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Benefits of E85 versus gasoline as low reactivity fuel for an automotive diesel engine operating in reactivity controlled compression ignition combustion mode

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ABSTRACT

This work shows the capabilities of E85 fuel to be used as low reactivity fuel in a high compression ratio light-duty diesel engine (17.1:1) running under reactivity controlled compression ignition concept. To do this, experimental steady-state engine maps are obtained in a single-cylinder engine with diesel-E85 fuel combination. The engine mapping was performed following the same procedure used in previous works with other fuel combinations to allow the results comparison. Considering the mechanical and emissions limits imposed during the engine mapping, it was found that with diesel-E85 the combustion concept is limited to the region defined from 2 to 7 bar at 1000 rpm, and from 1.5 to 9 bar indicated mean effective pressure at 3000 rpm. This operating region was satisfied with nitrogen oxides, soot and pressure rise rate levels below 0.4 g/kWh, 0.01 g/kWh and 10 bar/CAD, respectively. The reactivity controlled compression ignition maps with diesel-E85 were obtained taking as reference the total fuel energy used in a previous work to map the engine with diesel-gasoline. The direct comparison of both combustion concepts (diesel-E85 and diesel-gasoline) revealed that E85 allows to extend the engine map around 2 bar indicated mean effective pressure towards the high load region. Moreover, the minimum load achieved at high engine speeds was decreased down to 1.5 indicated mean effective pressure. Finally, the differences in terms of emissions and performance between both reactivity controlled compression ignition concepts are highlighted by doing the difference between the maps of several variables.

1. Introduction

Excellent combination of engine efficiency and performance has made compression ignition (CI) engines a widely used technology for transportation worldwide. However, the biggest challenge that CI engines are facing is the existing trade-off between nitrogen oxides (NOx) and smoke emissions during the combustion process [1]. Due to the evolution of the emissions regulations towards more stringent scenarios, the engine manufacturers are forced to mitigate these pollutants by different means. In this sense, current CI engines operated under conventional diesel combustion (CDC) have included aftertreatment equipment in order to reduce the emissions generated during combustion. Aftertreatment systems for CI consists of a selective catalyst reduction (SCR) for NOx emissions, diesel particulate filter (DPF) for the soot content at the exhaust gases, and diesel oxidation catalyst (DOC) to reduce hydrocarbons (HC) and carbon monoxide (CO) emissions.

Aftertreatment systems allow the harmful emissions be reduced below the limitations imposed by the emissions regulations [2]. However, these elements provide an increase of the engine complexity and imply an extra cost at the production budget [3]. In addition, these systems usually make use of exhaust fluids such as urea to enhance the SCR reduction capacity and diesel fuel for the DPF regeneration. Although it has been dedicated a huge effort to minimize the operational costs of the aftertreatment systems [4], they also provoke an inherent fuel consumption penalty due to the back pressure increase caused in

Abbreviations: ASTM, American Society for Testing and Materials; ATDC, After Top Dead Center; CAD, Crank Angle Degree; CA50, crank angle at 50% mass fraction burned; CDC, Conventional Diesel Combustion; CI, Compression Ignition; CO, Carbon Monoxide; CO_2, Carbon Dioxide; DOC, Diesel Oxidation Catalyst; DI, Direct Injection; DMDF, Dual-mode Dual-fuel; DPF, Diesel Particulate Filter; EGR, Exhaust Gas Recirculation; EVO, Exhaust Valve Open; FSN, Filter Smoke Number; GF, Gasoline Fraction; GIE, Gross Indicated Efficiency; HC, Hydro Carbons; HCCI, Homogeneous Charge Compression Ignition; HRF, High Reactivity Fuel; IMEP, Indicated Mean Effective Pressure; IVC, Intake Valve Close; LHV, Lower Heating Value; LRF, Low Reactivity Fuel; LTC, Low Temperature Combustion; MPRR, Maximum Pressure Rise Rate; NOx, Nitrogen Oxides; PER, Premixed Energy Ratio; PFI, Port Fuel Injection; P_{max}, Maximum Pressure; PPC, Partially Premixed Charge; RCCI, Reactivity Controlled Compression Ignition; RoHR, Rate of Heat Release; RON, Research Octane Number; SOC, Start of Combustion; SOI, Start of Injection; SCE, Single Cylinder Engine; SCR, Selective Catalytic Reduction

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the exhaust line [5].

To reduce the aftertreatment necessities, alternative combustion processes should be implemented to minimize the emissions levels generated during the combustion process [6]. In this sense, the low temperature combustion strategies (LTC) have been proved to be able to achieve high engine efficiency while reducing simultaneously NOx and smoke emissions [7,8]. These strategies use highly diluted fuel-air mixtures [9] increasing the mixing time prior to the start of the combustion [10], which provide a simultaneous reduction in both harmful pollutants. Moreover, the efficiency is improved due to fast combustion processes and reduced heat transfer [11].

Homogeneous charge compression ignition (HCCI), partially premixed combustion (PPC), reactivity controlled compression ignition (RCCI) or dual-fuel combustion [12,13] are the most studied LTC concepts by the research community. Diesel HCCI has been widely investigated during the recent years [14]. The high fuel-air premixing levels used with HCCI allow reducing the NOx and soot formation to virtually zero [15]. In addition, due to the fast heat release, high thermal efficiency can be achieved in the operating range in which the combustion process can be controlled. However, since chemical kinetics dominate the combustion onset, the operating range is too small, being limited by the appearance of high pressure gradients and combustion noise. In particular, the operating range of the HCCI strategy is limited to partial engine load [16]. Additionally to these problems, HCCI presents another challenges such as cold start and excessive CO and unburned HC [17] levels, which limited its potential to be used in real engines.

Gasoline PPC has been deeply studied as a possible solution for the shortcomings found with HCCI [18,19]. PPC strategy allows setting more delayed injection timings thanks to using a fuel with lower reactivity than diesel. This enables better control of the combustion phasing as load increases, which makes possible the reduction of the knocking phenomenon [20]. As the combustion phasing results in a better control due to the use of gasoline-like fuels, the control of the heat release was also improved and allowed this concept to reduce the NOx emissions compared to conventional combustion strategies [21,22]. Additional studies under PPC mode were carried out in order to improve the understanding of using gasolines with different research octane number (RON), resulting that gasolines with ON higher than 91 produce excessive unburned HC. Installing a spark plug would solve the cycle-to-cycle dispersion that was causing such a high levels of unburned HC [23], but the benefits observed at the NOx and smoke emissions were missed [24].

Inagaki et al. [25] operated a dual-fuel premixed compression ignition (PCI) combustion strategy using two fuels of different reactivity. The in-cylinder reactivity was controlled by modifying the fuels percentages using two injector systems. This provided an excellent control of the combustion onset and extremely low NOx and soot emissions simultaneously. These conclusions were later confirmed by Kokjohn et al. [26], which named this dual-fuel LTC technique as reactivity controlled compression ignition (RCCI).

RCCI has been found to be the most promising LTC concept in terms of efficiency, emissions and engine load range of operation [27]. Major part of these benefits come from the use of two fuels with different reactivity. The possibility of adjusting the reactivity on demand provides an upgraded control of the combustion [28] and thereby, can overcome the main limitations of the LTC concepts, mainly the combustion stability. Nonetheless, RCCI has still several challenges to face, such as unburned HC and CO emissions during the low engine load operation [29]. Maximum pressure rise rates (MPRR) and in-cylinder peak pressure are reduced with RCCI due to the more sequential autoignition obtained thanks to the high reactivity fuel (HRF) stratification [30]. However, these two factors still limit the RCCI operating range to moderate loads, compromising its application under real engine conditions [31].

Several studies have been carried out in order to extend the application range of RCCI. The dual-mode concept implies to operate with other combustion mode when it is critical for RCCI [32]. Thus, dual-mode RCCI/CDC could become a great potential option when high compression ratios (≈17:1) are used, as demonstrated in both medium-duty [33] and light-duty engines [31]. Another route to solve the shortcomings of the RCCI mode is using a lower compression ratio (≈15:1) to implement a concept known as dual-mode dual-fuel (DMDF) [34]. This DMDF concept allows to operate the engine from low load to full load in dual-fuel conditions, using RCCI in the lower portion of the map and diffusive dual-fuel combustion from the 75% engine load to full load. In addition, DMDF shows an excellent potential due to its low NOx emissions (below 0.4 g/kWh up to 75% engine load) and ultra-low smoke emissions [35]. In both dual-mode concepts, the major benefits in terms of NOx and soot emissions reduction are obtained in the RCCI portion of the map [36]. Thus, it is crucial to extend pure RCCI operating range in order to improve the global engine map emissions.

As literature demonstrates, the RCCI concept can be implemented using a wide variety of fuels apart from diesel and gasoline [37]. The most widely used high reactivity fuel is diesel [38]. Some other fuels such as gasoline doped with a cetane improver [39] or diesel-gasoline mixtures have been tested without relevant improvements [40]. By contrast, it has been proved that the low reactivity fuel physical and chemical characteristics have greater effects on RCCI emissions and performance [41]. In this sense, it was found that using fuels with lower reactivity than gasoline such as ethanol [42,43], methanol [44] and other biofuels [45] can contribute to reduce the MPRR and P_{max} at higher loads. Considering this background, the objective of this work is to assess the capabilities of using ethanol (E85) as a low reactivity fuel to extend the RCCI operating range in a high compression ratio engine (≈17.5:1). To do this, experimental steady-state engine maps are obtained in a single-cylinder engine with diesel-E85 fuel combination. Later, the mapping results are compared to those obtained in a previous work using diesel-gasoline as pair of fuels [46] in the same engine platform.

2. Materials and methods

2.1. Engine and test cell description

The single-cylinder diesel engine (SCE) used for the experiments is based on a serial production light-duty 1.9 L platform. The engine has four valves driven by dual overhead cams. The piston used is the serial one, with a re-entrant bowl that confers a geometric compression ratio of 17.1:1. The swirl ratio was fixed at 1.4 using the tangential and helical valves located in the intake port [47], which is a representative value of that used in the stock engine configuration. Table 1 summarizes the more relevant characteristics of the engine.

The scheme of test cell in which the engine is operated is shown in Fig. 1. An electric dynamometer is used for the engine speed and load control during the experiments. The air intake line is composed of a screw compressor that feeds the engine with fresh air at a pressure up to 3 bar, heat exchanger and air dryer to modify the temperature and relative humidity of the air, airflow meter and a settling chamber sized to

Table 1
Engine characteristics.

Engine type	4 stroke, 4 valves, direct injection
Number of cylinders [–]	1
Displaced volume [cm^3]	477
Stroke [mm]	90.4
Bore [mm]	82
Piston bowl geometry [–]	Re-entrant
Compression ratio [–]	17.1:1
Rated power [kW]	27.5 @ 4000 rpm
Rated torque [Nm]	80 @ 2000–2750 rpm