Potential of water direct injection in a CAI/HCCI gasoline engine to extend the operating range towards higher loads

J. Valero-Marco[a,*], B. Lehrheuer[b], Dr. J. J. López[a], S. Pischinger[b]

[a]*CMT-Motores Térmicos*
Universitat Politècnica de València
Camino de Vera, s/n. 46022 Valencia, SPAIN
[b]*Institute for Combustion Engines VKA*
RWTH Aachen University
Forckenbeckstr. 4, Aachen, Germany

Abstract

CAI (Controlled AutoIgnition) systems, also named HCCI (Homogeneous Charge Compression Ignition), are a promising way to improve gasoline engines. This combustion mode is more efficient than the standard SI (Spark Ignition) combustion and, additionally, it has very low emissions, especially NOx emissions, which represent a source of problems nowadays. The main problem of this combustion mode is the constrained operating range, caused, on the one hand, by the difficulty to ignite the fuel since it has to be auto-ignited by the control of the mixture reactivity, and, on the other hand, by its high heat release rates, causing high pressure gradients and, in some circumstances, knocking combustion. In this paper, the possibility to use directly injected water into the combustion chamber as a reactivity suppressor in order to extend the constrained load range of CAI operation is evaluated. For

*Corresponding author
Tel: +34 963 876 548. Fax: +34 963 877 659. E-mail: *jorvamar@mot.upv.es*

this study, a four-stroke single-cylinder gasoline engine has been modified to allow CAI combustion by means of adapted valve trains enabling to keep hot residual gases inside the cylinder, which will provoke the fuel autoignition. Additionally, a water direct injection system has been installed in the engine to carry out this study. The results show that water injection is an efficient strategy to increase the maximum affordable load in CAI conditions, since the reactivity of the mixture can be suitably controlled, thus reducing the pressure gradients and the knocking tendency of the combustion process, also keeping good levels of combustion stability. Nevertheless, the engine has to be significantly boosted and the necessary intake pressure compared to a conventional SI operation mode in stoichiometric conditions is much higher.
Keywords: Controlled autoignition, CAI, HCCI, water injection, operating range

1. Introduction

Nowadays, there is an increasing worldwide interest in renewable and clean fuels, as well as in new combustion technologies. This interest is motivated by the limited energetic resources available, and by the challenges arising from the use of the fossil energy, such as the environmental pollution and the increasing levels of CO2 in the atmosphere. All these problems have motivated the increase of the pollutant restrictions and, as a consequence, the need to reduce the vehicle emissions, encouraging the attempts to study different engine configurations, as well as new combustion strategies. Controlled Autoignition (CAI) combustion, also known as Homogeneous Charge Compression Ignition (HCCI), despite not being a new concept, has recently re-emerged as a viable alternative combustion process to the conventional spark ignition (SI) or compression ignition (CI) processes for internal combustion engines. Currently, this technology attracts a lot of interest due to its potential for high efficiency and extremely low NOx and particulate emissions[1]. This combustion process is dominated by the chemical kinetics [2, 3] of the charge through the control of temperature, pressure and fuel-air mixture composition, achieving by this way the self-ignition of the mixture instead of using the traditional spark ignition on gasoline engines [4]. The first works performed in this field were carried out by Onishi et al. [5] and by Noguchi et al. [6], in the late 70s, with a two stroke engine where this peculiar operation of the engine was named "Active Thermo-Atmosphere Combustion (ATAC)". A significant number of studies have been done since then and, in the following lines, the main findings about the characteristics of the CAI combustion mode shall be summarized. Compared to SI, CAI combustion is a more stable process, with better combustion repeatability and fewer misfires. To illustrate this effect, two equivalent operation points at low load conditions (2000 rpm and 3 bar IMEP) operated as SI and CAI combustion modes are shown in Figure 1. The figure to the left corresponds to a SI combustion process, whereas the one to the right corresponds to a CAI combustion process.

The IMEP deviation is much lower in CAI conditions, and consequently significant improvements in the combustion process appear, leading to lower fuel consumption and to lower pollutant emissions compared to SI conditions [7]. However, these benefits come together with a number of difficulties. On the one hand, the pressure gradients generated can become very high at high

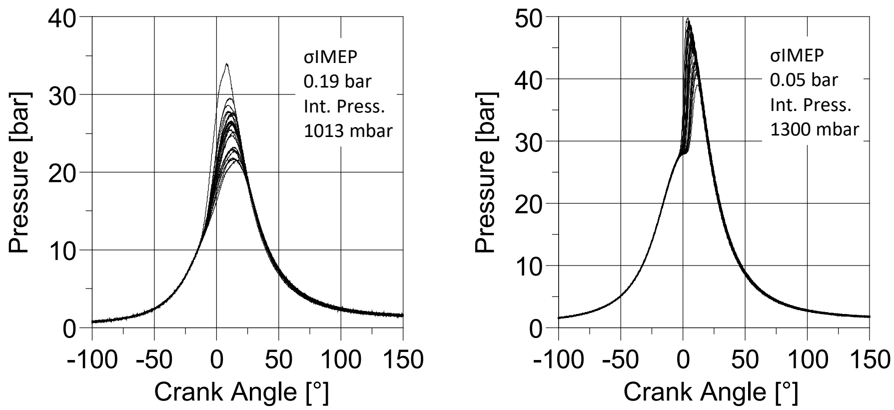

Figure 1: Instantaneous pressure measurements in SI model (left) and CAI mode (right).

loads due to an exceptionally high heat release rate, becoming harmful for the mechanical integrity of the engine. On the other hand, the autoignition conditions are very sensitive to small changes in the engine operating conditions. The combustion positioning is not as easy as in SI conditions, and the combustion process can start either too early, when the reactivity is very high, or it can be even extinguished if the reactivity is not high enough, thus limiting the CAI operating range. The GCAI (Gasoline-CAI) combustion process is currently under extensive research to overcome these limitations. Several works develop different strategies to achieve CAI operation on 4-stroke engines, such as: intake air heating, higher compression ratios, residual gas trapping and exhaust gas rebreathing [8–11]. These strategies have been proved most effective in achieving CAI combustion, and the corresponding works have demonstrated their potential to be incorporated in production gasoline engines. Despite these strategies, CAI combustion is still constrained at part load operating conditions because of the high heat release rates and the raised knocking tendency when the engine load is increased. In order to extend the load range of CAI combustion for automotive applications, different strategies have been investigated. Some of these strategies are well known, since they are commonly used in other combustion systems, like EGR [12, 13], whereas some others are more specific or unique for CAI, as modifying valve lift/timing configurations, fuel injection strategies or even adjusting the fuel composition [14–16]. Finally, there is an interesting strategy based on the introduction of reaction suppressors inside the combustion chamber to delay the autoignition process or slow down the excessively high combustion speeds. There are some works available in the literature dealing with these methodologies [17–21], where some low reactivity fuels or even non-reactive fluids, like water, are introduced via the fuel injection system as a blending, or with an independent injection system. The use of water in CAI engines has been experimentally investigated, since the cooling effect caused by its high latent heat of vaporization can be used to delay the ignition timing, and it is interesting to analyze whether it is possible to control the combustion phasing and slow down the combustion speed by these means [22, 23]. In the present paper, a direct (in-cylinder) and independent water injection has been installed in the engine used for the study, with the aim of reducing the local temperatures, which are assumed to be the cause for the excessive pressure gradients and the knock in the context of CAI combustion. The main objective of this research has been to explore the new operating range in CAI mode in a four stroke gasoline engine, analyzing how it could be enlarged towards higher loads thanks to the water cooling effect.

More precisely, the main studies carried out were the following:

- Analysis of the effect of water injection over the combustion process.
- Use of water addition to increase the maximum affordable load.
- Study of the different strategies, mainly water injection strategy and valve lift timing, to enlarge the CAI operating limits.

Throughout the paper the information in the different sections will be structured as follows:

- In the Experimental Facilities and Methodology section, the engine configuration selected and its peculiarities will be presented, as well as the boundaries of the test plan and the testing methodology followed.
- The Results and Discussion section is organized in four parts:
  - Potential of water as reactivity suppressor.
  - Study about the combustion controlling parameters at these new operating conditions.
  - Increase to the engine maximum affordable load.
  - Comparison: CAI vs SI. Benefits and main concerns.
- Finally, in the Conclusions section, a synthesis of the obtained results will be made to show the fundamental achievements reached during the ongoing project.

2. Experimental facilities and methodologies

In this section the necessary details to understand the results are given by detailing the different specifications of the installation, the test procedures and the results analysis.

2.1. Experimental Facilities

The project has been developed on an experimental basis, starting from a four-stroke gasoline single-cylinder research engine built to operate as a conventional SI engine. This engine has been adapted for the special requirements of this study by changing the valve train configuration and adding a direct injection system for the water addition. The main specifications of the engine and installation are given in Table 1.

Displacement/ Bore/ Stroke:	400cc / 75mm / 90.5mm
Compression ratio:	14.7 [-]
Fuel metering:	Direct injection system with solenoid injector
Water metering:	Direct injection system with solenoid injector
Intake:	Boosted intake with a roots external compressor and a heating system
Exhaust:	The exhaust pressure is regulated with a back-pressure electro-pneumatic valve
Fuel:	Commercial 95 RON gasoline with 10% of ethanol in volume

Table 1: Main specifications of the installation

The cylinder head has been designed to allocate the spark plug and the necessary two injectors. These two injectors are installed as shown in Figure 2, one for the fuel (green) and the other for the water (red). These injectors allow the direct injection of fuel and water, respectively, and they must be able to introduce both fluids at any crank angle position, giving

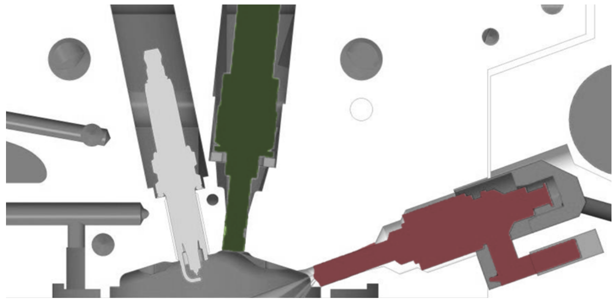

Figure 2: Cylinder head configuration.

enough flexibility to configure the injection timings as required. The spark plug installed is necessary to start the engine, to run it when it is cold and also to ignite the mixture when the engine is being operated out of the CAI operating range.

Valve train, intake and exhaust systems

The valve train had to be modified to get the CAI combustion by keeping a variable amount of hot residual gases inside the cylinder . These gases will be used to modify the reactivity of the mixture, allowing the control of its autoignition [24]. To achieve this, the camshafts have a reduced valve lift (Figure 3) compared to those used in the standard SI operation, and two variable valve timing systems (VVT), one for each camshaft [25]. When the engine is operating in CAI combustion mode, the configuration of the valves timing is different to the one used in standard SI engines. In a conventional engine, the EVO (Exhaust Valve Opening) is placed earlier than BDC (Bottom Dead Center, located at 180^o in the figure), and the EVC (Exhaust Valve Closing) takes place just after TDC (Top Dead Center, located at 360^o in the figure). Consequently the exhaust valve remains opened during the whole exhaust stroke. The IVO (Intake Valve Opening) is placed just before TDC (360^o in the figure) and the IVC (Intake Valve Closing) is located after the next BDC (540^o in the figure), and thus this valve remains opened during the whole intake stroke. In the GCAI engine, the camshaft positions are moved, and the two extreme angles (EVC and IVO) are modified. The EVC takes place well before TDC and the IVO well after TDC, thus creating an intermediate period at TDC during the end of the exhaust stroke and the beginning of the intake stroke where the intake and the exhaust valves remain closed. This action allows to hold a significant amount of hot residual gases inside the cylinder, which are recompressed during the period when the valves are closed and mixed with the fresh air when the intake valve is opened. The selected way to control the camshafts position is to control the distance between the EVC and the IVO (EVC-IVO), getting a parameter named Negative Valve Overlap (NVO). However, an increase in the absolute value of NVO would lead to bigger overlap periods (bigger negative values). And, on the contrary, a decrease in the absolute value of NVO would imply smaller overlap period (smaller negative values). It is worthy to underline that the two camshafts are moved symmetrically from TDC (i.e. if the exhaust is advanced 10^o, the intake is delayed 10^o).

Since the engine is going to be operated with high lambdas (high A/F equivalence ratios) and, because of the different valve timing strategies, the engine is less permeable, it will be necessary to increase the intake pressure to be able to get enough air in such a way that the initial objective of increasing the maximum affordable load with CAI conditions could be achieved [26]. The intake and the exhaust pressures are going to be set to the same value in order to simulate an eventual implementation of a turbocharger as the

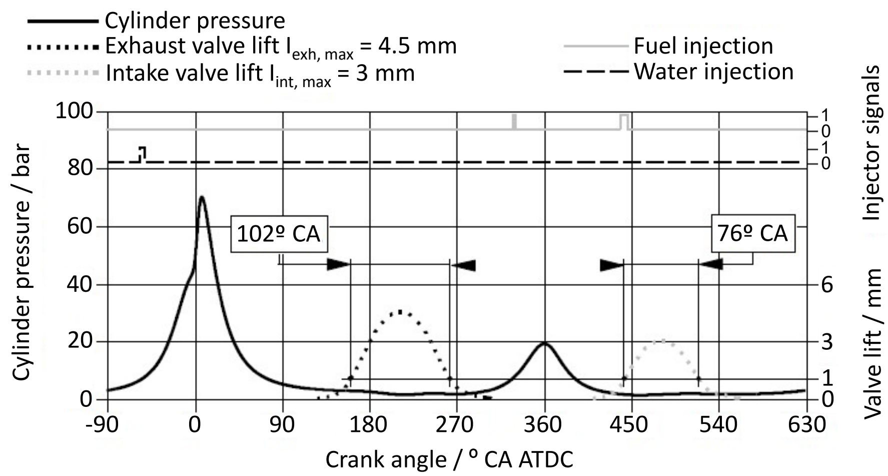

Figure 3: GCAI valve timing and injection strategy.

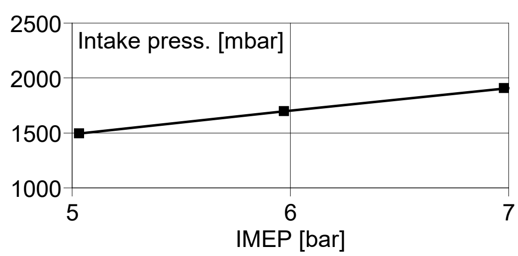

Figure 4: Intake pressure as a function of engine load.

boosting element. The intake pressures have been optimized for each load, and a nearly linear increase with the IMEP was found, as illustrated in Figure 4.

Fuel injection system

For the fuel metering, as already said before, there is a direct injection system with a solenoid injector and an externally driven gasoline pump. The fuel injection strategy is predefined and fixed for all the tests to avoid the influence on the data analysis of different fuel injection strategies. The fuel pressure selected is 150 bar, and the injection is split in two parts: the first part is injected during the recompression of the exhaust gases, and represents a small fraction of the fuel introduced, and the second (and main) injection is injected early during the intake stroke in order to premix the fuel as much as possible (Figure 3, top). This strategy has been selected to make the operation in CAI conditions easier, since the first injection is made to ensure a high reactivity of the charge [27]. During the tests, different loads are tested, defined by a given IMEP value. This is done keeping the first injection constant and the duration and start of the second fuel injection are modified to adjust the load and optimize the combustion, respectively.

Water injection system

The installed water injection system is a direct system with a solenoid injector and an externally driven water pump. This system uses distilled water as operating fluid, and it is capable of making multiple injections per cycle if required. The injected water mass has been determined from the measurement of the water mass flow rate in a gravimetric balance. In the results that will be shown later, this parameter will be presented together with the fuel mass as the 'water/fuel ratio', i.e. as a percentage of the total fuel injected in each operating point. The water injection pressure was fixed based on previous tests to define which the best compromise was. The results of those tests are presented in Figure 5, and they show a significant improvement of the engine behavior when the water injection pressure changes from 50 to 100 bar: the maximum pressure rise rate (dPmax) and the knock are reduced strongly, the efficiency is improved, and the stability remains inside the limits (all these parameters are going to be explained in the next sub-

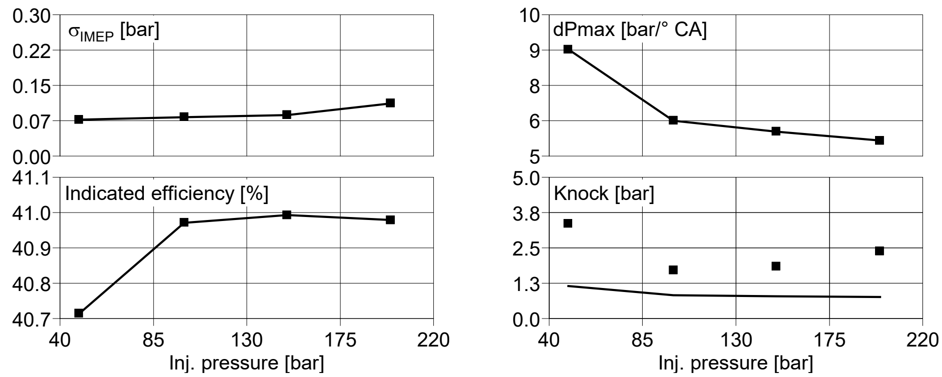

Figure 5: Effect of water injection pressure on the engine behavior.

section). A further increase in water injection pressure has not a significant benefit, as also shown in the figure. Taking in account these results, the water pressure has been fixed at 100 bar, since this pressure is high enough to allow the water introduction inside the combustion chamber at any crank angle position without any restriction. Besides, higher pressures give problems to inject small amounts of water.

The graph layout shown in Figure 5 will be used extensively to present the main results of the present paper. The peculiarities of this layout will be described in the following lines. All the graphs are structured in a set of preselected plots to show all the useful information in a compact way. The abcisas axes are the same for all plots, and the corresponding title is indicated only in the bottom row, whereas the ordinate axes are specific for each plot, and their title is located inside the corresponding plot. Finally it has to be pointed out that the plots are built with symbols, which correspond to measured data, linked with a line, to better illustrate the trends. On the plot showing the knock results there is a peculiarity: the points and the line are separated. This is because the line represents the *average* knock parameter (MAPO -Maximum Amplitude of Pressure Oscillations- in this case) for all the measured cycles, whereas the symbols represent the *maximum* MAPO among all measured cycles.

2.2. Methodoloy

In this section the measurement procedures, the most important variables and the fundamental points to understand the complete paper will be summarized and clarified.

This work has been performed at a constant engine speed of 1500 rpm. Secondly, as already known, in CAI conditions the intake temperature is an important parameter to modify the mixture reactivity. In all the tests performed, the intake temperature has been fixed at 50°C, since the purpose of this study is not to analyze the effect of this parameter. Finally there are three operating limits defined to be controlled during the tests:

- Maximum pressure rise rate (dPmax): this parameter has to be controlled online, since in CAI conditions the combustion is very fast, and the pressure gradients can be too high so as to damage the engine if the combustion is not controlled. The prefixed limit for this parameter in this study is 7 bar/CA. Beyond this limit the point is considered dangerous and not allowable.

- Combustion stability (referred as σ_{IMEP}): the combustion stability is evaluated with the standard deviation of the IMEP (Indicated Mean Effective Pressure), which is also monitored online. The defined limit to consider a stable CAI combustion in the present research is 0.15 bar.

- The combustion knock: this value is obtained from the instantaneous variations of the filtered pressure signal (band-pass filter between 5 to 20 kHz) taking the maximum value of each cycle (MAPO, Maximum Amplitude of Pressure Oscillations) [28, 29]. A knock of 3 bar is considered light knock (i.e. it is allowable to have a few cycles with this knock value) and when the value reaches 6 bar, then it is considered hard knock and not allowable [30].

Each measured point has got two different sets of measurements: instantaneous and mean measurements. For the instantaneous ones, 150 consecutive cycles are recorded, whereas for the mean ones, the time taken for averaging is 30 seconds.

3. Results and discussion

In this section, the obtained results of the performed work will be shown and discussed. These results are structured following the order of the main tasks carried out during the study.

The start of the investigation was to test the CAI conditions without adding any water, in order to know which the starting point of the investigation was. Once this limit was defined, an analysis of the water effect was made evaluating its influence on the combustion process compared to the points without water. After that, the different variables to adjust the new operating points were analyzed in order to understand the effect of each variable over the combustion control. After all this work, the results of the maximum affordable load will be shown together with the strategies followed to expand the CAI operating limits. And finally, as a closing point of this study, all the results obtained under CAI mode will be compared to equivalent load operating points but under spark ignition mode, trying to find out the utility and/or the improvement of CAI results against typical SI results in the same engine.

3.1. Potential of water as a reactivity suppressor

As a starting point of the study, the maximum affordable engine load without water addition has been tested, and the results are shown in Figure 6 (black data). As the engine load is increased, the reactivity of the mixture (intake gasses, fuel and hot residual gasses) is higher, since there is more pressure and higher temperatures at the end of the compression stroke. Thus, to allow further load increasing the mixture temperatures must be decreased with the aim of mitigating the raise of knock and the rate of pressure rise as much as possible. With this purpose the NVO is reduced, affecting the initial temperature of the mixture, which is lowered since the hot residual gases retained inside the cylinder are reduced. This lower initial temperature translates into a reduced mixture reactivity, which approaches its autoignition limit, and thus the combustion process starts to be erratic and the stability drops quickly.

Based on the results shown in the figure, the maximum affordable load without water is 3.5 bar of IMEP.

For the same engine configuration, a water injection has been performed in order to see the effect of water on the operating points that are close to the load limit. In Figure 6 (blue data) results are shown for the same operating points than before, but with an amount of water equivalent to a 20% of the fuel introduced. The results show significant differences in terms of

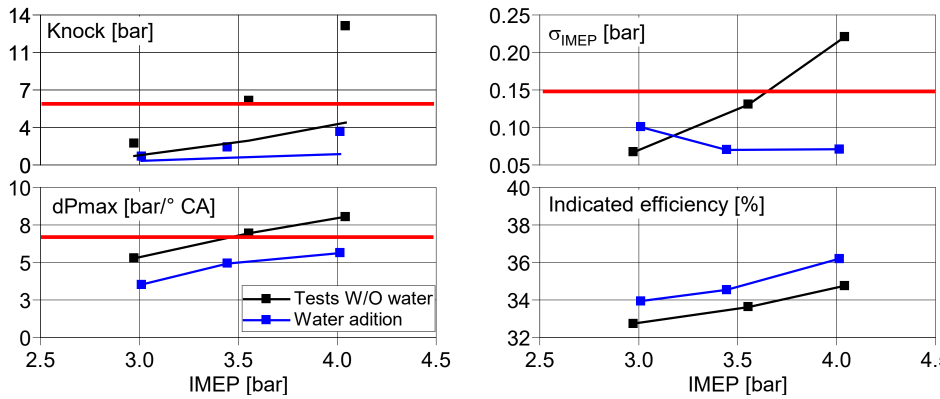

Figure 6: Maximum load without water injection (black), and with water injection (blue).

engine behavior between the cases with and without water: the pressure rates and the knock are strongly reduced, as expected, making now these points feasible, and the stability of the points is kept constant on acceptable values due to the higher amount of residual gases inside the cylinder allowed by the water addition. Regarding the indicated efficiency, two observations can be made from these results. On the one hand, as already known, at relatively low engine loads, the efficiency increases with an increase in engine load. And, on the other hand, the efficiency also improves with the addition of water. This latter effect can be explained by analyzing the combustion process. To start with, at the operating conditions without water, the combustion position is located very early due to the excessive reactivity of the mixture (see CA50, i.e. the crank angle where 50% of the heat has already been released, in Figure 7), which comes together with a very fast combustion process. This fast process is due to the different combustion nature compared to standard SI combustion: in CAI conditions, this one is governed by chemical kinetics and combustion starts in many different points, whereas in SI it starts in

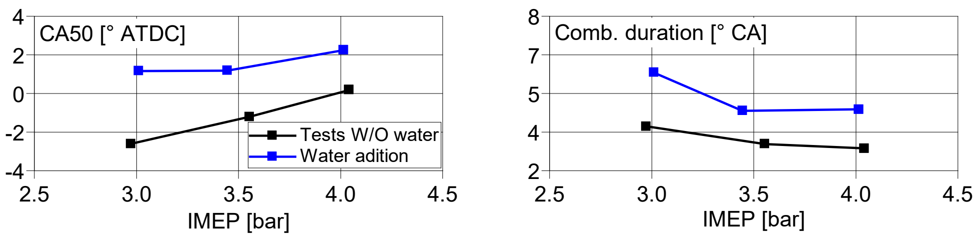

Figure 7: Effect of water addition on the combustion phasing and duration.

a single point, and depends on the flame front evolution. With the water introduction, the mixture reactivity is decreased due to the temperature reduction provoked by the water evaporation. This effect delays the start of combustion and slows down the combustion process (its duration, computed as CA90-CA10, increases), allowing a better phasing of the combustion that justifies the improvement of the indicated efficiency observed in the figure.

It is important to remark that the effect discussed above about the increased indicated efficiency depends on the operating point under analysis, and it can't be extrapolated to all the different cases. The water, by itself, is not the direct responsible of the observed improvement: it is a consequence of the combustion performance, which is altered by the water addition, moving the combustion phasing to a better position. But if the water amount is further increased, the point might be worsened because the combustion process would be modified again and moved to a worse location.

As a summary, the effects of water when added to the combustion chamber are:

- Reduction of the pressure rise rate.

- Mitigation of knock.

- Delay of the combustion phasing.
- Reduction of the combustion speed.

It can be concluded that water addition reduces the reactivity of the mixture, and gives an extra variable to control the combustion process. Its effect allows to continue increasing the maximum affordable engine load in CAI combustion mode. In the forthcoming sections, the new scenario allowed by the water addition will be studied, with the aim of trying to expand the maximum affordable load under CAI combustion.

3.2. Study about the combustion controlling parameters in the new scenario

The main control parameters over the combustion process in the frame of this study, taking into account that the intake pressure has been prefixed, are the valve train configuration, which is used to adjust the in-cylinder residual hot gases, the amount of water and the time when the water is injected.

In this section the results shown have been taken at an IMEP of 5, 6 and 7 bar. It is also important to underline that these loads were not achievable when operating without water, and consequently all these points belong to the increased operating CAI range.

Amount of water

This is, perhaps, the central parameter regarding this paper. The water amount affects the mixture reactivity by decreasing the in-cylinder temperature when it is evaporated.

During these tests the rest of parameters have been kept constant, except the amount of fuel, which needs to be continuously adapted to keep constant the IMEP. The different effects of the increase of the water amount can

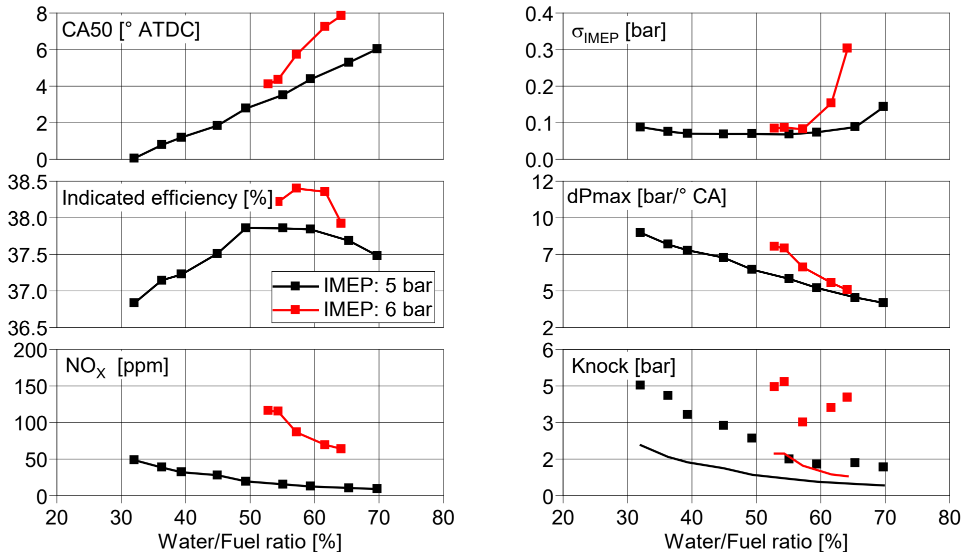

Figure 8: Effect of water amount variation.

be observed clearly in the Figure 8. As the amount of water is increased, the combustion is retarded and its duration increases due to the decrease in reactivity caused by the temperature reduction. Besides, the pressure gradients and the knock are reduced accordingly, but the σ_{IMEP} increases since there is too much water worsening the mixture autoignition.

Regarding the averaged knock, it always decreases. However, as observed for the 6 bar IMEP case, with the rise of the σ_{IMEP}, the residuals would contain unburned fuel from the previous cycle and the peak to peak knock starts to increase also with the increase of the water amount.

The influence of the water amount on the efficiency strongly depends on the combustion phasing at each operating point. To understand the results, there are two considerations to take into account: on the one hand, the combustion phasing (CA50) increases as the water amount increases (in this case getting closer to the best efficiency location); but, on the other hand, the stability of the point is critical for the efficiency, and if the σ_{IMEP} rises, the indicated efficiency will drop quickly.

Finally, the NOx emissions are included, since these emissions are a hot topic and one of the main advantages of this peculiar combustion mode. As it can be seen, the emissions level is decreased with the addition of water with the decrease of the in-cylinder temperature.

Start Of Injection

The timing of the water injection is also important to ensure the correct homogenization of the water and to control when the reactivity of the mixture is modified. Figure 9 shows the effect of SOI (Start Of Injection) variations at two operating points, one at 5 bar IMEP with a water/fuel ratio of 45% and the other at 6 bar IMEP with a water/fuel ratio of 60%.

As it can be seen in the plots, a delayed injection timing generates the need to introduce a higher amount of water compared to the optimum timing since the water homogenization has not taken place completely and the decrease of the temperatures inside the combustion chamber takes place too late. This is seen on the plots because the pressure rise and the knock levels are increased, and the combustion phasing advanced. These effects could be compensated by introducing a higher amount of water, thus forcing the water evaporation, but this would unnecessarily increase the water consumption.

On the other hand, the water injection can be over-advanced. It can be seen that there is a significant difference between the results at 5 and 6 bar IMEP. First, the 5 bar case is going to be analyzed. If it is considered that the amount of water is correctly adjusted, it can be seen that the effect is

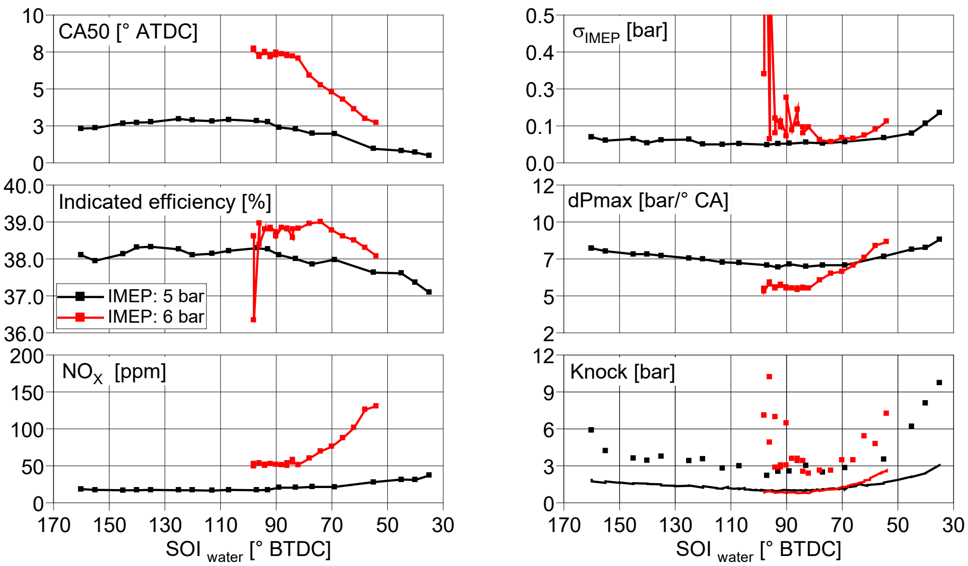

Figure 9: Effect of SOI variation.

not as inconvenient as when the injection is delayed. However, over-advanced injections provoke some undesirable effects like an increase of the pressure rise rate and the knock, as well as an earlier start of combustion. These effects imply that the mixture reactivity is increasing again because the cooling effect is not well performed, probably due to the water impingement on the cylinder walls.

Now, if the 6 bar IMEP case is considered, as the injection advance is increased, the engine behavior worsens due to a fast increase of the engine instability, the combustion takes place later in the engine cycle and the indicated efficiency drops. This fact, together with the significantly more delayed CA50 (compared to the 5 bar IMEP case), means that the water amount of the 6 bar IMEP case was too high, and it can hinder the start of the autoignition. It is also worthy to note that the limits of pressure rise rate and knock have to be respected during the tests, and at certain operating points it won't be possible to reduce the water amount, even readjusting the rest of parameters, since it will be necessary to decrease the combustion speed during the cycle.

Regarding the trend of the NOx emissions, it is consistent with the CA50 trend: the increase is due to an earlier combustion. This one starts progressively earlier and earlier with the delay of the water injection because the reactivity of the mixture is too high. At these conditions, the heat release rate becomes very high, reaching high pressures and temperatures, and in these conditions, the NOx emissions quickly rise. In this particular situation, lower NOx come together with higher efficiency, since the combustion was too advanced and it has been delayed with the water injection.

Adjustment of the Negative Valve Overlap

The Negative Valve Overlap (NVO), as explained in previous sections, allows to modify the mixture reactivity by adjusting the amount of hot residual gases retained inside the cylinder, allowing the correct conditions for the CAI combustion [31]. This parameter is closely related to the water addition that has been just analyzed. Prior to any analysis, it is worthy to remark some aspects about the values of the parameter under analysis (NVO).

Figure 10 shows three tests, at 5, 6 and 7 bar IMEP. All these three tests show the same general trends with the variation of the NVO. Higher NVOs increase the reactivity of the mixture by means of a higher amount of residual hot gases and lower lambdas inside the cylinder (because the intake pressure is fixed and a part of the fresh air is replaced by residual hot gasses). In the opposite way, by reducing the NVO, the amount of residual gases is lower, thus reducing the initial in-cylinder temperature and giving an extra input of fresh air, both effects reducing the reactivity of the mixture. The NVO variations are limited, on the one hand, by the drop of the stability if the NVO is decreased and, and on the other hand, by the raise of the pressure rise rates and knock if the NVO is increased. Taking a look into the combustion performance and the indicated efficiency, the combustion is advanced with an increase in the NVO, as expected, due to the higher mixture reactivity. However, as the NVO is reduced, the retained hot residual gases are being reduced and, beyond a certain point, the combustion starts to be delayed (CA50 increases). Consequently, the autoignition of the mixture is delayed and starts to be erratic, thus increasing the engine instability, as can be seen on the σ_{IMEP} plot (5 bar IMEP case in Figure 10).

Regarding the indicated efficiency, it decreases for two reasons: one reason is the decrease of the lambda when the NVO is higher, and the other reason is the combustion position, which has been extremely advanced with the increase of the hot burnt gases inside the cylinder.

Finally, the NOx emissions, as expected, follow the same trend as the amount of residuals inside the cylinder, since the in-cylinder temperature increases with this parameter.

In a similar way as with the water amount, through the hot residual gases rate control, the in-cylinder reactivity can be further optimized for CAI operation. Usually, the best choice is to set the NVO to the lowest as possible (without exceeding the σ_{IMEP} limit), since higher lambdas allow better combustion and indicated efficiencies, optimizing later the mixture reactivity with the amount of water.

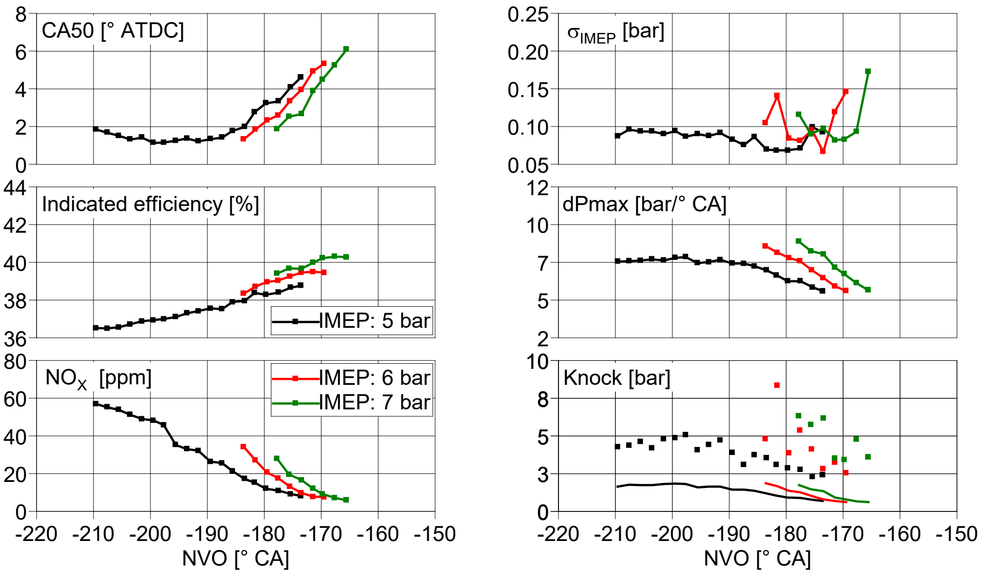

Figure 10: Effect of NVO variation.

It has to be noted that the NVO parameter needs to be selected so as to keep enough hot residual gases to ensure the autoignition process. With this parameter alone it is not possible to hold a stable autoignition process and, at the same time, avoid knocking problems and excessive pressure rise rates when the load is increased. Consequently, the water addition remains necessary to keep the combustion inside the defined limits.

Now a brief synthesis of this section is presented, summarizing the main effects of each control parameter.

- Water amount:
  - Decreases the reactivity of the mixture, reduces the dPmax, knock and the combustion speed, but the stability of the combustion is worsened.

  - Allows the increase of the engine load.

- SOI:
  - There is a minimum advance of the SOI necessary to allow the complete evaporation of the water.
  - The importance of the SOI adjustment increases with higher water/fuel ratios and higher loads, since the amount of water introduced in the cylinder is higher.

- NVO:
  - Modifies the amount of hot burnt gases inside the cylinder, affecting the reactivity of the mixture.
  - As the intake pressure is fixed, the lambda is also affected.
  - It affects the mixture reactivity, like the water does. But NVO has to be adjusted to ensure a stable combustion, and the amount of water has to be adjusted to keep the combustion inside the operating limits.

Finally, once the tests at the different loads have been performed and analyzed, it is find out that, as the engine load is increased, the range available to adjust the points is narrower, the instabilities appear sooner and the knock and the rate of pressure rise are more sensitive, making progressively more difficult to operate the points at higher loads. This means that there will be a load limit, beyond which the operation wouldn't be possible with this engine configuration.

3.3. Increase of the maximum affordable load

Once the effect of the main adjustments has been defined, the next step of this study is to determine which the maximum load achievable with these new strategies is.

To get this result, a set of tests at different loads has been arranged. With all this information, the best efficiency points and the relationship between them has been analyzed, in order to define a logical operating strategy to be able to increase the engine load limit.

Finally, the selected points shown in Figure 11 are the best results obtained among all the tests performed at each load, keeping the dPmax, knock and stability inside the limits.

Based on the information shown in Figure 11, the evolutions of the final configurations for the water, valve train position, lambda and intake pressure can be analyzed to define the most suitable strategy to follow if the engine load is increased:

- The water/fuel ratio is not constant: it needs to be increased progressively with the engine load.
- The SOI of the water injection needs to be increased faster than linearly, because the actual amount of water also increases faster than linearly, since it increases with the water/fuel ratio (which increases linearly) and with the engine load (the fuel amount also increases).
- The results show a lambda of 1.8 or 1.9 for the optimum configurations.
- The intake pressure has to be increased proportionally to the engine load.

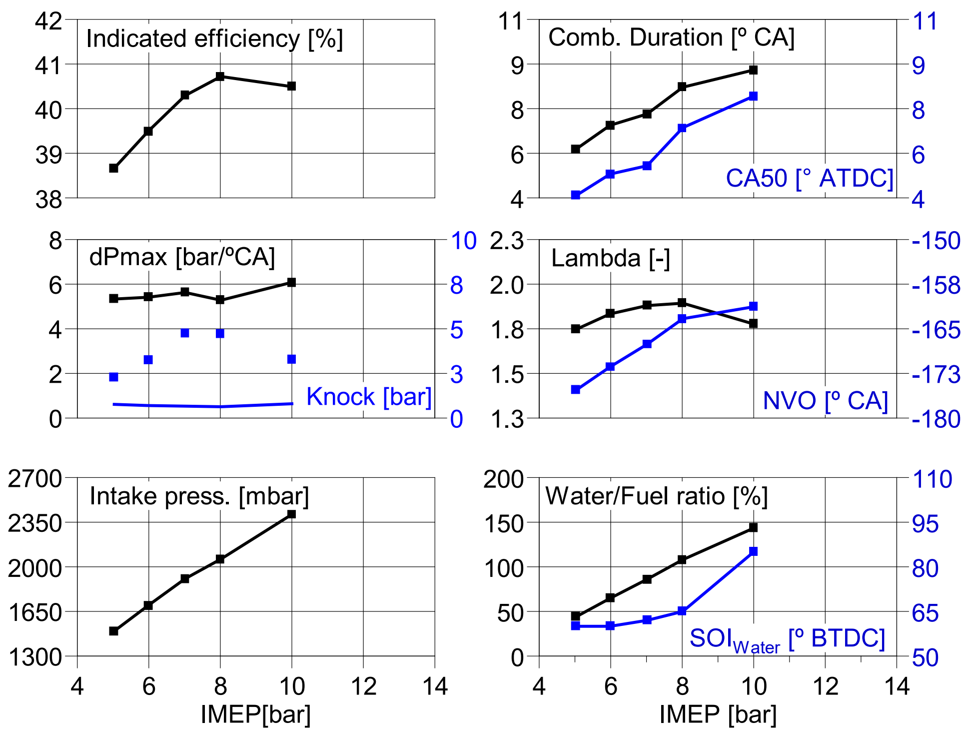

Figure 11: Best points at different engine loads.

- The NVO is reduced progressively in order to decrease the reactivity of the mixture.

With all these actions, both the dPmax and the knock are under control in all the tested range, and do not represent a major problem. The stability has not been shown because there is nothing relevant in this parameter: all the measured points were inside the limits.

The combustion phasing, as can be seen, is delayed progressively because of two reasons: firstly, because there is more fuel to burn; and, secondly, because the problems with knock and dPmax are higher as far as the engine load is increased. For this reason, for the initial loads, the combustion location is the optimum within the cycle, but for higher loads, it has to be delayed to be able to control the knock and the dPmax.

Regarding the indicated efficiency, it increases at the first part of the plot, as expected, since increasing the engine load comes together with an increase in engine efficiency provided that the combustion location is the optimum one. However, this trend is broken beyond a given engine load, because at higher loads it was impossible to place the combustion process on its optimum location, and it had to be delayed to be able to operate the point under CAI combustion mode.

Finally, at 10 bar IMEP, the tests were stopped since the adjustment range of the point was extremely narrow, and it was very difficult to achieve a correct engine operation. The load of 11 bar IMEP, for instance, although from the results shown in Figure 11 it might seem that could be achieved, it was observed that it was impossible to be reached without exceeding the acceptable limits of dPmax and knock.

Comparison: CAI vs SI. Benefits and main problems

All the results obtained with the engine operated in CAI conditions presented before give raise to certain questions as, for instance, what the benefit of these results compared to a conventional SI combustion mode is. In this case, there is some data available at equivalent engine loads from the same engine operated in spark ignition mode, which would help to give an answer to this question.

Prior to any analysis, it is important to point out that this engine, when operated in spark ignition mode, has two main differences respect to the CAI configuration: the first one is the use of different camshafts, without reduced valve lift and with different valve timings, because in this scenario it is not interesting to keep any residual gases inside the cylinder unlike for the CAI configuration. And, secondly, the engine is not boosted, since it is operated in stoichiometric conditions and there is no need of extra pressure in the intake to achieve these loads. These changes are necessary because it is impossible to get CAI combustion and achieve these loads with the same exact engine configuration.

To analyze the differences between these two different combustion modes, the results presented in Figure 12 will be discussed. At first sight, the most interesting differences are the indicated efficiency, the NOx emissions and the unburned hydrocarbons emissions. The indicated efficiency is improved in around 3 to 3.5 points in percentage, the NOx emissions are nearly suppressed and the hydrocarbons are halved. Concerning the indicated efficiency, there are some considerations to take into account when both engine configurations, or both operating modes, are compared: at the CAI configuration, the engine is boosted, and the back pressure is regulated assuming that the theoretical turbocharger is able to have the same pressure at the compressor outlet and at the turbine inlet.

The combustion process is also very different in CAI conditions. Its duration is much shorter, since the combustion starts spontaneously at different locations and grows on a different way (governed by chemical kinetics) than the traditional SI combustion, where there is a single starting point and a flame front propagation. The combustion phasing is also advanced, because the combustion is faster and placed as close to the optimum phasing as possible. But in some cases, the necessary minimum reactivity of the mixture to allow a stable CAI combustion forces an earlier than desired combustion phasing.

Regarding the engine control, it is by far more complex under CAI conditions, because there are more variables to control. Some variables are added (as, for instance, the water addition –mass and timing– and the control of the NVO) and, besides these new variables, the operation flexibility of the CAI combustion is smaller, since the control of the combustion process depends on the mixture reactivity. All these circumstances make necessary to control with more accuracy the mass flows (air, fuel and water) and temperatures to get a stable and efficient combustion process.

4. Conclusions

The aim of this publication has been to show the potential of the direct injection of water as a valid method to expand the maximum affordable load in an engine when operated under CAI conditions. The final results have

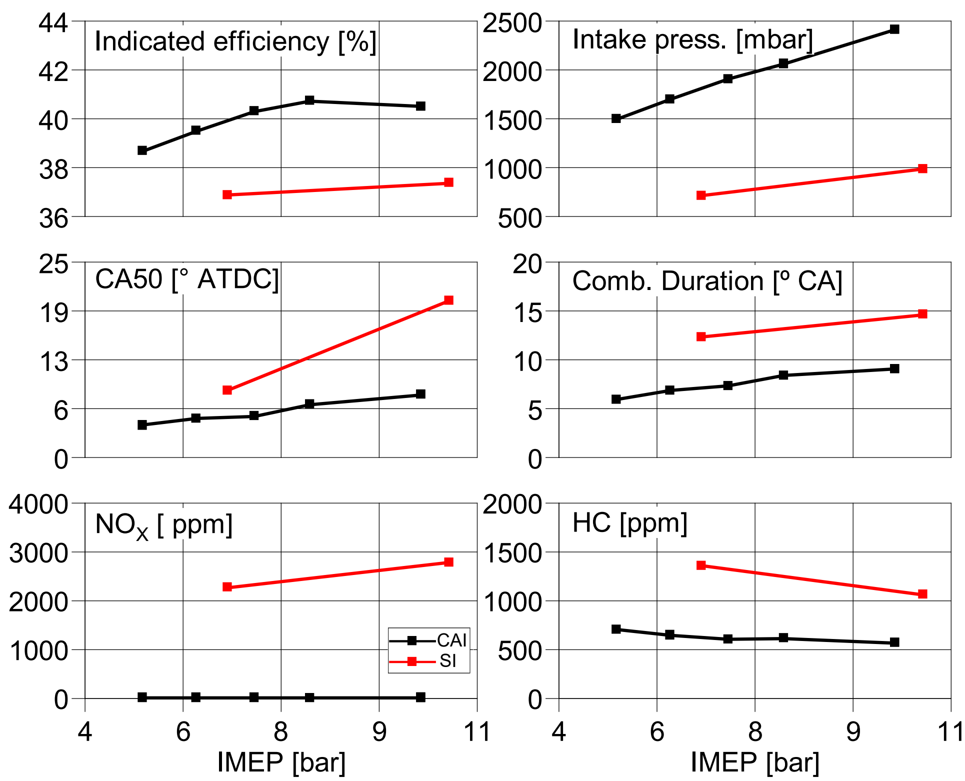

Figure 12: CAI vs SI.

shown an increase from an initial allowable load of 3.5 bar IMEP without water injection, to a final load of 10 bar IMEP with water addition, which implies an increase of almost three times in maximum affordable load.

The water has been shown to be an effective way to reduce and control the reactivity of the in-cylinder charge during CAI combustion operation. This reduces the pressure rise rates and the knock, makes lower the combustion speeds and delays the combustion phasing. But, nevertheless, it can also introduce a loss in combustion stability, since the reactivity of the charge is decreased.

As far as the engine load is increased the required amount of water also increases, reaching some non-negligible values (at the maximum load achieved in this study, for instance, the required water mass flow is 1.5 times higher than the fuel mass flow). Moreover, with the increase of the engine load, the available range to adjust the points is narrower, and it is very difficult to find configurations with a stable combustion inside the limits of pressure rise and knock. In this paper a great deal of information about the general trends of the main engine outputs when employing different strategies are presented, which are also discussed and explained in order to allow transferring this knowledge for future studies and continuations of the work.

Finally the results compared to SI combustion show the associated benefits of the CAI combustion applied to higher loads, allowing much lower pollutant emissions (HC and, especially, NOx) and improved fuel efficiency. The main problems for the implementation of these type of systems are a higher complexity of the engine control, as well as the need of some additional parts to include in a traditional SI engine, such as a direct injection system to introduce water, and a system to boost the engine up to the intake pressures required to operate in CAI mode at the higher loads. At high loads, the intake pressure needs to be really high (∼2.4 bar at 10 bar IMEP). This could be, perhaps, one of the main limitations of the CAI mode compared to the SI mode.

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Notation

BDC	Bottom Dead Center
CA10	Crank Angle where 10% of the fuel mass has been burned
CA50	Crank Angle where 50% of the fuel mass has been burned
CA90	Crank Angle where 90% of the fuel mass has been burned
CA90-CA10	Duration of the combustion estimated as CA90-CA10
CAI	Controlled Autoignition
CI	Compression Ignition
dPmax	Maximum pressure rise rate
EGR	Exhaust Gases Recirculation
EVC	Exhaust Valve Closing
EVO	Exhaust Valve Opening
GCAI	Gasoline CAI
HCCI	Homogeneous Charge Compression Ignition
IMEP	Indicated Mean Effective Pressure
IVC	Intake Valve Closing
IVO	Intake Valve Opening
MAPO	Maximum Amplitude of Pressure Oscillations
NOx	Nitrous Oxides
NVO	Negative Valve Overlap
SI	Spark Ignition
$\sigma IMEP$	Standard deviation of the IMEP
SOI	Start Of Injection

TDC	Top Dead Center
VVT	Variable Valve Timing

References

[1] B. Albayrak Çeper, M. Yildiz, S. O. Akansu, and N. Kahraman. Performance and emission characteristics of an ic engine under si, si-cai and cai combustion modes. *Energy*, 136:72–79, 2017.

[2] H. J. Curran, W. J. Pitz, C. K. Westbrook, C. V. Callahan, and F. L. Dryer. Oxidation of automotive primary reference fuels at elevated pressures. *Symposium (International) on Combustion*, 27(1):379–387, 1998.

[3] S. Tanaka, F. Ayala, J. C. Keck, and J. B. Heywood. Two-stage ignition in hcci combustion and hcci control by fuels and additives. *Combustion and Flame*, 132(1-2):219–239, 2003.

[4] H. Zhao. *HCCI and CAI engines for the automotive industry.* HCCI and CAI Engines for the Automotive Industry. 2007.

[5] S. Onishi, S. H. Jo, K. Shoda, P. D. Jo, and S. Kato. Active thermo-atmosphere combustion (atac) - a new combustion process for internal combustion engines. *SAE Technical Papers*, 1979.

[6] M. Noguchi, Y. Tanaka, T. Tanaka, and Y. Takeuchi. A study on gasoline engine combustion by observation of intermediate reactive products during combustion. *SAE Technical Papers*, 1979.

[7] Y. Zhang and H. Zhao. Investigation of combustion, performance

and emission characteristics of 2-stroke and 4-stroke spark ignition and cai/hcci operations in a di gasoline. *Applied Energy*, 130:244–255, 2014.

[8] H. Zhao, Z. Peng, and N. Ladommatos. Understanding of controlled autoignition combustion in a four-stroke gasoline engine. *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*, 215(12):1297–1310, 2001.

[9] Y. Li, H. Zhao, N. Brouzos, and B. Leach. Managing controlled auto-ignition combustion by injection on a direct-injection gasoline engine. *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*, 221(9):1125–1137, 2007.

[10] A. Uyumaz. An experimental investigation into combustion and performance characteristics of an hcci gasoline engine fueled with n-heptane, isopropanol and n-butanol fuel blends at different inlet air temperatures. *Energy Conversion and Management*, 98:199–207, 2015.

[11] C. Cinar, A. Uyumaz, H. Solmaz, F. Sahin, S. Polat, and E. Yilmaz. Effects of intake air temperature on combustion, performance and emission characteristics of a hcci engine fueled with the blends of 20n-heptane and 80 *Fuel Processing Technology*, 130(C):275–281, 2015.

[12] K. Lee, S. Cho, N. Kim, and K. Min. A study on combustion control and operating range expansion of gasoline hcci. *Energy*, 91:1038–1048, 2015.

[13] J. Jang, Y. Lee, C. Cho, Y. Woo, and C. Bae. Improvement of dme hcci engine combustion by direct injection and egr. *Fuel*, 113:617–624, 2013.

[14] P. A. Caton, H. H. Song, N. B. Kaahaaina, and C. F. Edwards. Residual-effected homogeneous charge compression ignition with delayed intake-valve closing at elevated compression ratio. *International Journal of Engine Research*, 6(4):399–419, 2005.

[15] J. Yang. Expanding the operating range of homogeneous charge compression ignition-spark ignition dual-mode engines in the homogeneous charge compression ignition mode. *International Journal of Engine Research*, 6(4):279–288, 2005.

[16] K. Yoshizawa, A. Teraji, H. Miyakubo, K. Yamaguchi, and T. Urushihara. Study of high load operation limit expansion for gasoline compression ignition engines. *Journal of Engineering for Gas Turbines and Power*, 128(2):377–387, 2006.

[17] H. Ogawa, N. Miyamoto, N. Kaneko, and H. Ando. Combustion control and operating range expansion with direct injection of reaction suppressors in a premixed dme hcci engine. *SAE Technical Papers*, 2003.

[18] A. B. Dempsey, B. Das Adhikary, S. Viswanathan, and R. D. Reitz. Reactivity controlled compression ignition using premixed hydrated ethanol and direct injection diesel. *Journal of Engineering for Gas Turbines and Power*, 134(8), 2012.

[19] A. Megaritis, D. Yap, and M. L. Wyszynski. Effect of water blending on bioethanol hcci combustion with forced induction and residual gas trapping. *Energy*, 32(12):2396–2400, 2007.

[20] T. Steinhilber and T. Sattelmayer. The effect of water addition on hcci diesel combustion. *SAE Technical Papers*, 2006.

[21] N. Kaneko, H. Ando, H. Ogawa, and N. Miyamoto. Expansion of the operating range with in-cylinder water injection in a premixed charge compression ignition engine. *SAE Technical Papers*, 2002.

[22] Y. Iwashiro, T. Tsurushima, Y. Nishijima, Y. Asaumi, and Y. Aoyagi. Fuel consumption improvement and operation range expansion in hcci by direct water injection. *SAE Technical Papers*, 2002.

[23] M. Christensen and B. Johansson. Homogeneous charge compression ignition with water injection. *SAE Technical Papers*, 1999.

[24] B. Wolk, I. Ekoto, W. F. Northrop, K. Moshammer, and N. Hansen. Detailed speciation and reactivity characterization of fuel-specific in-cylinder reforming products and the associated impact on engine performance. *Fuel*, 185:341–361, 2016.

[25] R. Vallinayagam, A. S. Alramadan, S. Vedharaj, Y. An, J. Sim, J. Chang, and B. Johansson. Low load limit extension for gasoline compression ignition using negative valve overlap strategy. *SAE Technical Papers*, 2018-April, 2018.

[26] M. Canakci. Combustion characteristics of a di-hcci gasoline engine running at different boost pressures. *Fuel*, 96:546–555, 2012.

[27] M. Tongroon and H. Zhao. Thermal and chemical effects of fuel direct injection on kinetically controlled combustion of alcohol and gasoline fuels. *International Journal of Engine Research*, 16(8):982–993, 2015.

[28] X. Zhen, Y. Wang, S. Xu, Y. Zhu, C. Tao, T. Xu, and M. Song. The engine knock analysis - an overview. *Applied Energy*, 92:628–636, 2012.

[29] Z. Wang, H. Liu, and R. D. Reitz. Knocking combustion in spark-ignition engines. *Progress in Energy and Combustion Science*, 61:78–112, 2017.

[30] P. Haenel, P. Seyfried, H. Kleeberg, and D. Tomazic. Systematic approach to analyze and characterize pre-ignition events in turbocharged direct-injected gasoline engines. *SAE Technical Papers*, 2011.

[31] J. Hunicz. An experimental study of negative valve overlap injection effects and their impact on combustion in a gasoline hcci engine. *Fuel*, 117(PART A):236–250, 2014.