Analysis of the potential of a new automotive two-stroke gasoline engine able to operate in Spark Ignition and Controlled AutoIgnition combustion modes

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Abstract

The need to reduce the emissions coming from automobiles encourages the attempts to study different engine configurations and new combustion strategies. In this case, a two stroke engine able to operate in Controlled AutoIgnition (CAI) and Spark Ignition (SI) combustion modes is studied, with the purpose of getting lower NO_x and CO_2 emissions than with other currently employed solutions. The engine configuration retained for the research is a uniflow scavenging configuration with intake ports in the cylinder liner and exhaust valves in the cylinder head. These valves are controlled by a Variable Valve Timing (VVT) system. The scavenging is guaranteed by an external blower driven by the crankshaft. Finally, the fuel supply is performed by a direct injection (DI) air-assisted fuel injection system.

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Through this paper the adjusting parameters to control the engine operation, as well as their influence on the CAI and SI combustion modes have been studied, providing the most relevant information and knowledge for controlling and optimizing the engine performance. Once these controlling parameters were studied, an EGR system was introduced in order to analyze the effect of this other parameter over the combustion process, as well as to determine the potential benefits of introducing such a system in this type of engines.

Keywords: Two-stoke engine, CAI, HCCI, Spark ignition, New combustion modes, Air-assisted injection

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1. Introduction

Two stroke (2S) engines have always had their fashions, and these go on stage once in a while [1]. Traditionally, 2S engines had been cheaper than their competitors (four stroke -4S- engines), and also have had some advantages like less weight and more specific power, for example. But these engines also present some important troubles that hinders their development (scavenging difficulties, fresh air short-circuit, excessive thermal load on some components, etc.).

However, technological advances evolve and make now easy some tasks that were considered too complex a few years ago [2–4], giving room to solve the problems indicated before, which impede 2S engines development. Perhaps now it is a good time to reconsider the feasibility of this type of engines.

Additionally to this, nowadays engines have different ways to improve

their fuel efficiency [5]. A significantly promising way is CAI combustion (Controlled AutoIgnition), where the combustion process is achieved by self-ignition through the control of the temperature, pressure and composition of a premixed charge, instead of the traditional spark ignition for gasoline engines [6]. This is one of the most promising combustion technologies to reduce fuel consumption and NO_x emissions in gasoline engines. Currently, however, CAI combustion is constrained at part load operating conditions because of misfire at low load and knocking combustion at high load.

Some additional benefits of the CAI combustion process, compared to the traditional SI combustion process, is that it is more stable, thus getting better repeatability of the combustion cycles, together with fewer misfires. In Figure 1, an operating point at low load (2000 rpm with an IMEP - Indicated Mean Effective Pressure- of around 3 bar) for the engine studied in this work is presented in two graphs, where the equivalence ratio is varied to obtain different combustion modes: on the left a spark ignition combustion mode, whereas on the right a CAI mode. From the figure, it can be seen that the standard deviation of the IMEP (σIMEP) is much lower under CAI conditions, thus achieving significant improvements in the combustion process leading to lower fuel consumption. Besides, thanks to the different combustion process, lower pollutant emissions compared to a traditional SI combustion process are obtained [7].

Nevertheless, despite the attractive advantages underlined before, this combustion mode has also got some limitations. On the one hand, the in-cylinder pressure gradients tend to increase with the engine load, and CAI gradually becomes knocking, which is harmful for the mechanical integrity of

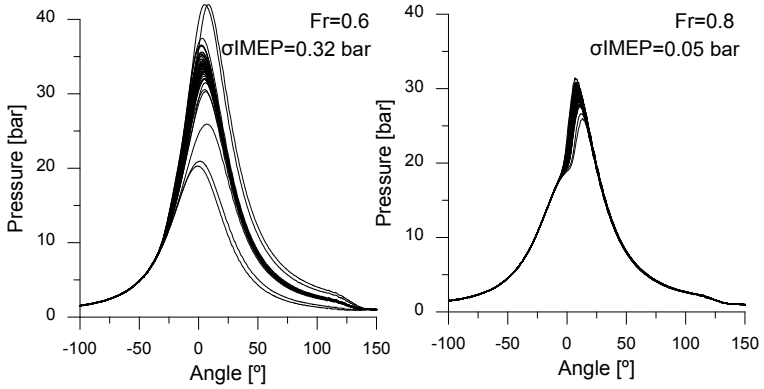

Figure 1: Same operating point (2000 rpm, IMEP ∼3 bar) with different combustion modes. **Left**.- SI mode. **Right**.- CAI mode.

the engine, thus limiting the CAI operating range. On the other hand, the classical control of the combustion onset by the spark is lost, and the control of this type of combustion mode becomes a real challenge, since the autoignition conditions are very sensitive to small changes in the engine operating conditions.

Looking back in time, CAI combustion technology was first applied successfully in 2S gasoline engines by Onishi et al. [8] and by Noguchi et al. [9]. This particular combustion mode can be achieved more easily in 2S engines thanks to their operation singularities, since one of the main operation principles of CAI is to warm up the intake charge in order to increase the mixture reactivity [10], and this can be better done in 2S engines. In fact, in 4S engines, the intake and exhaust processes take place in two separate strokes, and the exhaust gases are almost completely scavenged from the cylinder, whereas in 2S engines these two strokes are substituted by a scavenging process, which is carried out around bottom dead center (BDC), allowing to

heat up the initial in-cylinder charge by "simply" controlling the amount of hot residuals that remain in the cylinder. Some authors have tried to get this effect in 4S engines making an exhaust re-breathing [11, 12], but they faced two important problems: the significant complexity of the distribution system required to produce this effect, and the heat losses during the re-breathing process.

The purpose of this paper is to present the potential of a new 2S gasoline engine concept able to operate in SI and CAI combustion modes, which is able to compete with current, more standard gasoline engines in terms of fuel consumption for the same peak power and cost. This engine concept can be applied either to emerging markets or to more stringent markets. In the first case (emerging markets), the NO_x limit would be achieved without any specific after-treatment system for this pollutant, making use of the advantages of the CAI combustion in terms of NO_x emissions. In the second case (more stringent markets), an EGR system will be required as a way to fulfill the NO_x limit. In both scenarios, the traditional three way catalyst used in gasoline engines will be removed to allow the operation under lean mixture conditions, which would improve fuel efficiency. The only exhaust after-treatment system that will be used is an oxidation catalyst, the use of which is unavoidable in any premixed combustion engine to remove CO and UHC emissions.

Throughout the whole paper, the influence of the different engine configurations and parameters on the two operating modes (SI and CAI), as well as the transition between them, will be analyzed, showing at the same time the results in terms of combustion performance, fuel consumption and NO_x

emissions.

The paper structure is as follows: the next section deals with the engine configuration, how the experimental data is processed and which testing methodology is used. Later, the main results of the current research are presented in two parts: first, the engine behavior without EGR is shown, focusing on the effect of the fuel/air ratio, the VVT position and the injection parameters; and, second, the effect on the combustion process of adding EGR to all the previously analyzed parameters is studied. Finally, the main conclusions of the study will be enunciated.

2. Experimental facilities and methods

In this section, the following information will be given:

- The description of the final engine configuration, as well as the main reasons that justify the different selections.
- The test bench with the prototype engine (single-cylinder engine).
- Some details about the data processing, showing how the main parameters used in the analysis are determined.
- The testing methodology.

2.1. Engine Configuration Selection

The developed engine will be a 2S uniflow-scavenged twin-cylinder engine with Variable Valve Actuation (VVA) and direct injection. Besides, it will be a relatively small engine, with around 0.6 liters of total displacement volume,

suited for small vehicles. The main reasons behind the selection of the engine configuration finally retained will be discussed in this subsection.

Since the most common configuration in 4S engines is overhead valves, the first idea coming to mind is to keep this configuration to take advantage of all the cumulated know-how about this way to design automotive engines. However, 2S engines require a very high permeability because they have much less time to perform the intake and exhaust processes, hence the interest in the use of ports in these engines. Among the different possibilities, the uniflow scavenging configuration with overhead poppet valves is chosen, with intake ports and exhaust valves (Figure 2), since it is a widely employed configuration in applications other than automotive, with extremely good results [13–16]. The scavenging quality is greater than in other configurations, and permeability is much higher (at the intake by the incorporation of a high number of ports along the cylinder periphery, and at the exhaust because the space available in the cylinder head is greater, since there are only exhaust valves, and they can be larger).

Among the different VVA systems, a Variable Valve Timing (VVT) system has been considered to be installed in the camshaft actuating on the exhaust valves, enabling the modification of its angular position (Figure 3). In the figure and further on, the camshaft angular position is depicted by the Exhaust Valve Closing (EVC). The selection of this system was made paying attention to its cost. Such a system allows optimizing the scavenging process at different engine speeds, as well as modifying the effective compression and expansion strokes. It is important to note that with this system the valves opening and closing will be modified at the same time, and the lift duration

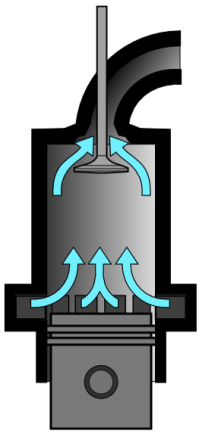

Figure 2: Uniflow scavenged engine (from Wikipedia.org).

will remain the same.

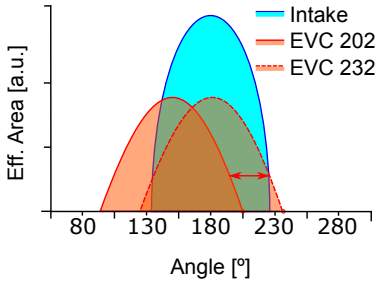

Figure 3: Distribution scheme where the two extreme positions of the VVT are shown.

Since the intake of the fresh air is not induced like in 4S engines, some scavenging pump is needed. To allow the maximum flexibility with a reasonable cost, a mechanical blower, driven by the crankshaft, will be used.

As in any 2S engine, air short-circuiting is unavoidable, and consequently it is strongly recommended to inject the fuel directly inside the combustion chamber once the ports and valves are already closed. Thus, a fuel direct injection system is required, but paying attention to the cost of the system,

finally an air-assisted fuel injection system will be retained. This system consists of two injectors: the first one introduces the gasoline inside a pressurized chamber, filled with air, located just above the second injector. Later, this premixture is introduced in the cylinder (Figure 4). Such a system allows a good enough atomization without the need of high (and "expensive") pressures.

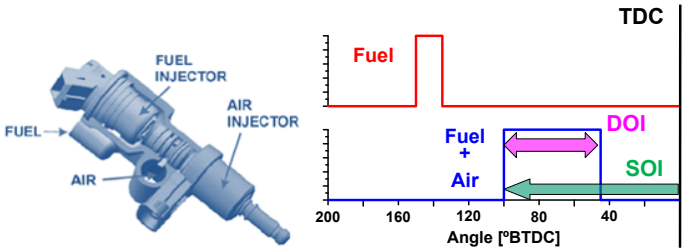

Figure 4: Air assisted injector sketch (left) and operation scheme (right).

The operating scheme shown in Figure 4 to the right, shows the preinjection of the fuel inside the air injector and, after a prefixed delay, the injection of this premixture (air + fuel) inside the cylinder. The main parameters to be set are the following: the amount of fuel introduced, the starting angle of the premixture injection (SOI, Start Of Injection) and the duration of this injection (DOI, Duration Of Injection). With this last parameter (DOI), the amount of air injected together with the fuel can be modified.

2.2. Experimental facilities

Even if, as indicated before, the definitive engine will be a twin-cylinder engine, the engine used for the current research is a single-cylinder, prototype engine, with the same configuration as in the final twin-cylinder engine. This choice is very common in basic research activities, since a single-cylinder

engine enables a much better analysis of the obtained results. The intake and exhaust lines of this single-cylinder engine have not been optimized from the acoustic point of view because, anyway, all these phenomena will be significantly different to those of the final (twin-cylinder) engine. However, the exhaust system has got a controlled back-pressure to resemble the one existing in the real engine, and some acoustic shock absorbers have been installed in the intake and exhaust lines not to optimize but, at least, to mitigate the possible adverse effects of the pressure waves.

The instrumentation installed in the engine and in the test cell to drive and control the engine, and to extract the necessary data from it during each test is as follows: an optical highly precise encoder is connected to the crankshaft to know exactly its instantaneous position. The in-cylinder pressure is measured twice with two piezoelectric sensors (Kistler 6061B U20). This redundant measurement is done to be sure about the proper operation of these key and critical sensors. To get a reference for the in-cylinder pressure, a piezo-resistive pressure sensor was installed in the liner near BDC (Kistler 4007B).

The single-cylinder engine can't run by itself, since many of the elements available in the real and definitive engine are not mounted on it. These missing elements need to be installed in the test cell, a complete sketch of which can be seen in Figure 5:

- The dry sump lubrication system as well as the cooling system are externally assisted with pressure, flow and temperature control.

- The intake system has got an external screw compressor to emulate the effect of the mechanical blower from the real engine. In this facility the

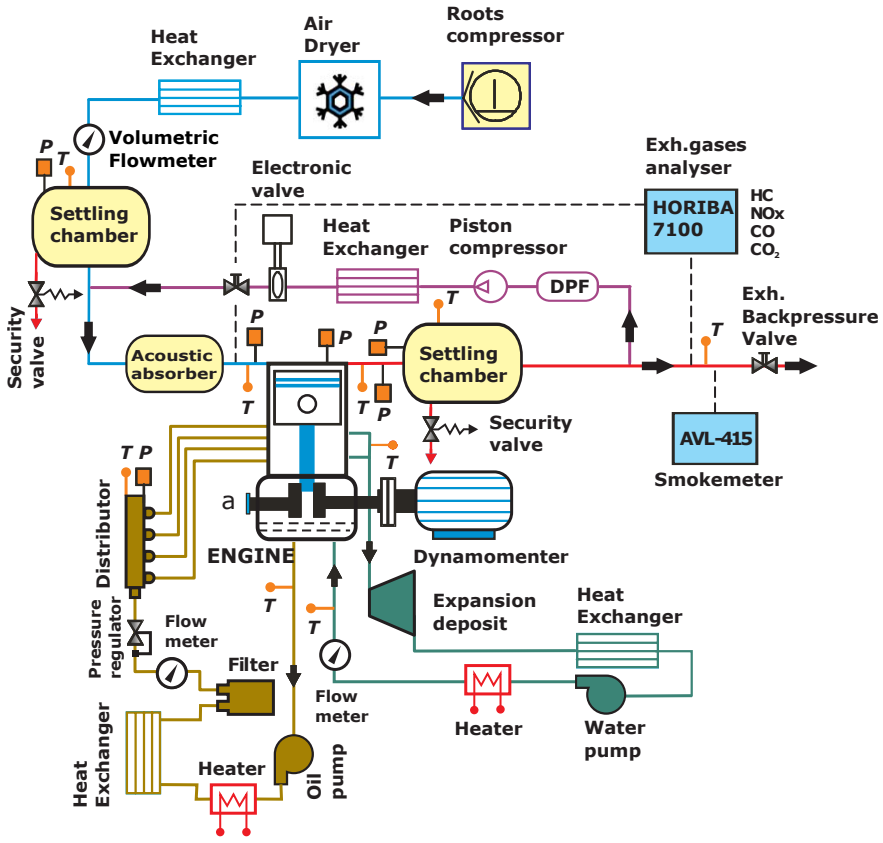

Figure 5: Test cell sketch.

temperature, humidity, pressure, mass flow and chemical composition (EGR) of the gas are controlled.

- In the exhaust line a pneumatically-controlled valve is used to control the backpressure, maintaining its value to that equivalent in the real engine.

- The EGR (External Gas Recirculation) facility is an external, low pressure system. It has got a particulate filter to suppress the soot particles,

a cooling system, a compressor, and a flow regulator, controlled by the CO_2 content of the intake gases, so as to control the EGR rate.

Finally, the fuel mass consumption is measured by a Horiba FQ2100 device, which is suitably adapted for small consumptions (it is important to keep in mind that the engine is a small, single-cylinder engine), and the exhaust gases composition is measured by a Horiba MEXA 7100.

2.3. Data processing

In this subsection, how the main parameters, later used in the analysis, are deduced from the experimental measurements is described.

2.3.1. BSFC estimation

The single-cylinder prototype engine used in this research is far from the real engine in many aspects. For example, it has not a mechanical blower driven by the crankshaft (the real engine needs to have one), and it has two counter-rotating balance shafts for inertia compensation (these will not be used in the real engine), leading to a set of mechanical losses that are not representative of the real engine. Despite these differences, it is important to estimate the main engine parameters for the real engine, in such a way that these parameters are more fair and useful for comparison. For this reason, the BMEP (Brake Mean Effective Pressure) is estimated from the IMEP (Indicated Mean Effective Pressure) taking into account the following mechanical losses:

- The friction losses + driving power of the water pump, using a suitable correlation as a function of the engine speed.

- The work performed by the mechanical blower supplying the intake air, using an experimental correlation after testing the real blower.

- The work performed by the high pressure compressor supplying the air for the fuel injection system, using an experimental correlation after testing the real compressor.

After considering all these losses, the BMEP and, thus, the BSFC of the twin-cylinder engine can be properly estimated.

2.3.2. IGR estimation

The IGR (Internal Gas Recirculation) is the amount of residual hot gases remaining inside the cylinder after the gas exchange process is over. To calculate this parameter, first, it is necessary to know the fresh charge amount (i.e. air + EGR gases) trapped in the cylinder, which is measured by injecting methane in the intake manifold (as a gas tracer), and measuring its content in the exhaust, as described in [17]. This allows computing the trapping ratio or trapping efficiency. Once this is known, the trapped residual gases are estimated with the equation of state and an in-cylinder enthalpy balance at the beginning of the compression process (also described in [17]), with the following assumptions: the trapped air is assumed to have the temperature measured in the intake manifold (i.e. around the intake ports) and its quantity is known thanks to the trapping ratio determined previously; the residual gases temperature is calculated from the temperature measured in the exhaust manifold, taking into account the possible amount of short-circuited fresh charge; the in-cylinder pressure is assumed to be the one measured in the cylinder at the corresponding crank angle. To have this last

information, as already mentioned above, a piezo-resistive pressure sensor was installed near BDC, to be able to measure this pressure. Finally, with the two equations mentioned above, the mass of residuals is determined.

2.3.3. Other parameters

EGR rate measurement

The measurement of the EGR rate is performed based on the CO_2 content in the intake and in the exhaust gases. With this twofold measurement, the EGR content in the intake charge can be determined.

Heat Release Rate (HRR)

The HRR's are calculated by the in-house code Calmec [18, 19] using a classical procedure, which makes use of the instantaneous in-cylinder pressure, applying the first law of Thermodynamics and a heat losses model based on the Woschni equation.

Knock detection

The MAPO (Maximum Amplitude of Pressure Oscillations) parameter is the one selected to detect knock, which is based on the analysis of the high frequency amplitudes in the in-cylinder pressure signal [20]. To obtain the MAPO, the raw pressure signal is band-pass filtered (between 5 and 20 kHz), and the maximum amplitude of the resulting signal is taken. When the MAPO value surpasses a given limit, knock is considered to occur. The MAPO limit is considered to depend linearly with the engine speed, being 1.8 bar at 2000 rpm and 3.8 bar at 4000 rpm (these values were determined by in-house experience). Finally, it is worthy to underline that the MAPO parameter is checked on-line for engine surveillance when performing the different tests.

Estimated performance in standard cycles

Among the different results shown in this research, the performance of the engine, mounted on a vehicle, in some standard cycles will be given. To perform this estimation, a group of operating points spread all along the engine operating map are selected, and they are tested in the engine test bench. Later, with the information available on each of these points, the main engine parameters (e.g. BSFC, BSNOx, etc.) are estimated on the whole engine map by interpolation/extrapolation. Finally, with all this information, the results for any given standard cycle can be predicted. Another additional data important to perform the prediction is the vehicle mass (700 kg), the engine peak power (30 kW) and the gearbox definition, which has been optimized to minimize NO_x emissions in the Bharat cycle. The standard cycles considered in the frame of this work are Bharat (currently in force in India), NEDC (New European Driving Cycle) and WLTP (Worldwide harmonized Light vehicles Test Procedure).

2.4. Testing methodology

For this study, each operating point is defined by an engine speed and a given fuel mass. There are two main reasons for choosing the fuel mass as an indicator of the engine load, instead of the BMEP: first, because BMEP is not a direct measurement, and consequently doing tests at a given BMEP would take a lot of time and effort; and, second, because in 2S engines the engine behavior depends quite a lot on the combustion characteristics (strongly dependent on the fuel mass), since it significantly affects the scavenging process. Consequently, changing the fuel mass in equivalent operating points would complicate the comparison. However, when presenting the results, an

illustrative value of the IMEP will be given, at each operating point, as a reference of the engine load (please note that the corresponding IMEP of an equivalent 4S engine will be approximately twofold).

Once the test point is defined (engine speed and fuel mass), the following degrees of freedom are explored to see which influence they have on the combustion process:

- Equivalence ratio (Fr): This parameter is adjusted by modifying the amount of air introduced in the engine (since the fuel mass flow is fixed for a given operating point). This equivalence ratio is calculated from the measured engine flows (air -including the injected air- and fuel), and it is equivalent to the one that can be measured in the exhaust by the exhaust gas analyzer.

- EVC position, which can be adjusted between 202º and 232º.

- Injection parameters: SOI and DOI (referred to the direct injection event, i.e. the air + fuel injection). To select these two parameters, attention should be paid to the in-cylinder pressure at EOI (End Of Injection), to guarantee that the injection pressure is above this other pressure during the whole injection event. In fact, the EOI is limited to 40 to 50º BTDC (Before TDC), depending on the operating conditions.

- The spark timing. Under SI operating mode, this parameter is intended to be placed at MBT (Maximum Brake Torque), even if, usually, this location is unachievable because of knock.

- The EGR rate.

Finally, each test has two sets of measurements: instantaneous and mean measurements. For the instantaneous ones, 250 consecutive cycles are recorded, and for the mean ones the averaging time is 60 seconds.

3. Results and discussion

The presentation of the results will be as follows. First, the peculiarity of this engine, regarding its ability to operate in different combustion modes, will be presented. On a next step, the effect of the Fr (fuel/air equivalence ratio) and VVT position will be reviewed, as well as the effect of the injection settings (SOI and DOI, in both cases referred to the direct injection of the fuel + air mixture), trying to summarize the results already presented in two previous publications [21, 22]. Finally, as a new contribution of the present paper, the effect of using EGR on the performance of this engine will be analyzed.

3.1. The different combustion possibilities of this engine

Because of its characteristics, this engine can operate in two different combustion modes: SI and CAI. And recently it has been proven that it can switch from one mode to the other without any trouble [23]. The operation in one or the other mode mainly depends on the IGR ratio and the initial temperature of the in-cylinder charge. In fact, both parameters are strongly affected by the engine load, as illustrated qualitatively in Figure 6. As far as the engine load is reduced, the amount of air required by the engine is lower, and the scavenging process is less efficient, thus explaining why the IGR ratio increases. Regarding the initial temperature of the in-cylinder charge, it is the result of an enthalpy balance between the fresh charge (at

the intake temperature) and the IGR gases (at the exhaust temperature in a first approach). The increase in IGR ratio as far as the engine load is reduced explains why the initial temperature of the charge increases initially. But at very low loads, this trend is reversed, which is justified by the decrease in exhaust temperature (i.e. the temperature of the IGR gases) due to the decreasing fuel mass burnt.

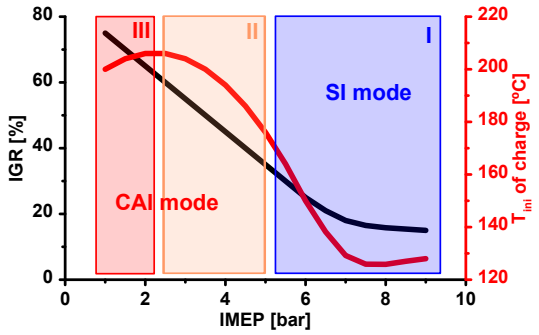

Figure 6: Evolution of the IGR ratio and the initial temperature of the in-cylinder charge with the engine load.

Based on the above explanation, three different regions have been indicated in Figure 6:

- Region I: at high loads, where the IGR and the initial temperature of the charge are moderate, the combustion process is controlled by the spark, as in any conventional SI engine. In this scenario, the autoignition of the mixture is dangerous for the mechanical integrity of the engine (knock), and consequently it needs to be avoided.

- Region III: at low loads, the fuel burns in purely controlled autoignition (CAI) mode because of the enhanced reactivity of the in-cylinder charge

caused by its high temperature (this is demonstrated in Appendix A). This CAI operation mode introduces some advantages that will be further discussed later: increased engine stability (as already shown in the introduction, see Figure 1), misfiring avoidance (which commonly takes place in engines operating with high percentages of residual gases), low NO_x, etc.

- Region II: between the two previous regions, at medium loads, a transition between SI and CAI takes place.

Because all these three scenarios are significantly different, they will be analyzed separately in the upcoming subsections. The operating points selected to present and analyze the results are shown in Figure 7. Point 1, at maximum torque, has been taken as representative for region I because of its importance in the engine map. Points 2a and 2b have been taken as representative for region II, and Point 3 for region III. These points were selected because of their relevancy in the standard cycles.

Finally, it is important to point out that the data used to present the results were selected trying to better show the effect of the analyzed parameters, and consequently they do not necessarily correspond to the best results obtained with the engine.

Now the influence of the different engine settings on the combustion process will be analyzed.

3.2. Effect of Fr and VVT position

In 2S engines, the in-cylinder trapped gas composition at the beginning of the compression stroke can be very different depending on the scavenging

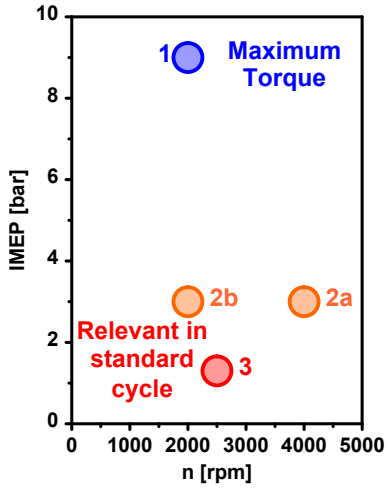

Figure 7: Selected points to present and analyze the results.

process. This gives further implications to the effect of the equivalence ratio and the VVT position at each operating point, compared to what would happen in 4S engines.

Now, taking into account the description shown in section 3.1, the results will be structured in the different combustion modes presented there.

3.2.1. High load – SI mode

At high loads the engine works like any SI engine. At these conditions, the autoignition of the charge is not controllable, and the corresponding pressure oscillations can damage the engine integrity. Thus, under these circumstances, the autoignition is considered as knock, and needs to be avoided.

In a high load operating point, like the one under analysis now, usually pollutant emissions are not a big concern, since it is outside of the usable region during the standard cycle. Consequently, the most important parameter for the analysis in the present conditions is the fuel efficiency, that can

be quantified by the BSFC. Figure 8 shows that BSFC correlates well with the CA75 angle (Crank Angle where 75% of the fuel mass has been burned), getting better fuel consumptions with earlier positions of the CA75. The different points in the figure correspond to different EVC's and Fr's, and the spark timing has been set for each case up to reach the MBT (Maximum Brake Torque) without exceeding the knock limit. Based on the figure, the CA75 seems to be the key parameter to understand how BSFC can be optimized, since it combines the combustion phasing -CA50- and its duration: what is important is to have, at the same time, a fast combustion (i.e. short duration) and a well-phased combustion (i.e. a small CA50), and this twofold information is suitably summarized in the CA75 parameter.

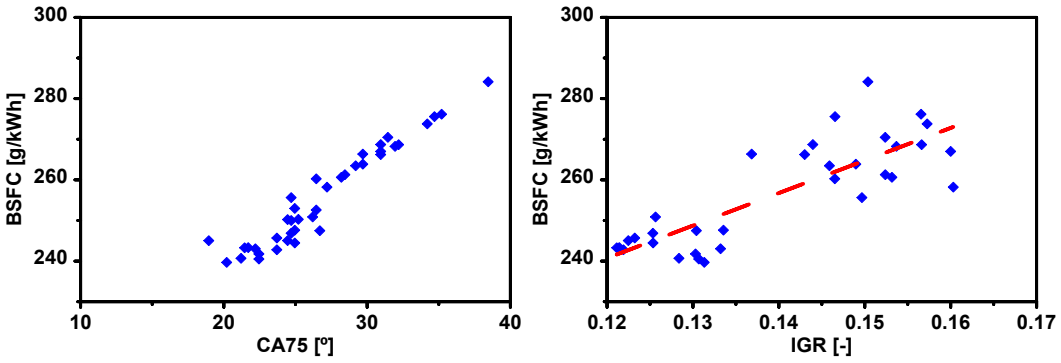

Figure 8: **Left.-** Evolution of BSFC with CA75 at Point 1. **Right.-** For the same tests, correlation between BSFC and IGR ratio.

The start of combustion is very important to advance the CA75 and get better fuel consumption. In the case of a spark controlled operating point, as the one under analysis now, this start of combustion is controlled by the spark timing, which should be commanded earlier to improve BSFC. However, the appearance of knock limits the spark timing advance, thus limiting BSFC.

The result shown in Figure 8 seems to indicate that something might change in the engine when either Fr and/or EVC are modified. With the aim of finding an explanation to this behavior, Figure 8 was built, showing that BSFC is also well correlated with the IGR rate: as the IGR rate decreases, the fuel efficiency is improved. An increase in IGR rate has two opposed effects: on the one hand, the temperature of the charge is increased thus increasing the mixture reactivity; but, on the other hand, the oxygen content is reduced, thus reducing the mixture reactivity. Appendix A shows some calculations made with Chemkin in order to evaluate the autoignition delay of a mixture composed by fresh air and IGR. As a conclusion, for moderate IGR rates the effect of the temperature increment is stronger than the oxygen reduction, and consequently it can be said that the mixture reactivity increases with an increase in IGR rate. The results shown in Figures 8 and 8, then, can be interpreted as follows: BSFC is improved when the IGR rate is reduced, and this is because the mixture reactivity is reduced, which avoids the appearance of knock and allows an earlier spark timing and, consequently, an earlier CA75.

Therefore, at high loads, where the engine works in SI mode, the appropriate configuration of the VVT and Fr will be the one that minimizes the amount of IGR.

3.2.2. Mid-load – SI / CAI transition

As far as the engine load is decreased, the amount of fuel to burn is smaller and the IGR rates are higher, leading to higher dilution of the in-cylinder charge. Thus, pressure gradients generated by the combustion and autoignition processes become more controllable and admissible for the engine.

At these operating conditions CAI combustion starts to be an option to operate the engine. The transition between the SI and CAI combustion modes is based on the control of the mixture reactivity by means of the IGR rate modification. In Figures 9 and 10, the transition between both modes is shown with an Fr variation. In view of these results the following comments can be introduced:

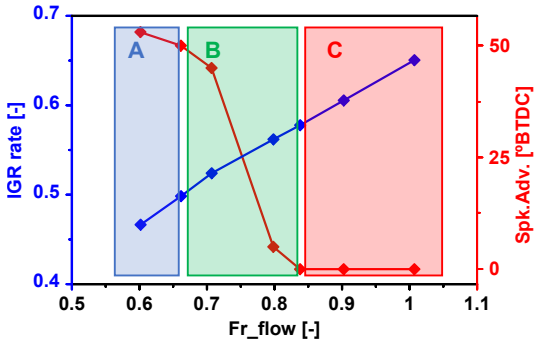

Figure 9: Transition from SI to CAI conditions at Point 2a.

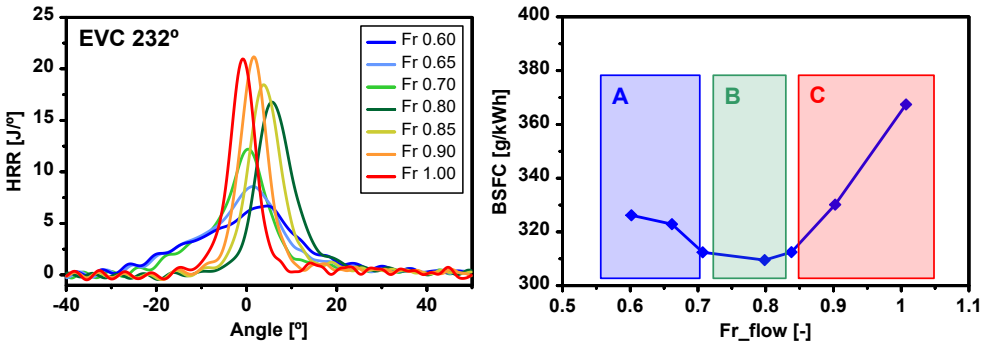

Figure 10: **Left.-** Effect of Fr on the HRR at Point 2a. **Right.-** Effect of Fr on BSFC for the same conditions. The tests are the same as in Figure 9.

- At low Fr (region A) the mixture reactivity is not enough to reach stable autoignition conditions. Under these conditions the combustion needs

to be initiated by the spark, and the spark timing should be commanded early enough to get a stable and well centered combustion.

- At high Fr (region C) the mixture reactivity is much higher, and the autoignition conditions are reached. The spark ignition control is lost (in fact, the spark event could be even removed), the combustion process is much faster (narrower HRR) and it starts to be placed too early, thus penalizing BSFC (as shown in Figure 10).

- Finally, at region B, placed between the two previously explained cases, there is an optimum value for Fr where the mixture reactivity is the best suited. For this particular value of Fr the combustion process is, at the same time, fast and well located in the cycle, and consequently the BSFC reaches a minimum value.

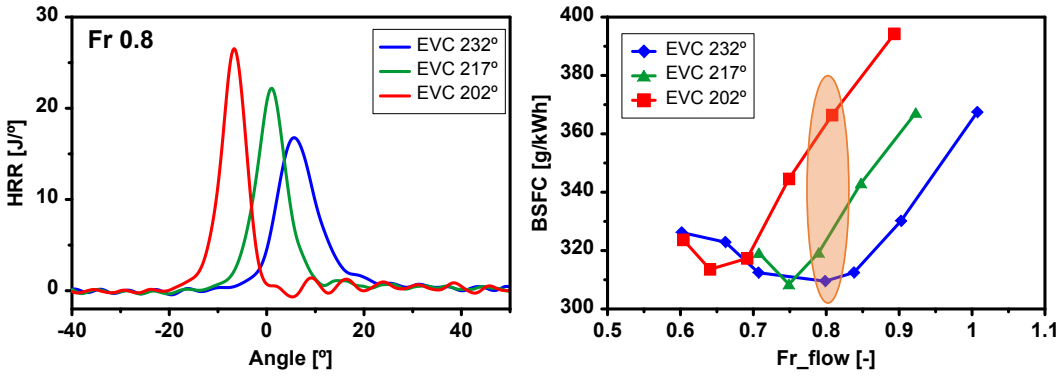

Figure 11: Effect of VVT on the HRR (**left**) and BSFC (**right**) at Point 2a operating in CAI mode.

Once the combustion control by the spark is lost (regions B and C), it can be recovered by the VVT position, as illustrated in Figure 11. This effect is

because of the influence of the VVT position on the scavenging process and on the in-cylinder thermodynamical conditions (see Figure 12). An early VVT position, for instance (e.g. EVC 202º), leads to a reduced effective expansion stroke and, thus, a higher exhaust temperature. This means that the IGR gases will be hotter. At the same time, this early VVT position will increase the effective compression ratio (which is defined as the ratio between the volume in the cylinder when the exhaust valves are closed and the one when the piston is at TDC), as demonstrated by the in-cylinder pressure at -40º ATDC shown in the figure. Both effects (hotter IGR and higher effective compression ratio) would lead to a much reactive mixture, thus explaining the trend observed in Figure 11.

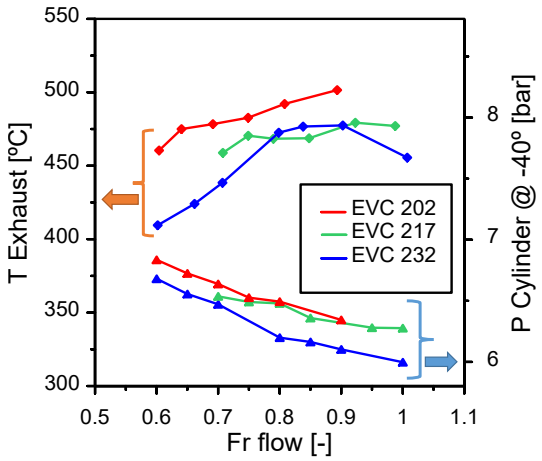

Figure 12: Exhaust temperature and in-cylinder pressure -40° ATDC for all the cases presented in Figure 11.

3.2.3. Low load – "Pure" CAI mode

At low load conditions, the IGR rate is well above 50% (i.e. the residual gases mass is bigger than that of the fresh air). These IGR rates lead to

a significantly high temperature in the in-cylinder charge together with an extremely low oxygen concentration. Such a mixture may present difficulties for being ignited by a spark, but can autoignite with no danger for the engine mechanical integrity. Thus, at these conditions, the control of the combustion onset by the spark is completely lost. This is shown in Figure 13, where a wide swept of spark timings has been performed, including an extra case with no spark ignition, at the operating conditions corresponding to Point 3 (see Figure 7). It can be observed that the combustion process is not affected by the spark timing, nor by the spark ignition, at all. The only role of the spark under these conditions is just to start the engine and to hold the combustion process when the engine is still cold. And, as indicated before, the combustion onset can be controlled with the parameters that control the mixture reactivity, namely Fr and VVT position.

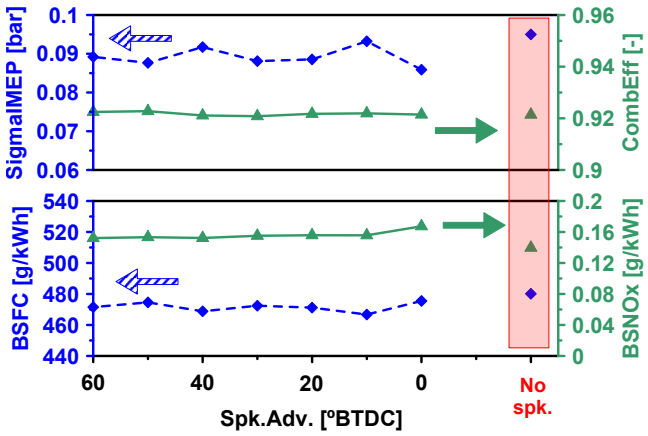

Figure 13: Effect of the spark timing on different engine outputs at Point 3.

Finally, two additional remarks from the previous figure: at these conditions the NO_x emissions are extremely low (below 0.2 g/kWh), which is

due to the low combustion temperatures (mainly because of the very low initial oxygen concentration in the in-cylinder charge), and the engine stability is outstanding ($\sigma IMEP$ below 0.1 *bar*), which is a typical characteristic associated to the CAI combustion mode, as indicated in the introduction section.

3.3. Effect of the injection parameters (SOI-DOI)

In section 2.1 the fuel injection system was described. It is an air-assisted direct injection system, where the start of injection (SOI) and its duration (DOI) can be adjusted. It is worthy to remind here that both SOI and DOI refer to the air+fuel injection process. As already said before, for a given operating point the fuel amount is prefixed, and a modification of the DOI affects the air quantity injected together with the fuel.

On the one hand, based on the analysis of all the available results, it can be said that, in general, an advanced SOI helps to improve the combustion process, since the time available to homogenize the mixture is higher. However, when the engine load is reduced, this trend is not always observed (see Figure 14). A possible explanation could be that, for advanced SOI's, the fuel is excessively spread in the IGR gases (please note that a reduction in the engine load considerably increases the IGR rate, see Figure 6), thus hindering the ignition of the mixture.

As a conclusion, regarding the effect of the SOI, unfortunately there is not a well defined general trend linking this parameter with the correct engine operation for all the tested points.

On the other hand, the effect of the DOI on the combustion process shows that there is an optimal value for this parameter that depends on the engine

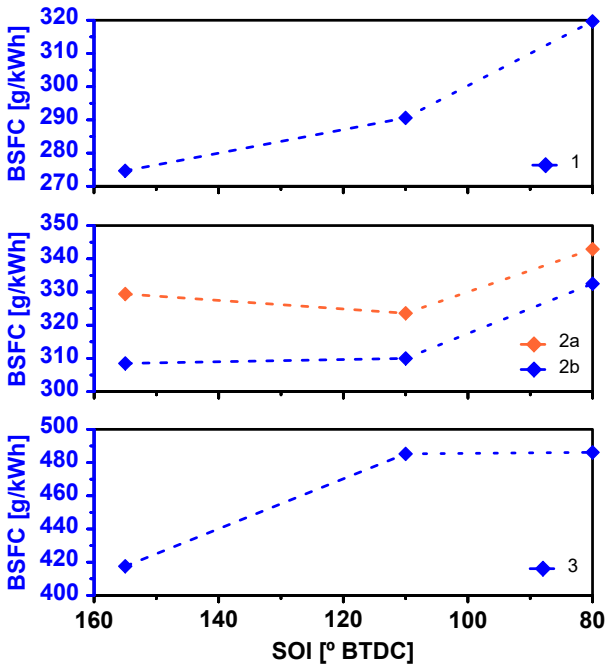

Figure 14: Effect of SOI on BSFC at high (**top**), mid (**center**) and low load (**bottom**).

speed and load. The key parameter to analyze the relationship between the DOI and the combustion behavior is the IAR (Injected Air Ratio)[22], which relates the injected air mass with the total air mass trapped in the cylinder, Equation 1.

$$IAR = \frac{m_{air\_HP}}{m_{air\_HP} + m_{air\_LP\_trapped}} \tag{1}$$

In Figure 15, some key engine parameters, as BSFC, $\sigma IMEP$ and combustion efficiency (calculated from the measured CO and UHC content in the exhaust gases), are plotted vs. the IAR for all the operating conditions analyzed in this paper (see Figure 7). It is important to remark that, whereas the values of $\sigma IMEP$ and combustion efficiency are comparable among the

different operating points, this is not the case for the BSFC. For this reason, this last parameter has been normalized (with a linear transformation), to make it comparable among all the operating points, but (of course!) keeping the trends of the parameter. The following observations can be done:

- For each parameter, all the operating points show a similar trend, which is indicated by a dashed red line.

- For a value of IAR of around 0.09, all three parameters are optimized at the same time: BSFC is minimum (maximum fuel efficiency), the IMEP variability is minimum (maximum stability) and the combustion efficiency is maximum.

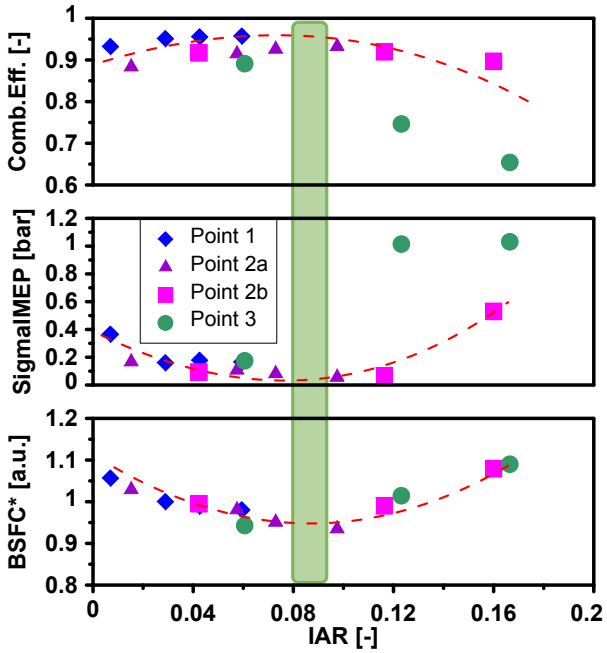

Figure 15: Effect of the IAR on BSFC, $\sigma IMEP$ and combustion efficiency.

This result indicates that, for each operating condition (engine speed and load) there is an optimum value of the DOI to optimize the engine behavior.

3.4. Influence of EGR on the combustion process

With the current engine and its controlling parameters presented up to now, its performance in some standard cycles has been evaluated using the methodology already presented in subsection 2.3 (results shown in Table 1). Based on these estimations, it can be seen that, without any particular technique to reduce NO_x emissions, this engine would be able to fulfill the Bharat standards, currently in force in an emerging country, which is significantly less demanding than in any other more developed country. But for the progress of the engine design, and to allow its application in more stringent scenarios, it is compulsory to include some NO_x reduction strategy. As already said before, an EGR system will be implemented to allow further reductions in NO_x emissions. The reason for choosing an EGR system instead of an exhaust after-treatment system (an SCR, for example) is, among others, the advantage to win another control parameter for the engine operation. In fact, the effect of introducing EGR on the combustion process and its control is what will be analyzed now.

The EGR introduction affects in different ways the engine performance depending on the operating point. Therefore, the results will be presented divided in the different engine loads, as in the previous sections.

3.4.1. High load

At high load operation, an increase in the equivalence ratio leads to a limitation in the CA50 advance due to knock, since the reactivity of the

Cycle	Fuel cons. l/100km	NO_x mg/km	NO_x Limit mg/km
Bharat	3.58	82.0	80
NEDC	3.87	209.2	60
WLTP	3.85	298.5	60

Table 1: Results from the performance of a vehicle powered by the engine without EGR in 3 different standard cycles.

mixture is increased. This effect prevents the correct positioning of the combustion event in the cycle. In this context, thanks to the EGR introduction the knock is mitigated and, thus, the combustion can be further advanced (Figure 16 to the right). This leads to an improvement in BSFC despite the loss in combustion efficiency associated to the lower O_2 content in the intake gases caused by the EGR introduction (see Figure 16 to the left): fuel consumption is improved in 2.3% even assuming a loss of 4% in combustion efficiency, and NO_x emissions are reduced in 75.6%. These results show the strong potential of EGR to reduce NO_x, which is a more than usual result. But, at the same time, EGR also serves to improve BSFC, because of its knock mitigation capability. This capability will be further analyzed in the following paragraphs.

Figures 17 and 18 show a cycle-to-cycle analysis from a tested point with the same configuration as before (Fr=0.75), where the spark advance and the EGR rate are modified, respectively, in order to advance the combustion position (i.e., the CA50). These two figures include two graphs in both cases: the main graph shows the instantaneous MAPO (which quantifies knock and

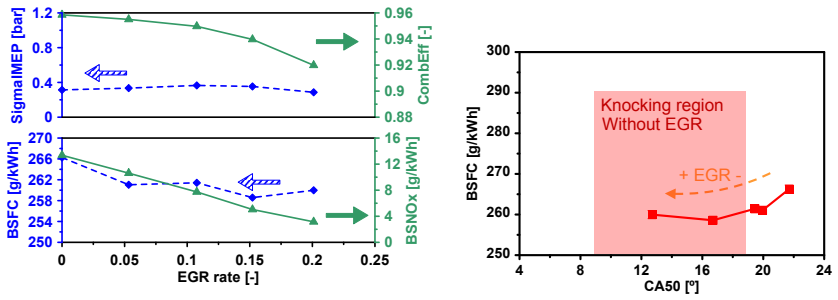

Figure 16: Point 1. High equivalence ratio case (Fr=0.75). Evolution of different parameters as a function of the EGR rate (**left**). For the same tests, correlation between BSFC and CA50, and detail of the knocking region when no EGR is used (**right**).

was defined in section 2.3) for each measured cycle (250 cycles per test) vs. the corresponding CA50, whereas the sub-graph shows the same information but averaged (MAPO mean vs. CA50 mean). The first case (Figure 17) shows an attempt to advance the combustion position without EGR, only advancing the spark timing. As can be seen, in this case the MAPO increases with the combustion advance (CA50), as expected.

However, in Figure 18, the combustion process, for the same operating point in similar conditions (Fr=0.8), is advanced by the introduction of a 15% EGR rate. The result is an advance of the combustion process without any significant change in knock (in fact, the mean value of the MAPO is even slightly lower than in the starting point). This result illustrates the strong potential of EGR to mitigate knock, thus allowing significant improvements in terms of fuel economy [24] and NO_x emissions.

Still at high load, but with lower equivalence ratios (Figure 19), where knock is not a problem for the correct combustion positioning, the introduction of EGR does not bring the clear benefit seen before, since the combustion

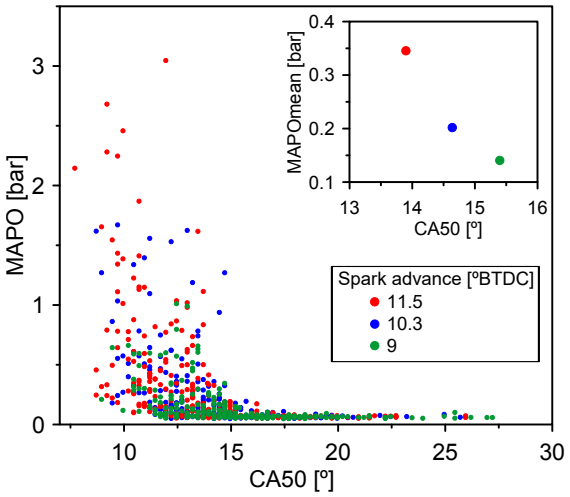

Figure 17: MAPO vs. CA50 for all individual cycles (main graph) or averaged (detailed graph) for a spark timing swept.

onset is placed even too early in the cycle. Thus, in this other case, the NO_x reduction is attained at the expense of an increase in BSFC. This BSFC deterioration can be explained as follows: the dilution effect of the mixture and the reactivity loss, both caused by EGR, lead to a decrease in combustion stability that needs to be compensated by an advanced spark timing (CA50 is placed too early) to hold a stable enough combustion process. This, coupled to the deterioration in combustion efficiency, leads to a significant increase in BSFC.

Given the widely-known NO_x reduction effect of EGR and all the previous explained effects, it is found that, at high loads, the EGR strategy improves considerably the NO_x-BSFC trade-off (shown in Figure 20). In view of this figure, it can be stated that NO_x emissions can be reduced in a factor of 4 without any increase in BSFC, or even slightly improving it.

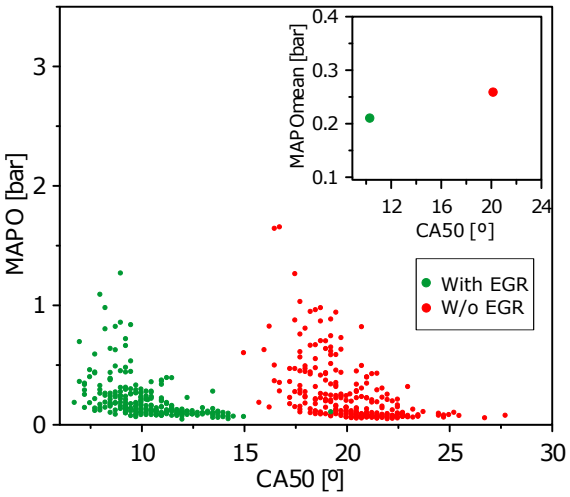

Figure 18: MAPO vs. CA50 for all individual cycles (main graph) or averaged (detailed graph), for two different cases: with (green) and without EGR (red).

3.4.2. Medium and low load

As far as the engine load is reduced, the limitation in CA50 position caused by knock disappears. Besides, in contrast with the previous scenario, because the combustion mode switches to CAI mode, the spark timing looses its controlling capability over the combustion process. Thus, now the CA50 can be correctly positioned by some means (e.g. Fr and/or VVT position, not the spark timing) other than EGR. Consequently, the role of EGR in this other scenario will be less important.

As in the previous subsection, the effect of the EGR will be analyzed at two different Fr's. In Figure 21, to the left, the evolution of different engine parameters as a function of the EGR rate is presented, for Point 2b (see Figure 7) at low Fr (0.65). The trends are similar to the ones already seen in the previous subsection, for higher loads. The following comments can be

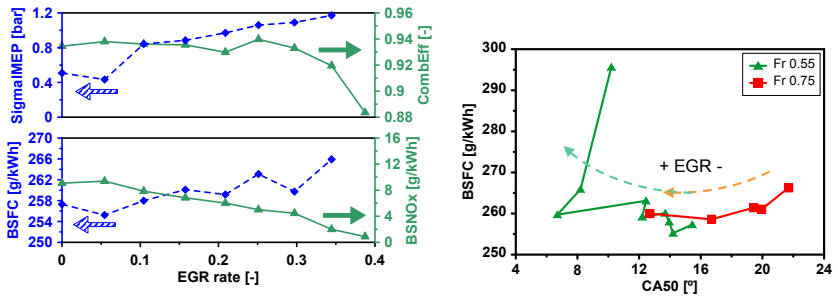

Figure 19: Point 1. Low equivalence ratio case (Fr=0.55). Evolution of different parameters as a function of the EGR rate (**left**). For the same tests, correlation between BSFC and CA50. In this case, the results for the Fr=0.75 case are also shown (**right**).

done:

- EGR significantly reduces NO_x emissions. However, the starting levels of NO_x are already low, and the use of EGR is much less justified than at higher loads. Remember that at CAI operating mode, the IGR rate is significantly high, and NO_x emissions are naturally low.

- BSFC worsens with EGR. On the one hand, as observed in Figure 21, to the right, this is because the combustion process is delayed (CA50 moves away from TDC). The reason for this is the lower reactivity of the in-cylinder mixture, and cannot be compensated with the spark timing, since it does not control the combustion onset anymore. On the other hand, this is also the result of the loss in combustion efficiency caused by EGR that can be observed in the figure as well.

- The lower mixture reactivity also explains the decrease in combustion efficiency, as well as the increase in $\sigma IMEP$ (the engine is more unstable).

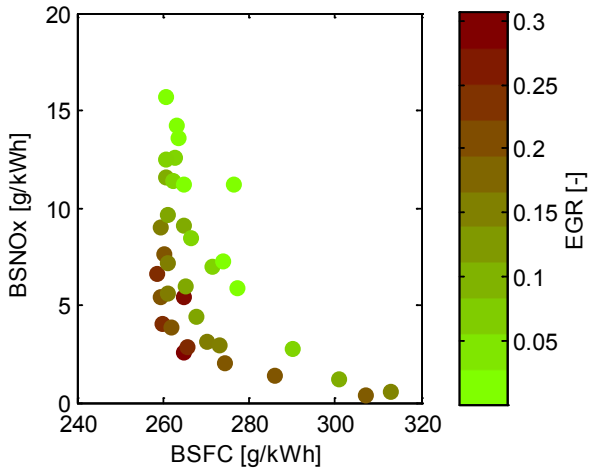

Figure 20: NO_x-BSFC trade-off. The color scale gives information about the EGR rate.

For higher Fr's (e.g. 0.75 in 21), because of the higher mixture reactivity, the combustion onset takes place too early, and the engine operates in the non-interesting region shaded in the figure to the right. The operation in this region needs to be avoided because, depending on the operating point, either the combustion onset is placed too early or some knock can appear. In this other case, EGR can help to place the combustion onset on the right place, as shown in Figure 21 to the right, thus allowing to meet, together, lower NO_x and BSFC.

Therefore, at the operating points where the combustion process was already well placed in the cycle, EGR only serves to reduce NO_x emissions. But taking into account the initially low NO_x emissions because of the CAI combustion, it might be not necessary to add EGR in most cases.

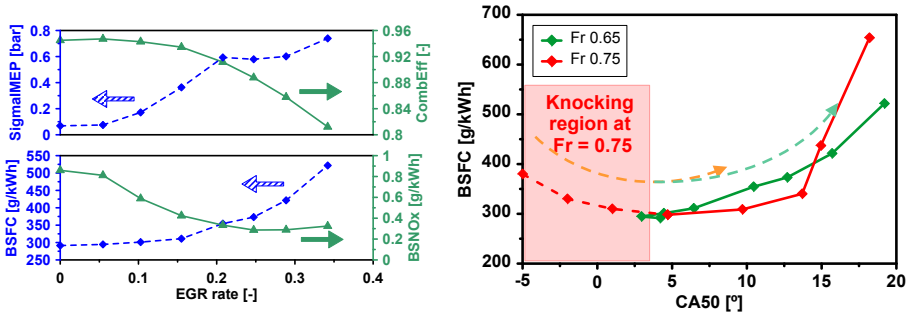

Figure 21: Point 2b. Low equivalence ratio case (Fr=0.65). Evolution of different parameters as a function of the EGR rate (**left**). For the same tests, correlation between BSFC and CA50. In this case, the results for a higher Fr (Fr=0.75) case are also shown (**right**).

As a conclusion, the use of EGR, as well as reducing NO_x emissions, allows the exploration of operating points that were previously impossible to achieve due to the excessively high knock level or the too early combustion onset. At the operating points where the excessive mixture reactivity represents a problem for the point optimization, the introduction of EGR decreases this reactivity and allows the correct combustion positioning, thus improving the fuel efficiency with a lower level of NO_x emissions. With this new degree of freedom in the engine, a wider range is now available at each operating point to look for lower NO_x together with the same or even lower BSFC. Besides, even the engine peak power can be increased (based on some other results not presented here, the increase in peak power can be around 25%).

Last, the results of the estimated performance of the engine, but now with an EGR system, on the different standard cycles has been reviewed, and they are shown in Table 2. Now, even the most stringent standards are fulfilled in terms of NO_x, even if the overall fuel consumption has increased respect

to the cases already shown in Table 1. This is a proof of the potential of this engine concept to be applied even in the most stringent markets, where it can help to reduce CO_2 emissions respect to the use of standard 4S gasoline engines.

Cycle	Fuel cons. l/100km	NO_x mg/km	NO_x Limit mg/km
Bharat	4.18	30.8	80
NEDC	4.47	47.5	60
WLTP	4.38	52.8	60

Table 2: Results from the performance of a vehicle powered by the engine with EGR in 3 different standard cycles, to be compared to those without EGR already presented in Table 1.

4. Conclusions

The present engine, as shown along the whole paper, has several characteristics that make it unique. Its capability to operate in different combustion modes together with the fact of being a 2S engine, make it much more complex (in terms of operation, but not of cost) than any traditional 4S engine. First, the IGR control (hot residual trapped gases) plays a fundamental role, since it strongly affects the mixture reactivity. Depending on the operating conditions, a high IGR rate will be interesting to achieve a CAI combustion mode, or, on the contrary, when the autoignition of the mixture can be dangerous for the engine mechanical integrity (knock), the IGR rate needs to be minimized to allow the correct placement of the combustion onset without any autoignition of the mixture.

The autoignition conditions are mainly controlled by means of the equivalence ratio, which has two effects that go in the same direction: on the one hand, it affects the mixture reactivity by changing the fuel/air ratio but, on the other hand, it also affects the IGR ratio. Remember that, for a given operating point, the mass of fuel is fixed, and the equivalence ratio is modified by changing the amount of air: more air (i.e. lower Fr) means less IGR, both leading to a lower mixture reactivity; and, on the contrary, less air (i.e. higher Fr) means more IGR, both leading to a higher mixture reactivity. Additionally, with the EVC modification, the effective compression and expansion strokes can be modified. This modification allows the correct adjustment of the combustion location in the cycle when the operating points are in CAI mode, or it can reduce the knock intensity when operating in SI mode, since this change affects the thermal in-cylinder conditions (temperature and pressure).

Regarding the injection parameters, on the one hand, advanced SOI's seem to optimize the engine behavior in most operating conditions. On the other hand, an optimum relationship between the air mass flow introduced through the injection system and the one introduced through the intake has been found, which is the key factor to find the optimum DOI.

Finally, in relation to the use of EGR, besides of decreasing the NO_x emissions, it has also revealed to be a key parameter to control the combustion process at high loads. With EGR, the knock can be mitigated thanks to the loss in mixture reactivity, introducing the possibility to operate in new conditions (where the operation was previously impossible) and giving the opportunity to get better fuel efficiency. Furthermore, an additional benefit

of this strategy can be, on the one hand, the decrease of the maximum air mass flow (due to the increase of the Fr at high load), which allows a reduction in the blower requirements and, with this, a general reduction in fuel consumption; and, on the other hand, for the same maximum air mass flow, to increase the engine peak power (up to around 25%).

Throughout all the present research, this engine concept has shown its big potential. In emerging markets (e.g. India), it can fulfill the emissions levels without any particular device or strategy to reduce NO_x, thanks to the advantages of the CAI combustion mode, thus being a really low cost solution in this scenario. And in more stringent markets, with the introduction of an EGR system (which implies the use of a particulate filter), also the corresponding standards can be fulfilled. In this other case, the concept can be still attractive from the economical point of view, especially when compared to other solutions with more expensive systems (SCR or similar), and also from the point of view of CO_2 reduction compared to more standard 4S gasoline engines.

Acknowledgments

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Notation

2S	Two Stroke
4S	Four Stroke
ATDC	After Top Dead Center
BDC	Bottom Dead Center
BSFC	Brake Specific Fuel Consumption [g/kWh]
BSNOx	Brake Specific NO_x emissions [g/kWh]
BTDC	Before Top Dead Center
CA50	Crank Angle where 50% of the fuel mass has been burned
CA75	Crank Angle where 75% of the fuel mass has been burned
CAI	Controlled AutoIgnition
DOI	Duration Of Injection (air + fuel injection)
EGR	External Gas Recirculation
EOI	End Of Injection (air + fuel injection)
EVC	Exhaust Valve Closing
Fr	Fuel/air equivalence Ratio
HRR	Heat Release Rate
IAR	Injected Air Ratio (ratio between the injected air and the total trapped air)
IGR	Internal Gas Recirculation (i.e. residual gases)
IMEP	Indicated Mean Effective Pressure
MAPO	Maximum Amplitude of Pressure Oscillations
MBT	Maximum Brake Torque

NEDC	New European Driving Cycle
SCR	Selective Catalytic Reduction
SOI	Start Of Injection (air + fuel injection)
TDC	Top Dead Center
VVA	Variable Valve Actuation
VVT	Variable Valve Timing
WLTP	Worldwide harmonized Light vehicles Test Procedure

Appendix A. Appendix A: effect of IGR on the ignition delay

In this appendix the ignition delay of a mixture of air + IGR wants to be assessed as a measure of its knocking tendency. The motivation is to find out which effect is stronger: the effect of the oxygen content reduction when the IGR rate increases, which would reduce the mixture reactivity; or the effect of the increase in mixture temperature when the IGR increases, which would increase the mixture reactivity.

For the study, the following hypotheses and assumptions were taken:

- The air in the intake manifold is at 323 K. This is assumed to be the initial temperature of the air trapped in the cylinder.
- The exhaust gases in the exhaust manifold are at 827 K. This is also assumed to be the initial temperature of the residual gases (IGR) trapped in the cylinder.
- With the two previous assumptions, the initial temperature of the mixture is calculated, as a function of the IGR rate, taking into account the c_p, initial temperature and mass fraction of both air and IGR.

- The mixture at known initial conditions is assumed to be compressed up to TDC (consequently both the temperature and pressure are significantly increased), and then introduced in a PSR (Perfectly Stirred Reactor) reactor (i.e. a perfectly homogeneous reactor), where the autoignition process is going to be studied with Chemkin. It is worthy to note that the compression up to TDC is necessary to ensure the mixture autoignition.

- The ignition delay is considered to happen when 50% of the total temperature increase in the reactor takes place.

- The fuel considered was iso-octane (the chemical mechanism considered was that from Curran et al. [25]), and the equivalence fuel/air ratio was 1 (for Fr=0.6, it was checked that the conclusions are exactly the same).

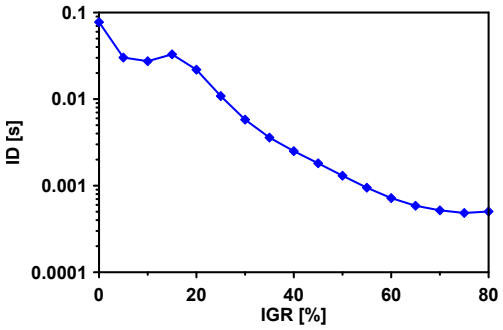

Figure A.22: Effect of the IGR rate on the ignition delay.

The results of the simulations are presented in Figure A.22. It can be seen that, in the main range of IGR rates, the higher the IGR rate, the lower the ignition delay, meaning that the thermal effect is stronger than the dilution

effect. This trend is clearly broken at very high IGR rates (above 75%; for an Fr=0.6, this shift in the trend takes place at an IGR rate of 90%), and at low IGR rates (below 15%). It should be taken into account, however, that the minimum IGR rate achievable in the engine is between 10 to 15%, whereas the maximum IGR rate is between 80 to 90%. Consequently, in the possible range of IGR allowed in the engine, the higher the IGR, the higher the mixture reactivity.

References

[1] S. Barclay. Lotus omnivore engine - 10% better fuel economy than current leading gasoline engines. *Automotive Industries AI*, 189(12), 2009.

[2] M. Nuti. Direct fuel injection: An opportunity for two-stroke si engines in road vehicle use. *SAE Technical Paper 860170*, 1986.

[3] M. M. Rahman, A. K. Ariffin, S. Abdullah, M. M. Noor, and A. B. Rosli. Durability assessment of cylinder block for two stroke free piston linear engine using random loading. *American Journal of Applied Sciences*, 6(4):726–735, 2009.

[4] G. A. Lustgarten and K. Aeberli. Improved liner and ring wear properties of two-stroke diesels. *Diesel and Gas Turbine Worldwide*, 20(3):45–48, 1988.

[5] O. A. Kutlar, H. Arslan, and A. T. Calik. Methods to improve efficiency of four stroke, spark ignition engines at part load. *Energy Conversion and Management*, 46(20):3202–3220, 2005.

[6] H. Zhao. *HCCI and CAI Engines for the Automotive Industry*. Woodhead Publishing Limited, 2007.

[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] 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 Paper 790501*, 1979.

[9] 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 Paper 790840*, 1979.

[10] D. Flowers, S. Aceves, R. Smith, J. Torres, J. Girard, and R. Dibble. Hcci in a cfr engine: Experiments and detailed kinetic modeling. *SAE Technical Paper 2000-01-0328*, 2000.

[11] A. C. Alkidas. Combustion advancements in gasoline engines. *Energy Conversion and Management*, 48(11):2751–2761, 2007.

[12] C. H. Lee and K. H. Lee. An experimental study of the combustion characteristics in scci and cai based on direct-injection gasoline engine. *Experimental Thermal and Fluid Science*, 31(8):1121–1132, 2007.

[13] E. Sigurdsson, K. M. Ingvorsen, M. V. Jensen, S. Mayer, S. Matlok, and J. H. Walther. Numerical analysis of the scavenge flow and convective heat transfer in large two-stroke marine diesel engines. *Applied Energy*, 123:37–46, 2014.

[14] K. M. Pang, N. Karvounis, J. H. Walther, and J. Schramm. Numerical investigation of soot formation and oxidation processes under large two-stroke marine diesel engine-like conditions using integrated cfd-chemical kinetics. *Applied Energy*, 169:874–887, 2016.

[15] E. Mattarelli, C. A. Rinaldini, and M. Wilksch. 2-stroke high speed diesel engines for light aircraft. *SAE International Journal of Engines*, 4(2):2338–2360, 2011.

[16] M. I. Lamas and C. G. Rodríguez Vidal. Computational fluid dynamics analysis of the scavenging process in the MAN B&M 7S50MC two-stroke marine diesel engine. *Journal of Ship Research*, 56(3):154–161, 2012.

[17] J. Benajes, R. Novella, D. De Lima, P. Tribotté, N. Quechon, P. Obernesser, and V. Dugue. Analysis of the combustion process, pollutant emissions and efficiency of an innovative 2-stroke hsdi engine designed for automotive applications. *Applied Thermal Engineering*, 58(1-2):181–193, 2013.

[18] M. Lapuerta, O. Armas, and J. J. Hernández. Diagnosis of di diesel combustion from in-cylinder pressure signal by estimation of mean thermodynamic properties of the gas. *Applied Thermal Engineering*, 19(5):513–529, 1999.

[19] M. Lapuerta, O. Armas, and V. Bermúdez. Sensitivity of diesel engine thermodynamic cycle calculation to measurement errors and estimated parameters. *Applied Thermal Engineering*, 20(9):843–861, 2000.

[20] 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.

[21] J. J. Lopez, R. Novella, J. Valero-Marco, G. Coma, and F. Justet. Evaluation of the potential benefits of an automotive, gasoline, 2-stroke engine. *SAE Technical Paper 2015-01-1261*, 2015.

[22] J. Benajes, J. J. Lopez, J. Valero-Marco, G. Coma, and F. Justet. Design and evaluation of an automotive, low cost, gasoline, 2-stroke engine. In *The low CO_2 spark ignition engine of the future and its hybridization, SIA Powertrain Versailles*. SIA, Société des Ingénieurs de l'Automobile, 2015.

[23] J. Benajes, J. J. Lopez, J. Valero-Marco, G. Coma, and C. Libert. Transition between si and cai operating modes in an automotive, low cost, gasoline, 2-stroke engine. In *The low CO_2 spark ignition engine of the future and its hybridization, SIA Powertrain Versailles*. SIA, Société des Ingénieurs de l'Automobile, 2017.

[24] J.M. Luján, H. Climent, R. Novella, and M.E. Rivas-Perea. Influence of a low pressure egr loop on a gasoline turbocharged direct injection engine. *Applied Thermal Engineering*, 89(5):432–443, 2015.

[25] H.J. Curran, P. Gaffuri, W.J. Pitz, and C.K. Westbrook. A comprehensive modeling study of iso-octane oxidation. *Combustion and Flame*, 129(3):253–280, 2002.