Energy Conversion and Management 149 (2017) 263–274

Contents lists available at ScienceDirect

Energy Conversion and Management

journal homepage: www.elsevier.com/locate/enconman

Evaluating the emissions and performance of two dual-mode RCCI combustion strategies under the World Harmonized Vehicle Cycle (WHVC)

Antonio García [a,*], Javier Monsalve-Serrano [a], Vinícius Rückert Roso [b], Mario Eduardo Santos Martins [c]

[a] CMT – Motores Térmicos, Universitat Politècnica de València, Camino de Vera s/n, 46022 Valencia, Spain
[b] CTM – Centro de Tecnologia da Mobilidade, Universidade Federal de Minas Gerais, Belo Horizonte, MG, Brazil
[c] Grupo de Pesquisa em Motores, Combustíveis e Emissões, Universidade Federal de Santa Maria, Santa Maria, RS, Brazil

ARTICLE INFO

Article history:
Received 10 May 2017
Received in revised form 7 July 2017
Accepted 15 July 2017

Keywords:
Reactivity controlled compression ignition
Dual-fuel combustion
Dual-mode concept
EURO VI emissions
Engine cycle simulation

ABSTRACT

This work compares the emissions and performance of two dual-mode reactivity controlled compression ignition (RCCI) combustion strategies under the World Harmonized Vehicle Cycle (WHVC), a chassis dynamometer version of the World Harmonized Transient Cycle (WHTC) test proposed by the EURO VI emission regulation for heavy-duty engines. The major difference between the two dual-mode combustion strategies investigated is that, while one of them relies on covering with conventional diesel combustion (CDC) the part of the map that cannot be covered by RCCI regime (RCCI/CDC dual-mode), the other does it relying on dual-fuel diffusion combustion (dual-mode dual-fuel).

The influence of the gear shifting strategy on the emissions and performance over the WHVC is discussed first. Later, both dual-mode concepts are compared considering the optimal gear shifting strategy. The results suggest that dual-mode dual-fuel concept allows reducing the specific fuel consumption by 7% in average versus RCCI/CDC concept. Moreover, NO_x emissions are around 87% lower with dual-mode dual-fuel, meeting the EURO VI requirements without the need for an SCR aftertreatment system. In counterpart, HC and CO emissions are near 2 and 10 times greater, respectively, for dual-mode dual-fuel than for RCCI/CDC.

1. Introduction

Consumption of petroleum products in the European Union (EU) increased by 13% between 1990 and 2005. This trend was mainly driven by the fuel consumption of transportation sector. Between 2005 and 2012, the oil consumption decreased in all sectors due to the recession, but transport sector experienced the smallest decrease by only 8%. In fact, in 2012 the transport industry had 76% share of the petroleum consumption in the EU, of which near 74% belonged to road transport. This means that internal combustion engines (ICE) for transportation are responsible of near 60% of the energy consumption of petroleum products in Europe [1].

Engine and vehicle manufacturers are continuously working on developing new technologies [2] to increase fuel economy and reduce pollutant emissions [3,4] according to requirements imposed by both users and regulations [5]. Historically, the higher efficiency of compression ignition (CI) engines than spark ignition (SI) engines has led them to lead heavy-duty transportation. However, in spite of conventional diesel combustion (CDC) offers high

Abbreviations: ATDC, After Top Dead Center; BMEP, Brake Mean Effective Pressure; BSFC, Brake Specific Fuel Consumption; B7, Diesel fuel with 7% of biodiesel content by volume; CAD, Crank Angle Degree; CDC, Conventional Diesel Combustion; CI, Compression Ignition; CO, Carbon Monoxide; CR, Compression Ratio; DOC, Diesel Oxidation Catalyst; DMDF, Dual-Mode Dual-Fuel; DPF, Diesel Particulate Filter; ECU, Electronic Control Unit; EGR, Exhaust Gas Recirculation; EU, European Union; E20-95, Fuel blend of 95 ON gasoline and 20% of ethanol by volume; FSN, Filter Smoke Number; GIE, Gross Indicated Efficiency; HC, Hydro Carbons; HCCI, Homogeneous Charge Compression Ignition; IMEP, Indicated Mean Effective Pressure; ICE, Internal Combustion Engines; IVC, Intake Valve Close; IVO, Intake Valve Open; LRF, Low Reactivity Fuel; LTC, Low Temperature Combustion; MON, Motor Octane Number; PCI, Premixed Compression Ignition; PFI, Port Fuel Injection; PPC, Partially Premixed Charge; PRR, Pressure Rise Rate; RON, Research Octane Number; RCCI, Reactivity Controlled Compression Ignition; SI, Spark ignition; SCE, Single Cylinder Engine; SCR, Selective Catalytic Reduction; TDC, Tope Dead Center; WHTC, World Harmonized Transient Cycle; WHVC, World Harmonized Vehicle Cycle.

* Corresponding author.
E-mail address: angarma8@mot.upv.es (A. García).

http://dx.doi.org/10.1016/j.enconman.2017.07.034
0196-8904/

efficiency, it requires using complex and costly exhaust aftertreatment systems to reach the NO_x and soot emissions levels imposed by the currently in force EURO VI regulation. The technological solution adopted by the majority of heavy-duty engine manufacturers for moving from EURO V to EURO VI regulation is quite similar [6]. It consist of a two-stage NO_x reduction; exhaust gas recirculation (EGR) first, followed by selective catalytic reduction (SCR) exhaust aftertreatment, and diesel particulate filter (DPF) to remove particulates [7]. Unburned hydrocarbons (HC) and carbon monoxide (CO) emissions are reduced using a diesel oxidation catalyst (DOC). As a result of the modifications introduced to fulfill the different regulatory stages, a EURO VI heavy-duty vehicle equipped with a 12 l engine has experienced an increase of approximately 6000 € as compared to its EURO II equivalent [8].

Considering this scenario, the research community is working on developing alternative combustion concepts that allow high efficiency and low NO_x and soot emissions simultaneously, thus contributing to reduce the aftertreatment necessities. In this sense, different low temperature combustion (LTC) strategies such as the homogeneous charge compression ignition (HCCI) [9], premixed compression ignition (PCI) [10], diesel partially premixed combustion (PPC) [11], gasoline PPC [12] and PPC spark assisted [13,14] have been studied up to date. These strategies base on promoting an air-fuel premixed environment before the start of combustion by injecting the fuel in the early instants of the compression stroke and high EGR levels, which results in extended ignition delay periods and low local flame temperatures. However, the use of a single fuel confines the operating region to a very narrow range, either because of excessive pressure gradients or due to misfire conditions.

The dual-fuel LTC concept so-called Reactivity controlled compression ignition (RCCI) has been deeply investigated over the last decade [15]. This combustion concept can be easily implemented by adding a port fuel injector (PFI) to a CI engine to inject a second fuel with different reactivity than that injected directly into the combustion chamber [16]. Experimental and simulation studies have demonstrated that RCCI is capable of achieving diesel-like or better efficiency [17] together with near-zero NO_x [18] and soot emissions [19,20]. To achieve this, the effect of different variables on RCCI efficiency and emissions has been deeply investigated. In this sense, the engine settings [21,22], piston geometry [23], compression ratio [24], fuels used [25,26], auxiliary systems [27] and air management conditions [28] have been optimized. The majority of the investigations found in literature rely on RCCI to achieve NO_x values in steady-state conditions below EURO VI regulation (0.4 g/kW h) and soot emissions in the range of 0.1–0.2 FSN [29]. These results would confirm the great potential of the RCCI concept, meaning that near 60% of the total aftertreatment costs of heavy-duty vehicles could be reduced by removing the selective catalytic reduction (SCR) system [8]. However, these emissions constraints can be only fulfilled in a certain portion of the engine map [30], thus requiring the implementation of a dual-mode concept to cover the whole engine operating range.

Previous investigations have demonstrated the capabilities of two dual-mode RCCI concepts in steady-state operation, dual-mode RCCI/CDC [31] and dual-mode dual-fuel (DMDF) [32]. The major difference between these combustion strategies is that, while one of them relies on covering with conventional diesel combustion the part of the map that cannot be covered by RCCI regime, the other does it relying on dual-fuel diffusion. Both strategies have shown clear potential over the world harmonized stationary cycle (WHSC), in which NO_x emissions, urea consumption and diesel particulate filter (DPF) regeneration were substantially reduced versus CDC operation. The objective of the current work is to compare the potential of both combustion modes in transient conditions, also required by the EURO VI type approval process. For this purpose, performance and emissions of both dual-mode concepts are evaluated over the World Harmonized Vehicle Cycle (WHVC) [33], a chassis dynamometer test developed based on the same set of data used for the development of the World Harmonized Transient Cycle (WHTC) [34], defined in the EURO VI regulation [35]. Cycle simulations were carried out using a dedicated vehicle model and steady-state experimental maps of emissions and performance. Before performing the direct comparison, the effect of seven different gear shifting strategies is analyzed. Later, both dual-mode combustion concepts are compared using the optimum gear shifting strategy, analyzing their differences in terms of emissions and performance.

2. Materials and methods

2.1. Test cell characteristics and engine description

Engine tests were carried out in a single-cylinder medium-duty EURO VI diesel engine, whose main characteristics are depicted in Table 1. The dual-mode RCCI/CDC was implemented using the stock compression ratio (17.5:1), while to develop the dual-mode dual-fuel, the engine CR was reduced down to 15.3:1 by means of the piston bowl modification. The piston bowl geometry was designed following the guidelines found in previous researches [36,37].

As shown in Fig. 2, the SCE was built on the base of the production Volvo D5K240 multi-cylinder engine by isolating one cylinder. The remaining cylinders work in CDC regime controlled by the ECU to balance the cylinder-to-cylinder pressure peaks. This engine layout makes not possible to isolate torque measurements from the SCE, so that values acquired from the experiments are provided in indicated basis. The figure shows that both sides of the engine (SCE and other cylinders) are fully instrumented. To isolate the SCE side, the stock air-loop was replaced by different elements such as the screw compressor, dryer, heater and settling chamber to allow a full control during the engine tests. Same job was done in the exhaust line, where a backpressure valve was installed to reproduce the backpressure generated by the turbine in the stock configuration. Moreover, an additional air-loop was designed to redirect the EGR flow to the intake line. Gaseous emissions were sampled at the end of the exhaust line using a five gas Horiba MEXA-ONE-D1-EGR analyzer. Emissions samples were averaged during 40 s once achieved steady-state conditions. Smoke emissions were measured using an AVL 415S Smoke Meter. In this case, three samples of a 1 L volume each with paper-saving mode off were acquired for each operating point. The accuracy of the different devices used to acquire data is summarized in Table 2.

Two injection systems were used to implement the dual-fuel concept. The diesel fuel was injected using the stock solenoid injector and common-rail system. However, the stock ECU was replaced by an in-house developed Drivven controller to allow the calibration work. The main characteristics of the diesel injection hardware are depicted in Table 3. An additional fuel injection system

Table 1
Engine characteristics.

Style	4 Stroke, 4 valves, direct injection
Manufacturer/model	VOLVO/D5K240
Maximum in-cylinder pressure	190 bar
Bore × stroke	110 mm × 135 mm
Connecting rod length	212.5 mm
Crank length	67.5 mm
Unitary displaced volume	1275 cm^3
Compression ratio (nominal)	17.5:1
Compression ratio for RCCI	15.3:1