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Two strokes Diesel engine - Promising solution to reduce CO2 emissions

Pascal TRIBOTTE[a], Frederic RAVET[a], Vincent DUGUE[a], Philippe OBERNESSER[a], Nicolas QUECHON[a]
Jesus BENAJES[b], Ricardo NOVELLA[b], Daniela DE LIMA[b]
a-Renault SAS, 1 Avenue du Golf, 78288, Guyancourt, FRANCE
b-CMT-Motores Térmicos, Universitat Politècnica de València, Camino de Vera s/n, 46022 Valencia, SPAIN

Abstract

Two-stroke engines have dropped out of the automobile market for a long time due to severe drawbacks. Unfortunately, the comparison with the performances of four-stroke engines was not in favour of two-stroke ones. Nevertheless, the needs of a more compact engine with a better ratio of the mass and size versus power motivated research efforts at the beginning of the 90’s. Regrettably, these efforts did not result in commercial success and automobile manufacturers kept four-stroke engine architecture as base architecture.
However, the two-stroke engine is a highly favourable concept for downsizing and cost reduction by reducing the number of cylinders without NVH penalties. All that added to the maturity of CFD calculations and the availability of high power electronic for control and fuel injection encouraged the Renault research division to have a closer look into its architecture.

The study deals with a two-cylinder Diesel engine based on two-stroke valves engine. Air admission and exhausts gas are done through four valves per cylinder; fuel injection is done through ten holes nozzle at 1800 bar of pressure. The company Delphi, partner of the project, provided the injection system.

The displacement of engine is 730 cm3 and the engine is designed for a range of power of about 35-50 kW and a range of torque of 110 – 145 N.m.

The design of the scavenging was achieved with the help of 3D simulation based on the best 3D simulation tools available. More than 250 calculations were completed to determine the best design of the cylinder-head reaching the objective of scavenging performances. At the end of this step, the best compromise was determined between the mass of fresh gas and the mass of burnt gas in the cylinder respecting combustion and engine efficiency criterias.
Air supercharging system was designed in cooperation between the partners Renault, Le Moteur Moderne and University of Technology of Prague.
The injection and combustion testing and optimisation were done on a one cylinder engine at CMT, University of Valencia, partner of the project. The set of parameters was explored and optimised to reach the best compromise between the combustion efficiency and the target of pollutant emissions to reach the EURO 6 standards.
Finally, the optimisation of the engine, including the air loop system, calibration and control is planed in 2012 and will be done at IFPEN, partner of project.

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The objectives, the design process, the major technical breakthrough as well as its detailed results will be presented in this paper.
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Project full title: POWERtrain for Future Light-duty vehicles

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http://www.powerful-eu.org/

Contacts : Pascal.tribotte@renault.com

Keywords: Type your keywords here, separated by semicolons ;

List of Symbols

EGR	External Gas Recirculation
EU	European Union
IGR	Internal Gas Recirculation
IMEP	Indicated Mean Effective Pressure
HCCI	Homogeneous Combustion Compression Ignition
HSDI	High Speed Direct Injection
NEDC	New European Driving Cycle
NVH	Noise, Vibration and Harshness
OHV	Over head Valves
TDC	Top Dead Center
VVT	Variable Valve Timing
VGT	Variable Geometry Turbine

1. Motivations

In 2007 and 2008, European Council decided of commitments to fight climate change and promote renewable energy. Among these, there are proposed legislation establishing mandatory limits on CO2 emissions from new passengers cars and promoting the use of renewable sources. The CO2 cars proposal is the most important element of EU's strategy to improve the fuel economy of cars, which account for about 12% of the EU's carbon emissions. Under the CO2 cars proposal, the average CO2 emissions for new cars in the EU would reach the objective of 130 g/km in 2012. This objective must comply with the more and more stringent regulation on pollutant emissions such as EURO 6 standards and the future EURO 7. For compression ignition engine (CI), such as Diesel, improve the fuel consumption, meet the target of pollutant emissions and limit the over cost of the engine are one of the most difficult challenge for the last decades. Renault decided to investigate an innovative solution: air valves two strokes engine. First, about 20% of fuel consumption saving has been already identified in comparison with a classical four strokes engine. In addition, two stroke engine open the possibility to overcome classical limitations of advanced combustion mode as homogeneous one. So, two strokes engine can take a combination of the advantages in fuel consumption and low emissions combustion [1, 5].

2. Architecture / Engine characteristics and architecture

2.1. Architecture requests

The Renault K9K is taken as our reference (1,46 liters, 4 cyl. Diesel), and all our target are taken in comparison to this reference. We defined our target as follow :

20% CO2 improvement compared to K9K, on NEDC cycle ; EURO6+

No or minimum investments.

Lower cost, weight and volume.

Better or equivalent NVH.

Same durability.

Better or equivalent torque response than K9K engine (low power spec.).

The basic idea behind the chosen architecture is to downsize more than the state of the art, and enable the access to high efficiency, low pollutant combustions. As we had a recent history of HCCI 4 stroke, where one of the identified shortcoming was noise, we started investigations on 2 stroke cycle as a way to have the same power with lower IMEP in HCCI mode [2]. We chose to concentrate on the 35kW to 50kW power range, where it is extremely difficult for Diesel engines to be competitive (cost, consumption, NVH, customer value). Cylinder-exhaust-ported engines were rejected for reliability and investment reason. Uniflow (or more exotic) architectures were rejected for investment reason.

2.2. Chosen architecture

description is detailed in table 1 :

Power, Torque	45 kW, 145 N.m
Capacity	730 cm3
Maximum rpm, max power rpm	4000 rpm, 3500 rpm
Bore x Stroke	76mm, 80,5mm
Number of cylinder, number of valves	2 cyl, 4v per cyl
Type of scavenge	Poppet valve loop scavenge
Injection	1800 bar, 10 holes; 80µ
Boost	T/C + Supercharger

2.3. Remarks on the chosen architecture

Potentially, this architecture has the same NVH behavior as a 4 cylinder 4 stroke engine [10], smaller, lighter, downsized by a factor of two, and the VVTs should allow for the control of the Internal Gas Residuals (IGR) and compression ratio, allowing for some combustion control.

But some questions are fully open

- How to achieve a good scavenge w/o ruining the combustion chamber design ?

- How to burn well with an aerodynamic governed by scavenge ?

- What kind of mechanics to withstand the 2 stroke particularities ?

3. Air management

The design of the air management loop for POWERFUL [4] two-stroke engine is driven by the following points: downsizing to reduce fuel consumption, two-stroke scavenging process, and reduction of NOx (nitrogen oxides) emissions. These points are explained in details below.

The advantage of engine downsizing is now proven to reduce fuel consumption. One issue to downsize engines is to increase enough the intake air density. Our target is to meet a 4.5 bar absolute inlet pressure at peak power.

In two-stroke cycle engines, a power stroke happens every cycle. The time to supply the engine with fresh air and to clear the cylinder from burned gases is greatly reduced (compared to four-stroke engines). The intake and exhaust phases are thus, very short and take place simultaneously. This phenomenon is called scavenging. Then one of the most challenging aspect for the two-stroke engine operation is to supply the engine with fresh air at high-enough pressure whereas the exhaust valves are open. The fresh air stays in the cylinder while the burned gases from the previous cycle are pushed out. In other words, the intake pressure has to be higher than the exhaust pressure over the complete range of engine speed.

The engine has to be supplied with EGR at part-load operation to reduce the temperature of combustion and consequently NOx emissions. However, two-stroke cycle implies to operate with a high rate of internal gas recirculation (burned gases which are not expelled from the cylinder during the exhaust phase). IGR also participates to the reduction of combustion temperature and thus on NOx reduction, so we need to manage the balance between IGR and EGR.

Considering these guidelines and especially the need of a 4 to 5 bar absolute intake pressure at maximum power, focus is given to the two boosting stages configurations. A list of possible configurations was set up at the beginning of the study. They are represented in fig. 1.

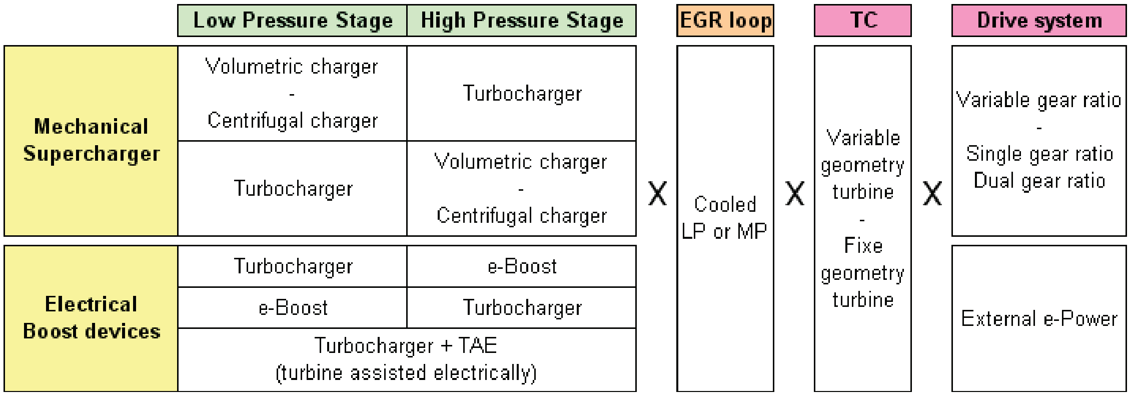


Fig. 1. Air loop management configuration investigation overview

To orientate technical solutions for the air loop system, a large amount of 1D stationary simulations had been carried out. The base model was calibrated on preliminary test bench results and includes several assumptions related to scavenging, combustion and thermal characteristics. The comparison of the performances of each configuration is performed on operating points at full load and part load. The part load points have been chosen to be representative of the vehicle consumption on NEDC cycle. [11, 12]

Table 2. Part load points

		Point 1	Point 2	Point 3	Point 4	Point 5	Point 6
Engine speed	RPM	1250	1000	1500	2000	1500	2500
Torque	N.m	25	50	50	50	90	130
BMEP	bar	2,15	4,3	4,3	4,3	7,7	11,2
EGR rate	%	20	20	20	20	15	15
Richness	-	0,4	0,6	0,6	0,6	0,7	0,7

Table 3. Full load points

		Point 1	Point 2	Point 3	Point 4	Point 5	Point 6
Engine speed	RPM	1000	1500	2000	2500	3000	3500
Torque	N.m	116	143	143	143	143	122
BMEP	bar	10	12.3	12.3	12.3	12.3	10.5
EGR rate	%	0	0	0	0	0	0
Richness	-	0,9	0,7	0,7	0,7	0,7	0,7

However, even with two boosting stages, these requirements lead to unusual high pressure ratios for low mass flow rates. Therefore, at least one of the boost components is out of range of usual providers availabilities. Thus in parallel to simulation task, the feasibility of the different tested configurations is evaluated toward consultations.

Finally, the best trade off regarding air loop feasibility, power achievement and fuel consumption is a configuration combining a mechanical volumetric charger set downstream to a turbocharger with a waste-gate. The charger is driven by a mechanical dual speed system, equivalent to a gearbox with two gear ratios. Even if the dual speed system provides more flexibility than a single gear ratio transmission, the engine airflow needs to be controlled with a bypass of the charger.

The main drawback of this solution is the high and efficient air cooling necessity between both boosting stages to low down the volumetric charger outlet temperature (150°C limitation). Moreover, the procurable chargers are larger than required (250cm3 instead of idealistic 150cm3) which would lead to operate in non-optimal efficiency areas.

For all architectures, the requirements concerning the turbine with a waste-gate are similar. The situation is identical with variable geometry turbine, but no device covering the full mass flow range seems procurable. Compared to a turbine with a waste-gate, a VGT would roughly afford 1% fuel benefits. Still, for full loads, a VGT would bring more engine control facilities to satisfy simultaneously scavenging and boost requirements.

Concerning configurations with a volumetric charger, a variable drive system shows no benefit compared to a dual drive system essentially due to higher mechanical friction losses. In case of configurations with a centrifugal charger, a mechanical single drive transmission is clearly insufficient to ensure the full load target torque.

According to simulations, for final selected solution, a mid pressure EGR loop leads to lower fuel consumption (fig. 2) [6] .

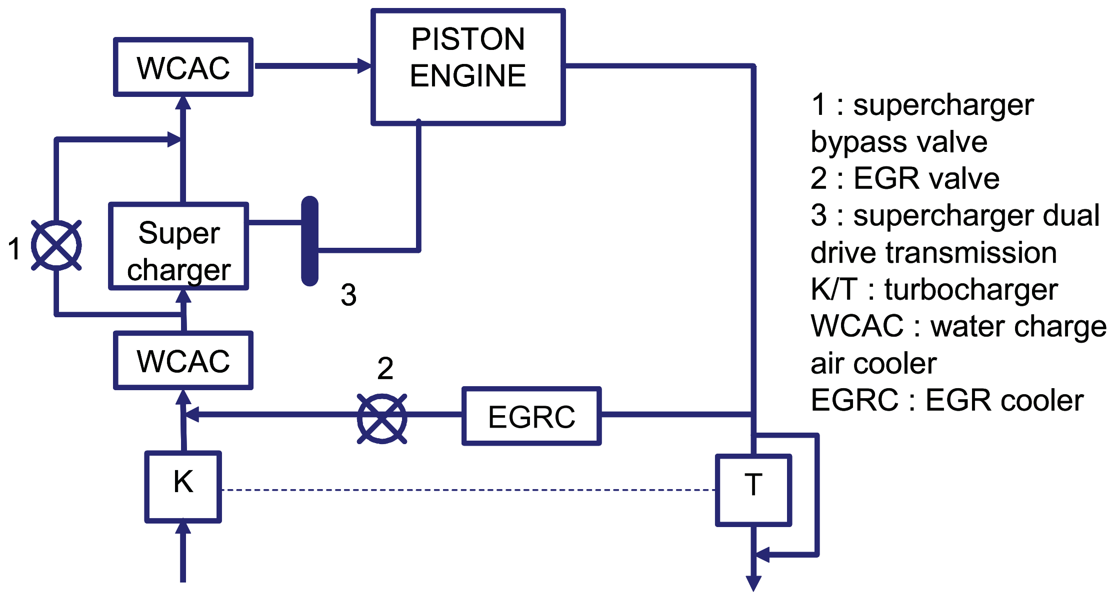


Fig. 2. Final air management architecture

4. Scavenging

Two main architectures are encountered in two-stroke engines:

- The “Valve Loop system” is a four-stroke like valve engine where intake and exhaust are performed through poppet valves. Therefore, gases enter and exhaust the cylinder throughout the head of cylinder. Obviously, the scavenging is not naturally efficient. This solution is advantageous on a mechanical point of view. Moreover, this system leads to the same possibilities of controlling the valve lifts than on a four-stroke engine.
- In the Uniflow system, intake is done through port in the cylinder, and exhaust is done trough port/valves in the head of cylinder. This system is more efficient for scavenging and load circulation; but incompatible with the existing production lines.

Considering a downsized Diesel engine (heavily loaded engine), the project POWERFUL [3] was directed towards a reliable and cheap to industrialize solution. Considering actual direct injection, up-to-date / innovative supercharging system and variable valve timing possibilities, it was supposed easier to cop with the scavenging and trapping difficulties, native to poppet-valve systems.

In this section, scavenging will be explored. However, this information is not available by experiment. Only integrated results of scavenging efficiency or trapping ratio may be exploited based on the experiments. Therefore, 3D simulations are setup to evaluate and look at the way gases are scavenged through the system during the cycle. These 3D simulations use the CFD code Star-CD (finite volume method). In these simulations, combustion is mimicked in order to save computation time. Therefore, during the combustion phase, source terms are used in order to produce the right amount of energy and to transform all fresh air and fuel into burned gases [7]. Only three scalars are therefore transported: fresh air, fuel and burnt gases. A usual modeling package for compressible aerodynamic (piso algorithm) is used with a standard k-epsilon turbulent model [8].

4.1. Definitions

In this section, we recall some usual definitions useful in a two-stroke engine context (cf. table 4).

Table 4: Usual definitions dedicated to two stroke engines

Name	Definition	Formula
Charging efficiency	enclosed mass of fresh gas in the cylinder divided by mass of air at ambient conditions that is possible to enclose in the cylinder volume at BDC	$m_{FG}^{cyl}/m_{FG}^{ref}$
Delivery ratio or volumetric efficiency	mass of fresh gas coming from intake duct divided by mass of air at ambient conditions that is possible to enclose in the cylinder volume at BDC	$m_{FG}^{int}/m_{FG}^{ref}$
Trapping efficiency or trapping ratio	charging efficiency divided by delivery ratio	$m_{FG}^{cyl}/m_{FG}^{int}=1-m_{FG}^{exh}/m_{FG}^{int}$
Scavenging efficiency	mass of fresh gas enclosed in the cylinder divided by the total mass enclosed in the cylinder (fresh gas + residual burnt gas)	m_{FG}^{cyl}/m_{cyl}

In order to evaluate the way the scavenging operates during the cycle we will usually present it giving the mass fraction of burned gases exhausted as a function of the mass fraction of burned gases in the cylinder (cf. fig. 3):

- On the perfect scavenging line, intake fresh gas pushes perfectly exhaust gases outside the cylinder.
- On the perfect mixing line, at each time intake fresh gas and in-cylinder gases are perfectly mixed and this mixing is exhausted.
- On the exclusive by-pass line, intake fresh gas is systematically exhausted.
- Between these lines, a favorable zone and an unfavorable zone take place.

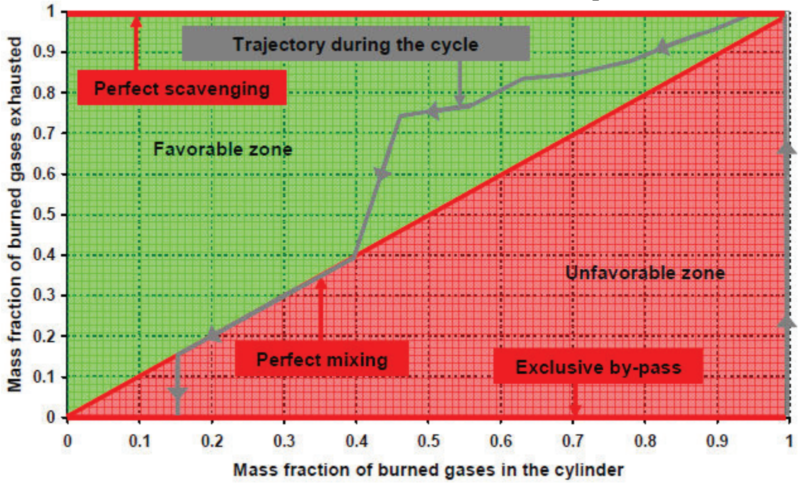


Fig. 3: Scavenging efficiency diagram

4.2. Cylinder head definitions

A reference was done experimentally, based on a 1.5L dci (4-strokes, 4-cylindres) Renault engine operating in 2-strokes mode with few modifications (4 valves/cylinder). But the measurements highlighted that on the one hand the permeability was too weak and on the other hand the scavenging was really unfavorable. 3D Simulations evaluate the evolution of the scavenging efficiency during a cycle and show the same conclusion (cf. fig 4).

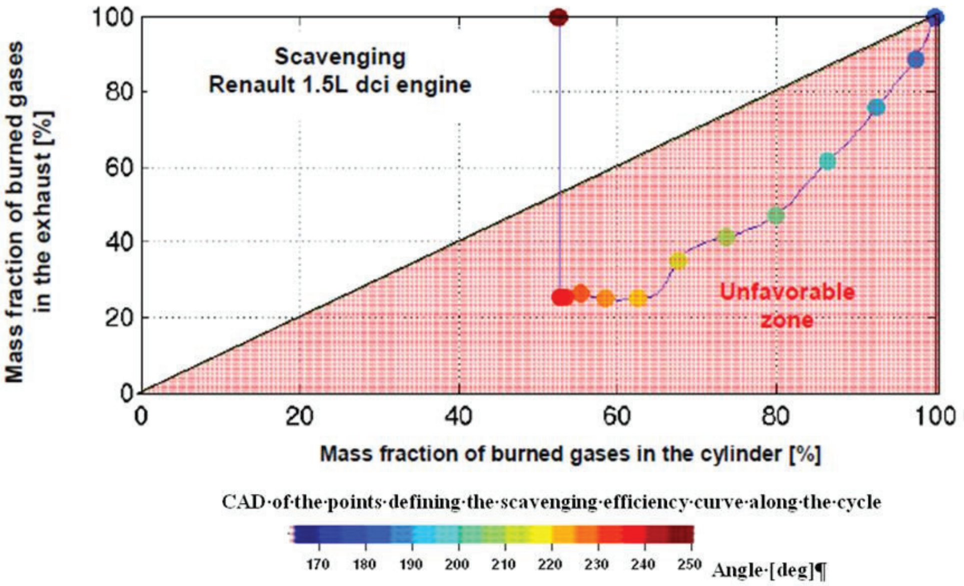


Fig. 4: Scavenging efficiency on four stroke like basic head of cylinder

Three different kinds of evolution where tested based on this reference:

- Duct evolution
- Mask at the intake
- Staged roof

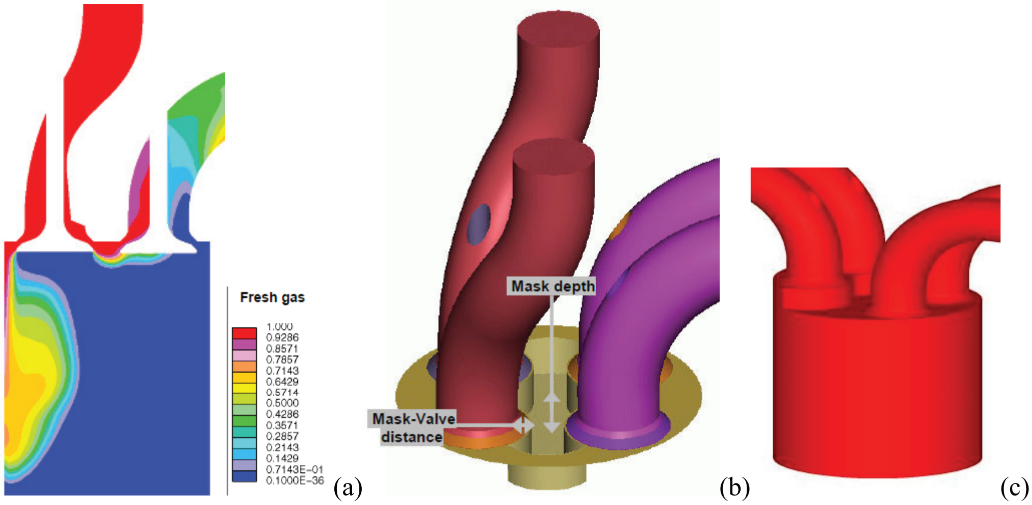


Fig. 5: Head of cylinder possible evolutions (intake on left, exhaust on right), (a) duct, (b) mask, (c) roof/staged head of cylinder

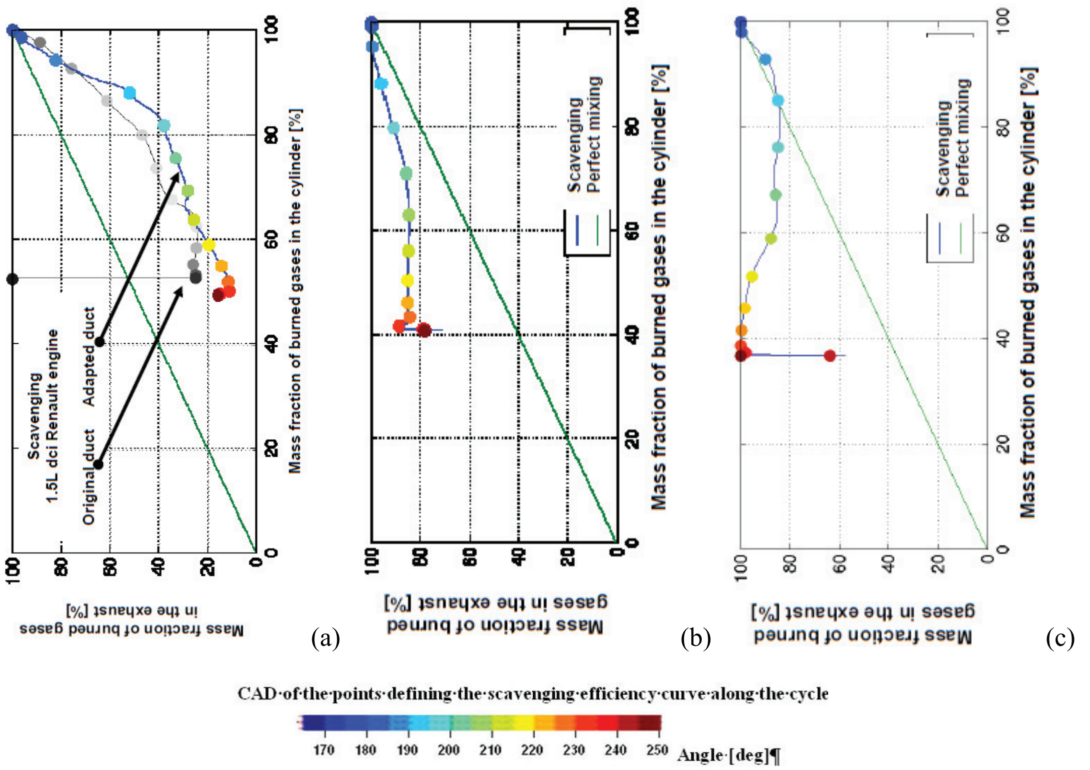


Fig. 6: Scavenging efficiency along the cycle, (a) duct, (b) mask, (c) roof/staged head of cylinder

At first, the duct was modified to be more favorable to scavenging (cf. fig. 5 (a)). This change did not modify the scavenging (cf. fig. 6 (a)). As a result, it was decided to investigate masking (cf. fig. 5 and 6). Almost the same kind of conclusion is achieved both for the intrusive mask and for the staged roof. Masking the intake considerably improve scavenging. At low lift condition, the distance between the mask and the valve is of first order to improve scavenging. When valve-mask distance is small enough (blanking off intake in the direction of the exhaust), it is not necessary to have a mask more intrusive than the maximum lift. However, if this distance is less important, the mask is baffling the intake flux only in the direction of the exhaust. As a result, it may be necessary to have a deeper mask. Between the roof solution, the staged head of cylinder and the mask, the following proposition was designed (cf. fig. 7). This solution is a compromise mostly improving scavenging, but also trying not to limit too much air flux through intake, feasible and not too distorted to preserve the combustion chamber.

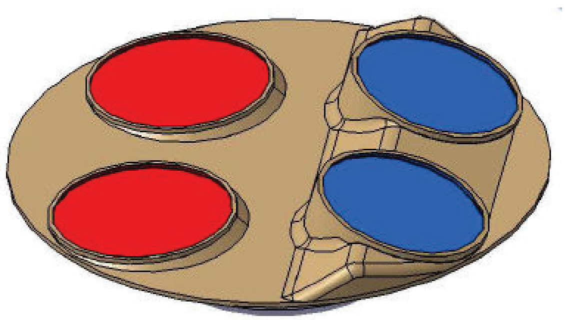


Fig. 7: Shape of the final head of cylinder [9]

The final cylinder-head concept is finally tested experimentally on a multi-cylinder real engine (with some acoustic shortcomings). Five experimental points are chosen in the experimental database (cf. table 5), fully exploited with 0D/1D tools and reproduced with 3D simulations. 3D simulation is used to evaluate the scavenging efficiency along the cycle, and how it evolves in the different conditions simulated. Two loads, two valve lift profiles (cf. fig. 8) and two speeds are investigated.

Table 5: Simulated points to characterize scavenging

Speed [rpm]	Load (IMEP) [bar]	Torque [N.m]	Cam lift law
3000	4	nc	D
3000	6	100	D
3000	6	100	D3
1250	6	100	D
1250	6	100	D3

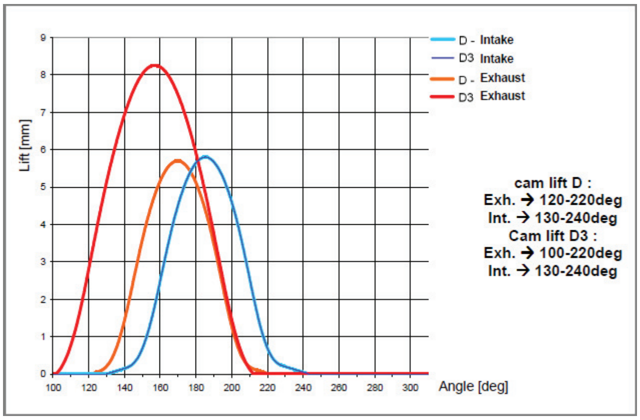


Fig. 8: Cam lift profiles

Finally, tendencies are mostly coherent between 3D and experiments post-treated by 1D, when the differences between two points are sensible (fig. 10). The backflow effects are probably not taken into account exactly in the same way between those tools (3D, experiment + 1D), as the less coherent point between the 5 points is one with a strong backflow (cf. fig. 9 and 10, point 1250rpm, cam lift D, 6 bar). In addition, it seems clear that the cylinder-head characterizes the shape of the scavenging (cf. fig. 9). However, cam lift profile and speed define how much of scavenging characteristic curve that will be used. It results in a scavenging efficiency and a trapping ratio (both are mostly correlated).

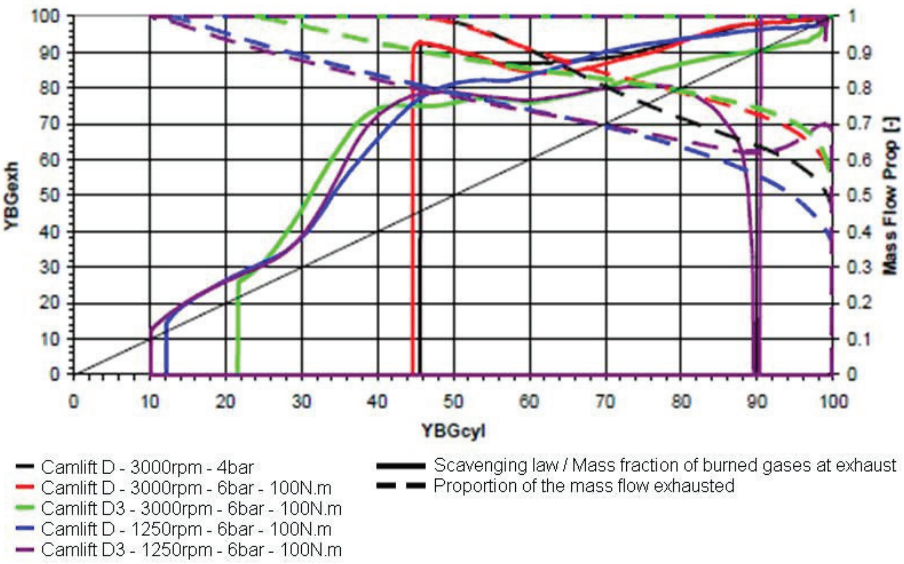


Fig. 9: Resulting scavenging evolutions

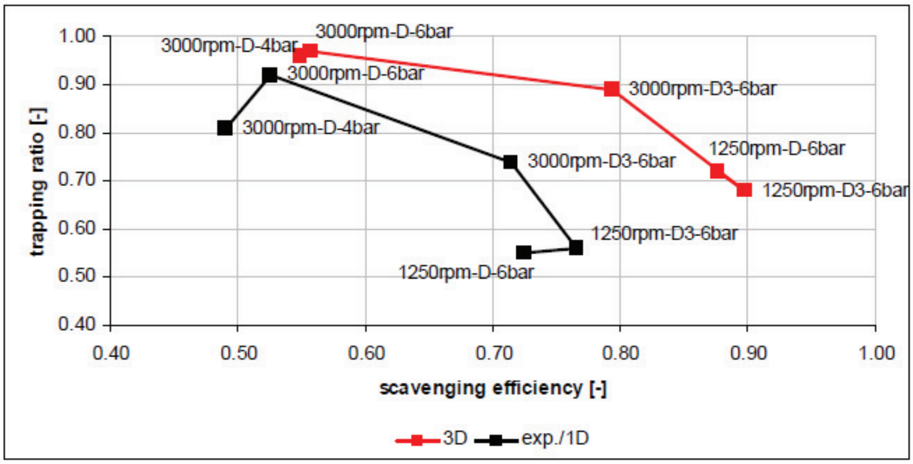


Fig. 10: Trapping ratio and scavenging efficiency

5. First results on single cylinder

The feasibility of the innovative two-stroke HSDI engine concept was evaluated by testing two operating conditions under the strictly controlled environment provided by a single-cylinder engine experimental facility. The main objective consists of confirming the potential of this engine architecture for providing suitable air management characteristics, evaluated in terms of trapping ratios and IGR levels, and competitive results in terms of engine performance and fuel consumption, with acceptable levels of pollutant emissions.

The methodology followed along this research activity was designed to isolate the optimization of the air management settings from that of the injection settings. This optimization is focused on attaining the best fuel consumption, but keeping the trapping ratio at realistic values. Therefore, at each operating condition, a design of experiments was initially performed to find the best settings for the air management related parameters, which are the intake pressure, the pressure drop across the engine (Δp) and the intake/exhaust valve timings, keeping constant the injection settings. In a second step, injection timing was swept to check the influence of combustion phasing not only in engine performance and emissions, but also in air management conditions.

5.1. Results at 1250 rpm and imep 3.1 bar

According to the modeling results previously discussed in section 4, the air management characteristics of the engine operating at low engine speeds and loads is compromised, decreasing the trapping ratios and increasing the IGR levels compared to other operating conditions. Thus, this first activity will provide an interesting feedback of the engine performance in this critical operating condition. Following the methodology, initially a design of experiments was performed considering the air management parameters included in Table 6, together with their ranges of variation.

Table 6. Air management related parameters included in the design of experiments, including their ranges of variation.

Air management related parameter	From	To
Intake valve timing [cad]	238	258 (+20)
Exhaust valve timing [cad]	222	242 (+20)
Intake pressure [bar]	1.3	1.5 (+0.2)
Δp [bar]	0.2	0.4 (+0.2)

The results obtained after testing the combination of parameters included in the design of experiments are useful for providing an overview of the behavior of the trapping ratio and the IGR, and their relations with engine efficiency. Fig 11 shows how at this operating condition, the levels of trapping ratio attained are comprised between 50% and 90%, while the IGR moves between 45% and 60%, being both inside the expected ranges. Additionally, Fig 11 confirms how the settings that decrease the trapping ratio, such as increasing the valve overlap or the pressure drop across the engine, have a positive impact on the scavenge process reflected by the reduction in IGR. However, a practical lower limit for IGR of 45% has been found, being impossible to go below this level even accepting further reductions in trapping ratio.

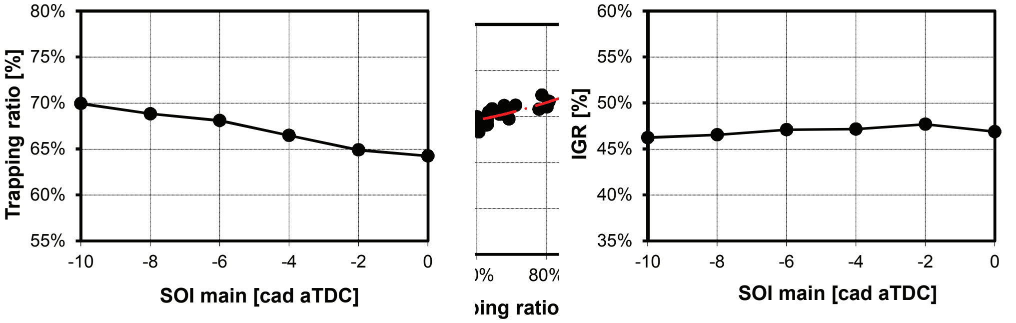


Fig. 11. Relation between the trapping ratio and the IGR level.

Describing the relation between the air management parameters, trapping ratio and IGR, and the engine efficiency (isfc) is also of interest. Fig 12 confirms the key role played by the air management on the engine efficiency and, as a result, on isfc. According to Fig 12, engine efficiency is intrinsically related with both trapping ratio and IGR, so acceptable levels of isfc are only attainable by decreasing both trapping ratio and IGR. However, as in the case of the IGR, 208 g/kWhi was the lower limit found for the isfc and values below this cannot be reached even reducing the trapping ratio below 60%.

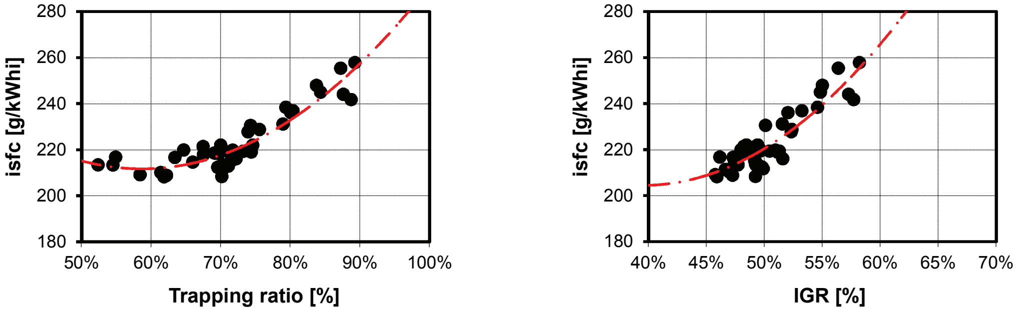


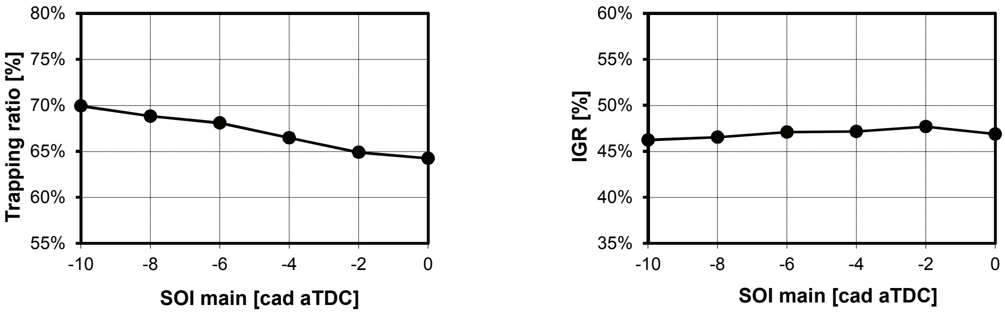
Fig. 12. (a) Relation between the trapping ratio and the isfc; (b) Relation between the IGR and the isfc

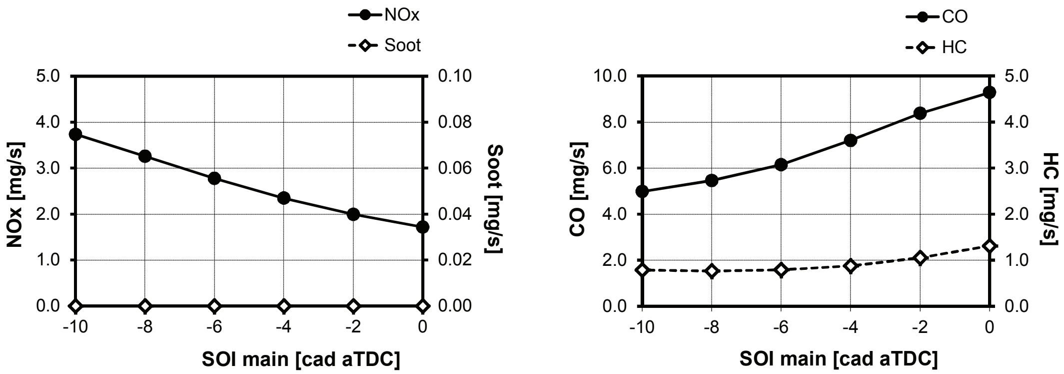
After the optimization process, the final air management settings corresponds to intake valve timing 256 cad, exhaust valve timing 242 cad, intake pressure 1.3 bar and Δp 0.3 bar. With these settings, the measured trapping ratio is 67% and IGR is 47% so the point can be easily compared to the results included in Fig 11 and Fig 12 to see how it is coherent with them.

Observing the optimum settings in detail, the exhaust valve timing corresponds to the maximum exhaust stroke duration, corroborating the important influence of lengthening the expansion stroke to increase the engine efficiency up to the desired levels. Intake valve timing is also shifted to the compression stroke to avoid extreme valve overlaps, which results in a non-useful reduction in trapping ratio since neither the IGR nor the isfc will decrease.

As a second step, the influence of injection timing was investigated by shifting both the pilot and main injections from -18/-10 to -8/0 cad aTDC, so the combustion phasing (50% fuel burnt angle) shifts from -0.4 to 9.4 cad aTDC. Results included in Fig 13 demonstrate the moderate effects of injection timing on air management, since the trapping ratio changes by less than 5%, while the IGR is kept almost constant.

Fig. 13. (a) Effect of injection timing on trapping ratio; (b) Effect of injection timing on IGR.





Then, this result support the methodology followed in this investigation since air management and injection settings can be sequentially optimized.

In terms of pollutant emissions, Fig 14 shows how NOx emissions follow the conventional trend of being reduced as the injection timing is delayed. However, despite the high levels of IGR the NOx emissions are not as low as expected, so the benefit of reducing in-cylinder gas oxygen concentration provided by IGR competes with the drawback of increasing the in-cylinder gas temperatures since the IGR cannot be externally cooled. As a consequence, probably external and cooled EGR should be introduced to attain NOx emissions levels below EURO VI, since retarding the injection timing is not sufficient and it has negative impact on engine efficiency (see Fig 15). On the contrary, soot emissions are always extremely low and almost negligible even at the most retarded injection timing, so high tolerance to EGR is expected without producing unacceptable levels of soot. Results in terms of CO and HC show how these pollutants increase by shifting the injection towards later timings.

Fig. 14. (a) Effect of injection timing on NOx and soot emissions; (b) Effect of injection timing on CO and HC emissions.

Fig 15 evidences the moderate influence of injection timing on engine efficiency, at least compared to the effect of air management settings previously observed in Fig 12. The improvement provided by the optimum combustion phasing in isfc is less than 2% considering both advanced/retarded phasing. Additionally, retarding the injection timing results in a slight reduction in combustion efficiency due to the increment in CO and HC emissions, previously shown in Fig 14.

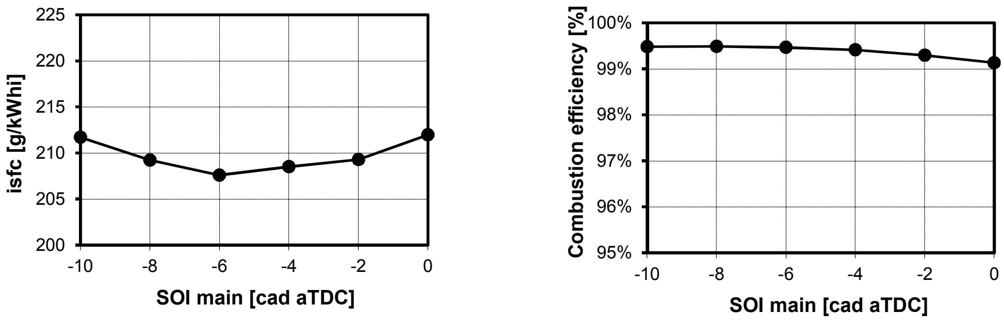


Fig. 15. (a) Effect of injection timing on isfc; (b) Effect of injection timing on combustion efficiency.

As a final overview of the results at this low engine speed and load condition, an acceptable air exchange process was experimentally confirmed, and the close relation between the air management parameters and the engine efficiency was established. Promising results have been obtained in terms of pollutant emissions and engine efficiency after optimizing also the injection settings, but probably additional strategies will be required to attain EURO VI limits, such as introducing EGR.

5.2. Results at 2000 rpm and imep 5.8 bar

The modeling results previously discussed in section 4 predicts and improvement in the air management characteristics of the engine at this medium engine speed and load compared to those at the previous operating condition, so higher trapping ratios and lover levels of IGR are expected. Thus, the design of experiments was performed considering the air management parameters and ranges included in Table 7.

Table 7. Air management related parameters included in the design of experiments, including their ranges of variation.

Air management related parameter	From	To
Intake valve timing [cad]	224	244 (+20)
Exhaust valve timing [cad]	208	228 (+20)
Intake pressure [bar]	1.7	1.9 (+0.2)
Δp [bar]	0.4	0.6 (+0.2)

Fig 16 shows how the levels of trapping ratio attained experimentally at this operating condition are undoubtedly higher than those obtained at the previous operating condition. In this case, trapping ratio ranges between 85% and 95% (despite the higher Δp included in this design of experiments), while the lower IGR value was 32% and the highest was 45%. Then, the scavenge strategy selected for this engine concept, based on keeping the OHV configuration, has been proven to be fully suitable for assuring the engine breathing in medium to high loads. Additionally, analog relations to those shown in Fig 12 have been found between the air management parameters and the engine efficiency, so as previously, so decreasing both trapping ratio and IGR is still mandatory to attain acceptable levels of isfc.

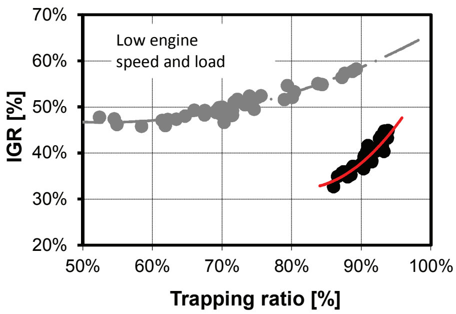


Fig. 16. Relation between the trapping ratio and the IGR level.

The best settings obtained through the optimization process set the intake valve timing at 244 cad, the exhaust valve timing at 228 cad, the intake pressure at 1.8 bar and the Δp at 0.6 bar. With these settings, the measured trapping ratio is 87% and IGR is 33% so the point can be found easily in Fig 11 and it is fully compatible with the results shown there. The best valve timing configuration is still what allows the most extended expansion stroke, highlighting its relevance to get competitive fuel consumption levels.

Following the research flow, the injection timing was optimized to check the influence of combustion phasing on engine breathing characteristics, pollutant emissions and engine performance, sweeping it from SOI pilot/SOI main -19/-13 to -13/-7 cad aTDC, so combustion phasing is comprised between -1.8 and 3.4 cad aTDC. Initially, negligible effects were found in air management parameters, so trapping ratio and IGR were kept almost constant, as it was evidenced by Fig 13. So this result corroborates experimentally the low relation between injection and air management settings.

Regarding pollutant emissions, Fig 17 shows how NOx emissions follows the usual trend and they decrease when the injection timing is delayed, but at this operating conditions seems still necessary the induction of external EGR to attain EURO VI levels, corroborating the impossibility of reaching the required reduction in NOx only by IGR, at least with an acceptable engine efficiency. Soot emissions are still controlled and kept at relatively low levels even at retarded injection timings, while CO and HC emissions are higher than desired, but they do not sharply increase by delaying combustion phasing.

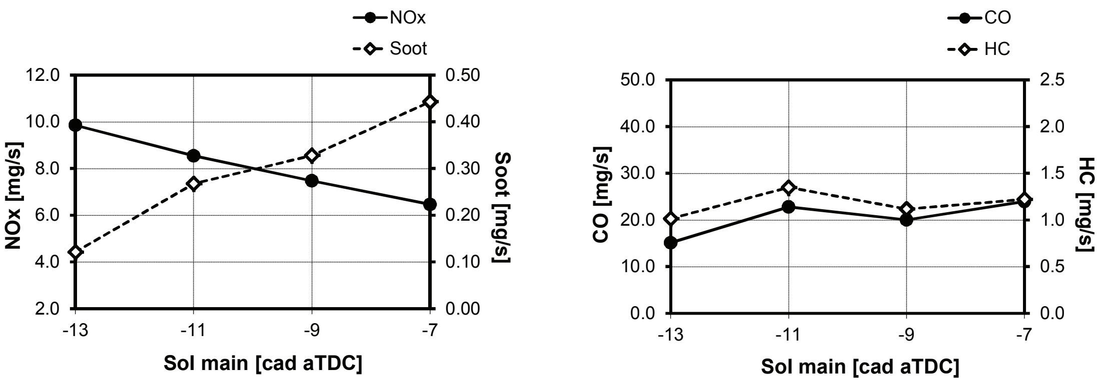


Fig. 17. (a) Effect of injection timing on NOx and soot emissions; (b) Effect of injection timing on CO and HC emissions.

Fig 18 shows the influence of injection timing and combustion phasing on isfc and combustion efficiency. These results are coherent with those observed in Fig 15 since in both cases the benefits attained by optimizing combustion phasing were lower than 2%. Then, attaining a breakthrough in terms of reducing the fuel consumption is associated to improvements the air management characteristics, while injection system only plays a secondary role. The results of this research included in Fig 18 also prove the feasibility of the engine concept and its suitability for producing efficient combustion processes at different engine loads and speeds.

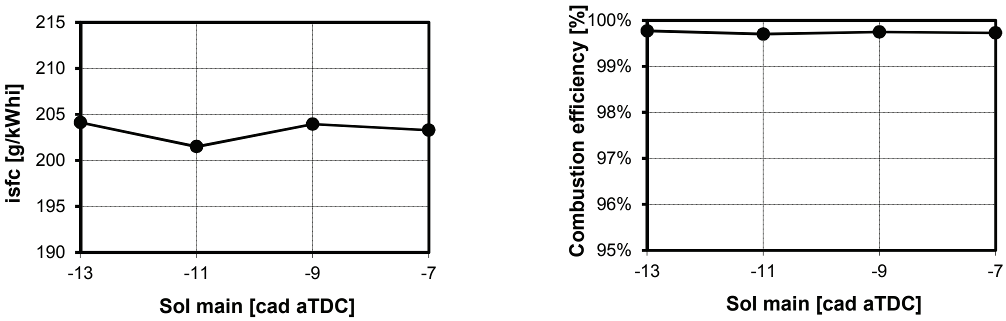


Fig. 18. (a) Effect of injection timing on isfc; (b) Effect of injection timing on combustion efficiency

As a final summary of the information gathered at this medium engine speed and load condition, an improvement in air exchange process was experimentally confirmed compared to the lower engine speed and load condition, while the close relation between the air management parameters and the engine efficiency was corroborated. Pollutant emissions and engine efficiency results attained after optimizing the injection settings are encouraging, but probably additional strategies such as introducing EGR will be still required to attain EURO VI limits.

6. Conclusion and future work

The development of a two strokes engine was motivated by the target of 20 % CO2 reduction on real cycle ; and the more and more stringent regulation in pollutant emissions. In addition, innovation should be acceptable by the car market and the introduction of expensive technological concept must be limited. In these conditions, the development of poppet-valve two strokes engine is an exciting challenge.

The development was split into two main topics: the development of the air path based on advanced super-charging concepts and the development of combustion chamber. To limit the delay and the cost on these developments ; a numerical design process was developed, based on large number of 1D and 3D simulations and coupling between these tools. Thus many configurations of air path were investigated, considering the data for many possible suppliers, to determine the one to reach the expected level of power and the highest in term of CO2 reduction. Only one configuration will be characterized and implemented on the engine. Simultaneously, the injection system and the piston profile were designed for the combustion chamber using full C3D computations. A limited number of configurations have been validated on single cylinder engine to check the isfc and the level of pollutant emissions.

As future work, the first step will be to validate the air path on specific test rig to characterize the efficiency and validate the 1D model. The second step is to manufacture a complete engine and test it. Firstly, these tests should validate the 1D approach and validate the different assumptions and secondly these tests must calibrate the engine for vehicle test to qualify the concept in regards of the expected CO2 reduction.

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