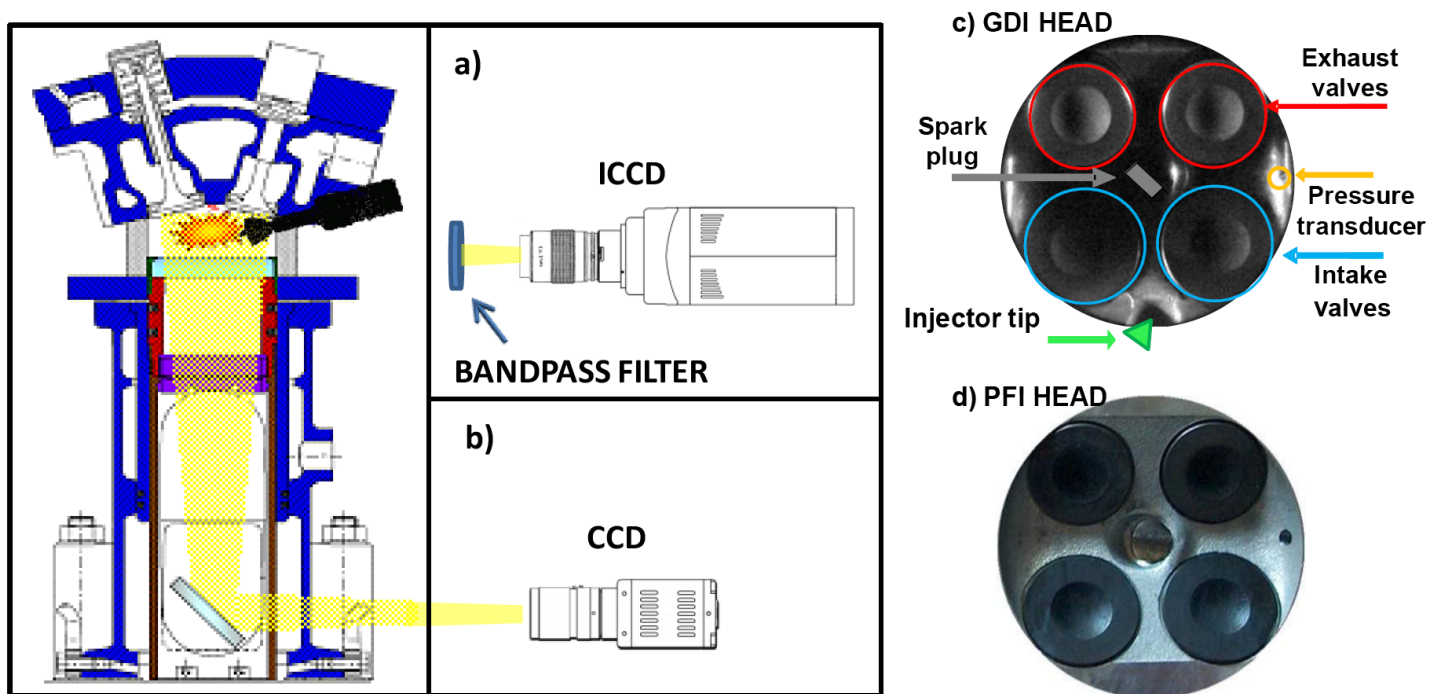


Figure 1. Optical setup for (a) radical measurement, (b) digital imaging, (c-d) field of view of the GDI-PFI combustion chamber.

A quartz pressure transducer was flush-installed in the region between intake-exhaust valves. Combustion pressure measurements were performed for all the selected operating conditions. The in-cylinder pressure, the rate of chemical energy release and related parameters were evaluated on an individual cycle basis and/or averaged on 400 cycles from the cylinder pressure data using conventional interpretation models [11].

Table 2. Specifications of the single cylinder spark ignition engine.

Displacement [cm ³]	250
Bore [mm]	72
Stroke[mm]	60
Connecting road [mm]	130
Compression ratio	10.5:1

OPTICAL APPARATUS

Figure 1a-b shows the experimental apparatus for optical investigations. The injection and combustion phases were detected through a wide sapphire located in the piston and they were reflected toward the optical detection assembly by a 45° inclined UV-visible mirror located in bottom of the engine. In order to characterize the injection phase an intense strobe lamp was used. The light from sapphire window was focused by a 78 mm focal length, f/3.8 UV Nikon objective onto the intensified cooled CCD camera

(ICCD). The ICCD had an array size of 512×512 pixels and 16-bit dynamic range digitization at 100 kHz. The ICCD spectral range spread from UV (180 nm) until visible (700 nm). The ICCD operated at a digitizer offset of about 230 counts, but the dark noise fluctuation in the background was much smaller, less than 50 counts. Dark noise and photon statistical noise were both smaller compared to the measured intensity. Radicals spatial distributions were acquired with the UV objective, ICCD camera, and band pass filters chosen at characteristic wavelength of OH (wavelength=309 nm), and CH (wavelength =431 nm) [10]. Filters had different transmission ratio 15% for OH, and 45% for CH.

2D flame visualization was obtained by digital CCD color camera system and a 50 mm focal length, f/3.8 Nikon lens (fig. 1b). The optical assessment allowed a spatial resolution around 0.1 mm/pixel. Spatial distribution of soot temperature and concentration was obtained by the two-color method. The soot emission wavelengths were selected by edge filters. This procedure allows estimating a KL-factor, which is proportional to the soot amount. More details about this methodology are reported in ref. [9, 12].

Finally, in order to evaluate the flame front speed, a retrieving procedure of the optical data was realized. The luminous signal locally in each point of the combustion chamber was evaluated processing each image. In this way, it was possible to identify the flame front, its mean diameter and then the flame front mean propagation velocity.

Table 3. Engine operating condition using PFI and GDI engine head.

Test case	[rpm]	Pinj [bar]	SOS [deg]	SOI [deg]	DOI [deg]	IMEP [bar]	CoV IMEP [%]
Gas-PFI	2000	3,5 (intake port)	-17	-360	65	5,7	2,8
Gas-GDI	2000	70 (combustion chamber)	-18	-285	35	6,0	2,7
Eth-PFI	2000	3,5 (intake port)	-17	-360	95	5,8	2,7
Eth-GDI	2000	70 (combustion chamber)	-20	-305	55	6,1	2,6

For all the optical measurements, the synchronization between the cameras and the engine was made by the Crank Angle Encoder signal through a unit delay. The exposure time of the cameras intensifier was fixed at 83.3 μ s. It corresponds to 1° crank angle (CA) at engine speed of 2000 rpm. The dwell time between two consecutive images was set at 83.3 μ s. The cameras were not cycle resolved detectors. In this work, each image was detected at a fixed crank angle of different engine cycles.

EXHAUST MEASUREMENTS

Steady-state measurements of CO, CO₂, O₂, HC and NO_x were made in the raw exhaust by AVL analyzers. CO, CO₂ and HC were measured by non-dispersive infrared detectors; NO_x and O₂ were detected by means of electrochemical sensor. The particulate mass concentration was measured by an opacimeter. Opacimeter is a partial-flow system that measures the visible light attenuation (wavelength=550 nm) from the exhaust gases. By an empirical relations it is possible to convert the opacity percentage in particulate mass concentration [13].

RESULTS AND DISCUSSION

The aim of this paper has been the evaluation of the performance and emissions of a two-wheel small port fuel injection (PFI) spark ignition engine modified using a direct injection system. Before to start with the analysis of the performance of GDI engine we have compared PFI and GDI configuration at 2000 rpm. Investigated engine operating conditions are summarize in table 3. With PFI head the fuel injection occurred in the intake manifold

at fixed pressure of 3,5 bar, with the GDI head the fuel injection pressure was set at 70 bar. Measurements were carried out at an engine speed of 2000 rpm-wide-open throttle, the Duration Of Injection (DOI) was chosen to obtain a stoichiometric equivalence ratio and the spark timings were always fixed to operate at the maximum brake torque. The effect of ethanol fuel was evaluated, too.

Figure 2 shows the in-cylinder pressure, Rate Of Heat Release (ROHR) and Heat Release (HR), averaged curves for the engine equipped with PFI and GDI head, respectively. From the analysis of data, it is evident that the direct injection strategy creates higher densities in the cylinder during the intake stroke, which increases the volumetric efficiency of the engine. Moreover, GDI combustion is faster than PFI one, it is due to the turbulence in the cylinder that is increased by the change of the intake ducts geometry. In particular, the GDI engine head was modified in order to obtain a stronger tumble motion into the combustion chamber, which is necessary to help mixing and homogenization of the charge. Observing the HR curve, it can be noted that in GDI engine the combustion induces less efficient fuel consumption during the flame front propagation and hence more fuel could be available to HC and soot production. This is also confirmed by HC exhaust emission reported in figure 3. Indeed, the real fuel consumption of a GDI engine could be minor than the PFI version if we consider transient operating condition, better regulation and eventually use of global lean (stratified charge) mixture.

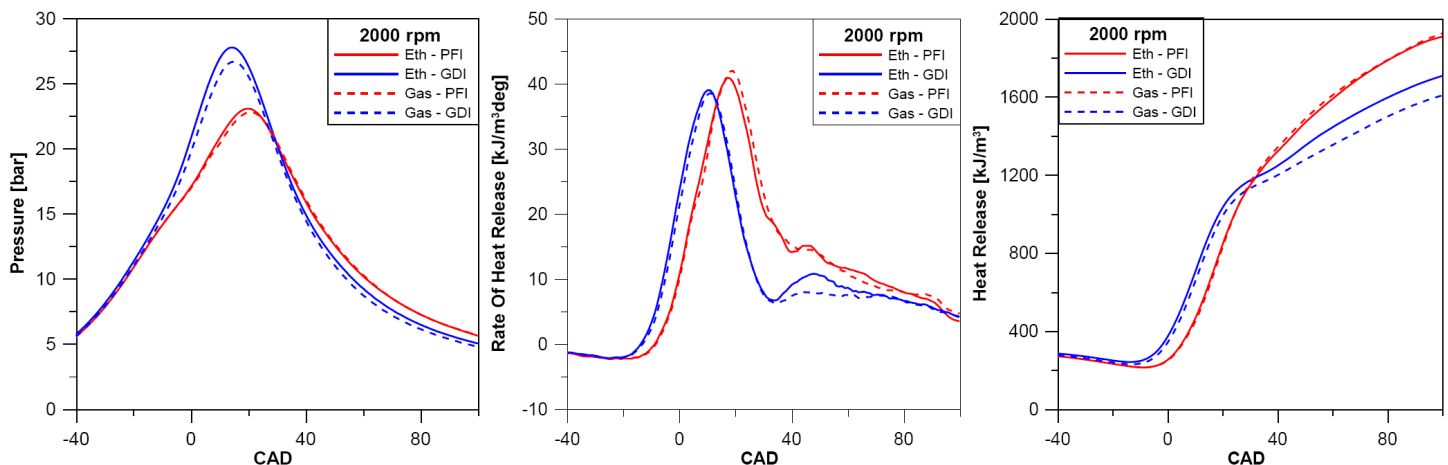


Figure 2. In-cylinder pressure, ROHR and HR averaged curves for the engine equipped with PFI and GDI head, fuelled with ethanol and gasoline at 2000 rpm and full load.

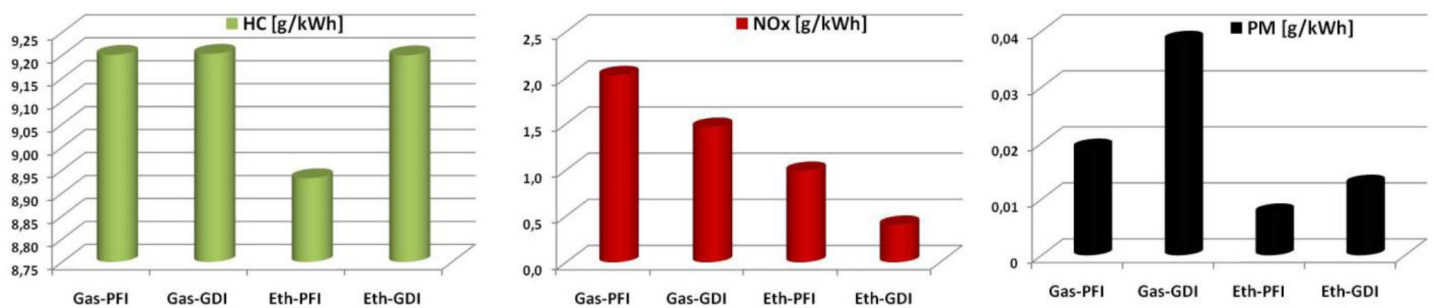


Figure 3. HC, NO_x and Particulate Matter exhaust measurements for GDI and PFI version fuelled with ethanol and gasoline at 2000 rpm and full load.

Table 4. GDI Engine Operating Conditions.

Operating Condition	[rpm]	SOS [deg]	SOI [deg]	EOI [deg]	DOI [deg]	IMEP [bar]	CoV IMEP [%]
Gas-Pinj30bar	2000	-24	-283	-250	33	2,6	16,6
Gas-Pinj120bar	2000	-24	-266,5	-250	16,5	2,9	2,7
Gas-Pinj30bar	4000	-21	-355	-250	105	6,5	3,4
Gas-Pinj120bar	4000	-21	-305	-250	55	7,0	2,0
Eth-Pinj30bar	2000	-24	-296	-250	46	2,6	12,2
Eth-Pinj120bar	2000	-24	-275	-250	25	2,8	2,6
Eth-Pinj30bar	4000	-21	294	-250	176	6,4	3,7
Eth-Pinj120bar	4000	-21	-337	-250	87	6,8	2,8

Summarizing GDI version shows reduction of NO_x emissions, an evident increase of particulate, and a light one of HC in the exhaust. This can be further improved fuelling with ethanol, which contributes to the abatement of NO_x and produces a further reduction of particulate matter in GDI operating condition. Since the ethanol has a larger heat of vaporization than gasoline (841 kJ/kg vs.360 kJ/kg), the amount of energy available per kg of stoichiometric mixture to cool the charge is therefore 84.1 kJ/kg vs.23 kJ/kg and it permits also the increase of densities in the cylinder and of the volumetric efficiency.

In order to better evaluate, the performance, emissions, and in-cylinder processes of GDI engine, several operating conditions were investigated. They were chosen in order to allow both optical measurements such as spray and combustion imaging and to test the engine in real conditions. All the next results presented were carried out at engine speed of 2000 rpm and low load and 4000 rpm medium load. The spark timings were fixed to operate in the MBT condition. The fuel injection occurred directed into the combustion chamber at two injection pressure, 30 and 120 bar. For all the test cases, the duration of injection was chosen to obtain stoichiometric equivalence ratio, as measured by a lambda sensor installed at the engine exhaust. Finally, gasoline and pure ethanol fuel were investigated. More details about the engine operating conditions are reported in [table 4](#).

[Table 4](#) lists the engine parameter and the related Indicated Mean Effective Pressure (IMEP) and Coefficient of

Variation (CoV) for all the selected conditions. An improvement in terms of the combustion stability at higher injection pressure is observed for both engine speeds and fuels. In particular, at 2000 rpm-low load, it is possible to note that the ethanol combustion stability is higher than gasoline one; it is probably due to the better charge homogenization originated by ethanol higher volatility. In this condition, IMEP are similar for both fuels. Instead, at 4000 rpm-medium load, gasoline's IMEP is lightly higher. In fact, the lower duration of injection than ethanol can create a less fuel impinged on the piston surface. [Figure 4](#) reports the in-cylinder pressure, ROHR and HR, averaged curves at 2000 rpm.

At 2000 rpm, the emission measurements, reported in [figure 4](#) and [table 5](#), corroborate these considerations. In particular, HC and particulate matter emissions are lower for ethanol than gasoline and decrease with the injection pressure.

Gasoline HC emission at $P_{inj}=120$ bar and 30bar conditions are similar. This effect could be due to the higher pressure that leads to a longer penetration of the jets and so a major impingement effect [14].

Analogous results were detected at 4000 rpm condition in terms of exhaust emissions and performance ([figure 5](#) and [table 6](#)). A further decrease of HC and soot emission with injection pressure increment was observed. This is explained probably considering that longer penetration

Table 5. CO, CO₂, and NO_x emissions at 2000 rpm

	CO [g/kWh]	CO ₂ [g/kWh]	NO _x [g/kWh]
Gas-Pinj30bar	54,68	625,98	0,60
Gas-Pinj120bar	43,16	563,67	0,57
Eth-Pinj30bar	35,19	639,61	0,14
Eth-Pinj120bar	31,62	586,17	0,16

Table 6. CO, CO₂, and NO_x emissions at 4000 rpm medium load.

	CO [g/kWh]	CO ₂ [g/kWh]	NO _x [g/kWh]
Gas-Pinj30bar	38,92	512,11	1,69
Gas-Pinj120bar	37,82	488,44	1,80
Eth-Pinj30bar	18,83	502,58	0,12
Eth-Pinj120bar	41,01	451,02	0,15

of the plums is balanced by the considerable reduction of the duration of injection and air motion. Nevertheless, the pressure measurements and derived engine parameters give global data and do not allow following locally the combustion process, optical techniques have been adopted in order to detail information on thermo and fluid dynamic phenomena that occur into the combustion chamber [9].

Figure 6 reports a selection of images for the engine running at 2000 rpm-low load with ethanol and gasoline

fuels and at injection pressure of 30 and 120 bar. Using gasoline fuel, the spray shows a single cloud due to the rapid vaporization of the fuel and a good homogenization of the charge. On the other side, the ethanol jets are more evident; it is possible to see the six individual plumes. It is due probably from differences in the liquid properties of alcohol fuels compared to the gasoline. In this condition, the throttle is quite closed and the piston movement mainly determines the air motion, in particular it provides a low tumble motion. The absence of a strong air motion into the combustion chamber does not provide a fast mixing [14].

Even if both fuels have enough time for a good evaporation and mixing, the effect of the injection pressure increment is not only positive. The duration of injection at low load is very short and the end of injection was fixed at 250 cad BTDC in every condition, in order to be distant from the start of spark. The long penetration, given by the highest pressure, leads to the formation of larger fuel deposits especially for gasoline fuel, as it can be observed in figure 6. In figure 7 at 4000 rpm-medium load the higher tumbling motions, due to the opening of throttle valve, enhance the fuel vaporization and fuel-air mixing. Nevertheless, the longer DOI, compared to 2000 low load condition, causes higher quantity of fuel impinged on the piston surface. These deposits of fuel generate locally fuel-rich zones that could ignite when the normal flame front reaches them.

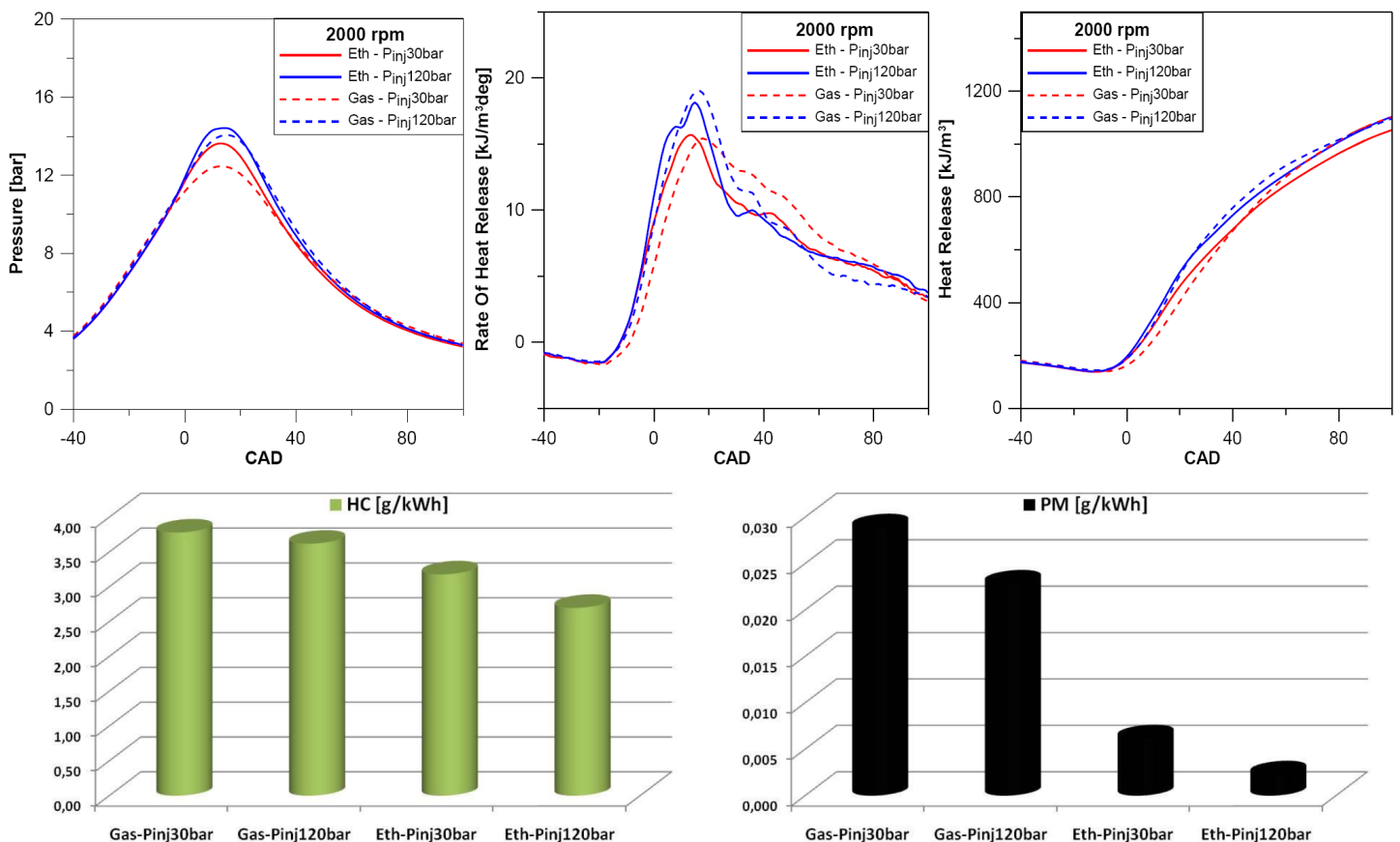


Figure 4. In-cylinder pressure, ROHR and HR averaged curves, HC and Particulate Matter (PM) at 2000 rpm low load for ethanol and gasoline and both injection pressures.