A RCCI Operational Limits Assessment in a Medium Duty Compression Ignition Engine Using an Adapted Compression Ratio

Jesús Benajes, José V. Pastor, Antonio García* and Vicente Boronat

CMT - Motores Térmicos, Universitat Politècnica de València, Camino de Vera s/n, 46022 Valencia, Spain

Corresponding author (*):

Dr. Antonio García (angarma8@mot.upv.es)

Phone: +34 963879659

Fax: +34 963877659

Highlights

RCCI with CR 12.75 reaches up to 80% load fulfilling mechanical limits

Ultra-low levels in NOx and soot emissions are obtained in the whole engine map

Ultra -high levels of CO and uHC have been measured overall at low load

RCCI improves fuel consumption from 25% to 80% engine loads comparing with CDC

Keywords

RCCI (Reactivity controlled compression ignition), Euro VI limits, CDC (Conventional Diesel Combustion), commercial fuels, compression ratio

Abstract

Reactivity Controlled Compression Ignition concept offers an ultra-low nitrogen oxide and soot emissions with a high thermal efficiency. This work investigates the capabilities of this low temperature combustion concept to work on the whole map of a medium duty engine proposing strategies to solve its main challenges. In this sense, an extension to high loads of the concept without exceeding mechanical stress as well as a mitigation of carbon oxide and unburned hydrocarbons emissions at low load together with a fuel consumption penalty have been identified as main Reactivity Controlled Compression Ignition drawbacks. For this purpose, a single cylinder engine derived from commercial four cylinders medium-duty engine with an adapted compression ratio of12.75 is used. Commercial 95 octane gasoline was used as a low reactivity fuel and commercial diesel as a high reactivity fuel. Thus, the study consists of two different parts. Firstly, the work is focused on the development and evaluation of an engine map trying to achieve the maximum possible load without exceeding a pressure rise rate of 15 bar/CAD. The second part holds on improving fuel consumption and carbon oxide and unburned hydrocarbons emissions at low load. Results suggest that it is possible to achieve up to 80% of nominal conventional diesel combustion engine load without overpassing the constraints of pressure rise rate (below 15 bar/CAD) and maximum pressure peak (below 190 bar) while obtaining ultra-low levels of nitrogen oxide and soot emissions. Regarding low load challenges, it has developed a particular methodology sweeping the gasoline-diesel blend together with intake temperature or exhaust gas recirculation maintaining constant the combustion phasing and ultra-low nitrogen oxide and soot emissions. As a result a drastic decrease carbon oxide and unburned hydrocarbons emissions is obtained with a slight fuel consumption improvement.

1. Introduction

Nowadays, for medium and heavy-duty applications, compression ignition engines are the most widely used all around the world. These engines are usually operated under conventional diesel combustion (CDC). This strategy has a clear diffusion combustion behavior governed by the injection timing. Thus, combustion phasing can be controlled with high precision. Consequently, high thermal efficiency is achieved. Despite compression ignition (CI) engines work with lean mixture, this strategy produces fuel-rich equivalences ratios due to mixture stratification. As a consequence, high combustion temperatures are achieved promoting nitrogen oxides (NOx) and soot formation.

In this sense, strict regulations have been introduced, in recent years, to limit pollutants emissions from CI engines. These limitations represent a challenge for the research community. Thus, present HD diesel engines require a huge exhaust after-treatment in order to meet emissions regulations, such as EURO VI. These systems are complex and imply a more expensive engine production. In addition, urea (known commercially as AdBlue) is needed to reduce NOx formations and make possible meeting the ultra-low NOx limitation. The use of these elements implies an extra cost in terms of fuel consumption, due to the penalty suffered from the DPF regeneration, and the consumable component urea from the SCR system.

In order to reduce after-treatment and fuel consumption costs [1], several advanced strategies have been developed to maintain the benefits of CDC operation, facing the trade-off between NOx and soot emissions and improving engine efficiency simultaneously [2,3]. In this sense, many researchers have focused on low temperature combustion strategies (LTC), which mitigate the NOx and soot formation while improves engine efficiency. This can be achieved due to heat transfer reduction provided by the premixing between fuel and air which generates long ignition delays and lower bulk gas temperatures. However, due to fuel premixing, chemicals kinetic controls the ignition timing and the heat release instead of mixing. Therefore, the stability of the combustion can be altered and the control of the combustion can be reduced.

Homogeneous charge compression ignition (HCCI) was widely investigated by the research community. This LTC strategy uses premixed charged of fuel and air. The combustion is dominated by the chemical kinetic due to the ignition, which depends of the pressure, temperature, equivalence ratio and fuel properties. HCCI provides higher or equal thermal efficiency than CDC mode and a huge reduction in terms of NOx and soot. However, the homogeneous cylinder charge provokes a rapid heat release occurring steeps pressure gradients. As a result the engine can be submitted under high engine stress and excessive combustion noise. Thus, HCCI has been limited up to partial load [4]. Regarding this limitation, Bessonette et al. [5] suggested that HCCI operation under different conditions would require different fuel reactivity's. In particular, low loads require high fuel reactivity and higher engine loads require low fuel reactivity.

Partially Premixed Combustion (PPC) strategies have been deeply studied [6-10]. PPC is presented with the idea of improving HCCI weaknesses in terms of controllability and knocking by using low reactivity fuels. So, PPC with gasoline allows controlling better the heat release rate providing NOx and soot emissions reduction [11-12]. By contrast, several fuel combustion studies made with different octane number fuels showed that the higher the research octane number (RON), the higher the unburning problems and dispersion cycle-to-cycle, being critical for gasolines with RON higher than 91. In addition, this problematic area overlaps with the area with major potential of the strategy in terms of NOx and soot reduction [13-14]. This resistance to ignite from the gasoline can be taken to increase the delay timing. On the other hand, this characteristic from the gasoline makes difficult to manage when it has to be burned at low load. Therefore, diesel ignites easier than the gasoline, so it is easier to burn at low load, requiring higher exhaust gas recirculation (EGR) rates while load is increased [15].

These results provided a detailed study by Park et al. [16], where the effects of fuel blends formed by diesel and gasolines were deeply studied. The study states that the gasoline in the fuel blend provides a reduction in density, kinetic viscosity and surface tension, improving the atomization process. In addition, it provides also high ignition delays enhancing a more homogeneous blend formation. As a result, the trade-off between NOx and soot is reduced. However, the emissions in terms of carbon monoxide (CO) and unburned hydrocarbons (HC) are increased. While the load is increased, the fuel blend tends to be moved forward to a high portion of diesel, worsening the benefits of this combustion previously mentioned. Regarding these conclusions, by using different fuels shows high combustion improvement potential.

Following that trend, Inagaki et al. [17] studied PCI combustion controlled by different ignitability fuels. It achieved low NOx and smoke emissions. Isooctane fuel was supplied by a port fuel injector and the diesel fuel was injected directly in the combustion chamber as ignition trigger. The ignition trigger was able to manage by modifying the portion of each type of fuel (low cetane number fuel at high load and high cetane number at low load), in other words, adjusting the reactivity of the fuel blend. Regarding these hypotheses, Kokjohn et al. [18] baptized as reactivity controlled combustion ignition (RCCI) combustion mode, injecting gasoline as a low reactivity fuel (low cetane number) and diesel as a high reactivity fuel (high cetane number). Port fuel injection (PFI) is used for gasoline and direct injection (DI) is used for diesel. Gasoline is injected generating a premixed blend of air and fuel, included EGR as well. Then, diesel is injected in one or two injections. As the high reactivity fuel is injected, added to the conditions at the combustion chamber, starts de ignition and derives into the burning of the premixed energy ratio as well. Therefore, it is possible to create different fuel blends in order to adjust the combustion phasing and the rate of heat release by controlling fuel reactivity [19].

Thus, RCCI operation mode shows a lot of potential in order to solve the main problems found at the LTC strategies. In addition, RCCI also provides ultra-low emissions in terms of NOx and soot simultaneously breaking the trade-off. This is achieved due to the premixing time, which avoids the formation of high equivalence ratios areas. Moreover, the combustion phasing is controlled by the direct injection of the high reactivity fuel and the rate of heat release is governed by the fraction of the fuels.

Despite the benefits obtained with RCCI concept, it has been appreciated some relevant challenges. In order to achieve ultra-low NOx and soot emissions at high or full loads a highly premixed combustion is needed. Thus, the maximum RCCI load is restricted by the high pressure rise rates. In this sense, it is stated the lack of RCCI experimental results in those loads. In addition, high levels of CO and unburned HC emissions have been stated in the whole engine map, but it should be highlighted its magnification at low loads. Thus, the main objective of the present work is to extend the RCCI concept to the maximum load without exceeding 15 bar/cad as the maximum pressure rise rate and fulfilling Euro VI soot and NOx limitations. Nonetheless, future works will be required in order to face the transition between different loads as well as engine speeds. In particular, the transition from one load to other load represents a challenge in terms of combustion stability.

Experimental configuration

2.1. Test cell and engine description

The experiments presented in this work were conducted using a fully instrumented test bench in which was installed the engine. The engine is a VOLVO D5K with 4 in-line cylinders and it has been modified in order to work the first cylinder as a single-cylinder diesel engine and the other 3 cylinders will work with the stock configuration. Main specifications of the engine are shown in the Table 1. The engine is a EURO VI medium-duty diesel engine developed for urban freight distribution purposes. Despite the engine has been presented as EU VI new engine, the after-treatment system has been removed, and even the high pressure EGR loop.

Table 1. Main specifications of the Volvo D5K diesel engine.

Style	*4 Stroke, DI diesel engine*
Manufacturer / model	*VOLVO / D5K240*
OEM EVO calibration	*EURO VI*
Piston bowl geometry	*Re-entrant*
Maximum power	*177 kW @ 2200 rpm*
Maximum brake torque	*900Nm@1200-1600 rpm*
Maximum in-cylinder pressure	*190 bar*
Maximum injection pressure/Nº injections	*2000 bar/3*
Bore x Stroke	*110 mm x 135 mm*
Connecting rod length	*212.5 mm*
Crank length	*67.5 mm*
Total displaced volume	*5100 cm^3*
Number of cylinders	*4*
Compression ratio (Stock)	*17.5:1*
Compression ratio tested for RCCI	*12.75: 1*

Regarding the test bench, it is fitted with all the equipment necessary to operate and control as it can be seen in Figure 1. The set–up found in this test bench is quite particular because of the hybrid solution developed to operate with a single-cylinder engine. The engine is not a conventional SCE research engine, it is a hybrid between a multi cylinder engine (MCE) and a SCE. A cylinder of the engine has been isolated and allows studying it as a conventional SCE. On the other hand, the three remaining cylinders are driven by the original equipment manufacturer (OEM) engine control unit (ECU) with the aim to balance the cylinder-to-cylinder maximum pressure and load. Similar test cell configurations are found in [21][22][28].

In Figure 1, it can be seen that the MCE is also fully instrumented allowing to have monitored the engine during its operation. In addition, the in-cylinder pressure from both parts of the engine (cylinders numbers 1 and 4) are monitored in real-time in order to balance the crankshaft forces. Therefore, conditions of the MCE were modified during the studies done at the SCE with the aim to provide similar maximum pressure, engine load and combustion phasing. It is worthy to remark that, the SCE was the only part of the engine studied. As one cylinder was removed from the stock configuration in terms of air management, EGR system was annulled to compensate the part of the inlet gas at the turbine from the isolated cylinder. It was done with the aim of preserving the surge phenomenon in the turbocharger due to a lack of mass flow. This compensation was done with the aim of preserve the potential of turbocharging of the engine by itself during the test campaign. With the 100% of pressure available from the turbocharger, forces along the crankshaft will be compensated by adjusting the same indicated mean effective pressure in both cylinders, number 1 (SCE) and the number 4.

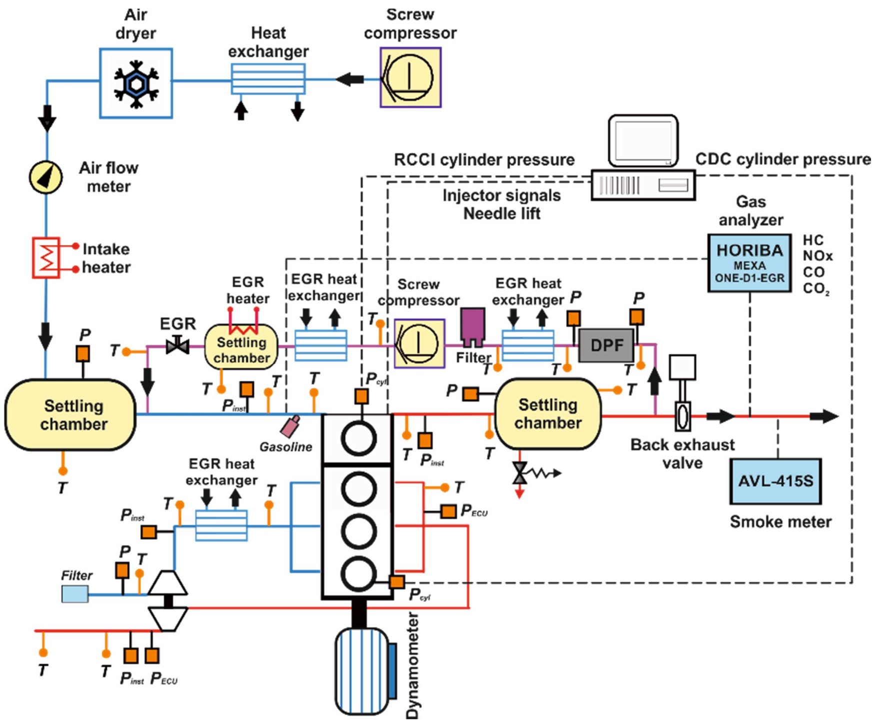

Figure 1. Test bench scheme.

As it has been said previously, SCE is needed to be fitted with a new air management system. This new system represents the most part of the work done to isolate the SCE. Thus, a screw compressor supplies the boost pressure required by the controller and is dried by an air dryer in order to ensure stable intake air conditions. Fresh air is measured by the flow meter and then is passed through a settling chamber to avoid intake engine pulses. As it can be observed in Figure 1, the intake air temperature is controlled in the intake manifold after mixing with the EGR flow. EGR is obtained from the exhaust line from the SCE. Therefore, low pressure EGR was produced taking exhaust gases from the exhaust settling chamber. As the SCE is isolated, the back pressure produced by the turbine in the stock engine is replicated by a valve placed in the exhaust system. This valve controlled the air pressure inside the settling chamber. Moreover, the exhaust gases are passed through an EGR conditioner before being introduced at the intake manifold. In order to replicate low pressure EGR and to be able to control EGR supply, EGR flow was cleaned with a DPF, condensates were removed and then it was pressured. It is worthy to note that it was not possible to introduce the EGR before the screw compressor, so it was needed to increase the EGR pressure over the intake manifold pressure to introduce the EGR flow in the intake manifold.

The determination of the EGR rate was carried out using an experimental measurement of intake and exhaust carbon dioxide (CO2) concentration. The equation used to determine the EGR concentration is presented below. Additionally, the symbols used are also detailed.

$$\mathrm{EGR}[\%] = \frac{\mathrm{CO2_{int}} - \mathrm{CO2_{amb}}}{\mathrm{CO2_{exh}} - \mathrm{CO2_{amb}}}$$

- $CO2_{int}$: CO2 concentration measured at the inlet manifold.
- $CO2_{exh}$: CO2 concentration measured at the exhaust manifold.
- $CO2_{amb}$: CO2 concentration at the ambient. This value is introduced by hand to the analyzer because it its standard in the atmosphere.

Emissions concentration of NOx, CO, unburned HC, intake and exhaust CO2, and oxygen (O2) were analyzed with a five gas Horiba MEXA-ONE_D1_EGR analyzer bench by averaging 40 seconds after attaining steady state operation. In addition, CO and unburned HC results were used to obtain the combustion efficiency as[29]:

$$\mathrm{Comb.Eff} = \left(1 - \frac{\mathrm{HC}}{\mathrm{mf}} - \frac{\mathrm{CO}}{4 \cdot \mathrm{mf}}\right) \cdot 100 \qquad (1)$$

Where HC is the measured unburned hydrocarbons; CO refers to the measured carbon monoxide and mf is the total fuel mass. The units for the parameters are g/s.

An AVL 415S Smoke Meter was used to measure smokes emissions. Measurements were averaged between three samples of a 1 liter volume each with paper-saving mode off, providing results directly in FSN (Filter Smoke Number) units. An AVL 415S Smoke Meter was used to measure smokes emissions. Measurements were averaged between three samples of a 1 liter volume each with paper-saving mode off, providing results directly in FSN (Filter Smoke Number) units. Therefore, particulate matter measurements of FSN were transformed into specific emissions (g/kWh) by means of the factory AVL calibration.

Regarding the in-cylinder pressure, the signal was measured with a Kistler 6125C pressure transducer coupled with a Kistler 5011B10 charge amplifier. In order to have a crank angle degree (CAD) reference, it was used a shaft encoder with 1800 pulses per revolution, providing a resolution of 0.2 CAD.

All sensors, transducers and analyzers were calibrated by applying traditional or manufacturers recommended methods. The table 2 shown below summarizes the accuracy of the instrumentation used in this work.

Table 2: Accuracy of the instrumentation

Accuracy of the instrumentation used in this work.			
Device	*Manufacturer and model*	*Variable measured*	*Accuracy*
Piezoelectric transducer	*Kistler 6125B*	*In-cylinder pressure*	*±1.25 bar*
Piezoresistive transducers	*Kistler 4045A10*	*Intake and exhaust pressure*	*±25 mbar*
Thermocouple	*TC direct K Type*	*Temperature in settling chambers and intake manifold*	*±2.5 □C*
Encoder	*AVL 364*	*Crank angle, Engine speed*	*±0.02 cad*
Gas analyzer	*HORIBA Mexa 7100DEGR*	*NOX, CO, HC, CO2, O2*	*4%*
Smoke meter	*AVL 415*	*FSN*	*±0.025 FSN*

Fuel balance	*AVL 733S*	*Fuel mass flow*	*±0.2%*
Air flow meter	*Elster RVG G100*	*Air mass flow*	*±0.1%*

2.2. Test fuels

For the present study, it has been selected regular European diesel (EN590) and regular 95 octane gasoline. This selection was made to study the potential of RCCI with CR(12.75) and with the commercial fuel which is possible to find in any petrol station. Their main properties according to auto-ignition are listed in Table 33. All the properties were obtained following RD 1700/2003 which is in charge for the commercial fuels.

Table 3. Physical and chemical properties of the fuels used along this study.

	Diesel EN590	*95*
Density [kg/m^3] (T= 15 °C)	*820*	*720*
Viscosity [mm^2/s] (T= 40 °C)	*2.00*	*-*
RON [-]	*-*	*95.0*
MON [-]	*-*	*85.0*
Biodiesel content by volume [%]	*<0.2*	*-*
Ethanol content by volume [%]	*-*	*5*
Cetane number [-]	*51*	*-*
Lower heating value [kJ/kg]	*42.97*	*42.4*

The engine was equipped with a mixed injection system in order to allow RCCI operation. RCCI requires gasoline injection and diesel injection in the same cylinder and in the same cylinder cycle. The scheme of the injection system used at the test bench can be observed in Figure 2. The mixed system allows varying the in-cylinder fuel blending ratio and fuel mixture properties for each engine operating condition. The engine control is done by a real time National Instruments powertrain control system with two controllers, combining a field-programmable gate array (FPGA) based synchronization of the injection, and a peripheral component interconnect (PCI) extensions for instrumentation (PXI) system is used for the in-cylinder pressure acquisition and processing. The control software was developed in-house and allows performing transitions between different combustion modes, and a closed loop control of the combustion characteristics.

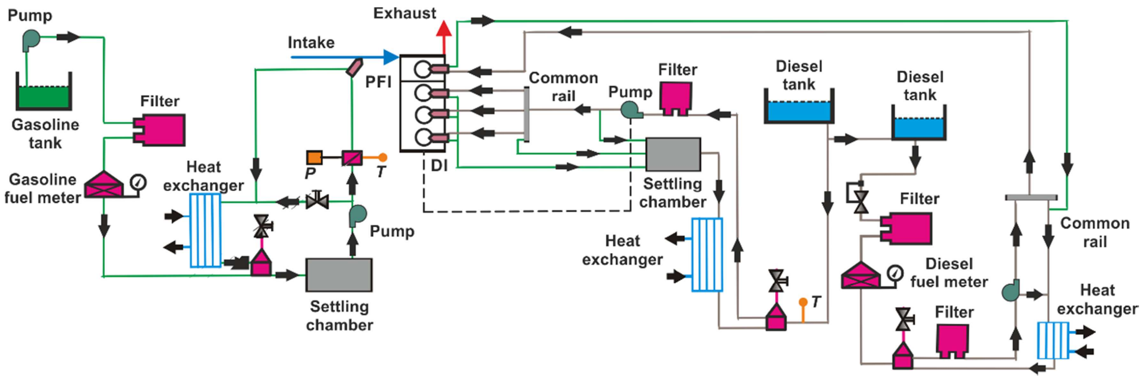

Figure 2. Mixed fuel injection scheme.

From the fuel injection scheme, it is important to note that there are two types of injector. There are a direct diesel injector and a port fuel injector. The diesel injector is the genuine of the engine. But for this case, it is driven by the controller developed in-house instead of being driven by the OEM ECU. Main properties of the diesel injector are presented in Table 44.

Table 4. Main characteristics of the diesel injector.

Actuation Type	*Solenoid*
Steady flow rate @ 100 bar [cm^3/min]	*1300*
Number of Holes	*7*
Hole diameter [μm]	*177*
Included Spray Angle [°]	*150*
Maximum injection pressure (bar)	*2000*

Regarding the port fuel injection (PFI) system, the fuel circuit was located at the intake manifold. The system used consisted of several parts in order to provide an injection and the measurements needed for the current study. Therefore, the port fuel injection system was fitted with a reservoir, a fuel filter, a fuel pump, a fuel meter, a heat exchanger and a commercially available port fuel injector.

In order to avoid fuel pooling, the injection timing was fixed at 10 CAD after the inlet valve opening (IVO). Thus, fuel flowed along the intake manifold crossing the distance between the PFI location and the intake valve seats.

This injector settings would avoid fuel pooling over the intake valve and the undesirable variability cycle-to-cycle produced by this phenomenon. In Table 5 are presented the main characteristics of the PFI.

Table 5. Main characteristics of the gasoline port fuel injector.

Injector Driver	Saturated
Steady flow rate @ 3 bar [cm^3/min]	980
Included Spray Angle [°]	30
Injection Pressure [bar]	5.5
Injection Strategy	Single
Start of Injection Timing	340 CAD ATDC

Both fuels use the same configuration and systems to measure the mass flow. The system used was an AVL733S flowmeter. This flowmeter operates by weighting with a gravimetrical balance the quantity of fuel which is introduced in the engine. Inside the device, there is a volume of 1 liter, in order to measure the time needed to empty the fuel. With these parameters, the flowmeter is able to calculate the mass flow of fuel used. This type of device decouples the temperature from the measurement, being dependent of the fuel density.

RCCI operating strategy and constraints

The aim of this study is to evaluate the potential of RCCI in an engine map using a SCE with a compression ratio of 12.75. In order to stablish a procedure to fulfill the whole engine map, as wide as possible, it has followed a similar procedure described in [20]. In this study is stated that maximum engine loads were limited by the pressure rise rates given during the combustion.

Moreover, in Figure 3, it can be seen the scheme used to drive the experimental tests and the constraints used to work under RCCI operation. Regarding the constraints, the maximum pressure was imposed by the engine manufacturer. The other constraints were self-imposed in order to fulfill EURO VI in terms of NOx (0.4 g/kWh) and soot (0.01 g/kWh) and to ensure the mechanical integrity of the engine, with a maximum pressure rise rate (MPRR) of 15 bar/CAD.

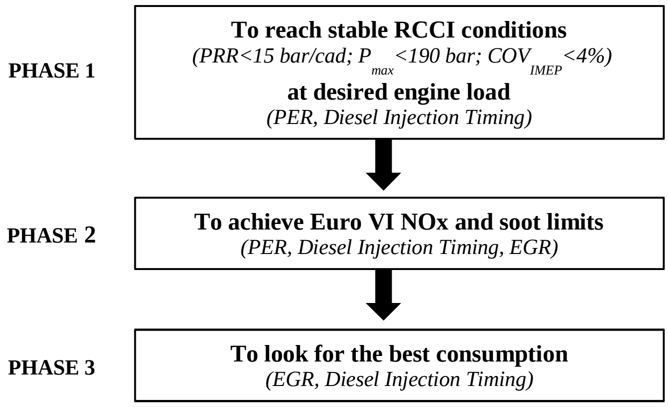

Figure 3. Strategy used to conduct the RCCI experimental tests.

Thus, the different phases are defined in Figure 3. The idea is to complete the constraints of each one before passing to the new phase. Phase 1 is in charge of reaching stable RCCI conditions in terms of MPRR and maximum pressure. Moreover, it determines the injection timing and the gasoline and diesel fraction needed to achieve the engine load. Second phase is in charge of driving RCCI into EURO VI limitations in terms of NOx and soot emissions. Finally, the third phase is used to reduce the fuel consumption trying to optimize CO and unburned emissions while the combustion process is improved.

It is worthy to note that CDC and RCCI comparison are made without considering the emissions reduction that after-treatment provides to CDC mode.

In table 6 are presented the main settings of the air management and the injection pressure used for all the engine speeds and engine loads. Those values have been kept constant during the test campaign.

Table 6. Air management settings of the tested points.

Engine Load[%]	Intake Pressure [bar]	Intake Temperature[°C]	Exhaust Pressure[bar]	Injection pressure[bar]
10	1.6	60	1.8	1200
25	1.6	60	1.8	1000
50	2.4	20	2.6	1200
80	3	20	3.2	1400

2.3. In-cylinder pressure signal analysis

The analysis of the combustion was performed with an in-house one-zone model named CALMEC, which is fully described in [24]. In order to diagnose the combustion, CALMEC mainly needs some mean variables (temperatures of the coolant, oil, inlet manifold and exhaust manifold, EGR and fuel mass flow, air mass flow, engine speed, and so on) and the in-cylinder pressure signal.

The pressure traces from 110 cycles were recorded in order to compensate the cycle-to-cycle dispersion during the engine operation. Thus, the individual pressure data of each engine was smoothed using a Fourier series low-pass filter. After the process of signal filtering, the collected cycles were ensemble averaged to yield a representative cylinder pressure trace, which was used to perform the analysis. Then, the first law of thermodynamics was applied between intake valve closing (IVC) and exhaust valve opening (EVO), considering the combustion chamber as an open system due to the blow-by and fuel injection. The ideal gas equation of state was used to calculate the mean gas temperature in the chamber. In addition, the in-cylinder pressure signal allowed obtaining the gas thermodynamic conditions in the chamber to feed the convective and radiative heat transfer models [25], as well as the filling and emptying model that provided the fluid-dynamic conditions in the ports, and thus the heat transfer flows in these elements. The convective and radiative models are linked to a lumped conductance model to calculate the wall temperatures.

The main result of the model used in this work was the rate of heat release (RoHR), the bulk gas temperature and the maximum pressure gradient in the combustion chamber. Moreover, several parameters were calculated from the RoHR profile. Particularly, the start of combustion (SoC) was defined as the crank angle position in which the cumulated heat release reached a value of 2% and combustion phasing was defined as the crank angle position of 50% fuel mass fraction burned (CA50).

3. RCCI Operational limits

3.1. Low load cases

Figure 4 presents indicated raw emissions and performance for the low load engine cases, in particular for 10% and 25% engine loads. In this sense, nitrogen oxides (NOx), soot, carbon monoxide (CO) and unburned hydrocarbons (HC) are presented. In addition, normalized indicated fuel consumption (ISFC), gross indicated mean effective pressure (IMEP), maximum pressure and pressure rise rate (PRR) are also shown. Both, emissions and performance, are directly compared between RCCI and CDC operation modes. EURO VI limits are included in the sub-figures as a reference to be compared. It is interesting to note that the RCCI results are obtained by using a reduced CR (12.75) and the CDC with the standard CR (17.5).

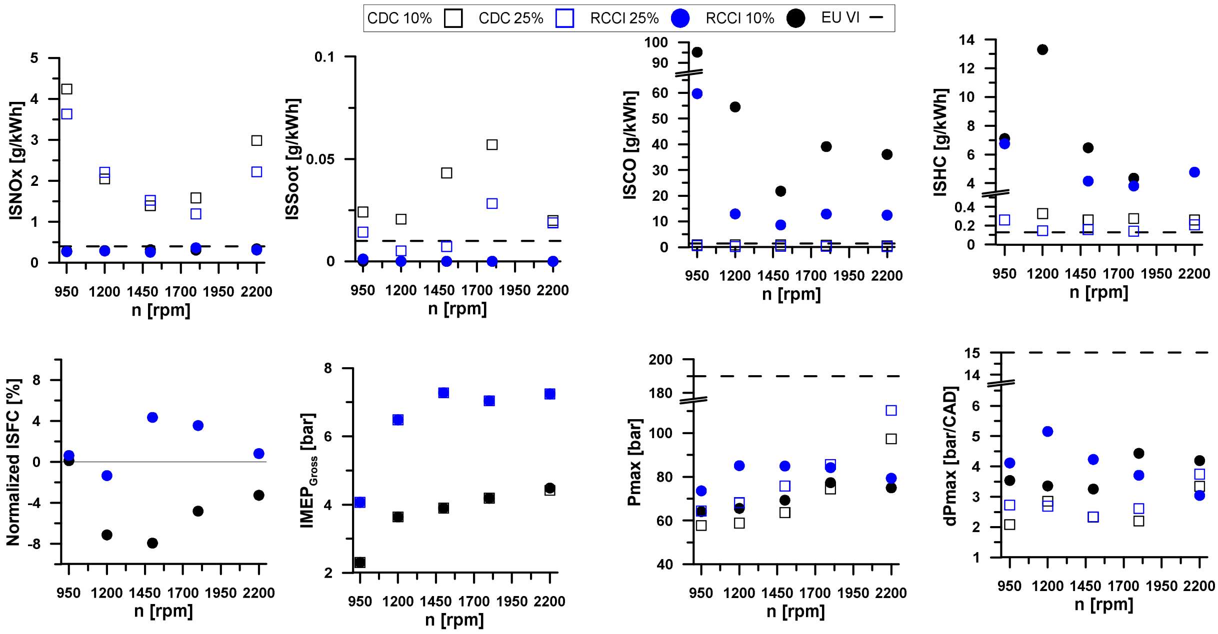

Figure 4. Comparison of emissions and performance between RCCI and CDC at 10% and 25% loads at different engine speeds

Considering NOx emissions, independently on engine load and speed, it is possible to state that RCCI values are below EURO VI limitation (0.4 g/kWh) as well as lower than CDC results. This behavior is explained by a better air fuel mixing and lower temperature for RCCI cases. Thus, observing Figure 5, it can be proved that, when RCCI is operated, the mixing time is increased (higher Ca10-EoI) and the mixture will become locally leaner than CDC. In addition, the in-cylinder gas temperature found for RCCI is also lower, in spite of having CA50 closer to TDC, due to less energetic combustion (lower RoHR peak). As a result, areas with high reactivity and high temperature practically disappear and therefore the NOx formation is mitigated.

Regarding soot emissions, for all engine load and speeds, it is worthy to note that all RCCI values fulfill Euro VI limit (0.01 g/kWh). In addition, lower results than CDC cases are also attained. These trends are explained by the extra mixing time (CA10-EoI) obtained for RCCI cases compared to CDC as it is shown in Figure 5. This promoted premixing behavior for diesel injection implies a reduction in soot formation and therefore lower soot emissions are achieved for RCCI cases.

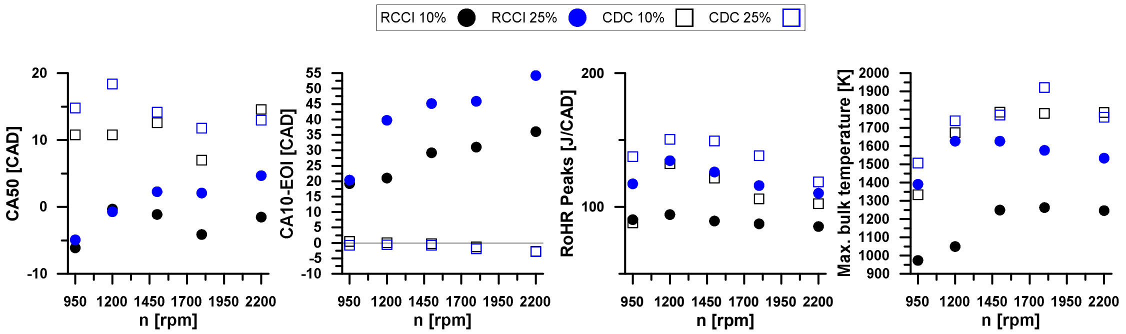

Figure 5. Combustion phasing (CA50), mixing time (CA10-EOI), rate of heat release peaks (RoHR) and maximum bulk temperature for 10% and 25% engine loads

Focusing on carbon monoxide (CO) and unburned hydrocarbons (HC), it is observed that the results are far from Euro VI limits at RCCI operation mode. Additionally, higher results than in CDC operation are obtained. This trend is more evident when engine load is decreased. Particularly for CO, RCCI results are explained due to the difference found at the bulk gas temperature presented in Figure 5. For the cases with low engine speed, 950 rpm and 1200 rpm and 10% engine load, the maximum bulk temperature is below 1200K, so it does not promote the oxidation process. As a result the higher values of CO emissions are obtained. This CO behavior with engine speed variation is also found at 25% load.

In Figure 6 the premixed energy ratio (PER), the EGR, the combustion efficiency, the coefficient of variation of the IMEP, the number of diesel injections and the start of injection (SoI) of both, pilot and main injections. rates are presented also for low load engine cases by operating under RCCI at both loads. As it can be observed in the figure 6, the premixed energy ratio follows a similar trend observed in the IMEP. This behavior is due to the fuel burning capabilities. At 10% load, combustion chamber has not enough high thermodynamic conditions to ensure the burned of all the gasoline, resulting in high unburned HC if the premixed energy ratio increases. At 25% engine load the trend is similar but the values of premixed energy ratio are higher. Nonetheless, once the IMEP is almost constant (between 1500rpm-2200rpm) the premixed energy ratio keeps quite constant as well. The trend observed in EGR rates is explained with the trend observed on the combustion phasing. Combustion phasing is adjusted with the reactivity of the fuel and with the EGR. As the combustion phasing is stable at the same point from 1500 rpm to 2200 rpm, the EGR decreases while the premixed energy ratio remains almost constant (at 10% is balanced the evolution of the premixed energy ratio and the EGR rate).

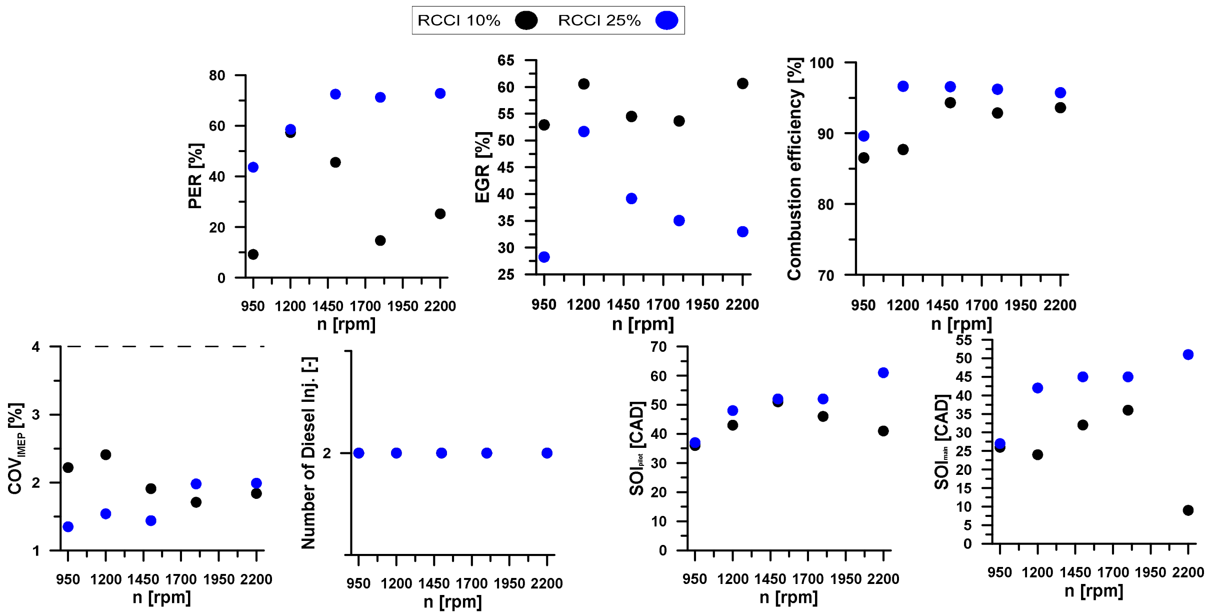

Figure 6. Premixed energy ratio (PER), EGR rate, coefficient of variation of the IMEP, number of injections and SoI for the pilot and main injections for 10% and 25% engine load

Regarding unburned HC, it is well stated that this raw emission correlates with the amount of gasoline in the fuel blend as well as with the maximum energy released during the combustion [23]. In this sense, the higher the reactivity of the fuel blend (lower amount of gasoline) and/or the higher the energy released in the combustion, the lower the HC. Thus, this is confirmed with the unburned HC measured at 10% and 25% engine loads.

Regarding ISFC measurement and for direct comparison of RCCI operation and CDC results without introducing deviations associated to the lower heating value

(LHV), the total injected mass for all specific parameters is calculated in grams of diesel energy equivalent as follows:

$$m_{fuel} = m_{Diesel} + m_{Gasoline} \cdot \left(\frac{LHV_{Gasoline}}{LHV_{Diesel}}\right)$$

For explaining RCCI ISFC results, it should be considered CA50, IMEP, bulk gas temperature, CO and HC. Thus, comparing with CDC and looking at Figure 5, it is possible to state that RCCI cases are presenting better combustion phasing (closer to TDC), lower heat transfer maintaining IMEP (lower bulk gas temperature) which would imply better ISFC. By contrast, drastically worse CO and HC emissions are attained for RCCI which would involve an increase in ISFC. Thus, in the 10% load cases seem that the worsening in the combustion efficiency (CO and HC) has more impact that the benefit in terms of combustion phasing and heat transfer. As a consequence, an increase in ISFC results is obtained. On the other side, for 25% load cases, the opposite trend is attained and therefore better ISFC is presented. To complete performance analysis, observing Figure 4, similar P_{max} and PRR values for RCCI cases than in CDC are obtained. The PRR has been maintained below 15 bar/cad for all cases, guaranteeing the mechanical engine integrity.

3.2. Medium and high load cases

As it has been presented in the previous study, in the current section, medium and high loads have been studied for different engine speeds. Figure 7 presents indicated raw emissions which contain NOx, soot, CO and unburned HC results. Regarding the engine performance, it has been attached to the Figure 7 the maximum in-cylinder pressure (Pmax), gross indicated mean effective pressure (IMEP), pressure rise rate (PRR) and fuel consumption normalized by the CDC fuel consumption.

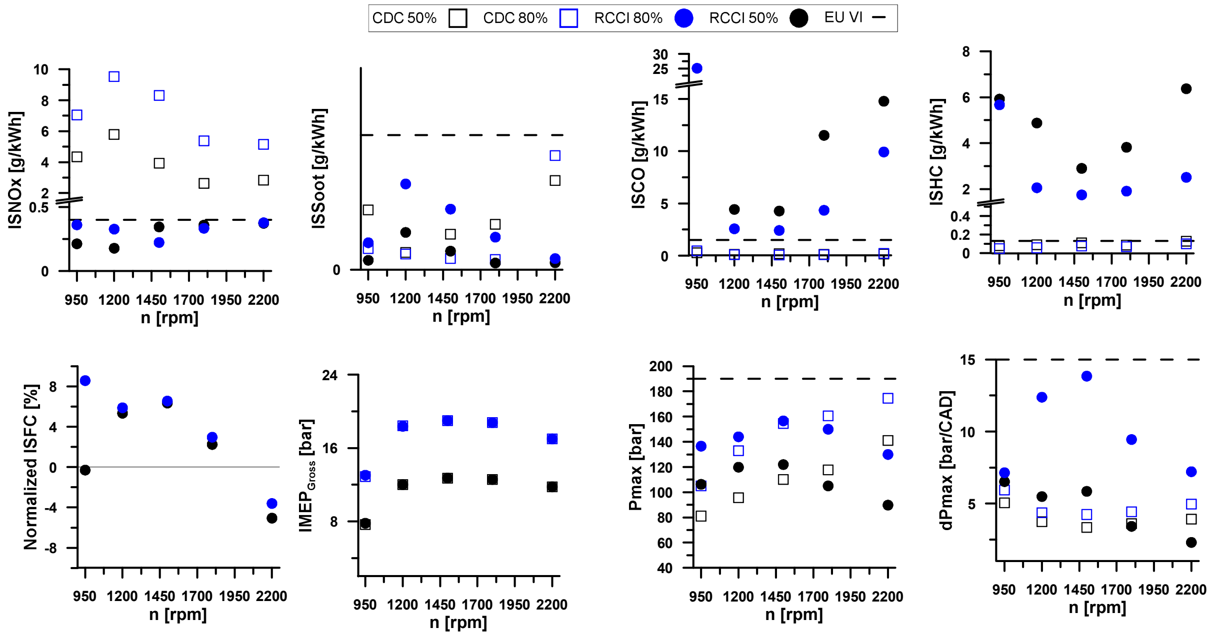

Figure 7. Comparison of emissions and performance between RCCI and CDC at 50% and 80% loads at different engine speed

Focusing on NOx emissions, it is seen that when the engine load is higher, it becomes to be more critical to fulfill Euro VI limitations. Nonetheless, the limit is accomplished in all the cases. Thus, considering that the production engine calibration implies the use of SCR it can be stated that the higher the load, the greater the NOx emissions when the engine is operating under CDC conditions. So, a drastic reduction is obtained with RCCI mode in all the values. This relevant NOx reduction has been achieved by using high EGR rates and high premixed energy ratio. The bulk gas temperature is only reduced, comparing to CDC mode, for some engine speeds. Thus, one of the most important factors to provide low NOx emissions is the mixing time [27]. The mixing time, as it can be observed in Figure 8, is increased as the engine speed does. In addition, premixing allows increasing EGR rates without exceeding soot limits. By increasing the mixing time, the mixture becomes more homogeneous avoiding fuel-rich regions and maximum fuel mass concentration. This combination of high EGR rates and higher mixing time promotes low NOx and soot formation, demonstrating that is capable to break the trade-off existing in CDC mode.

In soot emissions, CDC fulfills, at medium and at high engine loads, Euro VI limitation (0.01 g/kWh). In this sense, RCCI operation is not providing an additional benefit compared to nominal engine calibration. RCCI soot results are also below the limit; nevertheless, the mechanisms which lead to these results are different between both combustion modes. In the CDC case, the higher the load, the higher the in-cylinder temperature, promoting the soot formation and also the soot oxidation. In these conditions, the oxidation process mainly governs the soot balance and therefore the raw emissions. By contrast, in the RCCI cases, is the low soot formation which mainly governs the low raw emissions. Figure 8 shows the extra mixing time for the diesel injected. This extra mixing time avoids local rich equivalence-ratios, becoming the fuel blend more homogeneous. This mixture reduces drastically the soot formation, being more impressive at high engine speed, as it is presented in figure 7.

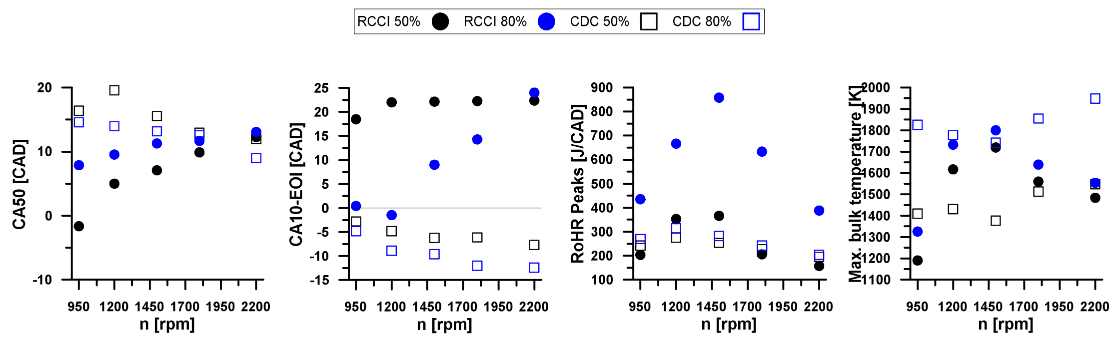

Figure 8. Combustion phasing (CA50), mixing time (SOC-EOI), rate of heat release peaks and maximum bulk gas temperature for 50% and 80% engine loads

Considering Figure 7, CO and HC results for RCCI present similar trends in both loads independently on the engine speed. Results are far from Euro VI limits as well as from CDC results. Comparing with low engine loads at RCCI operation, it is observed a great reduction in CO emissions. This great reduction is driven mainly by the maximum bulk gas temperature reached during the combustion; the bulk temperature has been over 1200K in all cases improving CO oxidation process. By contrast, HC levels are quite similar between low and medium loads decreasing for high loads. Thus, despite of increasing load improves the combustion process providing higher RoHR peaks and bulk temperatures, the HC levels remain similar since the amount of gasoline injected is also increased. In Figure 9, as literature demonstrates [28], the higher the gasoline amount, the higher the HC measured. As expected, the lower HC and CO emissions correspond to those engine speeds where the combustion stability and PRR are the highest (<15 bar/cad in all cases).

In Figure 9 is presented premixed energy ratio (PER), the EGR, the combustion efficiency, the coefficient of variation of the IMEP, the number of diesel injections and the start of injection (SoI) for the pilot and main injections for both RCCI loads. The trend followed by the premixed energy ratio and the EGR are clearly seen in the figure 9. These trends are explained due to the combustion phasing. At low engine speed, the combustion phasing must be delayed in order to reduce NOx emissions. In order to delay the CA50, the fuel reactivity must be lowered. Thus, the premixed energy ratio or the EGR must be increased (or both, depending of each particular situation). Particularly, at 950 rpm it is not possible to increase the premixed energy ratio due to high emissions in terms of CO and unburned HC. Once the engine speed increases, it is possible to increase the premixed energy ratio while the EGR rate remains quite constant. Consequently, CA50 is delayed. This is observed in figure 9 and the combustion phasing and the emissions are shown at figure 7 and 8.

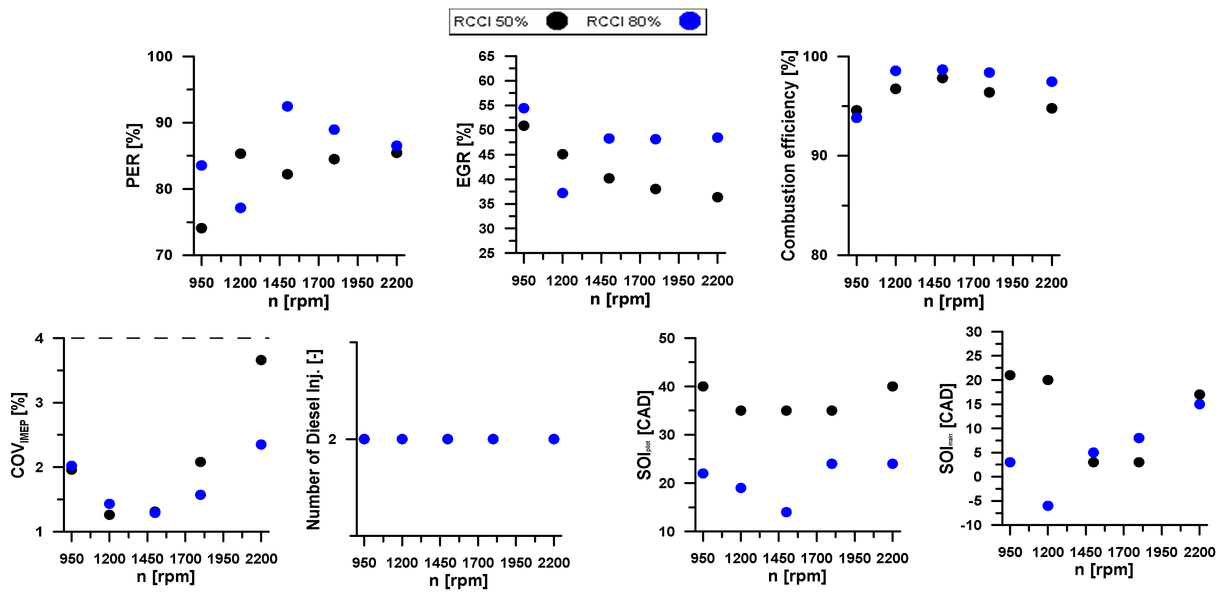

Figure 9. Premixed energy ratio (PER), EGR rate, coefficient of variation of the IMEP, number of injections and SoI for the pilot and main injections for 50% and 80% engine load.

Despite the worsening on the heat transfer, due to a higher rate of heat release peaks, it is possible to state that indicated fuel consumption is improved in almost all the cases studied. This is mainly due to a better combustion phasing (CA50 closer to top dead center (TDC)) for RCCI compared to CDC operation mode, as it can be observed at Figure 8. Additionally, similar “U-shape” trend observed for the normalized ISFC is also observed at CO and unburned HC emissions. Thus, a direct relationship between combustion efficiency and ISFC is also demonstrated. On the other hand, for the highest engine speed, the ISFC is worsened. This occurs owing to drastically worse combustion efficiency. The combustion phasing is delayed, compared to the CDC engine. As a result, as the maximum pressure as the pressure rise rate (PRR) decreases compared with the values obtained in other engine speeds. Compared with the CDC operation mode, the maximum pressure also is decreased. As a consequence, a reduction is observed at the bulk gas temperature, in Figure 8. Therefore, as it can be seen in Figure 7, the CO emissions and unburned HC are increased, confirming the worsening in the combustion process.

From the Figure 7, it is stated that RCCI engine mapping has been possible reached up to 80% load (load referred to CDC). As it can be seen, for all cases it is not exceeded the constraints of PRR < 15 bar/cad and the maximum pressure P_{max} < 190 bar. This fact implies that is not possible to attain RCCI full load conditions under current engine configuration.

4. Low load ISFC improvement

As it has been seen in the last results (Figure 4 and Figure 7), a low CR (12.75) allows fulfilling NOx and soot limits in almost the whole engine map. By contrast, at low loads, this CR worsens the indicated specific fuel consumption (ISFC) independently on the engine speed provoking also high amounts of CO and HC. At least for the operation conditions tested in the previously presented engine!. Thus, considering the first set of tests, 10% load cases are especially critical. In that load, the ISFC worsening is around 10% in all engine speeds. For the rest of engine loads and speeds (except for 2200 rpm), the ISFC is improved comparing with CDC mode.

With the aim of improving ISFC at 10% engine load, it has been proposed an additional test procedure [26]. CA50's obtained in previous tests (Figure 5) allow meeting Euro VI limitations in terms of NOx and soot. Moreover, this combustion phasing provides a guarantee for the mechanical engine components. Thus, it has been decided to keep constant the combustion phasing as well as main and pilot diesel injection timings sweeping the EGR rate and the intake temperature. In order to maintain the combustion phasing constant during the sweeps, the premixed energy ratio (PER) is adjusted.

4.1. Sweep operation procedure and analysis

Figure 10 presents two different sweeps which have been performed at 1200 rpm and at 10% engine load. In both sweeps, it is presented the normalized indicated fuel consumption (ISFC) and the emissions in terms of carbon monoxide, unburned HC, soot and NOx. Two different procedures have been used. Thus, for figure 10 left cases, diesel injections timings and the intake temperature are maintained constant. So, different EGR rates and premixed energy ratios have been adjusted to guarantee the same combustion phasing (CA50). For figure 10 right cases, the tests have been conducted with a similar procedure. For this case, the EGR rate has been kept constant instead of the intake temperature. Then, adjusting the PER, it has been possible to achieve the CA50 constant along the study. The rest of the engine settings are constant. Equivalence ratio has held constant also along the sweeps.

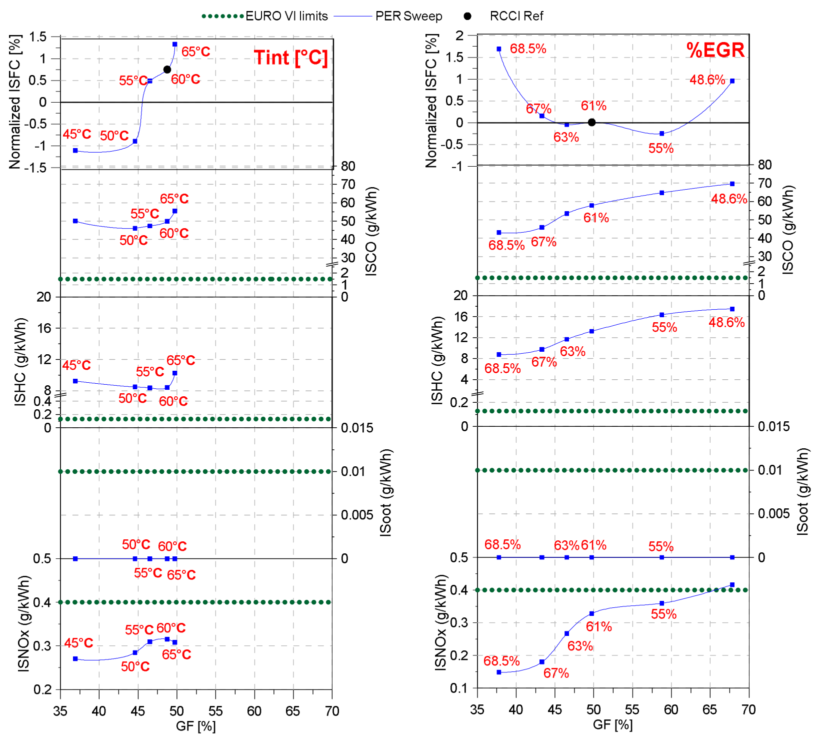

Figure 10. PER&EGR and Tintake&PER sweep at 1200 rpm and 10% load

Regarding EGR and PER sweep, 6 steps have been tested. Considering emissions in terms of NOx, CO and unburned HC, it can be observed that all of them follow a similar trend. All the emissions, except soot, increase as PER is increased and EGR is decreased. By contrast, soot emissions remain almost constant during all the study. Fuel consumption has been slightly improved, but it is not very sensitive with the EGR and the PER variations.

Focusing at the intake temperature and PER sweep, 5 steps have been tested varying 20 ºC the intake temperature. These steps only imply a variation of 15% in the premixed energy ratio to maintain the CA50. As expected, as the intake temperature is increased, the PER also is increased. Considering emissions, as previously stated, all of them show very low variation. Only NOx emissions clearly increase as the temperature and the PER also increase. Soot emissions remain almost constant during all the study. Fuel consumption tends to increase as the intake temperature is decreased. In addition, a slight improvement is observed when the intake temperature reaches the maximum temperature studied.

It can be stated that in the range tested, the best results are offered by the EGR and PER sweep. These results indicate that the best option to optimize the combustion is to reduce gasoline and increase EGR.

The same study has been also performed for 1500 rpm and 1800 rpm at 10% load. Both test procedures have been tested. Nonetheless, only the best fuel consumption result is presented for each engine speed in figure 10. In both cases, the combination of EGR rate and gasoline blends maintaining constant the intake temperature provides the best results. Thus, In figure 11 is presented the results of the different sweeps performed in terms of normalized fuel consumption for all the engine speeds tested. These results are directly compared with the previous results presented in Figure 4 and corresponding with 10% engine load.

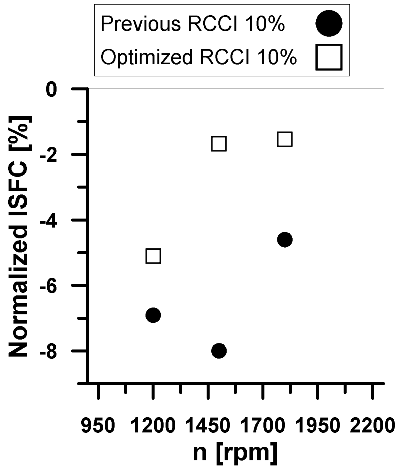

Figure 11. Comparison of the normalized ISFC for 10% load.

Despite the benefits demonstrated along the tests performed in the study, compared to previous RCCI results, it does not present an improvement is ISFC compared with CDC in any case. So, the consumption, at this low load condition, continues being a challenge when RCCI is operated using low CR.

5. Conclusions

The present study is focused on the RCCI capabilities of a SCE derived from a serial production EURO VI medium-duty multi-cylinder diesel engine. The SCE has been fitted with a CR of 12.75 with the aim of exploring the benefits along the whole engine map in terms of performance and raw emissions. A commercial 95 octane gasoline was used as a low reactivity fuel and commercial diesel was used as a high reactivity fuel. The most relevant conclusions are:

a. A specific engine procedure to ensure mechanical and environmental constraints has been performed. Thus, the RCCI mapping has revealed that the maximum load achievable for RCCI operation mode and CR=12.75, fulfilling all the constraints, was 80%.
b. In terms of raw emissions, RCCI combustion is able to meet EURO VI limitations in terms of NOx and soot limiting the exhaust after-treatment. By

contrast, high levels of CO and unburned HC were measured. These levels are higher at low load, where low reactivity fuel is more difficult to be oxidized.

c. In terms of normalized fuel consumption, RCCI operation mode provides an improvement in fuel consumption from 25% up to 80% engine loads comparing to the CDC operation mode. By contrast, fuel consumption is worsened at low load (10%) and in all engine speeds.
d. Regarding the high levels of CO and unburned HC, they should be after-treated with a DOC. The low temperature registered at the exhaust gases can reduce the DOC efficiency considerably, needing bigger systems for the same engine size. So, this is still a challenge that the concept should solve.

With the aim of improving fuel consumption, a specific methodology has been developed in order to optimize the combustion process at low load. In order to perform these tests, two sweep procedures were developed. In both procedures diesel injection settings and combustion phasing were constant. In the first procedure, intake temperature was also constant, adjusting the premixed energy ratio and the EGR rate in order to maintain the CA50 constant. For the second procedure case, EGR rate was constant and by modifying the intake temperature and the PER, the CA50 was maintained constant during the tests. Thus, the main conclusion is:

e. Slight improvement in fuel consumption has been achieved for the engine speeds tested, 1200 rpm, 1500 rpm and 1800 rpm. This reduction in fuel consumption has also a reduction in CO and unburned HC emissions. Only in one case, the fuel consumption is better than in CDC operation mode.

The study has provided these results for steady conditions, showing that RCCI operation mode is a reliable way to meet current EURO VI limitations in terms of NOx and soot. Moreover, is has been stated that RCCI can be extended for almost the whole engine map without generating mechanical stress. By contrast, future work is needed for optimization in CO, unburned HC emissions, fuel consumption for low engine load and to perform transient conditions tests.

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Abbreviations

ASTM: American Society of Testing and Materials

ATDC: After Top Dead Center

CAD: Crank Angle Degree

CA10: Crank angle at 10% mass fraction burned

CA50: Crank angle at 50% mass fraction burned

CDC: Conventional Diesel Combustion

CI: Compression Ignition

CO: Carbon Monoxide

CR: Compression Ratio

DI: Direct Injection

DPF: Diesel Particulate Filter

ECU: Engine Control Unit

EGR: Exhaust Gas Recirculation

EOI: End of Injection

EU: European Union

EVO: Exhaust Valve Open

FPGA: Field-programmable gate array

FSN: Filter Smoke Number

HC: Hydro Carbons

HD: Heavy-Duty

HCCI: Homogeneous Charge Compression Ignition

MCE: Multi cylinder engine

MPRR: Maximum Pressure Rise Rate

IMEP: Indicated Mean Effective Pressure

ISFC: Indicated Specific Fuel Consumption

IVC: Intake Valve Close

IVO: Intake Valve Open

LHV: Lower Heating Value

LTC: Low Temperature Combustion

MCE: Multi Cylinder Engine

MON: Motor Octane Number

OEM: Original Equipment Manufacturer

ON: Octane Number

PCI: Peripheral Component Interconnect

PER: Premixed Energy Ratio

PFI: Port Fuel Injection

PPC: Partially Premixed Charge

PRR: Pressure Rise Rate

PXI: PCI eXtensions for Instrumentation

RCCI: Reactivity Controlled Compression Ignition

RoHR: Rate of Heat Release

RON: Research Octane Number

SC: Screw Compressor

SCE: Single Cylinder Engine

SCR: Selective Catalytic Reduction

SOC: Start of Combustion

SOI: Start of Injection

TDC: Top Dead Center

Appendix

The calibration provided by AVL can be observed in the equation below, resulting soot emissions in [mg/m3].

$$\mathrm{Soot}\ \left[\frac{\mathrm{mg}}{\mathrm{m}^3}\right] = \frac{4.95 \cdot \mathrm{FSN}}{0.405} \cdot \mathrm{e}^{(0.38 \cdot \mathrm{FSN})}$$

Once, soot is obtained, it is transformed into [g/kwh] by using the following equation.”

$$\mathrm{Soot}\left[\frac{\mathrm{g}}{\mathrm{kWh}}\right] = \frac{\mathrm{Soot}\ \left(\frac{\mathrm{mg}}{\mathrm{m}^3}\right)}{1000} \cdot \frac{(\mathrm{m}_{\mathrm{air}} \cdot \mathrm{m}_{\mathrm{fuel}}) \cdot 3.6}{1.165 \cdot \mathrm{Ni[kW]}}$$

For the rest of the emissions values (NOx, HC and CO) it has been only carried a transformation of their units from [ppm] to [g/kwh] as it is shown below.

$$\mathrm{ISNOx}\left[\frac{\mathrm{g}}{\mathrm{kWh}}\right] = \mathrm{NOx(ppm)} \cdot \frac{(\mathrm{m}_{\mathrm{air}} + \mathrm{m}_{\mathrm{fuel}}) \cdot 1.587}{1000 \cdot \mathrm{Ni[kW]}}$$

Where 1.587 is the molecular weight of NOx.

$$\mathrm{ISCO}\left[\frac{\mathrm{g}}{\mathrm{kWh}}\right] = \mathrm{CO(ppm)} \cdot \frac{(\mathrm{m}_{\mathrm{air}} + \mathrm{m}_{\mathrm{fuel}}) \cdot 0.966}{1000 \cdot \mathrm{Ni[kW]}}$$

Where 0.966 is the molecular weight of NOx

$$\mathrm{ISHC}\left[\frac{\mathrm{g}}{\mathrm{kWh}}\right] = \mathrm{HC(ppm)} \cdot \frac{(\mathrm{m}_{\mathrm{air}} + \mathrm{m}_{\mathrm{fuel}}) \cdot 0.479}{1000 \cdot \mathrm{Ni[kW]}}$$

Where 0.479 is the molecular weight of HC.

For all these calculations the fuel mass is considered as explained in previous reviewer's combustion:

$$m_{fuel} = m_{Diesel} + m_{Gasoline} \cdot \left(\frac{LHV_{Gasoline}}{LHV_{Diesel}}\right)$$