Miller cycle for improved efficiency, load range and emissions in a heavy-duty engine running under reactivity controlled compression ignition combustion

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Abstract

The low temperature, premixed combustion strategies are being investigated in the recent years as a mean to break the NOx-soot trade-off appearing during the diffusive conventional diesel combustion. This approach relies on promoting premixed combustion events with shortened duration, which reduces the heat transfer losses, improves the thermal efficiency, and allows a simultaneous reduction of engine-out NOx and soot emissions. However, since the combustion onset only depends on chemical kinetics, most of these strategies cannot be implemented at medium and high loads due to excessive pressure gradients, which lead to unacceptable noise levels and reliability issues.

This experimental work investigates the potential of the Miller cycle as a strategy to minimize the aforementioned challenges when operating under reactivity controlled compression ignition combustion. Moreover, the coupled effect of the Miller cycle with the fuel reactivity modulation is also explored as a way for improving the combustion control. For this purpose, parametric studies varying the effective compression ratio and gasoline fraction have been done in a single-cylinder heavy-duty engine operating at 14 bar indicated mean effective pressure and 1200 rev/min as a baseline condition. The results show that this strategy allows better control of the in-cylinder thermodynamic conditions, enabling a simultaneous reduction of nitrogen oxides and soot emissions down to the EURO VI limits, while keeping a reduced fuel consumption and suitable in-cylinder maximum pressure gradients.

Keywords

Reactivity controlled compression ignition; Miller cycle; emissions control; engine efficiency

1. Introduction

Nowadays, conventional diesel combustion (CDC) is the main combustion strategy used for heavy-duty applications, mainly due to its reliability and fuel efficiency [1]. However, the CDC process entails a trade-off between nitrogen oxides (NOx) and soot emissions, from which the reduction of one pollutant provokes and increase of the other [2]. To fulfill the limits imposed in the emissions regulations [3], the compression ignition (CI) engines, mainly running on CDC, need using aftertreatment systems to cut the emissions before being emitted to the atmosphere [4]. Motivated to reduce the cost of these aftertreatment devices, the engine manufacturers are focusing part of their resources on developing new strategies to minimize the emissions directly from their source, i.e., during the combustion process [5].

In this scenario, the highly premixed combustion (HPC) strategies have arisen as combustion strategies in which the temperatures are too low for promoting the NOx formation [6][7], and the local equivalence ratios are enough lean to avoid the soot formation . Those lean, diluted conditions are attained by decoupling the fuel injection event from the combustion event [8]. The main challenges with the HPC strategies is that they lead to high levels of noise, hydrocarbons (HC) and carbon monoxide (CO) emissions. Moreover, the operating range becomes limited due to the lack of control over the combustion phasing, which also causes excessive maximum pressure rise rates (MPRR) as the engine operation load increases [9][10].

The improvements on the injection systems technologies have allowed to minimize the mentioned challenges, enabling the evolution of the HPC strategies. Firstly, the Homogeneous Charge Compression Ignition (HCCI) combustion was developed [11]. During HCCI operation, the diesel fuel is injected at the intake port and then it mixes with the air before entering into the cylinder. In the Premixed Charge Compression Ignition (PCCI) combustion strategy [12], the diesel fuel is injected directly into the cylinder, early during the compression stroke, to provide enough mixing time to generate lean equivalence ratios [13]. Finally, other strategies using multiple direct injection pulses with different type of fuels such as the Partially Premixed Combustion (PPC) [14]have been used to achieve equivalence ratio stratification [15][16].

Despite the evolution of the HPC concepts, the challenge about the narrow engine operating range was still unsolved [17]. Therefore, the proper operation with the HPC strategies requires using other additional strategies to control the ignition delay and in-cylinder pressure gradients [18]. One of the most effective ways to extend the ignition delay is the use of exhaust gas recirculation (EGR), which is based on increasing the in-cylinder specific heat capacity and reduce the oxygen content to slow down the temperature rise during the compression stroke [19]. Other effective methods arose thanks to the improvements in the air management technologies, such as the variable valve actuation systems [20]. These systems enabled the engine operation with variable compression ratio to control the in-cylinder temperature and pressure [21][22]. The fuel reactivity control is one of the latest strategies that has appeared inside the HPC concepts [23], which is used to tune the fuel auto-ignition characteristics according to the engine operating conditions [24]. This method for controlling the combustion process is the basis of the relatively new combustion concept commonly known as reactivity controlled compression ignition combustion (RCCI) [25].

The RCCI concept relies on injecting a low reactivity fuel through the intake port and a high reactivity fuel injected directly into the cylinder [26]. The use of one injection system for each fuel allows optimizing the global fuel reactivity according to changes in the engine operating conditions. This can be done almost on a cycle-to-cycle basis by varying the gasoline fraction (GF) [27]. Moreover, the fuel reactivity stratification inside the cylinder can be easily modulated through the direct injection timing modification [28]. This action promotes an equivalence ratio and octane number stratification [29], leading to a more sequential combustion event than other HPC strategies [30]. On the other hand, since the mechanism for starting the combustion process is the autoignition of the in-cylinder charge, the control of the thermodynamic conditions is also a key aspect for optimizing this type of combustion.

Literature demonstrates that RCCI combustion concept is able to modify the NOx-soot trade-off appearing with CDC, reducing both pollutants to near-zero levels with thermal efficiencies similar or better than CDC [31]. However, several studies have found the operational range of RCCI to be limited as load increases due to excessive in-cylinder pressure gradients [32][33]. The objective of this investigation is to analyze the capabilities of extending the RCCI operating range by modulating the in-cylinder thermodynamic conditions and fuel autoignition properties. In this sense, the coupled effect of both actions is expected to reduce the in-cylinder peak pressures and temperatures, which will improve the RCCI combustion control at relatively high engine loads. To prove this, the effective compression ratio of the engine is varied through the Miller cycle strategy application, and the global fuel reactivity is tuned by varying the gasoline-to-total fuel ratio. The engine tests have been performed in a single-cylinder, heavy-duty compression ignition engine equipped with a hydraulic variable valve actuation (HVA) system running at 14 bar indicated mean effective pressure (IMEP) and 1200 rpm as a baseline condition.

2. Materials and methods

2.1. Engine, fuels and test cell description

The main component used in this research is a single-cylinder heavy-duty compression ignition engine equipped with a hydraulic VVA system with a dedicated oil circuit, whose main specifications are given in Table 1.

Table 1. Engine specifications.

Engine type	Single cylinder, 4 stroke cycle
Bore x stroke [mm]	123 x 152
Connecting rod length [mm]	225
Displacement [cm^3]	1806
Maximum injection pressure [bar]	2300
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

The hydraulic VVA system allows to actuate over all the valves independently trough different hydraulic pistons mounted in each valve. These hydraulic pistons are controlled by means of a dedicated electronic control unit that allows managing the opening timing, the opening duration and lift of each valve facilitating the Miller cycle implementation. One characteristic of the HVA system is the fast-motion of the intake and exhaust valve lift profiles. When the speeds of the valves opening and closing events are increased there is a reduction in the gas energy losses because the time of the exhaust gas flow under sonic conditions is reduced [34]. On the other hand, a non-desired effect can appear if those fast-motion systems are used in engines with small clearance between the valves and the piston. In this sense, to avoid the direct mechanical interference when the piston is close to the top dead center (TDC), the valve overlap at TDC is cancelled causing small reductions of the engine volumetric efficiency.

In terms of fuel injection equipment, the engine has two different injection systems, one for each different fuel used. In this case, the high reactivity fuel is EN590 diesel and the low reactivity fuel is EN228 98 octane number gasoline. Their main properties are listed in Table 2. All the properties were obtained in the fuels laboratory at CMT-Motores Térmicos following the American society for testing and materials (ASTM) standards. The diesel fuel was injected using a common-rail direct injection system (Bosch CRSN 4.2). This system gives the opportunity to extend the numbers of injections up to five injections per cycle and serves at the same time to increase the common-rail fuel pressure using a hydraulic amplifier piston that is installed inside the injector. In the case of the low reactivity fuel, a port fuel injection system composed by two equal injectors located radially opposite at the intake port allowing injecting the gasoline, as is described in [35].

Table 2. Properties of the used fuels.

	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
Ethanol content [% vol.]	<5	-

The single-cylinder engine is installed in an instrumented test cell, as illustrated in Figure 1. In this layout can be observed all the auxiliary facilities required for its operation and control as well as the measurement and data acquisition equipment.

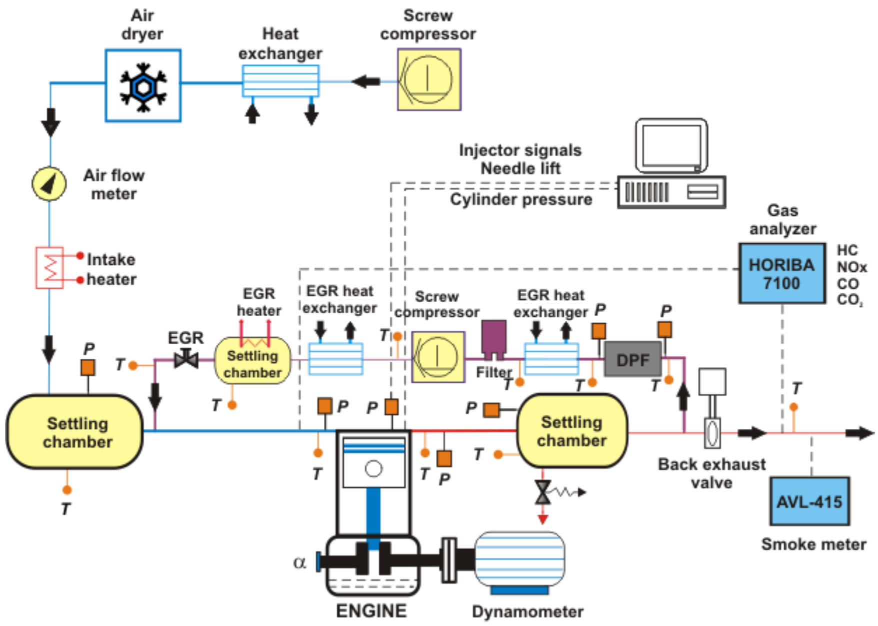

Figure 1: Test cell layout.

As shown in Figure 1, an externally driven screw compressor is used to supply the required boost pressure to reach the desired intake air conditions. After that, the air passes through an air dryer to eliminate all the humidity. Later, the air pressure is stabilized and controlled in the intake settling chamber, the same place where the temperature is measured. The exhaust loop is composed by a valve placed in the exhaust system which replicates the exhaust backpressure caused by the turbine mounted in the real multi-cylinder engine, while the pressure was controlled in the exhaust settling chamber in order to have stabilized pressure values.

The EGR system used in this work is a low pressure EGR circuit installed in the same engine test cell. An externally driven roots-type supercharger was in charge to boost the exhaust gas pressure in levels over the intake pressure but in this case before passing for the supercharger the solid particles and liquid droplets are removed from the exhaust gases through a DPF, heat exchanger and filter group. An EGR valve placed between the EGR settling chamber and the intake pipe allows to control the EGR rate. Finally, the exhaust gas mass flow is driven to the intake pipe depending on the each EGR rate desired. The EGR-fresh air mixture intake temperature regulation was performed by means of a temperature sensor located in the intake manifold.

In terms of measuring device, a variable sampling smoke meter (AVL 415) is used to quantify the smoke emissions. For this purpose, three consecutive measurements of 1 liter volume each with paper-saving mode off were acquired and averaged. The results provided by this equipment are directly transformed from Filter Smoke Number units (FSN) to dry soot mass emissions using the calculation procedure as the research done by Christian et al. [36]. The gaseous emissions were measured by a Horiba MEXA 7100 DEGR system which is one of the state-of-the-art analyzers, giving the concentrations of CO_2 (intake and exhaust), CO, NOx, O_2 and HC while the EGR rate was calculated using the experimental measurement of intake and exhaust CO_2 concentration. The accuracy of the instrumentation used in this work is summarized in Table 3.

Table 3. Accuracy of the instrumentation used in this work.

Variable measured	Device	Manufacturer / model	Accuracy
In-cylinder pressure	Piezoelectric transducer	Kistler / 6125B	±1.25 bar
Intake/exhaust pressure	Piezorresistive transducers	Kistler / 4045A10	±25 mbar
Temperature in settling chambers and manifolds	Thermocouple	TC direct / type K	±2.5 °C
Crank angle, engine speed	Encoder	AVL / 364	±0.02 CAD
NOx, CO, HC, O_2, CO_2	Gas analyzer	HORIBA / MEXA-ONE-D1-EGR	4%
FSN	Smoke meter	AVL / 415S	±0.025 FSN
Gasoline/diesel fuel mass flow	Fuel balances	AVL / 733S	±0.2%
Air mass flow	Air flow meter	Elster / RVG G100	±0.1%

For the combustion analysis, the in-cylinder pressure traces are acquired by means of a non-cooled piezo-electric transducer (Kistler 6125B) along 3 repetitions of 100 consecutives engine cycles using a shaft encoder with 1800 pulses per revolution, which provides a resolution of 0.2 CAD. Later, an in-house developed combustion diagnosis code (CALMEC) [37][38] is used to process and analyze the in-cylinder pressure signal data. This combustion diagnosis code provides direct use engine operating data such as the rate of heat release (RoHR) or entrance data for other models as the unburned gases temperature. This parameter is a key input data for the calculation of the adiabatic flame temperature (T_{ad}), which is calculated in this case in the same way than Way et al. [39]. The complete details for the T_{ad} calculation can be found in [40].

The tests repeatability was controlled measuring three times every operation point coupled with a control reference point measured before every testing session. Both verification measures guarantee the reliability of all the results provided along the study.

2.2. Methods

A set of parametric studies have been done to study the effect of the Miller cycle and the fuel reactivity variation on RCCI combustion characteristics. First, the effective compression ratio (CR_{ef}) has been modified by means of the Miller cycle strategy using a fixed gasoline fraction (GF), which is defined as the gasoline-to-total fuel mass ratio. The Miller cycle was implemented by advancing the intake valves closing (IVC) angle, as shown in Figure 2. The air mass flow reduction caused by the shorter intake event was compensated by increasing the boost pressure as depicted in Table 4. The second study tackles the effect of the in-cylinder fuel reactivity variation under different CR_{ef}. This was done by modifying the GF at different ratios while keeping constant the total amount of fuel injected. The conditions for both studies are summarized in Table 4.

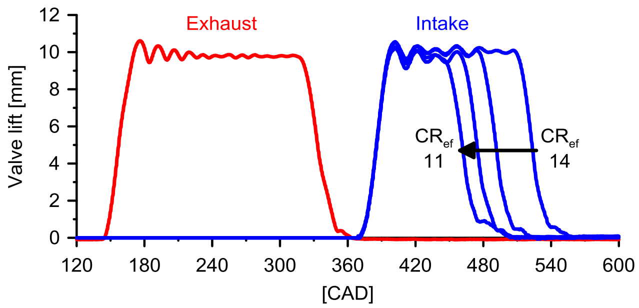

Figure 2. Valve lift profiles for the different Miller cycle.

Table 4: Engine settings for CRef and GF sweeps.

Engine speed [rpm]	1200
Total fuel mass [mg/cycle]	119
GF [%]	70, 80, 90
m_{air} [kg/h]	86
EGR [%]	45
P_{int} [bar]	2.2, 2.4, 2.62, 2.92
IDUR [CAD]	160, 128, 112, 99
CR_{ef} [-]	14, 13, 12, 11

The baseline condition selected for the engine tests was 14 bar IMEP and 1200 rpm, as it was found as an upper boundary condition for RCCI [41]. Highly premixed conditions were promoted using two diesel injection pulses with advanced injection timing: -60 and -40 CAD ATDC, respectively. The gasoline fuel was injected in the intake port 10 CAD after the intake valve opening. This injection strategy was selected because it was found to be able to provide ultra-low NOx and soot emissions with simultaneous high efficiency [42]. Table 5 summarizes the values for the parameters that were kept constant during the tests.

Table 5: Experimental conditions.

Engine speed [rpm]	1200
Diesel IP [bar]	800
Diesel SoI [CAD aTDC]	-60 (pilot), -40 (main)
EVC [CAD aTDC]	347
EDUR [CAD]	200
IVO [CAD aTDC]	375
Intake air temperature [ºC]	40
Coolant temperature [ºC]	80
Oil temperature [ºC]	90

3. Results and discussion

This section presents the main results of the current investigation. The first subsection is dedicated to evaluate the engine performance and emissions as the CR_{ef} is lowered from 14:1 to 11:1 by using the Miller cycle strategy. The second one discusses the role of the fuel auto-ignition qualities on RCCI combustion control, performance and emissions.

3.1. Evaluation of Miller cycle strategy

Figure 3 shows the diesel injection profiles and rate of heat release (RoHR) for different CR_{ef} and 80% GF. From Figure 3 it is seen that, with nominal compression ratio, the combustion event evolves entirely during the compression stroke. This is explained due to the combination of highly premixed conditions and high levels of pressure and temperature attained before TDC. In addition, it is shown that the Miller strategy is able to extend the ignition delay up to moving the RoHR peak near the TDC. This occurs because of the lowering of the in-cylinder temperatures histories as CR_{ef} is reduced. To support this statement, Figure 4 shows the in-cylinder averaged temperatures at IVC (T_{IVC}) and ignition delay as a function of the CR_{ef}. T_{IVC} becomes reduced around 24°C and the ignition delay becomes around 7 CAD longer. The slower chemical kinetics from the lower temperature histories also implies slightly larger combustion events, as Figure 4 quantifies.

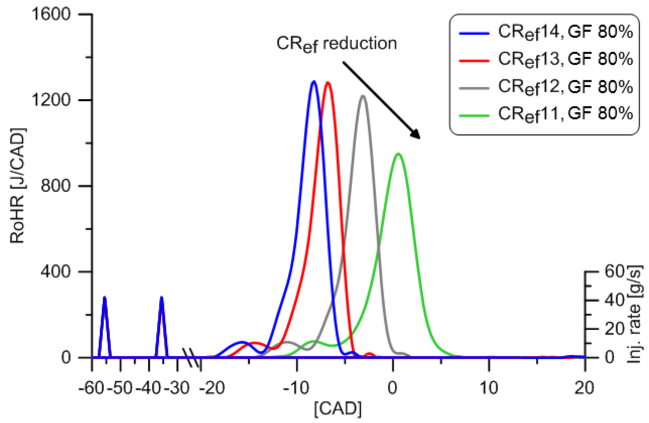

Figure 3: Diesel injection rates and RoHR for different effective compression ratios and fixed GF of 80%.

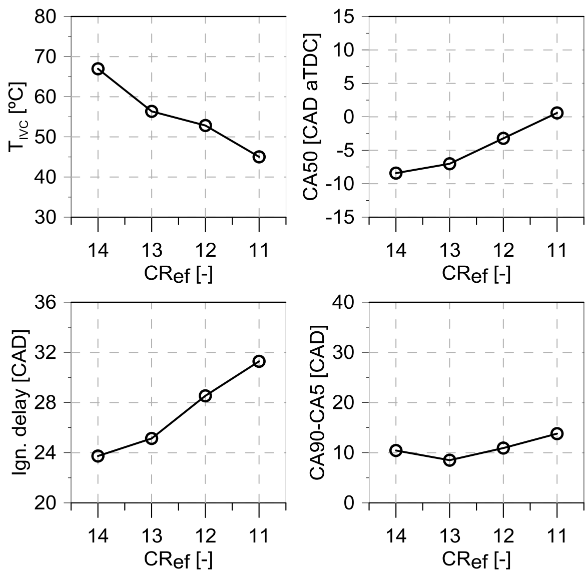

Figure 4: Temperature at IVC, ignition delay, CA50 and combustion duration as a function of the effective compression ratio at fixed GF of 80%.

As confirmed in Figure 5, the potential of Miller cycle strategy to control in-cylinder temperature histories resulted also in a mechanism to control NOx emissions. As literature demonstrates, NOx formation occurs at stoichiometric equivalence ratios and high temperatures [41]. Thus, as Figure 6 shows, the lower CR_{ef} the lower adiabatic flame temperatures, which agrees with the reduction of NOx emissions shown in Figure 5.

Looking at Figure 5, it can be stated that the use of the Miller strategy does not imply any noticeable change in terms of soot either in HC emissions. The soot level remains EURO VI compliant due to the lean local equivalent ratios attained from using this highly premixed combustion strategy. By contrast, the level of HC emissions is one order of magnitude above the EURO VI regulation. As literature demonstrates, high HC emissions levels are inherent during RCCI combustion due to the difficulty in burning the gasoline fuel located near the cylinder wall and crevice regions [42].

Despite the CR_{ef} lowering provided slightly longer combustion duration (which could imply longer time available for CO oxidation), as Figure 5 presents, the level of these emissions tend to increase with the lowering of the CR_{ef}. It is because the lowering of in-cylinder temperature history implies the worsening of CO oxidation processes, which has a stronger effect than the mentioned from longer combustion duration.

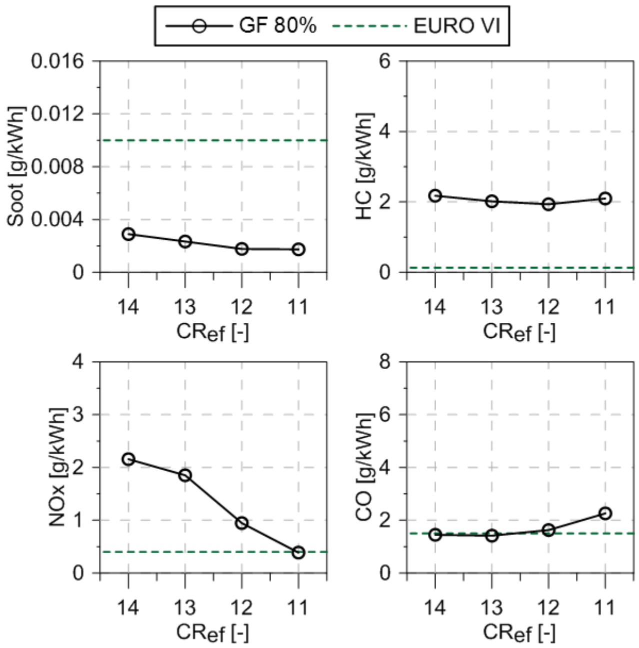

Figure 5: Engine-out emissions as a function of the effective compression ratio at fixed GF of 80%.

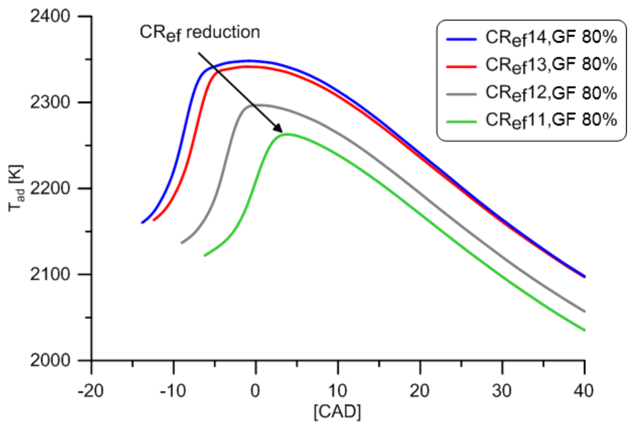

Figure 6: Adiabatic flame temperature for different effective compression ratios at fixed GF of 80%.

Figure 7 shows the results in terms of engine performance obtained from the CR_{ef} sweep. The fuel consumption has been represented in terms of percentage, where negative values mean an improvement with respect to the result obtained with GF 70% and CR_{ef} 14:1. Accordingly, Figure 7 highlights around 7% fuel consumption benefit from the use of the Miller strategy. Despite the lowering of the CR_{ef} implies slightly lower combustion efficiency, a more favorable combustion phasing (CA50) from the fuel-to-work conversion process standpoint is attained. This promotes an increase of the IMEP, which justifies the fuel consumption results. Another important result achieved from the Miller cycle sweep is that maximum pressure gradients are lowered, which is well related with the warmer temperature histories of the reduced CR_{ef}.

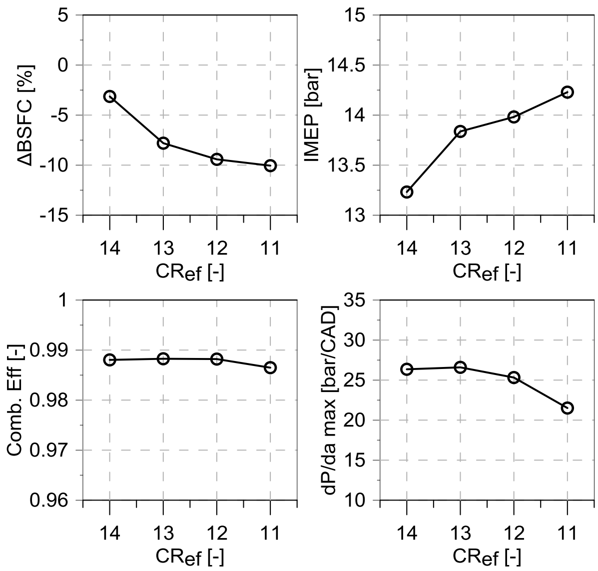

Figure 7: Engine performance as a function of the effective compression ratio at a fixed GF of 80%.

3.2. Evaluation of fuel reactivity control strategy

This subsection is focused on evaluating the role of the fuel auto-ignition qualities on the ignition delay and combustion development control. The engine performance and emissions are evaluated for three different GF (70%, 80% and 90%) using the same injection pattern and CR_{ef} levels than in the previous subsection.

Figure 8 shows the diesel injection profiles and RoHR from the experiments with different GF ratios at CR_{ef} 11:1. For the shake of clarity, only the traces corresponding to CR_{ef} 11:1 have been presented in the figure. These cases were selected because the lowered in-cylinder thermodynamic conditions enhance differences between RoHR of the fuel blending ratios tested. Nonetheless, the general behavior for the different CR_{ef} is similar. Looking to the RoHR profiles in Figure 8, it can be stated that the fuel reactivity tuning strategy is able to extend the ignition delay, firstly because of the T_{IVC} reduction from gasoline vaporization, and then mainly due to the global fuel reactivity reduction. On one hand, most of the case with the highest amount of diesel fuel evaluated (GF 70), burns during the compression stroke. On the other hand, as the amount of gasoline in the blend is increased combustion becomes delayed and RoHR peaks are lowered, attaining a combustion event placed entirely during the expansion stroke (GF90).

Figure 9 shows combustion Temperature at IVC, ignition delay, CA50 and CA90-CA5, for the different GF and CR_{ef} combinations. The effect of GF on combustion phasing is quite similar for the CR_{ef} tested. However, the combination of low in-cylinder temperatures and low global fuel reactivity (CR_{ef} 11:1 and GF 90%) implies much slower chemical kinetics and much longer combustion events than in the other cases.

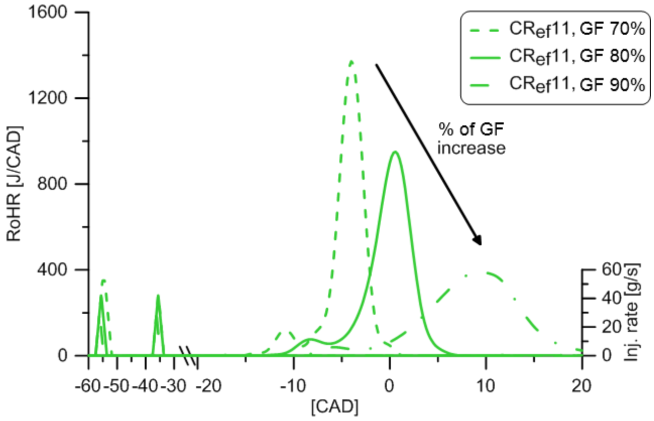

Figure 8: Diesel injection rates and RoHR for different gasoline fractions at a fixed CR_{ef} of 11:1.

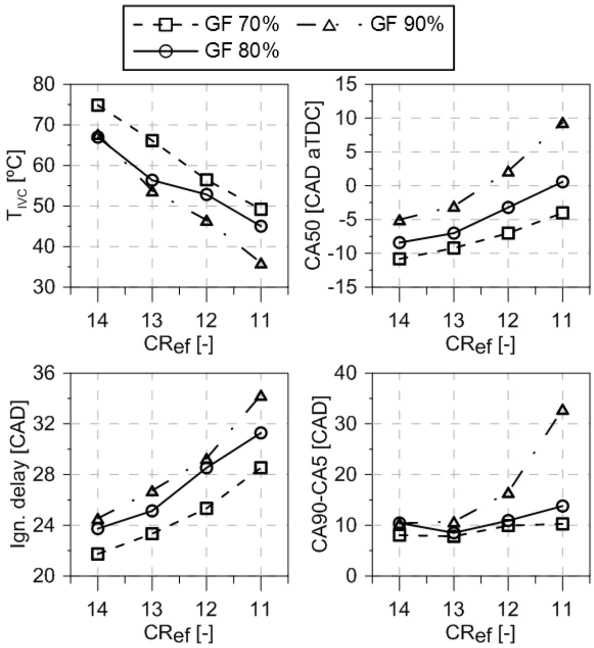

Figure 9: Temperature at IVC, ignition delay, CA50 and combustion duration as a function of the effective compression ratio for GF 70%, 80% and 90%.

An effect of the mentioned chemical kinetics lowering from the global fuel reactivity reduction is shown in Figure 10. The evolution of adiabatic flame temperatures shows lower maximum values (below 2200 K) that avoid NOx emissions formation. These calculations agree with the NOx emissions measurements included in Figure 11. The higher GF the lower NOx emissions level. It should be pointed out how the extreme in-cylinder temperature reduction from the combination of GF 90% and CR_{ef} 11 avoids NOx and also soot formation mechanisms, leading to almost zero emissions level.

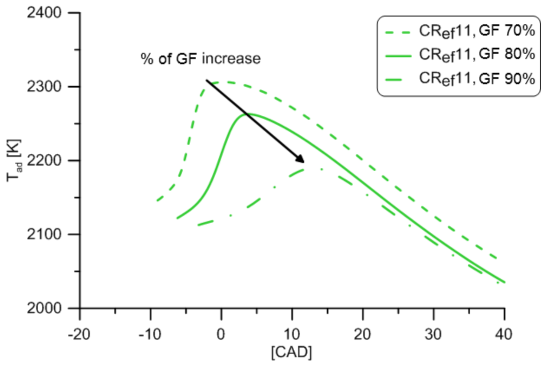

Figure 10: Adiabatic flame temperature for different gasoline fractions at a fixed CR_{ef} of 11:1.

As was noted in the previous subsection, for the RCCI strategy, the level of HC emissions mainly depends on the amount of premixed gasoline. The higher the GF, the larger the HC emissions level. However, the mentioned NOx- and soot-free conditions imply a sharp HC emissions increase. In addition, Figure 11 also shows a sharp increase of the CO emissions level at those conditions. This means that the extreme lowering of global fuel reactivity at reduced thermodynamic conditions results in a worsened combustion process. Results presented in Figure 12 agree with this statement. There is an important worsening of combustion efficiency reflected in longer combustion duration, delayed combustion phasing, lower maximum pressure gradients and lower IMEP. As a consequence, at these operating conditions fuel consumption changes its trend and starts being worsened instead of improved while the CR_{ef} gets reduced. By contrast, at the rest of CR_{ef} tested, higher GF ratios resulted in improved fuel consumption mainly because of better combustion phasing, well related with slightly higher IMEP. On the concern of maximum pressure gradients, the use of higher amounts of gasoline in the fuel blend tends to slow combustion which implies the lowering of maximum pressure gradients. This fact is not as noticeable as at CR_{ef} 14:1 because, at these highly premixed conditions, that effective compression ratio provides enough thermodynamic conditions to a fast onset of combustion, whatever the tested GF.

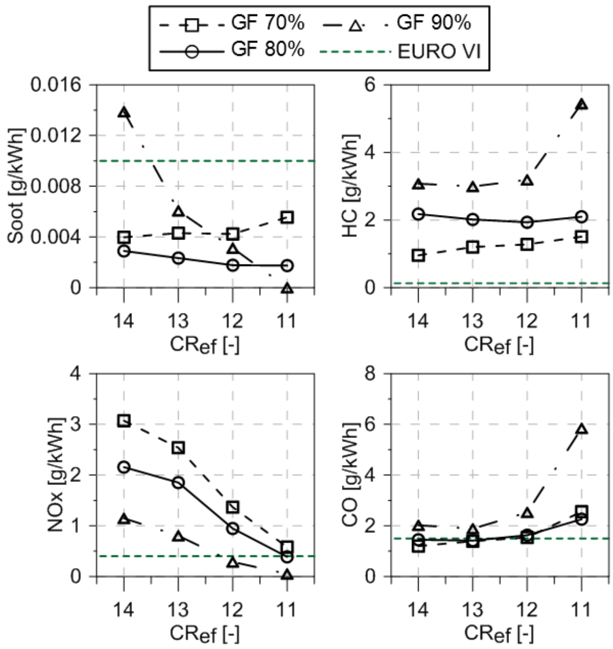

Figure 11: Engine-out emissions as a function of the effective compression ratio for GF 70%, 80%, 90%.

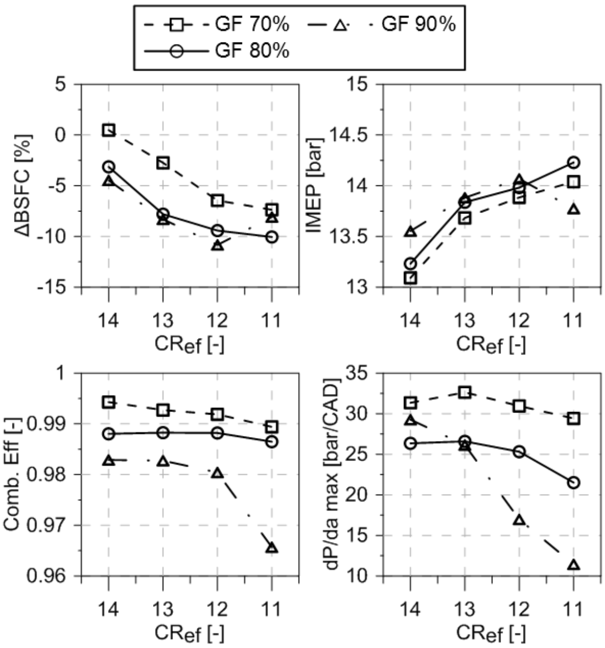

Figure 12: Engine performance as a function of the effective compression ratio for GF 70%, 80%, 90%.

4. Conclusions

A key parameter that governs the combustion control under premixed combustion regimes in compression ignition engines is chemical kinetics. Thus, in-cylinder thermodynamic conditions and fuel auto-ignition characteristics should be controlled to attain high efficiency and low emissions. This work explored the potential of using the Miller cycle to control the in-cylinder thermodynamic conditions when using a highly premixed RCCI strategy. In this sense, four CR and three different gasoline fraction conditions at 14 bar IMEP and 1200 rpm were studied. The most important findings from the study can be summarized as follows:

- The Miller strategy is able to extend the ignition delay and reduce pressure gradients because of the lowering of in-cylinder temperatures histories, from the reduction of the CR_{ef}.
- The potential of Miller strategy to control in-cylinder temperature histories resulted also in a mechanism to control NOx emissions.
- The level of soot emissions remains reduced due to the lean local equivalent ratios attained from using this highly premixed combustion strategy.
- Lowering in-cylinder temperature and global fuel reactivity implies and increase on CO emissions.
- HC emissions mainly depend on the difficulty to burn gasoline near the cylinder wall and crevices, inherent to this implementation of RCCI.
- Despite the lowering of the CR_{ef} and the higher GF imply slightly lower combustion efficiency, better combustion phasing is attained and it is reflected in higher IMEP and fuel consumption improvements.

Therefore, this research work evidences that the combination of Miller cycle with the RCCI combustion strategy is a promising way to extend the engine operating range, meeting future emissions regulations without expensive after-treatment systems, while keeping competitive levels of fuel consumption. However, further research is still needed to provide suitable strategies to operate this combustion concept all over the engine operating range.

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References

[1] Tauzia X, Maiboom A, Chesse P, Thouvenel N. A new phenomenological heat release model for thermodynamical simulation of modern turbocharged heavy duty Diesel engines. Applied Thermal Engineering, Volume 26, Issue 16, November 2006, Pages 1851-1857.

[2] Pierpont D, Montgomery D and Reitz RD. Reducing Particulate and NOx Using Multiple Injections and EGR in a D.I. Diesel, SAE Technical Paper 950217, 1995.

[3] Luján JM, Bermúdez V, Dolz V, Monsalve-Serrano J. An assessment of the real-world driving gaseous emissions from a Euro 6 light-duty diesel vehicle using a portable emissions measurement system (PEMS). Atmospheric Environment, Volume 174, Feb 2018, Pages 112-121.

[4] Posada F.; Chambliss S.; Blumberg K. Costs of emission reduction technologies for heavy-duty diesel vehicles. ICCT White paper 2016.

[5] García-Valladolid P, Tunestal P, Monsalve-Serrano J, García A, Hyvönen J. Impact of diesel pilot distribution on the ignition process of a dual fuel medium speed marine engine. Energy Conversion and Management, Volume 149, 1 Oct 2017, Pages 192-205.

[6] Fang O, Fang J, Zhuang J, Huang Z. Effects of ethanol–diesel–biodiesel blends on combustion and emissions in premixed low temperature combustion. Applied Thermal Engineering, Volume 54, Issue 2, 30 May 2013, Pages 541-548.

[7] Hasan MM, Rahman MM, Kadirgama K, Ramasamy D. Numerical study of engine parameters on combustion and performance characteristics in an n-heptane fueled HCCI engine. Applied Thermal Engineering, Volume 128, 5 January 2018, Pages 1464-1475.

[8] Ryan TW and Callahan TJ. Homogeneous Charge Compression Ignition of Diesel Fuel, SAE Technical Paper 961160, 1996.

[9] Liu H, Zhang P, Li Z, Luo J, Zheng Z, Yao M. Effects of temperature inhomogeneities on the HCCI combustion in an optical engine. Applied Thermal Engineering, Volume 31, Issues 14–15, October 2011, Pages 2549-2555.

[10] Epping K, Aceves S, Bechtold R and Dec J. The Potential of HCCI Combustion for High Efficiency and Low Emissions, SAE Technical Paper 2002-01-1923, 2002.

[11] Torres García M, José Jiménez-Espadafor Aguilar F, Sánchez Lencero T, Antonio Becerra Villanueva J. A new heat release rate (HRR) law for homogeneous charge compression ignition (HCCI) combustion mode. Applied Thermal Engineering, Volume 29, Issues 17–18, December 2009, Pages 3654-3662.

[12] Mei D, Yue S, Zhao X, Hielscher K, Baar R. Effects of center of heat release on combustion and emissions in a PCCI diesel engine fuelled by DMC-diesel blend. Applied Thermal Engineering, Volume 114, 5 March 2017, Pages 969-976.
[13] Hasan MM, Rahman MM, Kadirgama K, Ramasamy D. Numerical study of engine parameters on combustion and performance characteristics in an n-heptane fueled HCCI engine. Applied Thermal Engineering, Volume 128, 5 January 2018, Pages 1464-1475.
[14] Benajes J, Molina S, García A, Monsalve-Serrano J, Durrett R. Performance and engine-out emissions evaluation of the double injection strategy applied to the gasoline partially premixed compression ignition spark assisted combustion concept. Applied Energy, Volume 134, 2014, Pages 90-101.
[15] López J, García-Oliver JM, García A, Domenech V. Gasoline effects on spray characteristics, mixing and auto-ignition processes in a CI engine under Partially Premixed Combustion conditions. Applied Thermal Engineering, Volume 70, Issue 1, 5 September 2014, Pages 996-1006.
[16] Benajes J, Molina S, García A, Monsalve-Serrano J, Durrett R. Conceptual model description of the double injection strategy applied to the gasoline partially premixed compression ignition combustion concept with spark assistance. Applied Energy, Volume 129, 2014, Pages 1-9.
[17] Lee K, Cho S, Kim N, Min K. A study on combustion control and operating range expansion of gasoline HCCI. Energy, Volume 91, November 2015, Pages 1038-1048.
[18] Komninos NP. The effect of thermal stratification on HCCI combustion: A numerical investigation. Applied Energy, Volume 139, 1 February 2015, Pages 291-302.
[19] Peng Z, Zhao H and Ladommatos N. Effects of Air/Fuel Ratios and EGR Rates on HCCI Combustion of N-Heptane, a Diesel Type Fuel, SAE Technical Paper 2003-01-0747, 2003.
[20] Knop V, Mattioli L. An analysis of limits for part load efficiency improvement with VVA devices. Energy Conversion and Management, Volume 105, 15 November 2015, Pages 1006-1016.
[21] Liu M-B, He B-Q, Zhao H. Effect of air dilution and effective compression ratio on the combustion characteristics of a HCCI (homogeneous charge compression ignition) engine fuelled with n-butanol. Energy, Volume 85, 1 June 2015, Pages 296-303.
[22] Benajes J, Molina S, Martín J and Novella R. Effect of advancing the closing angle of the intake valves on diffusion-controlled combustion in a HD diesel engine, Applied Thermal Engineering, 29, 1947-1954, 2009.
[23] Bessonette PW, Schleyer CH, Duffy KP, Hardy WL, Liechty MP. Effects of fuel property changes on heavy-duty HCCI combustion. SAE paper 2007-01-0191, 2007.
[24] Inagaki K, Fuyuto T, Nishikawa K, Nakakita K, Sakata I. Dual-Fuel PCI Combustion Controlled by In-Cylinder Stratification of Ignitability. SAE Technical Paper 2006-01-0028, 2006.
[25] Kokjohn SL, Hanson RM, Splitter DA and Reitz RD. Fuel reactivity controlled compression ignition (RCCI): a pathway to controlled high-efficiency clean combustion, Int J Engine Res. 12:209-26, 2011.
[26] Benajes J, Molina S, García A, Monsalve-Serrano J. Effects of low reactivity fuel characteristics and blending ratio on low load RCCI (reactivity controlled

compression ignition) performance and emissions in a heavy-duty diesel engine. Energy, Volume 90, October 2015, Pages 1261–1271.
[27] Wang Y, Yao M, Li T, Zhang W, Zheng Z. A parametric study for enabling reactivity controlled compression ignition (RCCI) operation in diesel engines at various engine loads. Applied Energy, Volume 175, 1 August 2016, Pages 389-402.
[28] Benajes J, Molina S, García A, Monsalve-Serrano J. Effects of Direct injection timing and Blending Ratio on RCCI combustion with different Low Reactivity Fuels. Energy Conversion and Management, Volume 99, July 2015, Pages 193-209.
[29] Kokjohn S, Musculus M , Reitz R. Evaluating temperature and fuel stratification for heat-release rate control in a reactivity-controlled compression-ignition engine using optical diagnostics and chemical kinetics modeling. Combustion and Flame, Volume 162, Issue 6, June 2015, Pages 2729-2742.
[30] Luong MB, Yu GH, Chung SH, Yoo CS. Ignition of a lean PRF/air mixture under RCCI/SCCI conditions: Chemical aspects. Proceedings of the Combustion Institute, Volume 36, Issue 3, 2017, Pages 3587-3596.
[31] Benajes J, García A, Pastor JM, Monsalve-Serrano J. Effects of piston bowl geometry on Reactivity Controlled Compression Ignition heat transfer and combustion losses at different engine loads. Energy, Volume 98, March 2016, Pages 64-77.
[32] Wang Y, Zhu Z, Yao M, Li T, Zhang W, Zheng Z. An investigation into the RCCI engine operation under low load and its achievable operational range at different engine speeds. Energy Conversion and Management, Volume 124, 15 September 2016, Pages 399-413.
[33] Kavuri C, Paz J, Kokjohn S. A comparison of Reactivity Controlled Compression Ignition (RCCI) and Gasoline Compression Ignition (GCI) strategies at high load, low speed conditions. Energy Conversion and Management, Volume 127, 1 November 2016, Pages 324-341.
[34] Benajes J, Serrano JR, Dolz V, Novella R. Analysis of an extremely fast valve opening camless system to improve transient performance in a turbocharged high speed direct injection diesel engine. International Journal of Vehicle Design 2009;49:192-13.
[35] Benajes J, García A, Pastor JM, Monsalve-Serrano J. Effects of piston bowl geometry on Reactivity Controlled Compression Ignition heat transfer and combustion losses at different engine loads. Energy, Volume 98, March 2016, Pages 64-77.
[36] Christian R, Knopf F, Jasmek A, Schindler W. A New Method for the Filter Smoke Number Measurement with Improved Sensitivity. MTZ Motortechnische Zeitschift 1993;54:16-22.
[37] Lapuerta M, Armas O, Hernández JJ. Diagnostic of D.I. Diesel Combustion from In-Cylinder Pressure Signal by Estimation of Mean Thermodynamic Properties of the Gas. Applied Thermal Engineering, 1999;19:513–29.
[38] Payri F, Molina S, Martín J, Armas O. Influence of measurement errors and estimated parameters on combustion diagnosis. Applied Thermal Engineering 2006;26:226–36.
[39] Way RJB. Methods for Determination of Composition and Thermodynamic Properties of Combustion Products for Internal Combustion Engine Calculations. Proceedings of the Institution of Mechanical Engineers 1976;190(60):686-97.

[40] Benajes J, Novella R, García A, Arthozoul S. The role of in-cylinder gas density and oxygen concentration on late spray mixing and soot oxidation processes. Energy, Volume 36, Issue 3, March 2011, Pages 1599-1611.
[41] Benajes J, Molina S, García A, Belarte B, Vanvolsem M. An investigation on RCCI combustion in a heavy duty diesel engine using in-cylinder blending of diesel and gasoline fuels. Applied Thermal Engineering, Volume 63, Issue 1, 5 February 2014, Pages 66-76.
[42] Desantes JM, Benajes J, García A, Monsalve-Serrano J. The Role of the In-Cylinder Gas Temperature and Oxygen Concentration over Low Load RCCI Combustion Efficiency. Energy, Volume 78, December 2014, Pages 854–868.

Abbreviations

ASTM American Society for Testing And Materials

ATDC After Top Dead Center

BSFC Brake Specific Fuel Consumption

CAD Crank angle degree

CA5 Angle when 5% of the fuel is burnt

CA50 Angle when 50% of the fuel is burnt

CA90 Angle when 90% of the fuel is burnt

CDC Conventional Diesel Combustion

CO Carbon monoxide

CO_2 Carbon dioxide

CR Compression ratio

CR_{ef} Effective compression ratio

EDUR Exhaust Event Duration

EGR Exhaust Gas Recirculation

EVC Exhaust Valves Closing (angle)

FSN Filter Smoke Number

HC Unburned Hydrocarbon

HCCI Homogeneous Charge Compression Ignition

HPC Highly-Premixed Combustion

HVA Hydraulic Valve Actuation

GF Gasoline Fraction

IDUR Intake Event Duration

IMEP Indicated Mean Effective Pressure

IP	Injection Pressure
IVC	Intake Valves Closing (angle)
IVO	Intake Valves Opening (angle)
MPP	Maximum Pressure Rise Rate
NOx	Nitrogen Oxides
ON	Octane Number
O_2	Oxygen
PCCI	Premixed Charge Compression Ignition
P_{int}	Intake Pressure
PPC	Partially Premixed Combustion
RCCI	Reactivity Controlled Compression Ignition
RoHR	Rate of Heat Release
SoI	Start of Injection
T_{ad}	Adiabatic flame temperature
T_{IVC}	In-cylinder averaged temperature at IVC
TDC	Top Dead Centre
VVA	Variable Valve Actuation