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Investigation of performance and combustion characteristics of a four-valve supercharged two-stroke DI engine fuelled with gasoline and ethanol

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ABSTRACT

Worldwide emission legislations have promoted the development of more efficient internal combustion engines through engine downsizing and advanced lean-burn combustion concepts. However, these measures if solely employed in the next generation vehicles are still not enough to comply with future CO_2 limits, so powertrain hybridisation has been already set up. In this circumstance, internal combustion engines play a less important role in the vehicle's propulsion system serving mostly as a range extender or extra power supply unit. Power density (kW/dm^3) and power per unit mass (kW/kg) are as important as fuel consumption, so not only four-stroke engines may serve to this purpose but also two-stroke engines. Therefore, the present research evaluates the performance and combustion characteristics of a two-stroke cycle engine embedded in the architecture of a contemporary four-stroke supercharged engine, with direct fuel injection, intake and exhaust valves, and a wet sump. Commercial gasoline and anhydrous ethanol (E100) were tested at loads from 0.2 MPa to 1.0 MPa indicated mean effective pressure and speeds varying from 800 rpm to 2400 rpm. The results demonstrated that ethanol increased the overall indicated efficiency by about 10% compared to gasoline, while improving the burning process at very light loads thanks to its higher tolerance to diluted combustion. At higher engine loads the calculated supercharger power consumption largely increased, which reduced the corrected indicated efficiency for both fuels tested. The maximum in-cylinder pressure of 6.5 MPa was obtained using ethanol at 160 Nm/dm^3 of torque and 800 rpm. The short time available for air-fuel mixing at high loads and excessive combustion dilution at low loads deteriorated the combustion process. The lowest combustion efficiency of 0.80 was obtained at transitioning regions from spark ignition to spark assisted compression ignition combustion with gasoline. The 50% of mass fraction burned was found closer to top dead centre at several operating conditions due to a largely diluted combustion. Finally, the exhaust gas temperature presented a peculiar behaviour due to a competition between combustion rate and exhaust gas dilution by the intake air. This resulted in an exhaust gas temperature as low as 500 K at full load and 800 rpm, while the maximum gas temperature of 740 K was observed at mid-loads.

1. Introduction

The expanding markets around the world, particularly in Asia, are the main players for the expected increase in the light-duty vehicle fleet in the next decades, to be at about 1.8 billion by 2040 [1]. Consequently, several technological improvements such as engine downsizing and lean combustion, alongside the adoption of biofuels, are necessary to guarantee a balanced fuel supply and demand. The European Union

Abbreviations: aTDC, after top dead centre; CA, crank angle; CAI, controlled auto-ignition; CI, compression ignition; COV_{IMEP}, covariance of the IMEP; DI, direct injection; EVC, exhaust valve closing; EVO, exhaust valve opening; GDI, gasoline direct injection; HCCI, homogeneous charge compression ignition; NHRR, net heat release rate; IMEP, indicated mean effective pressure; IVC, intake valve closing; IVO, intake valve opening; KLS, knock limited spark advance; LHV_{fuel}, lower heating value of fuel; LHV_C, lower heating value of solid carbon; LHV_{CO}, lower heating value of carbon monoxide; LHV_{H_2}, lower heating value of hydrogen; LHV_{UHC}, lower heating value of unburned hydrocarbons; m/m, mass basis; $\dot{m}_{air}$, air mass flow rate; $\dot{m}_{fuel}$, fuel mass flow rate; $\dot{m}_{soot}$, mass flow rate of soot; $\dot{m}_{CO}$, mass flow rate of carbon monoxide; $\dot{m}_{H_2}$, mass flow rate of hydrogen; $\dot{m}_{UHC}$, mass flow rate of unburned hydrocarbons; MBT, minimum ignition advance for best torque; MFB, mass fraction burned; NA, naturally aspirated; NOx, oxides of nitrogen; NVH, noise vibration and harshness; p_a, ambient pressure; p_{int}, intake pressure; P_c, compressor or supercharger power consumption; P_i, indicated power; PFI, port fuel injection; PPC, partially premixed combustion; PRR, pressure rise rate; Q_{net}, net heat release; rpm, revolutions per minute; RON, research octane number; SACI, spark assisted compression ignition; SI, spark ignition; SOI, start of fuel injection; T_a, ambient temperature; TDC, top dead centre; UHC, unburned hydrocarbons; V, volume; γ, ratio of specific heats c_p/c_v; η_c, combustion efficiency; $\eta_i corr$, corrected indicated efficiency; η_{comp}, compressor efficiency; θ, crank angle

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has proposed a carbon dioxide (CO_2) emission limit between 68 and 78 g/km for passenger cars to be phased in by 2025 [2], while the USA has already approved the CO_2 emission target of 93 g/km for passenger cars by the same year. In this scenario, even diesel fuelled cars and small spark ignition (SI) vehicles may fail to meet such stringent regulations.

Engine downsizing has been widely accepted as an effective method to reduce fuel consumption at part load operation of four-stroke engines running on stoichiometric air/fuel ratios. Considering the nature of driving cycles such as the Worldwide Harmonized Light Vehicles Test Cycle (WLTC) [3], these improvements in the mid-low load range have a large impact over the vehicle's total CO_2 emissions. The gains in fuel consumption are usually reported in the range from 20% to 30% for a 50% downsized engine [4,5]. Better mechanical efficiency resulted from lower friction losses is also obtained [6], as well as lower heat rejection due to fewer cylinders [7]. However, the higher in-cylinder pressures and temperatures, as well as low speed pre-ignition (LSPI), are among the main issues compromising the engine operation and its durability [8,9].

Among the distinguished burning methods employed in gasoline engines, controlled auto-ignition combustion (CAI) has been extensively studied over the last decades, particularly homogeneous charge compression ignition (HCCI) combustion and partially premixed combustion (PPC). Among several studies, for instance, HCCI combustion was able to improve fuel economy by 21% and cut oxides of nitrogen (NOx) emissions by half compared to conventional SI operation [10]. Similarly, gasoline PPC has demonstrated Diesel like efficiencies with a BSFC around 190 g/kW h [11] and NOx emissions below 0.2 g/kW [12].

Biofuels such as methanol and ethanol present interesting characteristics such as greater knock resistance, faster laminar flame development and higher heat of vaporization than gasoline [13,14]. However, their reduced lower heating values (LHV) compared to gasoline or even superior alcohols e.g. propanol and butanol, result in higher volumetric fuel consumption [15,16]. Even then, moderate blends with gasoline tolerate higher compression ratios and allow better combustion phasing, which increases the thermal efficiency and may compensate the lack of LHV on a volumetric fuel consumption basis [17,18]. Ethanol is currently available from feedstock such as corn, sugar cane and sugar beet [19], while its production from cellulose and algae, known as second and third generation ethanol, respectively, has been subjected to extensive research [20,21].

In the case of series or parallel hybrid electric vehicles, the synergy among the various components of the powertrain i.e. internal combustion engine, power generator, electric motor(s) and batteries, depends on driving cycle requirements. In such situation gasoline engines may have a secondary role in passenger cars and only operate as a range extender in case of battery depletion [22,23]. Therefore, not only fuel economy is required but other characteristics such as great packaging, low weight, and reduced noise, vibration and harshness (NVH) are equally important [24].

In this scenario, two-stroke engines present several similarities to downsized four-stroke units, particularly considering their power per unit mass and high power and torque density as it can be observed in Table 1. This table presents a comparison among several contemporary two-stroke and four-stroke engines regarding maximum output power and torque, with all models supercharged and/or turbocharged fuelled with gasoline or diesel. If the engine operation is intermittently required due to battery recharging strategies, the reduced NVH of two-stroke engines may present a large advantage over the four-stroke counterparts [25,26]. Compared to a four-stroke engine of the same displacement and operating at the same speed, a similar output power can be obtained with lower in-cylinder pressures and hence less structural and thermal stresses. Several two-stroke engine concepts have

Table 1
Comparison of specific brake power and torque of selected contemporary turbocharged and/or supercharged two-stroke/four-stroke engines.

Engine model or manufacturer	Cycles	Max. Power (kW/dm^3)	Max. Torque (Nm/dm^3)	Refs.
Bosch MPE	Four-stroke	83	178	[5]
Fiat Multiair		73	169	[38]
JLR Ultraboost		142	257	[39]
Mahle 3 Cylinder		120	238	[6]
Ricardo HyBoost		95	230	[40]
Toyota ESTEC		71	155	[41]
Achates Power	Two-stroke	44	225	[30]
Renault (uniflow)		50	136	[42]
Renault (4-valve)		62	199	[43]
Ford-Ricardo 2/4	Two/four-stroke	94	230	[44]

received attention in the last years, particularly uniflow scavenged engines [27–29] including the opposed piston design [30–32], as well as poppet valve two-stroke engines [25,33] and free-piston engine concepts [34–36]. The more frequent combustion of the two-stroke cycle and its inherent residual gas fraction are responsible for higher in-cylinder temperatures, which may be important in the case of flex-fuel (gasoline-ethanol) hybrid electric vehicles [37]. In this regard, the intermittent operation of ethanol-fuelled engines poses challenges regarding thermal management and cold startability under urban driving cycles.

The two-stroke poppet valve engine, despite the modest scavenging performance compared to the uniflow system, has the advantage of sharing the same architecture of contemporary four-stroke engines, with valves in the cylinder head, a wet sump and no ports in the cylinder liner. The gas exchange process in such engines is performed through boosted intake air during a long valve overlap around bottom dead centre, with fuel injection usually occurring after all valves are shut to avoid its short-circuiting to the exhaust [45]. Differently from four-stroke engines where the load is controlled by the air mass induced via throttling (so pumping losses prevail at mid-low loads), in the two-stroke engine the load is proportional to the replacement of burned gases by fresh charge. Hence, at low loads the residual gas fraction can reach values above 65% [46]. This may also trigger controlled auto-ignition (CAI) combustion if enough in-cylinder temperature is reached [33,47].

Given these facts, the present research exploits the characteristics of a four-valve direct injection boosted engine operating in the two-stroke cycle with commercial gasoline (RON 95) and ethanol (E100). Fifty operating conditions with engine speeds varying from 800 rpm to 2400 rpm and loads in the range 0.2–1.0 MPa of indicated mean effective pressure (IMEP) were evaluated concerning performance and combustion characteristics.

2. Experiments

2.1. Experimental setup

A Ricardo Hydra Camless engine (Table 2) equipped with fully variable electrohydraulic intake and exhaust valves was employed in this research. The fuelling rate was measured by an Endress + Hauser Coriolis Promass 83A with a maximum error of ± 0.2%. Spark timing, injection timing and duration, and valve events were managed by a Ricardo rCube unit. A transient dynamometer enabled constant speed tests (± 5 rpm) while a LeineLinde incremental encoder with 720 pulses per revolution was used to record the crank-train position. Intake and exhaust valve timings were set according to the results obtained in previous studies [48], while their positions were recorded by four Lord