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Thermodynamic and optical characterizations of a high performance GDI engine operating in homogeneous and stratified charge mixture conditions fueled with gasoline and bio-ethanol

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ABSTRACT

UltraViolet–visible imaging measurements were carried out in a gasoline direct injection (GDI) engine in order to investigate the spray and combustion evolution of gasoline and pure bio-ethanol fuel. Two different starts of injection, early injection (homogeneous charge) and late injection (stratified charge), were tested in two different engine conditions, 1000 rpm idle and 1500 rpm medium load as representative point of urban new European driving cycle (NEDC).

Measurements were performed in the optically accessible combustion chamber made by modifying a real 4-stroke, 4-cylinder, high performance GDI engine. The cylinder head was instrumented by using an endoscopic system coupled to high spatial and temporal resolution cameras in order to allow the visualization of the fuel injection and the combustion process.

All the optical data were correlated to the in-cylinder pressure-based indicated analysis and to the gaseous and solid emissions. Wide statistics were performed for all measurements in order to take into account the cycle-to-cycle variability that characterized, in particular, the idle engine condition.

Optical imaging showed that gasoline spray was more sensible to air motion and in-cylinder pressure than ethanol's, for all the investigated conditions. The stratified flame front for both fuels was about 40% faster compared to homogeneous in the first phase, due to the A/F ratio local distribution. It leads to better performance in terms of stability and maximum pressure, even if the late injections produce more soot and UHC emissions due to fuel impingement.

Ethanol combustion shows less diffusive flames than gasoline. A lower amount of soot was evaluated by two color pyrometry method in the combustion chamber and measured at the exhaust.

1. Introduction

The growing transport field is considered to be one of the main reasons for failing to meet the Kyoto targets. In combination with the emission limits and new emission standards, the already very low carbon dioxide (CO_2) emission levels must be furthermore reduced to meet the Kyoto targets both for spark ignition and diesel engines. In particular, in Europe transport field accounts for more than 30% of the total energy consumption in European Community. It depends 98% on fossil fuels with the crude oil feedstock being largely imported and thus extremely vulnerable to any market disturbance [1].

It is mandatory, then, to search and use alternative/organic fuels with respect to fossil ones for internal combustion engines. In particular, the request for cleaner emissions and improvements of fuel consumption economy with internal combustion engines are important issues taking into account CO_2 exhaust regulations and the limited supply of crude oil. According to these issues, the development of new clean-gasoline engines, such as direct injection fueled with bio-fuel, is important because it has at the same time the advantages of higher thermal efficiency due to direct fuel injection and higher power output than conventional engines [2–4]. The gasoline direct injection (GDI) engine has also a better transient

Abbreviations: ABDC, after bottom dead centre; AFR, air fuel ratio; ASOS, after start of spark; ATDC, after top dead centre; BBDC, before bottom dead centre; Bio, biological; BMEP, brake maximum effective pressure; BTDC, before top dead center; CA, crank angle; CCD, charge coupled device; CDM, crank angle degree marker; CO, carbon monoxide; CO_2, carbon dioxide; COV, coefficient of variation; DI, direct injection; DOI, duration of injection; EVO, exhaust valve opening; FSN, filter smoke number; GDI, gasoline direct injection; ICCD, intensified charge coupled device; IHR, integral of heat release; IMPEH, indicate mean effective pressure high; IVO, inlet valve opening; λ, actual afr to stoichiometric afr; MBT, maximum brake torque; NEDC, new European driving cycle; NO_x, nitrogen oxides; nm, nanometer; PM, particulate matter; P_{inj}, pressure injection; ROHR, rate of heat release; RON, research octane number; rpm, revolution per minute; SI, spark ignition; SOI, start of injection; SOS, start of spark; TTL, transistor transistor logic; TDC, top dead center; UHC, unburned hydrocarbon; UV, ultra violet; VVT, valve variation timing.

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response, more precise control of the air–fuel ratio, an improvement of fuel economy, and a reduction of exhaust emissions thanks to ultra lean combustion due to stratification and a rich fuel–air mixture near the spark plug. Moreover, the higher compression ratio due to the reduced possibility of knocking, leads to an improvement of the output performance by using alcohol fuel [2,4].

Ethanol can be considered one of these alternative/biological fuels, because it can be used as a fuel extender for petroleum-derived fuels, oxygenate, an octane enhancer, and a pure fuel. The start up to ethanol production in the mid 1970s was due to the need to develop alternative supplies of motor fuel in response to the oil embargoes in 1973 and 1979. Then, its use was focused on special markets such as in Brazil or Sweden [3–5].

Many differences between ethanol and gasoline are reported in Table 1. In particular, ethanol has a heating value (LHV) about 60% lower than gasoline. However the amount of energy per kg of stoichiometric mixture is very close for both fuels as ethanol stoichiometric air/fuel ratio is also smaller of about the same amount. Moreover, ethanol has higher research octane number (RON). This parameter allows higher compression ratio, higher boost in turbocharged engine, and better knock limited spark advances. Ethanol also has a higher vaporization heat and in this way the available energy amount per kg of stoichiometric mixture to cool the charge is about three times bigger (3.65). This provides higher densities in the intake that may increase volumetric efficiency mainly in naturally aspirated engines with port fuel injection (PFI) system, or better cooling of the in-cylinder charge in naturally aspirated and turbo charged direct injection engines. Furthermore, this latter feature reduces the knock sensitivity.

If direct fuel injection and turbo charging are two of the most effective indications in enhancing the efficiency of gasoline engines, the use of pure ethanol can give additional improvements. In particular, the effect of inhibiting knock has been already tested [2,4–6]. Nevertheless, the chemical composition of ethanol requires larger amounts of fuel to be injected, thus leading to the danger of oil dilution with direct-injected combustion concepts [7].

Direct injection and turbo charging may therefore optimize pure ethanol-fueled engines to a level of performance that exceeds gasoline engine efficiency, taking full advantage of ethanol's higher octane number and vaporization heat.

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

Fuel property	Bio-ethanol	Gasoline
Formula	C_2H_5OH	C_4–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
Density15/15 °C (kg/l)	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
Aromatics (Vol%)	–	35
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	–

In order to enhance these improvements, some non intrusive measurements in the cylinder must be performed so that the related chemical and physical events can be assessed. 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 [3]. In particular, combined optical techniques were used in order to analyze the fuel spray distribution and evaporation, mixture preparation and self-ignition [8]. Few works have been carried out in real four cylinder engines using data of the ultraviolet–visible spectroscopy while combustion process involved in gasoline engine fueled with mono-components fuels has been widely studied [2,9].

The current study reports macroscopic imaging of spray and combustion and spectroscopic measurements obtained during tests carried out in a commercial 4-cylinder high performance GDI engine fueled with a multi-component fuel, namely standard European commercial grade gasoline (RON 95) and bio-ethanol produced from grape marc. Different injection strategies were tested moving the injection start from early to late injection, in order to obtain both stratified and homogeneous charge mixture. The best ones in these two conditions were widely investigated by non intrusive diagnostics. In particular, 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 has shown to be a powerful tool because radical species (OH, CH, HCO, CN), carbonaceous material and CO–O bands can be observed in this spectral range [10]. Finally, flame-kernel radius growth and motion from image processing, in-cylinder pressure history, and Mass Fraction Burned (MFB), as well as exhaust emission, are presented for the same engine operating conditions in order to compare the pure bio-ethanol fuel with gasoline.

2. Experimental apparatus and procedures

2.1. Engine

The experimental apparatus includes the following modules: the spark ignition engine, an electrical dynamometer, the fuel injection line, the data acquisition and control units as well as the emissions measurement system. The electrical dynamometer allowed operating both in motoring and firing conditions that was appropriate to detect the in-cylinder pressure data and to explore the engine behaviour in stationary and simple dynamic conditions.

A spark ignition direct injection (DI), inline 4-cylinder, 4-stroke, displacement of 1750 cm^3, turbocharged, high performance engine was used. It had a wall guided injection system with a six holes nozzle located between the intake valves and oriented at 70° with respect to the cylinder axis. The engine is equipped with a variable valve timing system in order to optimize intake and exhaust valve lift for each regime of operation. The engine was not equipped with any after-treatment device. Further details are reported in Table 2.

An optical shaft encoder was used to transmit the crank shaft position to the electronic control unit. The information was in digital pulses, the encoder had two outputs, the first is top dead center (TDC) index signal, and it had a resolution of 1 pulse/revolution. The second is the crank angle degree marker (CDM) 1 pulse/0.2°.

The engine is a 4-stroke and the encoder gives as output two TDC signal per engine cycle so to have the right crank shaft position, one pulse was suppressed via software.

A quartz pressure transducer was installed into the spark plug in order to measure the in-cylinder pressure with a sensitivity of 19 pC/bar and a natural frequency of 130 kHz. Thanks to its characteristics was obtained a resolution of about 0.06° crank angle (CA)