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Operating range extension of RCCI combustion concept from low to full load in a heavy-duty engine

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Abstract

Fuel reactivity controlled compression ignition (RCCI) concept has arisen as a solution to control premixed combustion (PCI) strategies, which avoids soot and NOx formation by promoting a lean air-fuel mixture and low temperature combustion. Thus, this study is focused on investigating the effects of different engine operating variables over combustion, to be able to suggest suitable strategies for extending the RCCI operation from low to full load, in a HD single-cylinder research engine.

Different strategies are implemented at low, medium and high load, varying fuel and air reactivity, by means of parametrical studies. Performance and emissions results are analyzed combining engine testing with 3D-CFD modeling. Based on those results, an *overlimit function* is used to select the best engine settings for each operating point. Finally, engine emissions and performance results from that RCCI operation are compared with conventional Diesel combustion (CDC).

Results suggest that double injection strategies should be used for RCCI operation from low to mid load. However, from high to full load operation, single injection strategies should be used, mainly to avoid excessive in-cylinder pressure gradients. In addition, it is confirmed the suitability of RCCI combustion to overcome the soot-NOx trade-off characteristic of CDC, from 6 to 24 bar of BMEP, while improving fuel consumption.

Keywords:

Reactivity controlled compression ignition; Dual fuel combustion; emissions control; engine efficiency

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1. Introduction

The volume of global transport could be double or even quadruple by 2050, according to a new study released by the International Transport Forum [1]. Strong increases in transport volumes mean strong growth of emissions from transport. In addition, estimates indicate that heavy duty vehicles are the second-biggest source of emissions within the transport sector, larger than both international aviation and shipping. As a consequence, the world's leading manufacturers of heavy-duty trucks and engines endorsed a harmonized global approach in improving energy efficiency and reducing harmful emissions from commercial vehicles [2]. Accordingly, the efforts of the scientific community are focused on a combination of in-cylinder reduction strategies and exhaust gas after-treatment technologies, because the difficulties faced for reducing the cost of after-treatment devices, highlight the potential of in-cylinder strategies to control emissions.

On the one hand, mixing-controlled low temperature combustion (MC-LTC) has been widely investigated as a combustion technology to avoid soot and NOx engine-out emissions [3]. As conventional Diesel combustion, MC-LTC strategy is based on the coexistence of injection and combustion events. Thus, combustion phasing can be controlled by means of the injection timing, whatever the operating conditions. To attain low temperature combustion (LTC), the use of high injection pressures and downsized nozzle diameters is needed (to reduce the equivalence ratio at lift-off), and a sharp decrease of the in-cylinder temperature or the intake oxygen concentration ($YO_{2,IVC}$) is also needed (to reduce combustion temperatures) [4]. Despite the MC-LTC ability to avoid soot and NOx emissions formation processes [5], real application in production engines is compromised. The use of massive exhaust gas recirculation (EGR) to reduce $YO_{2,IVC}$ implies an ISFC and therefore BSFC problem, and the use of the Miller cycle to reduce compression temperature still needs further development to reduce BSFC [6].

On the other hand, premixed combustion (PCI) strategies use volumetric auto-ignition to achieve high levels of efficiency, and combustion in lean or dilute mixtures which result in combustion temperatures too low for significant NOx formation, and air-fuel mixtures (A/F) too lean for soot formation [7]. In PCI strategies, combustion timing is mainly determined by chemical kinetics, so the entire history of temperature, pressure and composition of the in-cylinder charge affect to ignition timing. In this sense, the main challenges of PCI strategies are combustion phasing control, noise, HC and CO emissions reduction, and the operation range extension [8]. In order to address those challenges, mixture preparation has been investigated by improving the mixing rate of air and fuel, and by extending the ignition delay. Strategies to improve the air/fuel mixing rate are mainly the use of high injection pressures with small nozzle holes, to improve atomization and increase the relative velocity of the fuel injected in the cylinder and the surrounding air [9]; the use of high boost pressures, to increase the in-cylinder density [10]; and the use of multi-pulse fuel injection, to create mixture stratification which avoids excessive rates of heat release [11, 12]. Moreover, strategies to extend the ignition delay are mainly: the use of EGR, to reduce $YO_{2,IVC}$ and increase in-cylinder specific heat capacity, which slows down the temperature rise during compression stroke [13]; the use of variable valve actuation systems, to reduce the effective compression ratio, controlling in-cylinder temperature histories [14]; the use of fuel reactivity control strategies, to modify fuel auto-ignition characteristics [15].

In this research work, the above control strategies for PCI operation are not used alone because their combination is more effective, as reported in other investigations [16, 17]. This approach is commonly referred to as reactivity controlled compression ignition (RCCI) combustion, and relies on in-cylinder fuel blending of low reactivity fuel injected through the intake port, and high reactivity fuel injected directly in the cylinder. The use of one injection system for each fuel allows optimizing fuel reactivity on almost a cycle to cycle basis, by varying the ICFB ratio, according to changes in engine operating conditions.

Authors' previous work on RCCI combustion was focused on basic understanding of mixing and auto-ignition processes and its influence on engine-out emissions, at low load operating conditions [18]. It was found that in-cylinder fuel blending generates fuel reactivity stratification, which is accompanied by equivalence ratio stratification, promoting a staged combustion process, which avoids high peaks of rate of heat release.

On the side of the technological potential of RCCI combustion concept, several investigations have been carried out [19 - 22]. Using a HSDI engine to evaluate the concept all over the operating range, experiments highlight promising results at part load, but important issues with smoke emissions and in-cylinder maximum pressure gradients at high load operation [19]. In addition, using a HD Diesel engine, experimental work focused on extending the operating range of the concept resulted in high efficiency and clean combustion. However, those results were obtained in a range from low to medium load operation (up to 15 bar IMEP) [20, 21]. Moreover, when trying to attain higher load operation, experiments from other research work [22] show that they have to be carried out modifying engine hardware (reducing the compression ratio down to 12, to reach 20 bar BMEP). This fact leads to define the main objective of this research work.

2. Objectives and methodology

The investigation has been performed to increase the existing knowledge of advanced combustion strategies. In particular, the main objective of this research work is to extend the RCCI operating range, in a heavy duty Diesel engine. As well, the influence of different engine operating variables of this combustion strategy is analyzed in terms of combustion performance and engine-out emissions.

To find suitable RCCI operation strategies, research has been based on parametrical studies of ICFB, in-cylinder thermodynamic conditions and fueling strategies, at low (≈6 bar BMEP), medium (≈12 bar BMEP), high (≈17 bar BMEP) and full load (≈24 bar BMEP) operating conditions. The analysis has been carried out by single cylinder engine testing, supported by 3D-CFD modeling, so trends followed by pollutant emissions are related to local in-cylinder conditions along the combustion process.

In the light of the results obtained from the mentioned analysis, engine operating strategies have been proposed. Those strategies have not been rigorously optimized, they have been selected from the parametrical studies using an *overlimit function*. This is a mathematical function that provides a simple parameter to quantify the extent to which a given combustion strategy is able to simultaneously achieve all relevant constraints [23]. It is defined as Equation 1 shows, where F is the *overlimit function*, x_i is the value of the i^{th} constrained parameter, x_i^* is the constraint of the i^{th} parameter and i is the index over all of constraints. F will be zero if only the measured value is less than or equal to the specified limit.

$$F = \sum_i \max\left(0, \frac{x_i}{x_i^*} - 1\right)$$

Equation 1. Overlimit function definition.

In this application, constrained parameters are soot and NOx at the limitation from EURO VI regulation (0.01 and 0.4 g/kWh, respectively); also BSFC, at the level of fuel consumption measured from conventional Diesel combustion tests (variable depending on the engine operating conditions); and the last one, a mechanical constraint (pressure gradient < 22 bar/CAD), as recommended by engine manufacturer. Furthermore, if this function turns out to be zero for various strategies, a new *overlimit function* with two more constrained parameters is used to select the best one between them. Those parameters are HC and CO emissions, at the level of EURO VI limits (0.13 and 1.5 g/kWh, respectively).

To conclude the research work, engine performance and emissions results from the selected strategies at each operating condition, are critically compared with the results obtained using conventional Diesel combustion (settings from commercially available truck engines, to which is equivalent the SCE used in this investigation). This final comparison was conducted based on experimental measurements.

Basic engine operation conditions are defined by 1200 rpm and constant injected fuel mass, at each engine load mode. Further development, based on the results obtained from this research, will be needed to extend the RCCI concept to the whole engine operation map.

3. Experimental setup

In comparison with multi-cylinder engines, single-cylinder engines generate more accurate data [24]. Accordingly, a single-cylinder engine was used in this research work. The main characteristics of the engine and the test cell are described below.

3.1. Single cylinder engine

The engine is a single-cylinder, four-stroke, compression ignition research engine, representative of commercial truck engines. Basic specifications of this engine are given in Table 1.

The engine is equipped with a hydraulic VVA system, so all valves are independently actuated by different hydraulic pistons (one per valve), which are controlled by a specific electronic control unit. The main benefit of this VVA system is its high flexibility, including variable timing, duration and lift for each valve. Thus, using this system, many strategies can be performed in the engine (such as Miller cycle). As a counterpart, this system requires an adapted cylinder head different from the conventional design (although the structural changes are not extreme) and also a dedicated oil circuit in addition to the standard oil lubricating system.

One of the specificities of the hydraulic valve actuation is its fast-motion intake and exhaust valve lift profiles. Increasing the exhaust valves opening and closing speeds is interesting for reducing gas energy losses by shortening the time in which the exhaust gas flow evolves under sonic conditions [25]. However, this fast-motion together with the very little clearance between the piston and valves, when the piston is close to top dead

centre (TDC), make the valve overlap at breathing TDC impossible due to direct mechanical interference between the piston and the valves. This lack of valve overlap slightly reduces the engine volumetric efficiency.

Additionally, the engine is equipped with two different injection hardware, one for each different fuel used. Commercially available Diesel and gasoline 98 ON fuels were selected as high and low reactivity fuels, respectively. Their main properties are listed in Table 2. On the one hand, Diesel fuel was injected by means of a common-rail direct injection system (Bosch CRSN 4.2), capable to perform up to five injections per cycle and to amplify common-rail fuel pressure by means of a hydraulic piston directly installed inside the injector. On the other hand, gasoline was injected by means of a port fuel injection system, in which two injectors are placed at the intake port, as is described in [18].

3.2. Test cell

The engine is installed in a fully instrumented test cell, with all the auxiliary facilities required for its operation and control, as is illustrated in Figure 1.

To achieve stable intake air conditions, an externally driven screw compressor supplied the required boost pressure, before passing through an air dryer. Air pressure was adjusted in the intake settling chamber, while intake temperature was controlled in the intake manifold, after mixing with EGR. The exhaust backpressure produced by the turbine in the real engine was replicated by means of a valve placed in the exhaust system, controlling the pressure in the exhaust settling chamber.

A low pressure EGR circuit is also installed on the engine test bench. Once solid particles and liquid droplets were removed from the cooled exhaust gases, an externally driven roots-type supercharger was used to increase the exhaust gas pressure over the intake pressure. Then, the exact EGR rate was controlled by means of a valve between the EGR settling chamber and the intake pipe, so the required exhaust gas mass flow was introduced into the intake duct depending on the desired EGR rate. The temperature regulation was performed upon the EGR-fresh air mixture by means of a temperature sensor in the intake manifold.

The EGR rate was calculated using the experimental measurement of intake and exhaust CO_2 concentration. The concentrations of NO_X, CO, HC, intake and exhaust CO_2, and O_2 were measured with specific state-of-the-art analyzers (Horiba MEXA 7100 DEGR). Smoke emission was measured with a variable sampling smoke meter (AVL 415), providing results directly in Filter Smoke Number units (FSN) that were transformed into dry soot mass emissions by means of the correlation proposed by Christian et al. [26].

The in-cylinder pressure traces from a piezo-electric transducer (Kistler 6125B) were recorded during 100 engine cycles, in order to compensate for dispersion in engine operation. The recorded values of in-cylinder pressure were processed by means of a combustion diagnosis code (CALMEC) [27, 28], which provides valuable information such as the rate of heat release (RoHR) and the unburned gases temperature. The latest mentioned is a basic input for the adiabatic flame temperature (T_{ad}) calculation, according to the scheme proposed by Way [29].

To ensure the reliability of the provided results, every operation point was measured three times and a reference point was controlled before every testing session, to assure tests repeatability along the study.

4. CFD modeling approach

The computational model was built by means of CONVERGE CFD code [30]. Closed-cycle computations on sector grids with periodic boundaries were carried out to improve computational efficiency. In these calculations $1/9^{th}$ of the combustion chamber, due to the 9-hole nozzle, is modeled. The CFD code uses a structured cartesian grid with base cell size of 1.6 mm, an adaptive mesh refinement (AMR), as well as a fixed refinement within the spray region. In addition, an additional volume on the cylinder head is employed to match engine compression ratio. Figure 2 shows clips of the final mesh configuration, at two different time steps, to illustrate the mentioned additional volume and the AMR.

Calculations run from intake valve closing (IVC) with initial thermodynamic conditions, as well as wall temperatures, estimated from experimental data by means of CALMEC combustion diagnostics code.

The port-fuel injected gasoline is considered to be homogeneously mixed and vaporized at IVC. The Diesel injection process is simulated by the standard Droplet Discrete Model [31]. Spray atomization and break-up are simulated by means of the KH-RT [32] model. Diesel fuel physical properties are given by the *diesel2* fuel surrogate available in CONVERGE. Turbulent flow is modeled by means of the RNG k-ε model with wall-functions [33], in order to account for wall heat transfer.

Concerning combustion modeling, a direct integration of detailed chemistry approach was used by means of the CONVERGE code and the SAGE solver. A multi-zone model from Babajimopoulos et al.[34] is used to solve the detailed chemistry in zones, i.e., groups of cells that have similar thermodynamic state, in order to speed-up chemistry calculations. Cells are grouped based in two variables, temperature and equivalence ratio. Calculations are performed using a 5 K bin size for temperature and 0.01 bin size for equivalence ratio zones.

A Primary Research Fuel (PRF) (n-heptane + iso-octane) has been used as fuel surrogate. This kinetic mechanism [35] is composed by 45 species and 142 reactions, including NOx formation (thermal, N_2O and NO_2 pathways). Soot formation and oxidation is modeled by a phenomenological two-step model which uses acetylene as soot precursor [36]

5. Results and discussion

In previous studies [18], the use of an in-cylinder fuel blending ratio of 25% Diesel and 75% gasoline (ICFB 75) and a single Diesel injection strategy placed at -24 CAD aTDC, resulted in smooth and well-timed rate of heat release, at low load operation. Also working at low load conditions, other researchers have attained successful results, (high efficiency, low NOx and low soot emissions) from RCCI operation with double Diesel injection strategies [37]. Therefore, in this research work both mentioned strategies have been analyzed at different operating conditions, (combined with the sweeps of ICFB, injection timing/s and fuel distribution) to be able to provide suitable engine settings for extending the engine operating range of the RCCI concept from low to full load.

5.1. RCCI low load operation

Table 3 (a) summarizes the main operating conditions fixed and swept, following parametric studies, at low load. In this case, the most important engine variables fixed were speed, total fuel mass and air mass flow. Using a single injection strategy, ICFB and injection timing were swept. Using a double injection strategy, the first injection event (SoI1) was swept while the second injection event (SoI2) was fixed; also, keeping SoI1 fixed, SoI2 was varied; and then, Diesel fuel split amount between injection events was also swept.

Figure 3 contains the main results obtained from the mentioned sweeps, in terms of emissions and performance. The dashed lines mark the region of interest, according to EURO VI emissions regulations and fuel consumption results (in terms of percentages where negative values mean an improvement with respect to the low load neat Diesel combusting case). On the left side of the mentioned figure, the results from using a single direct injection strategy are included; while on the right side, are the ones from using a double injection strategy. Looking to the NOx-ΔBSFC subfigure, there is a trade-off when using a single injection strategy: delaying SoI and increasing the ICFB reduces NOx but increases fuel consumption. As is shown in Figure 4, those strategies imply later combustion phasing, longer combustion events and, as a consequence, lower flame temperatures. Thus NOx emissions get lowered, but BSFC is increased. However, the mentioned trade-off does not appear when using a double injection strategy, where advanced injection events and more fuel in the first one of them are able to reduce simultaneously NOx emissions and fuel consumption. It is mainly due to those strategies are able to phase combustion slightly towards the expansion stroke, without worsening the RoHR (keeping almost constant duration and the peak of heat release), as shown in Figure 4. In addition, as the rest of subfigures from Figure 3 show, both injection strategies are able to provide soot emissions compliant with EURO VI requirements, reduced pressure gradients and ~97% of combustion efficiency.

Important differences have been found between using single or double injection strategies. To deeply analyze that behavior, single injection at -24 CAD aTDC and double injection at -60/-35 CAD aTDC, using ICFB 75, have been modeled using 3D-CFD. To assure the quality of the model, their results have been validated versus experimental data, and as Figure 5 shows, fair predictions in terms of combustion development have been obtained. Figure 6 contains a sequence of in-cylinder equivalence ratio and temperature cut planes, focused from the start to the end of combustion. The higher level of NOx emissions obtained from the single injection strategy is justified by the existence of wider regions of high temperature, for longer in the cycle. In terms of soot emissions both cases show low levels, however it is caused by different mechanisms, as Figure 7 depicts. Using single injection strategies there appear rich regions which promote soot formation; however the high temperatures existing help to their oxidation resulting in low soot emissions. Using double injection strategies, the high equivalence ratio regions are strongly reduced, and therefore, the low level of soot emissions is ruled by avoiding its formation. The mentioned lower local equivalence ratios from double injection strategies provide later combustion phasing (from TDC to the expansion stroke) that contributes to increase IMEP and to improve fuel consumption.

Hence, at low load conditions high in-cylinder fuel blending ratios and double injection strategies are more adequate for RCCI operation.

5.2. RCCI medium load operation

The main operating conditions for the study at medium load are detailed in Table 3 (b). This study was developed by means of parametrical studies, keeping fixed speed, total fuel mass and air mass flow. As well as fuel reactivity, ambient reactivity effects over combustion performance and emissions have been analyzed, due to the increased engine load. Using a single injection strategy, the ICFB and the injection timing were swept. Using a double injection strategy, the ICFB, SoI1 (with fixed SoI2), SoI2 (with fixed SoI1); the Diesel fuel split amount between injection events; and the effective compression ratio (CR_{ef}) were swept.

It should be noted that, in this research, the CR_{ef} was varied by implementing an early Miller cycle. This strategy is based on the reduction of the CR_{ef}, and therefore, of compression temperatures, by shortening the duration of the intake event by advancing the intake valves closing (IVC), as Figure 8 shows. Consequently, the intake air mass is reduced; to overcome this fact, the intake pressure has been increased as advanced the IVC, keeping constant the intake air mass flow along the CR_{ef} sweep.

Figure 9 contains the main results obtained from the mentioned sweeps, in terms of emissions and performance. As in Figure 3, the region of interest is defined by EURO VI emissions regulation, neat Diesel fuel consumption at medium load and the limit of maximum pressure gradient.

As is shown in the left hand subfigures from Figure 9, using single Diesel injection strategies, a trade-off appears between NOx and fuel consumption, as in low load operation (the injection timing delay and the ICFB lowering reduces NOx emissions but increases fuel consumption). Moreover, an additional trade-off appears between soot and NOx emissions due to the worsening of combustion efficiency promoted by the injection timing delay and the longer Diesel injection rates from lower ICFB. By contrast, the use of double Diesel injection strategies is able to attain the NOx-ΔBSFC simultaneous reduction, as is shown in the right hand subfigures from Figure 9. Nevertheless, it should be remarked that an advanced double Diesel injection strategy is not enough to attain the region of interest, because of the early phasing of combustion, which results in high NOx emissions and fuel consumption. The last mentioned subfigures show that the strategies which shift combustion towards TDC and the expansion stroke are: the ICFB lowering, the advance of SoI1, the delay of SoI2, the use of fewer amount of fuel into the first injection event and the lowering of the CR_{ef}. Those strategies also provide soot emissions compliant with EURO VI requirements, reduced pressure gradients and combustion efficiency between 97 and 98 %.

The most representative extremes from the mentioned strategies ([CR_{ef} 14, SoI1 -60, **SoI2 -10** CAD aTDC]; [**CR_{ef} 11**, SoI1 -60, SoI2 -40 CAD aTDC]) have been compared with their baseline case ([CR_{ef} 14, SoI1 -60, SoI2 -40 CAD aTDC]), using 3D-CFD modeling. Figure 10 shows a good agreement on ignition delay though over-predicted peaks of RoHR between modeling and experimental results. The cause of this over-prediction may be due to chemical mechanism validity range, since higher pressure and temperature are found when increasing load. Also the lack of any turbulent-chemistry interaction model may also contribute to this result [38]. Nevertheless, predicted in-cylinder pressure and RoHR traces show the ability to modify combustion phasing from the delay of the SoI2 and the reduction of the CR_{ef} .

To further analyze this behavior, Figures 11 and 12 contain sequences of in-cylinder equivalence ratio and temperature cut planes, between the start and the end of combustion. The first one compares the cases with the same CR_{ef}, and shows that the later SoI2 triggers the onset of combustion. Thus, it provides a longer ignition delay which implies the later appearance of high temperature zones into the combustion chamber (closer to TDC, instead of earlier during the compression stroke). Therefore lower in-cylinder maximum temperatures are attained. On the other hand, the second injection event creates a high reactivity region, under-stratified, which promotes a faster RoHR, placed before TDC, at these operating conditions. Figure 12 compares the cases with the same fueling strategy and different CR_{ef}, and shows that both result in very similar equivalence ratio distributions. However, their main difference is when in-cylinder temperatures distributions get similar (9 CAD after for the lower CR_{ef} case), as Figure 10 shows. The CR_{ef} lowering reduces compression temperatures, enabling much longer ignition delay (CA25 takes place around -10 CAD aTDC for the higher CR_{ef} case, while around -4 for the lower CR_{ef}). As main consequence, combustion tends to take place during the expansion stroke, improving the level of NOx emissions and fuel consumption.

According to this analysis, high ICFB and double Diesel injection strategies, coupled with lowered in-cylinder thermodynamic conditions, provide the best results for medium load RCCI operation.

5.3. RCCI high load operation

As in previous subsections, Table 3 contains the main operating conditions, particularized for the study at high load in column (c). In this study, as well as speed, total fuel mass and air mass flow, $CR_{ef} = 11$ was also fixed (CR_{ef} was reduced on the light of the results from the medium load analysis and considering the increase on engine load). Using a double injection strategy, SoI1 was swept while SoI2 was fixed; and SoI2 was varied with a fixed SoI1. In both sweeps the fuel split ratio was set at 50 / 50 between injection events. Then, using a single injection strategy, the ICFB and the injection timing were also swept.

Figure 13 contains the emissions and performance results obtained from the mentioned sweeps, and the region of interest marked with dashed lines. Right hand subfigures show the results from using double Diesel injection strategies, while the left hand ones show the results from using single Diesel injection strategies. Despite the lowered in-cylinder thermodynamic conditions from the reduced CR_{ef}, when using a double injection strategy, the main challenge is the extremely high in-cylinder maximum pressure gradient. Even delaying both injection timings, pressure gradients still too high and combustion starts before the SoI2, burning the second injection event in a high temperature and fuel rich region, which promotes high soot formation. Regarding NOx emissions, mainly the SoI2 delay is able to reduce them, but remaining far away from the region of interest. Then, a single injection strategy was tested trying to improve high load engine operation. As found at lower operating loads, the NOx-ΔBSFC subfigure on the left side of Figure 13 shows a trade-off when lowering ICFB and delaying injection timing: the lower NOx, the higher fuel consumption. In terms of soot emissions, they remain below the EURO VI limit, mainly due to high load operation implies enhanced combustion efficiency and improved oxidation processes. Aside from those findings, the most important result is that single injection strategies, placed close to TDC, attain adequate in-cylinder maximum pressure gradients, enabling RCCI operation at high load.

The cases selected to perform the analysis of in-cylinder local conditions are a single injection strategy (-6 CAD aTDC) and a double injection strategy (-40/-10 CAD aTDC), both with ICFB 70 and CR_{ef} 11. Figure 14 shows a good agreement between experimental and modeling results despite of the overpredicted peak of RoHR, as found under medium load conditions. Moreover, this figure highlights the great difference in terms of combustion phasing between both strategies. Figure 15 contains a sequence of in-cylinder temperature and equivalence ratio cut planes, starting just after the SoC of the double Diesel injection case and finishing before the EoC of the single Diesel injection case. There is shown that, when using the double injection strategy, the first injection ignites before the second one and high temperatures are attained much earlier than in the single injection case. That is the main reason of the slight influence of SoI2 over combustion phasing, as Figure 16 reports. Following with the analysis from Figure 15, using a double injection strategy, as the second injection burns, high temperature regions are extended to a wider region of the combustion chamber. It is coherent with the high levels of NOx and the low levels of soot emissions measured. By contrast, when using a single injection strategy, combustion is triggered by the Diesel injection and progresses through the Diesel spray (high equivalence ratio and reactivity zones), keeping high temperature regions there. Thus, at these conditions, the single Diesel injection strategy is able to control RCCI combustion, as is shown in Figure 16.

Hence, at high load conditions single injection strategies provide better results for RCCI operation, mainly due to its capability to keep controlled in-cylinder maximum pressure gradients.

5.4. RCCI full range operation

In this subsection the RCCI operation is experimentally demonstrated over a range of loads from 25% (≈ 6 bar BMEP) up to 100% (≈ 24 bar BMEP).

Results presented here have been selected by using the *overlimit function* as detailed in section 2, from a different number of experiments. It means that the engine operating variables are not fixed over the load sweep. Moreover, these have not been rigorously optimized, so it is expected that different combinations of injection parameters, EGR levels and ICFB may yield similar results.

As has been proved in the previous subsection, in-cylinder maximum pressure gradients and combustion phasing become a concern when increasing engine load. Thus, to extend the operation range with safe combustion performance and adequate combustion phasing, in-cylinder fuel blending ratio, injection timing/s, injection pressure and fuel distribution between injection events, as well as engine effective compression ratio have been tuned, providing suitable RCCI operating strategies all over the load range. Figure 17 summarizes the main settings adopted for the variables tuned, represented versus BMEP, as load indicator.

Best results at low load operation, 6 bar BMEP, are attained by means of ICFB 75 and a double injection strategy, with 60% of the Diesel fuel mass injected into the pilot event at -60 CAD aTDC and the rest at -15 CAD aTDC. At medium load operation, 12 bar BMEP, the best results are provided by ICFB 80 and early double injection strategy (-60/-40 CAD aTDC), with the same amount of Diesel fuel in each event. It should be remarked that, this combustion strategy requires lowered CR_{ef} to maintain adequate combustion phasing and safe pressure gradients. Thus, up to medium load, RCCI

combustion results EURO VI compliant in terms of NOx and soot emissions, simultaneously.

As has been detailed before, high load operation using double Diesel injection strategies provides too high pressure gradients, which make this strategy not suitable for this operating condition, (at least keeping this hardware configuration). Accordingly, at 17 bar BMEP, the best results were attained with ICFB 60 and a single Diesel injection, close to TDC (-6 CAD aTDC). Further engine load increase, up to 24 bar BMEP (full load), needs a close to TDC single Diesel injection and also a reduction of the premixed gasoline ratio to keep autoignition under control. Hence, full load best results were obtained using ICFB 40 and single Diesel injection placed -8 CAD aTDC. The main drawback from the strategies used at high load is the increase in terms of NOx emissions, due to the high flame temperatures resulting from the coexistence of Diesel injection and combustion events, as the plot of maximum adiabatic temperature of Figure 18 shows.

5.5. RCCI comparison with CDC

In the previous subsection suitable RCCI operating strategies have been suggested to be used in the entire engine load range. Then, to complete the study, performance and engine-out emissions from those engine operating strategies are directly compared with conventional Diesel combustion (CDC), at the same speed and load conditions. CDC settings were provided from the equivalent commercial truck engine (which was designed to operate with EGR and SCR), by the engine manufacturer supporting this research work. Note that, to be a fair comparison, CDC experiments were conducted with the same SCE, detailed in section 3. The key operation parameters for the comparison are included in Figure 17. Performance and emissions results from RCCI and CDC are compared in Figure 18.

Looking to the last mentioned figure, the main drawbacks from RCCI combustion strategy are: higher in-cylinder maximum pressure gradients at high load operation and lower combustion efficiency all over the load range. Regarding pressure gradients, the much more premixed conditions from RCCI combustion strategy, implies higher pressure gradients when compared with the mixing-controlled strategy used in CDC. Concerning combustion efficiency challenge, it is mainly related with the incomplete combustion of low reactivity fuel existing at cold regions. On the other hand, Figure 18 highlights as main benefits from RCCI combustion lower soot and NOx emissions, and also competitive fuel consumption whatever the load conditions. Looking at soot results, RCCI is much better at full load operation, because this strategy promotes lean equivalence ratios that avoid soot formation, while mixing-controlled combustion (specially with medium injection pressures) results in a large soot formation rate. On the concern of NOx emissions, the lower adiabatic flame temperatures achieved by the RCCI mode avoid NOx formation processes, especially at medium-low load operation, where a double injection strategy is used. Regarding fuel consumption, RCCI mode only shows worse results than CDC at the 17 bar operating point. It is mainly due to the higher pumping work from the p-V low pressure loop that Miller cycle implies [6], and the higher RCCI global equivalence ratio (0.78 vs. 0.57) which deal with a lower polytropic index, and therefore worse fuel conversion efficiency. This issue could be solved by engine optimization methodology; however this is not the aim of this research and remains for further work on this topic. Moreover, despite both combustion strategies attain correctly phased CA50 whatever the engine load, RCCI combustion event is much

shorter, which makes the better thermal efficiency compensate the worse combustion efficiency, resulting in similar or better BSFC levels than CDC.

6. Conclusions

Present study focuses on extending the RCCI operating range from low to full load, using a heavy-duty single-cylinder research engine to analyze different engine variables, which lead to the definition of suitable operating strategies.

The research is based on different sets of parametric studies. However, multi-dimensional CFD simulations resulted essential to gain an insight of in-cylinder local conditions.

Engine performance and emissions have been analyzed at low, medium and high load conditions, by combining single cylinder engine testing with 3D-CFD modeling. This study provided the following conclusions:

- At low load conditions high ICFB and double direct injection strategies provide better results than single injection strategies. Mainly due to the longer ignition delays and the latter combustion phasing from the earlier injection timings, which help to reduce in-cylinder temperatures during combustion and to lower zones with rich equivalence ratios, reducing simultaneously soot and NOx emissions to almost zero.
- At mid load conditions, high ICFB and double direct injection strategies need to be coupled with lowered in-cylinder thermodynamic conditions to keep autoignition under control. This fact implies low maximum adiabatic flame temperatures that avoid NOx formation mechanisms, and lean equivalence ratios that avoid soot inception.
- At high load conditions, single direct injection strategies are able to trigger combustion by means of the injection timing. It permits to attain an adequate combustion phasing and to control in-cylinder maximum pressure gradients, from the two-staged heat release.

Then, using an *overlimit function* suitable settings for RCCI operation from low to full load are provided. In addition, their performance and emissions results have been compared with CDC, at the same speed and load conditions.

- From the high ICFB and double direct injection strategy used at low load, when increasing to medium load, RCCI operation needs to lower the effective compression ratio. Further increase to high load, also needs the use of single direct injection strategies, close to TDC. And to reach full load, is also necessary the lowering of the ICFB (reducing the amount of premixed gasoline).
- Comparing the results between RCCI and CDC, the main drawback from RCCI is its lower combustion efficiency (especially at low load) and its higher in-cylinder maximum pressure gradients (from medium to full load operation). By contrast, soot and NOx emissions levels are simultaneously better than CDC, whatever the load conditions. In addition, fuel consumption results are also improved at every operating condition, except around 17.5 bar BMEP where still competitive but 1% higher.

From this investigation, suitable settings have been provided to extend the RCCI operating range from low to full load. Moreover, it has been demonstrated how the RCCI combustion concept is capable to solve the soot-NOx trade-off characteristic of CDC, all

over the engine load range. However, the main challenge for RCCI application in production engines is the lowering of NOx emissions from high-to-full load operation. Thus, further research on this topic is still needed to improve the reactivity stratification at high load operating conditions, to control combustion progression and to reduce NOx emissions level, avoiding the need of SCR technologies.

7. Acknowledgments

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637 **Nomenclature**

AMR	Adaptive Mesh Refinement
aTDC	After Top Dead Centre
A/F	Air – Fuel ratio (mass)
BMEP	Brake Mean Effective Pressure
BSFC	Brake Specific Fuel Consumption
CAD	Crank angle degree
CA25	Angle when 25% of the fuel is burnt
CA50	Angle when 50% of the fuel is burnt
CDC	Conventional Diesel Combustion
CO	Carbon monoxide
Comb. eff	Combustion efficiency
CO_2	Carbon dioxide
CR	Compression ratio
CR_{ef}	Effective compression ratio
dP/da	Pressure Gradient
EGR	Exhaust Gas Recirculation
EoC	End of Combustion
FSN	Filter Smoke Number
HC	Unburned Hydrocarbon
HD	Heavy Duty
HSDI	High Speed Direct Injection
ICFB	In-Cylinder Fuel Blending
Ign. delay	Ignition Delay
IMEP	Indicated Mean Effective Pressure
IP	Injection Pressure
ISFC	Indicated specific fuel consumption
IVC	Intake Valve Closing (angle)
LTC	Low Temperature Combustion
MC-LTC	Mixing-Controlled low temperature combustion
NOx	Nitrogen Oxides
ON	Octane Number
O_2	Oxygen
P	Pressure
PCI	Premixed Compression Ignition
P_{int}	Intake Pressure
PRF	Primary Reference Fuel
RCCI	Reactivity Controlled Compression Ignition
RoHR	Rate of Heat Release
SCE	Single Cylinder Engine
SoC	Start of Combustion
SoI	Start of Injection
T	Temperature
T_{ad}	Adiabatic flame temperature
T_{int}	Intake temperature
TDC	Top Dead Centre
VVA	Variable Valve Actuation
$YO_{2,IVC}$	Oxygen concentration in the cylinder at the intake valves closing
3D-CFD	Tri-dimensional Computational Fluid Dynamics
ϕ	Equivalence ratio

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639 **List of Figures**

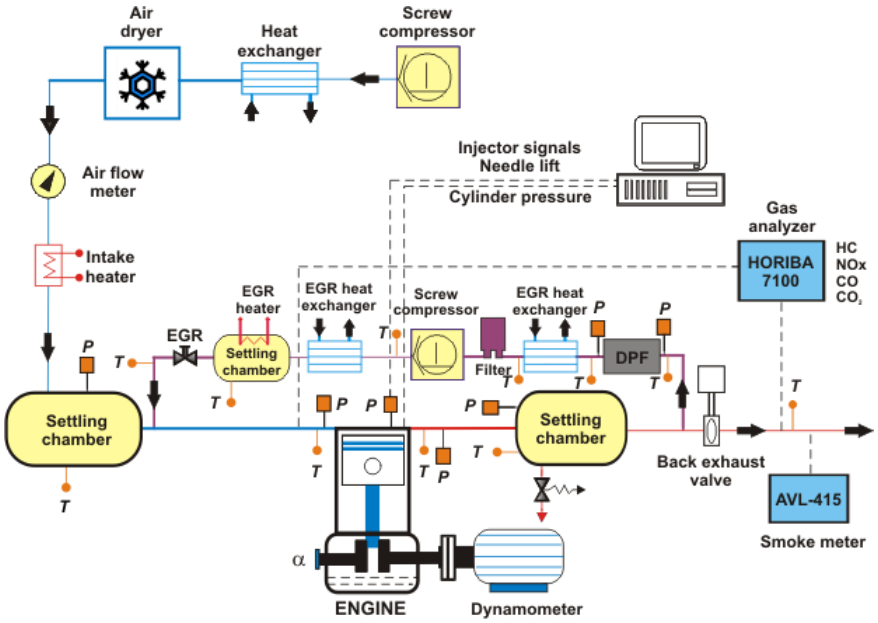

Figure 1: Complete test cell schema.

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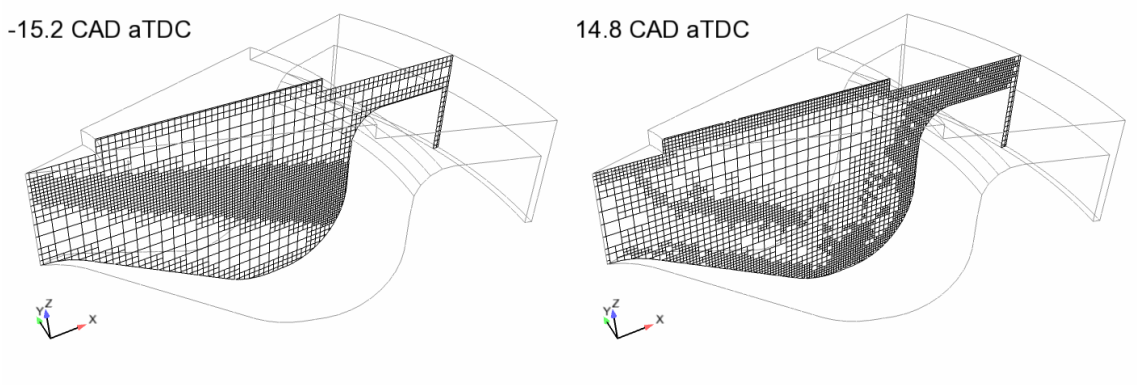

Computational grid at -15.2 CAD aTDC Computational grid at +14.8 CAD aTDC

655 Figure 2: Different grid refinements by CONVERGE adaptive mesh refinement.

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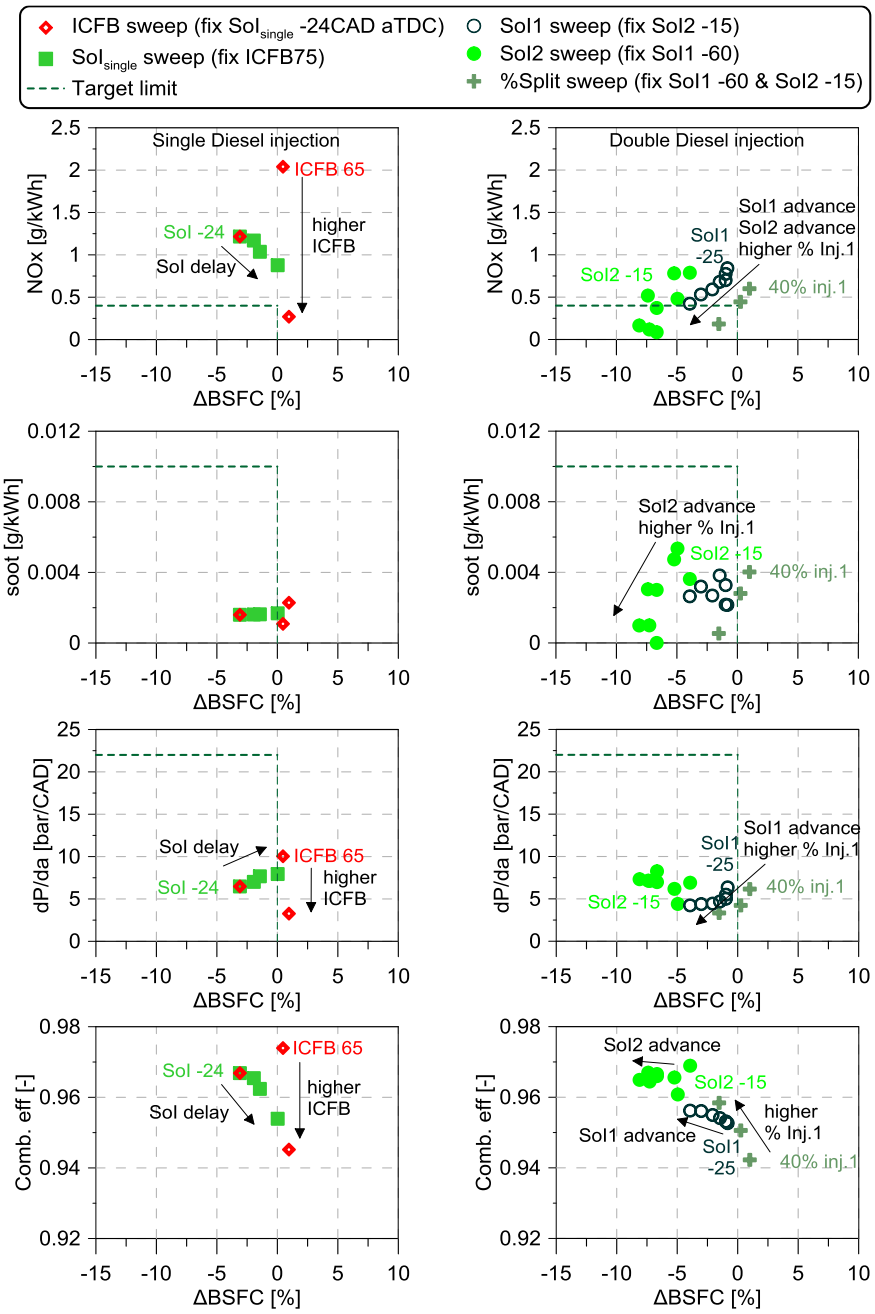
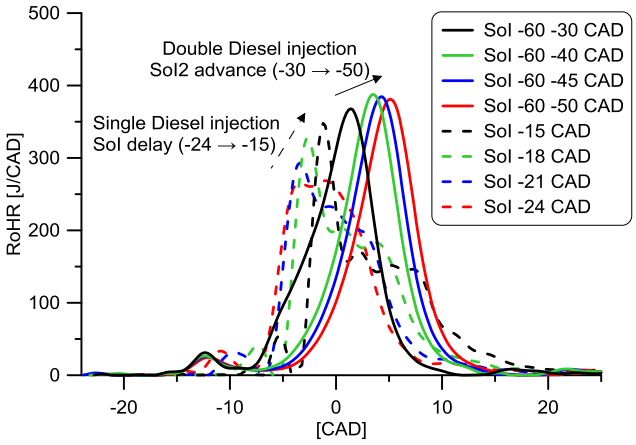


Figure 3. Experimental results from the different variables swept at low load operation, using single Diesel injection (left) and double Diesel injection (right).



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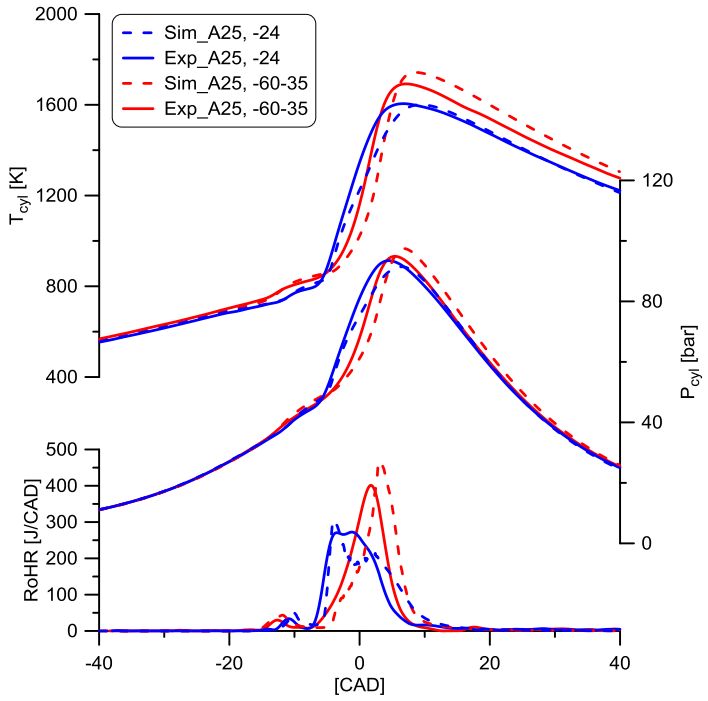

Figure 5. RoHR, in-cylinder mean pressure and temperature comparison between modeling and experimental results, for single and double injection strategies, at ICFB 75 and low load operation.

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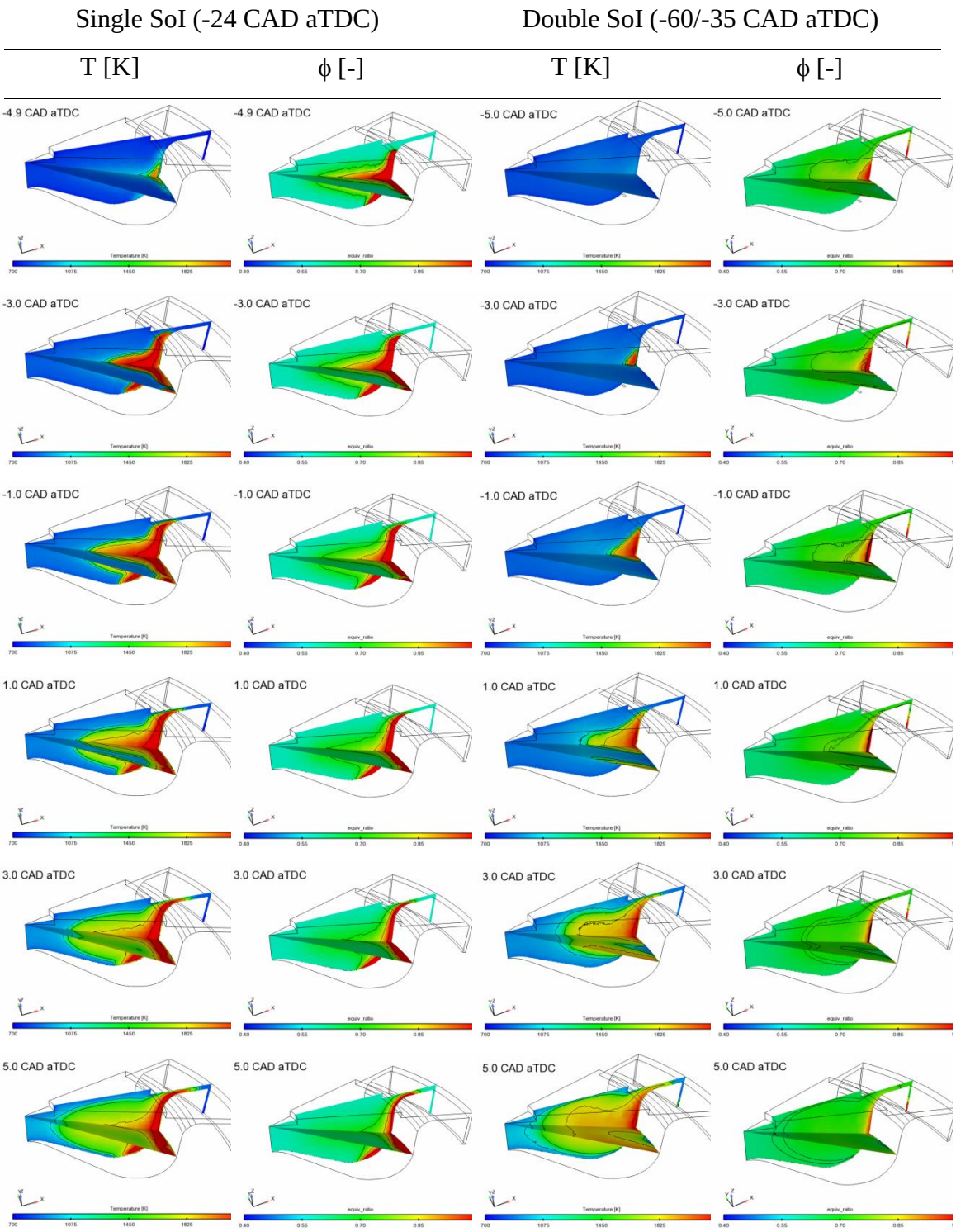

690 Figure 6. Calculated temperature and equivalence ratio cut planes for low load, single
691 Diesel injection and double Diesel injection.

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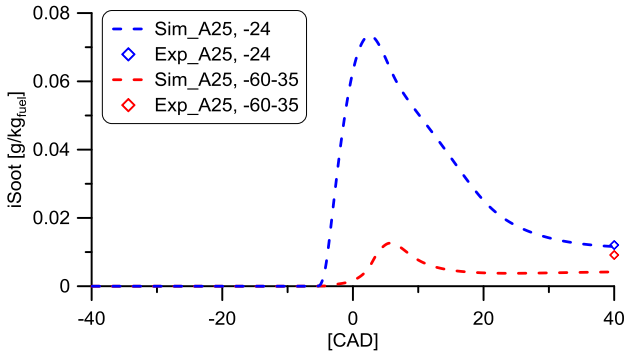

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701 Figure 7. Comparison between modeling and experimental results in terms of fuel

702 specific soot emissions, for single and double injection strategies, at ICFB 75 and low

703 load operation.

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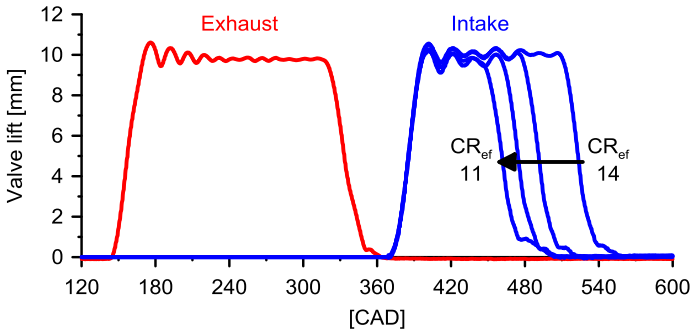

Figure 8. Valves strategy employed in realizing the Miller cycle.

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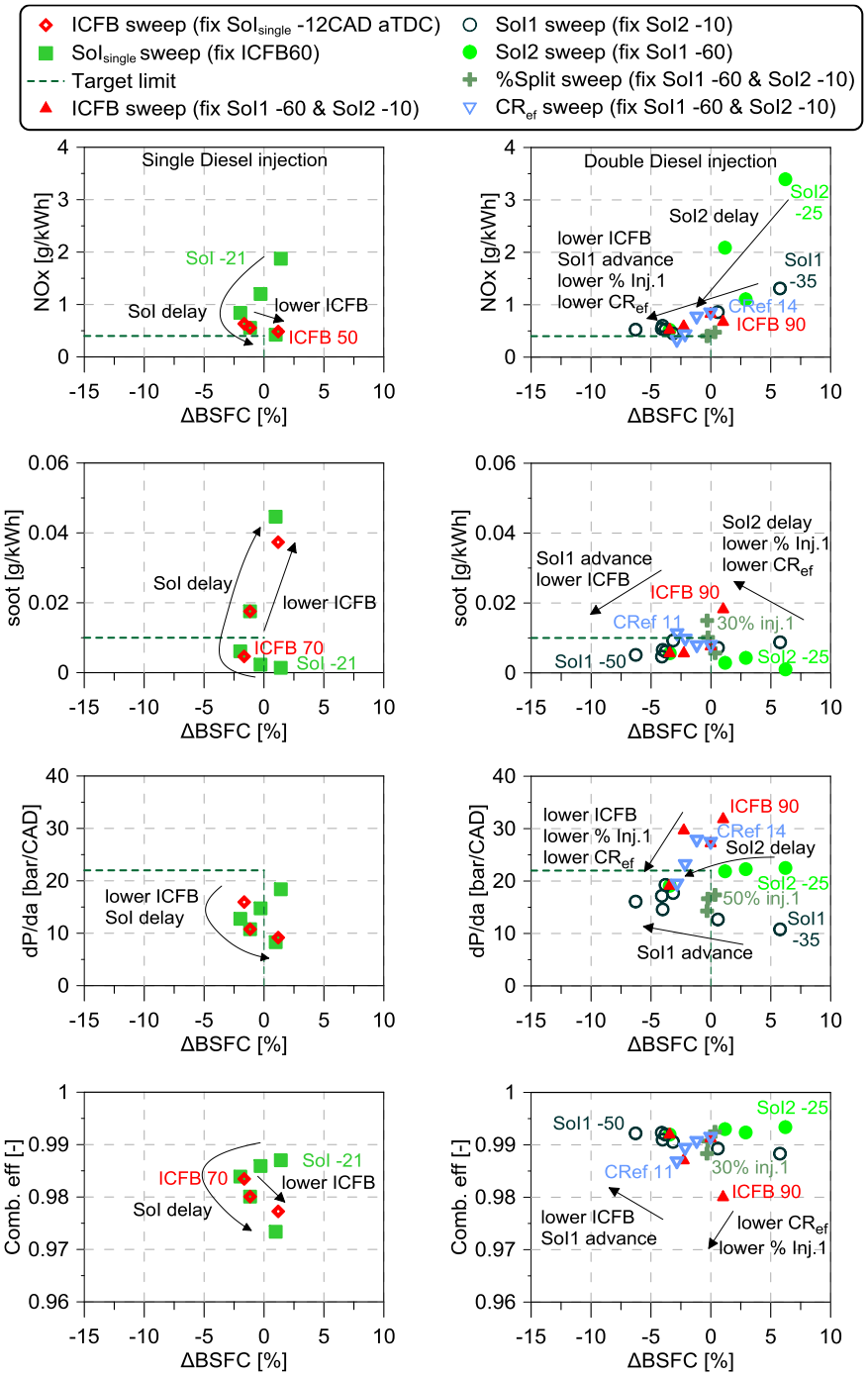

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718 Figure 9. Experimental results from the different variables swept at medium load

719 operation, using single Diesel injection (left) and double Diesel injection (right).

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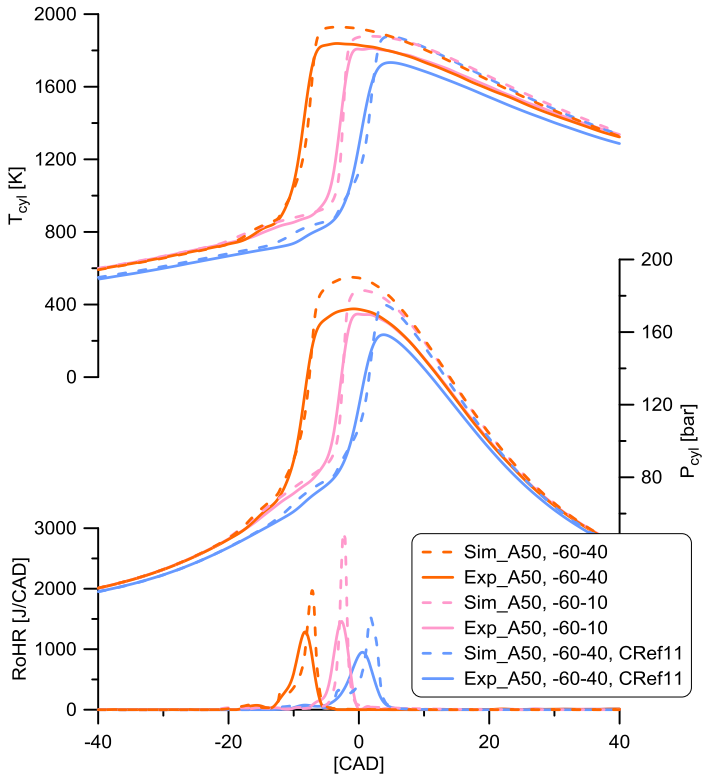

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728 Figure 10. RoHR, in-cylinder mean pressure and temperature comparison between

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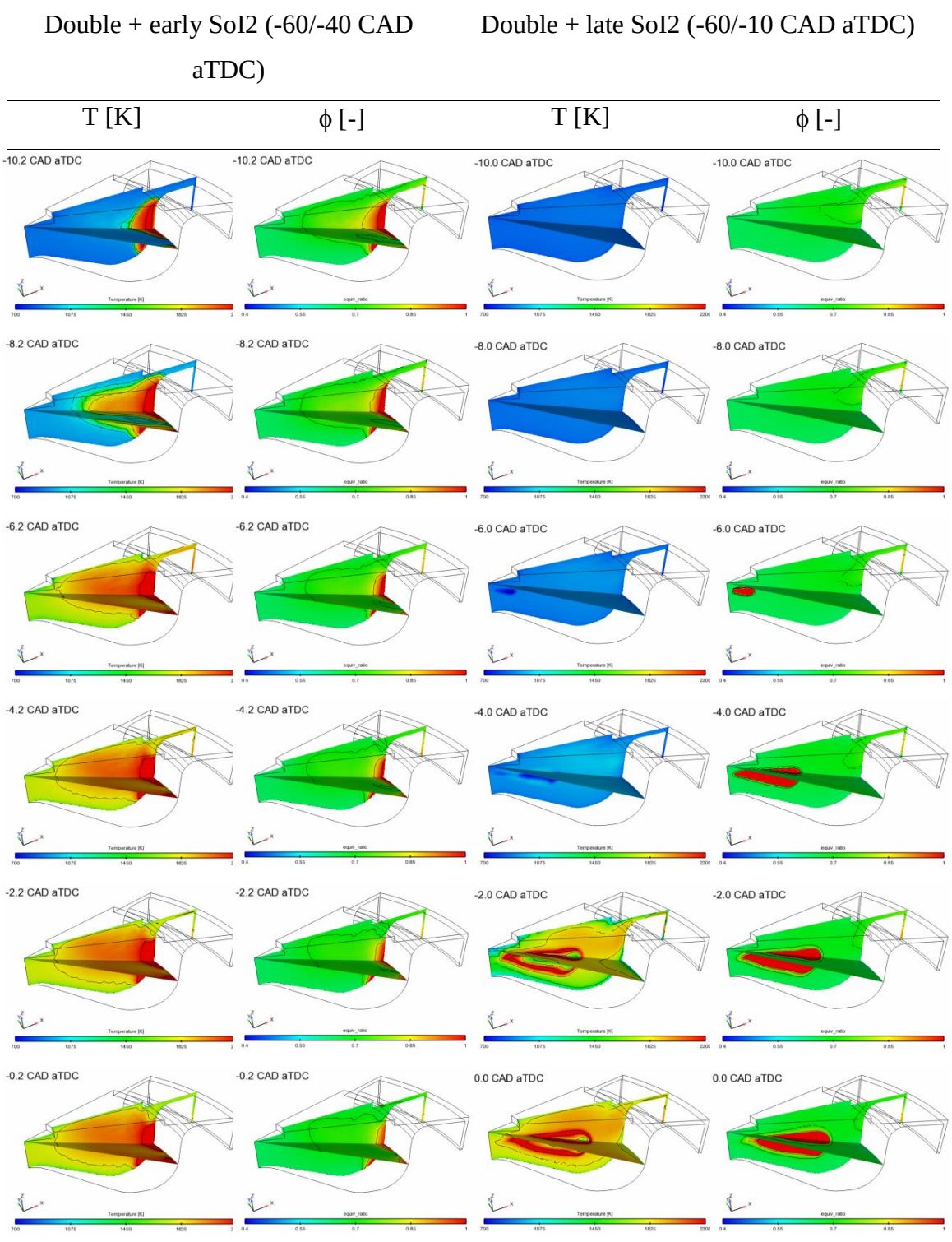

734 Figure 11. Calculated temperature and equivalence ratio cut planes for medium load,
735 double injection strategies, with early SoI2 (left) and close to TDC SoI2 (right).

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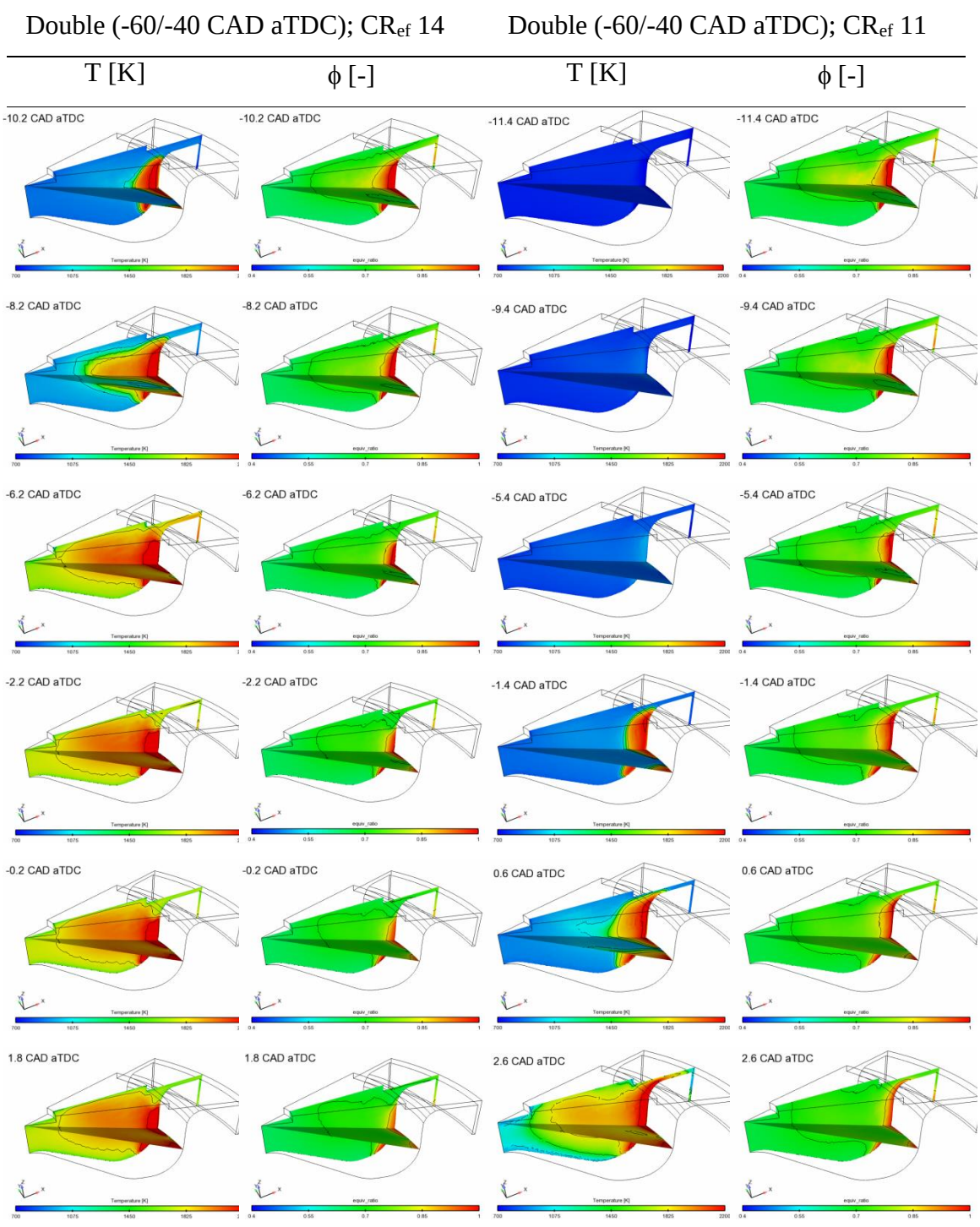

738 Figure 12. Calculated temperature and equivalence ratio cut planes for medium load,
739 double injection strategies, with CR$_{ef}$ 14 (left) and CR$_{ef}$ 11 (right).

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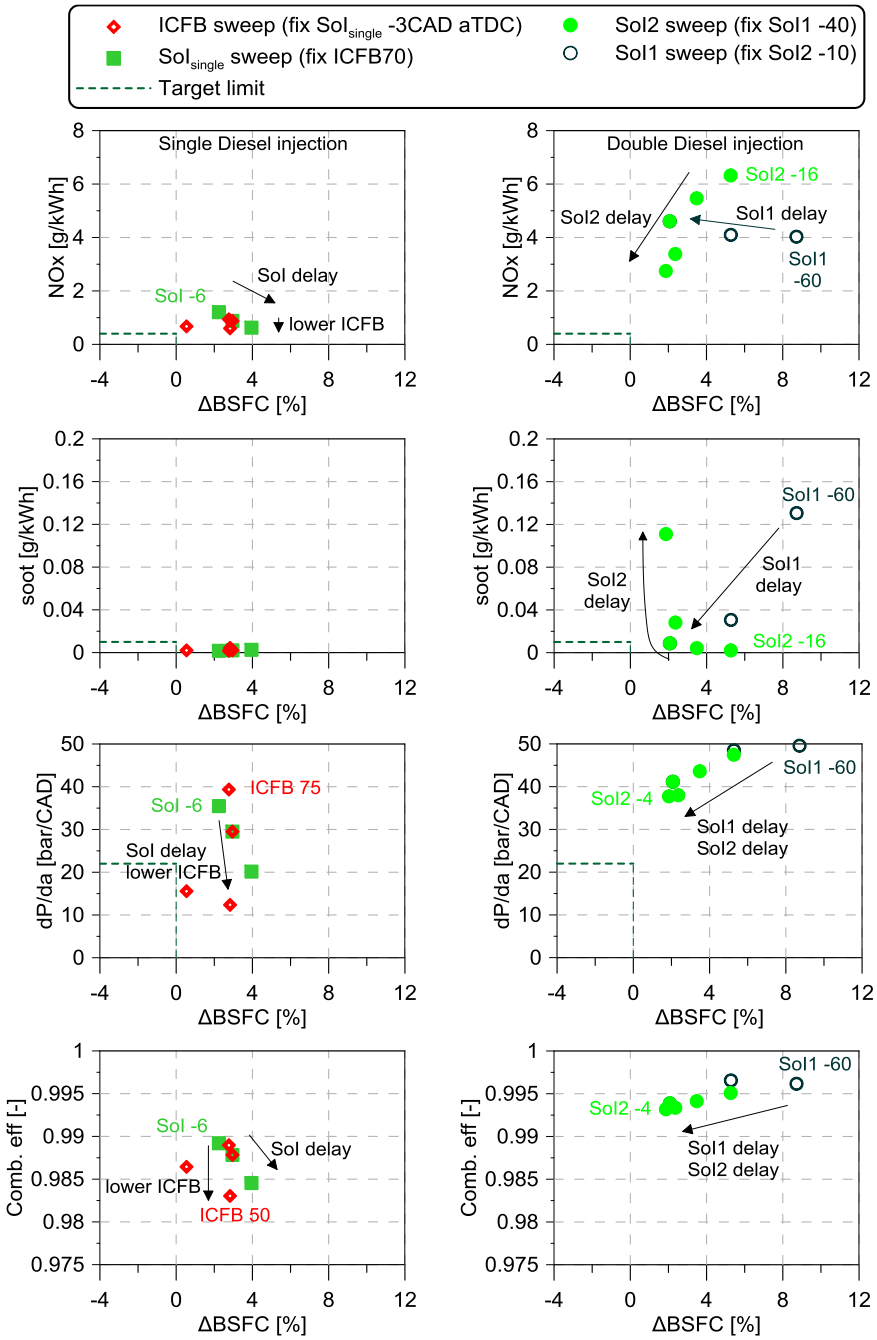

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744 Figure 13. Experimental results from the different variables swept at high load

745 operation, using single Diesel injection (left) and double Diesel injection (right).

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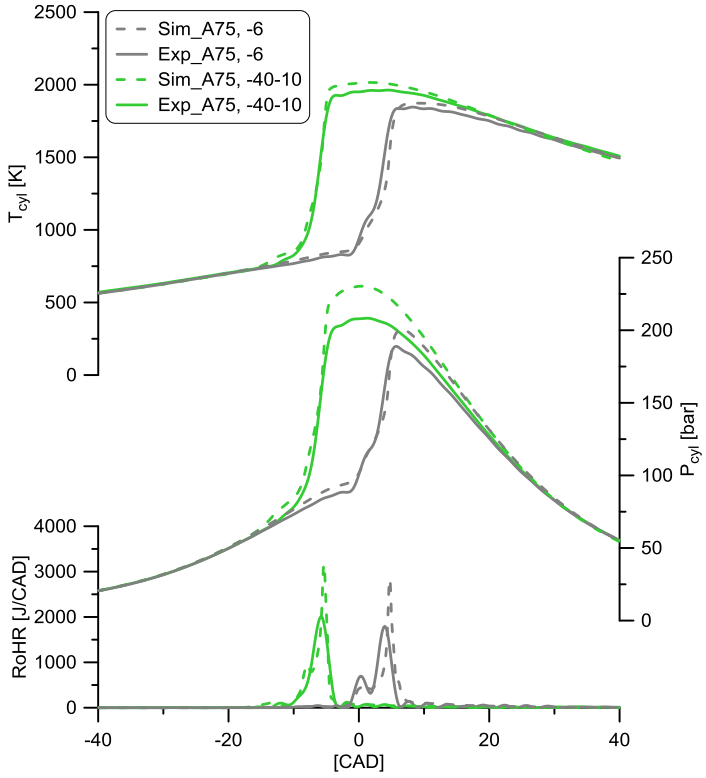

Figure 14. RoHR, in-cylinder mean pressure and temperature comparison between modeling and experimental results, for single and double injection strategies, at ICFB 70 and high load operation.

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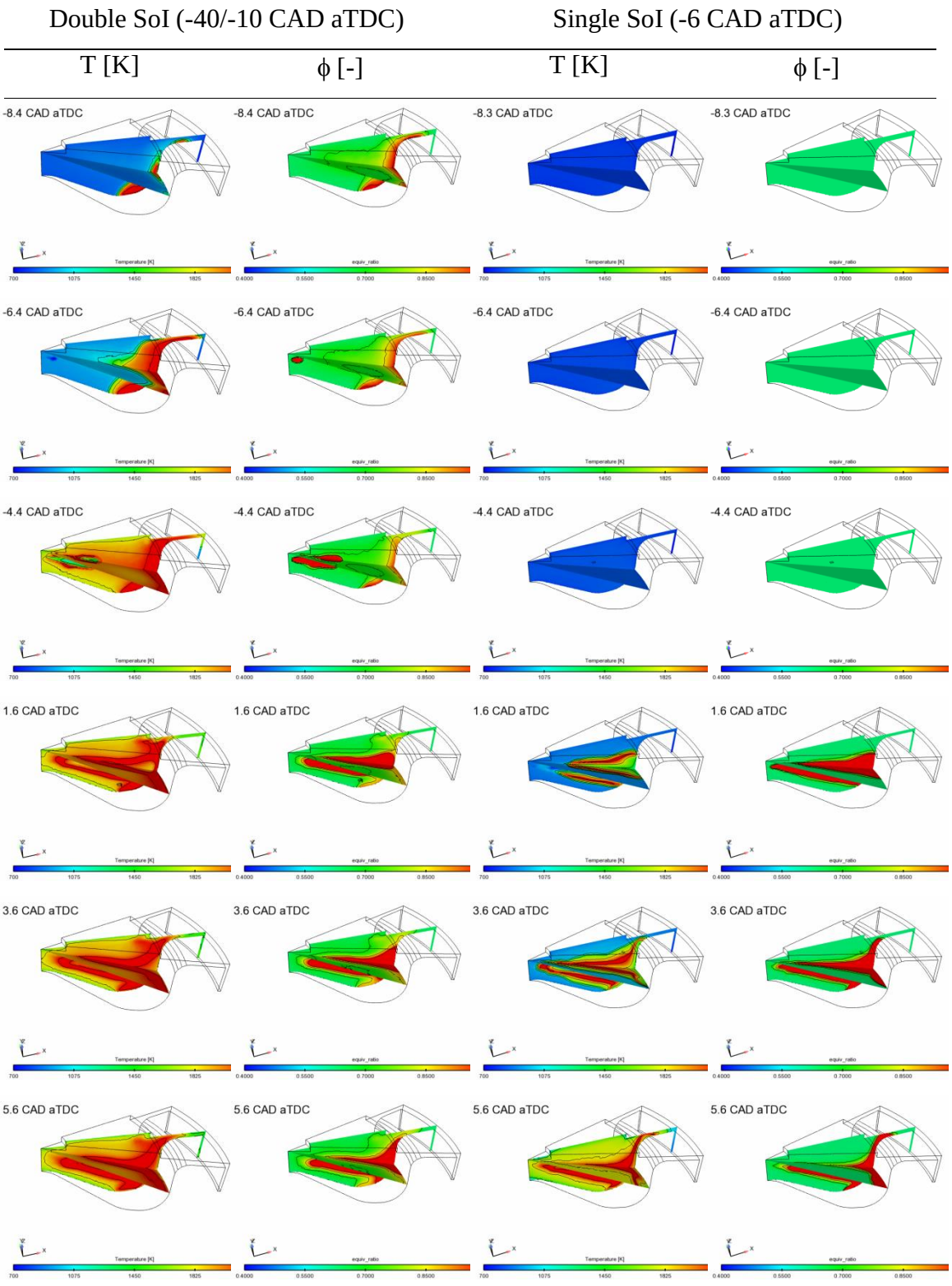

760 Figure 15. Calculated temperature and equivalence ratio cut planes for high load, double
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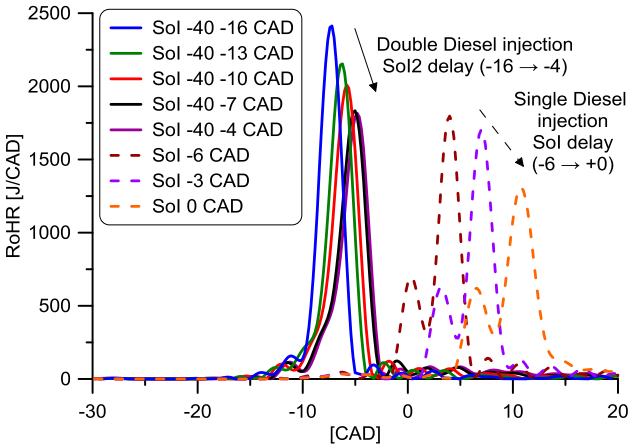

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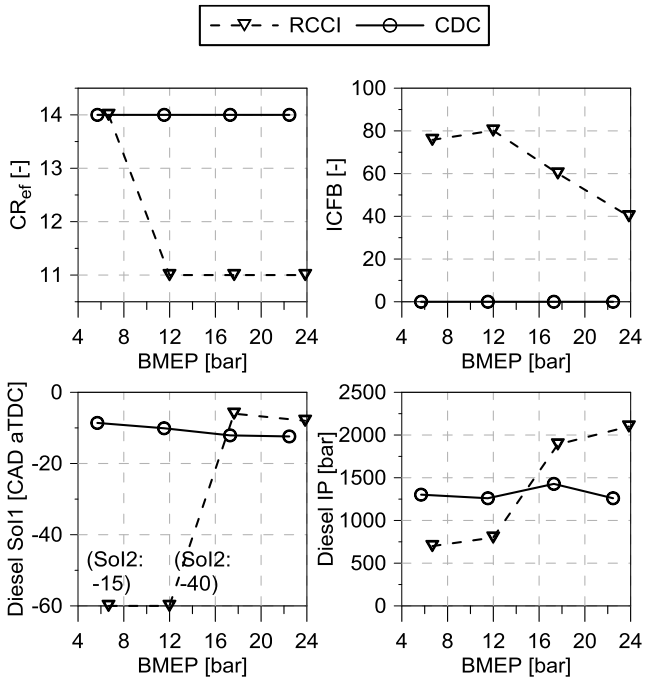

Figure 17. Best RCCI settings, selected for full range operation, and CDC settings provided by the engine manufacturer for analogue operating conditions.

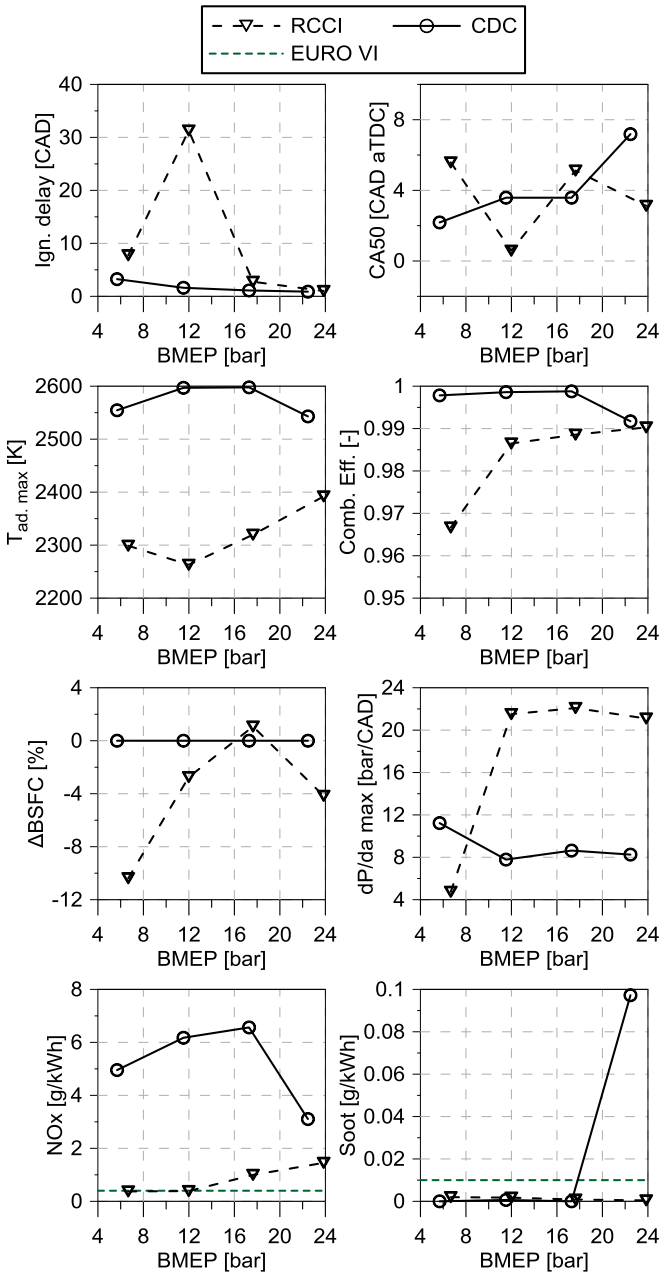

Figure 18. Experimental results from full range RCCI and CDC operation, in terms of performance and emissions.

List of Tables

Table 1. Engine basic specifications.

Engine type		Single cylinder, 4 stroke cycle
Bore x stroke	**[mm]**	123 x 152
Connecting rod length	**[mm]**	225
Displacement	**[cm^3]**	1.806
Geometric compression ratio	**[-]**	14.4:1
Number of valves	**[-]**	4
Valve actuation system		Camless HVA
Diesel fuel injection system		Bosch CRSN 4.2 direct injection Common-rail, Amplifier-piston
Diesel injection nozzle		9 holes, 0.168 mm, 142 deg
Gasoline injection system		Low pressure, 2 port fuel injectors
Port fuel injector model		Bosch EV14 KxT

Table 2. Fuels main properties.

		Gasoline	**Diesel**
Density (T=15ºC)	**[kg/m^3]**	735	824
Viscosity (T=40ºC)	**[cSt]**	0.45	2.8
Octane Number	**[-]**	98	-
Cetane Number	**[-]**	-	52
Lower Heating Value	**[kJ/kg]**	43950	42920

Table 3. Detailed conditions, fixed and (swept), for the parametric studies of the low, medium and high load operation, for the RCCI operation range extension.

		(a)	(b)	(c)
		Low load	**Medium load**	**High load**
Speed	**[rpm]**	1200	1200	1200
Air mass flow	**[kg/h]**	53.3	86	118
Total fuel	**[mg/cycle]**	70	119	175
ICFB (single inj.)	**[%]**	75 (65 → 85)	60 (50 → 70)	70 (50 → 75)
Diesel SoI (single inj.)	**[CAD aTDC]**	-24 (-15 → -24)	-12 (-9 → -21)	-3 (0 → -6)
Diesel IP (single inj.)	**[bar]**	1000	1175	1890
ICFB (double inj.)	**[%]**	-	60 (60 → 90)	-
Diesel SoI1 (double inj.)	**[CAD aTDC]**	-60 (-25 → -60)	-60 (-35 → -65)	-40 (-40 → -60)
Diesel SoI2 (double inj.)	**[CAD aTDC]**	-15 (-15 → -50)	-10 (-10 →-25)	-10 (-4 → -16)
Diesel ratio (double inj.)	**[$\%_{SoI1}$]/[$\%_{SoI2}$]**	(40/60 → 60/40)	(30/70 → 50/50)	50/50
Diesel IP (double inj.)	**[bar]**	700	800	900
CR_{ef}	**[-]**	14	(14 → 11)	11
P_{int}	**[bar]**	1.35	(2.2 → 2.92)	3.39
T_{int}	**[ºC]**	40	40	32
$YO_{2\ IVC}$	**[%]**	15.5	15.1	16.3