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Development of a emission compliant, high efficiency, two-valve DI diesel engine for off-road application

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Abstract

Nowadays, environmental concerns are posing a great challenge to DI Diesel engines. Increasingly tightening emission limits require a higher attention on combustion efficiency. A high efficiency Diesel engine can be developed only mastering all the parameters that can affect the combustion and, therefore, NOx and soot emissions. In this scenario, computational fluid-dynamics can prove its power guaranteeing a deeper understanding of mixture formation process and combustion.

In this work, the development of an engine in order to fulfill Tier 4i emission standard will be presented, the Tier 4i compliance must be reached without an excessive increase of the final cost of the engine. Originally, the engine was a two-valve engine supplied with a DPF, since no SCR aftertreatment is supplied, NOx emission target are achieved through external exhaust gas recirculation and retarding the start of injection.

Through combustion process simulations, performed with the CFD code KIVA3D, varying different geometric parameters and the intensity of the swirl ratio, the interaction between the swirl flow field, generated by the intake duct, the reverse squish motion, and motions aerodynamically generated by spray has been investigated leading to a better interaction between the flow field, the fuel spray and the piston bowl geometry and to the definition of a new engine lay-out. The study shows how, given the need of retarded injection for limiting NOx emission, the decrease of swirl ratio, when combined with a proper piston bowl design, allows a significant decrease of soot emissions and the achievement of Tier 4i emission standard.

The study has been validated comparing the intake phase simulations, performed with the CFD code Fire v2009 v3, followed by the combustion process performed with the KIVA3D code, with the experimental result obtained from the engine assembled following the developed design.

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Keywords: Two valve diesel engine, Swirl Ratio, Soot emission, CFD

1. Introduction

Since the design of Diesel engines became more challenging due to the environmental concerns and the emission regulations tightening, it is mandatory to develop a robust methodology for the CFD simulation of the combustion

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system in order to get the maximum benefit from in-cylinder processes. In fact, the fulfillment of all the engine targets (emission, specific consumption, power/torque, noise) can be achieved only through a deep understanding of the physical processes and the interaction between design choices and engine performances.

The next step for both European (Euro IV) and American (Tier 4) emission standard for off-road engine application (now almost aligned) will require a reduction of NOx and PM by about 90% and introduce a limit to the HC emissions.

To be able to comply with these standards upcoming DI Diesel engines must rely both on in-cylinder pollutant reduction strategies and the exhaust aftertreatment devices.

The traditional in-cylinder reduction, achieved mainly by controlling the combustion process via the use of high pressure injection system, turbo-charging, EGR and evolved control system, shows opposites tendencies between soot reduction and NOx increase (and vice versa). As in-cylinder strategies have an impact on the cost of the aftertreatment the possibility of the reduction of both the NOx and soot emission has been a focus for industrial and academic research over the last 30 years.

All the proposed techniques are based on the concept of Low Temperature Combustion (LTC) such as Homogeneous or Premixed Charge Compression Ignition (HCCI/PCCI).

PCCI [1] is a kinetically controlled combustion process of a very lean mixture obtained with very early direct injection. The mixture formation and control of the combustion process are still an open issue. To partially overcome these issues Simescu et al. [2] present a PCCI-DI concept. Applied to heavy duty engine, where only a part of the combustion is premixed and the other follows a diffusive combustion, this concept is successfully applied to light duty diesel engine using a CR system by Gary et al. [3] and experimentally reviewed against Euro IV regulamentation by Juttu et al. [4].

LTC concept require that the flame temperatures remain under 1700 K to prohibit NOx and soot formation, while at the same time allowing for premixed combustion to eliminate fuel rich combustion zones which further reduces soot formation, this is usually obtained with high EGR percentage and very early injection timing.

All the lower temperature combustion methods are capable to simultaneously reduce both NOx and soot emissions, but with a reduced fuel efficiency and an increase in both the carbon monoxide and HC emissions. Also the zones of the applicability for the low combustion method are limited with the knocking at high loads and the misfire at low.

The application to light duty engine has been extended to all the working conditions by Catania et al. [5] thanks to an optimization of the combustion chamber shape and compression ratio, and by Cursente et al. [6] with the NADI concept, a complete rethinking of the system based on the interaction and guide between the spray and the piston bowl walls. Kazuhisa et al. [7] for an automotive engine proposed a reduction of swirl and compression ratio and an increase of the number of injector holes, but for the high load and high power condition the PCCI has been substituted with a standard multi injection.

The application of all lower temperature strategies has been confined to low to medium load diesel engine.

2. Contribution

This work shows the development of a two-valve DI Diesel engine into a Tier 4i (Tier4 Interim) emission regulations complying engine by the use of the CFD simulations, according to prescribed lay-out and design constraints and to a cost-driven approach. The main challenge was to promote the emission (and BFSC) reduction, together with a slight increase of maximum power target, accomplished maintaining the engine head two-valve architecture and avoiding the installation of a De-NOx after-treatment device.

To comply with NOx emission limits without the use of a De-NOx after-treatment device, the new engine will require the use of external EGR and retarding the fuel injection, operating the main fuel injection around the top dead center, as a consequence the combustion is quite shifted during the expansion stroke.

In this work, the combustion process has been optimized following the consideration that the interaction between the in-cylinder swirl flow and the reverse squish flow might be a key controlling factor to improve air/fuel and air/soot mixing, and to reduce the raw soot emissions. The achievement of an axi-symmetric fluid dynamics condition for air/fuel mixing and combustion is mandatory in order to get the maximum benefit from:

1. The development and application of new swirl concept, avoiding overswirl effects even in low swirl flows.

2. The optimization of the piston bowl shape in order to promote the homogeneous air/fuel mixing by using the spray momentum.
3. The optimization of the injection system specifications (nozzle hole number, angle between sprays, flow rate, etc)

On the other hand the lowering of the engine cost require the maintaining the two-valve layout of the baseline engine. The two-valve arrangement has the main drawback to promote non axis-symmetric in-cylinder flow conditions, due to the valve accommodation influence on the injector location which likely penalizes any solution of spray targeting, oriented to improve the air/fuel mixing. Thus, a mandatory preliminary work has been performed in order to assess the influence of the injector location offset with respect to the bowl axis and the piston bowl offset on the engine raw emissions and IMEP.

In the second part of the work, the influence of the swirl ratio at IVC was assessed and a low swirl flow concept was found as the most effective in complying with the Tier 4i emission targets.

In the last part of the paper the configuration of the engine combustion system will be presented and compared with the baseline engine we started from by using both the CFD simulation results and the experimental data.

The results achieved will indicate that:

1. A two-valve engine head might provide a similar axis-symmetric in-cylinder flow conditions as those typical of a four-valve engine configuration.
2. Lower swirl port concept is effective to drop raw soot emissions and BSFC in engines operating with retarded fuel injection strategy required by the absence of the De-NOx after-treatment device.

2.1. Engine description

The developed engine is a four cylinder, two-valve, turbocharged, common rail DI diesel engine, targeted to both off-road and marine application, its main specifications are reported in Tab. 1. The engine was characterized by a swirl ratio at IVC of 1.8, by an injector-bowl offset of 3.5 mm and by a bowl-cylinder offset of 4.5 mm. The compression ratio and the maximum rail pressure were defined in advance and they must be considered as given.

In this paper, the results of the operating point (named Mode 1) corresponding to the maximum rated power condition (Tab. 2), which has historically proven to be the most challenging for the containment of raw soot emissions, are presented.

3. CFD simulation settings

In the current work, the CFD code KIVA3D has been used to perform the in-cylinder combustion simulations while the intake process simulations have been carried out by using the CFD code Fire v2009.

During the work, two different approaches have been followed:

1. The conceptual analysis, which is the focus of the first part of the paper and it is aimed to find the best engine head layout configuration and the target swirl ratio to be used, has been performed only by the KIVA3D code

Table 1. Engine main data

Bore	94 mm
Stroke	107 mm
Compression ratio	17.5
Number of holes	6
Spray angle	151 degree
Max injection pressure	150 MPa
Max power	77 kW@ 2300 rpm
Max torque	380 Nm@1800 rpm

Table 2. Tested conditions

Mode	Air [kg/h]	Fuel [kg/h]	Inlet pressure [bar]	EGR [%]
Mode 1	345.6	18.93	2.22	13.31

starting all the simulations at IVC. A velocity profile has been imposed to assign an initial swirl vortex with a prescribed swirl ratio (SR).

2. The calculation performed to check the final configuration of the engine before its manufacturing, which is the focus of the final part of the paper, includes the simulation of the intake process by using the CFD code Fire and the simulation of the combustion process by using the KIVA3D code. The latter starts from the fluid dynamics conditions at IVC provided by the intake stroke simulation. This complete simulation procedure was adopted in the steps focused on the optimization of the bowl shape and the spray targeting which are not showed in the present paper

The CFD code named KIVA3D is a customized version of the KIVA-3 code [8] developed at the University of Bologna since 1996. The turbulence RNG k-eps linear model model has been updated following the work of Han and Reitz [9], and Bianchi et al. [10]. The fuel liquid dispersed phase is treated according a Lagrangian approach using the hybrid spray breakup model proposed and validated by Bianchi et al. [11]. The latter accounts for both the atomization of the liquid jet and the droplet secondary breakup. The spray-wall interaction was modelled according to Cazzoli et al. [12]. The fuel auto-ignition is simulated using the Shell model in the implementation proposed by Kong et al. [13]. The high-temperature combustion part follows the characteristic-time combustion model developed by Abraham et al. [14], with the correction proposed by Bianchi et al. [15] to account for non-equilibrium turbulence effects of the energy cascade and thus on the turbulence time scales. Finally the extended Zeldovich mechanism is used to model NOx formation while the Hiroyasu's formation model [16] and the Strickland-Constable's model for the oxidation [17] have been used to predict in-cylinder soot.

As far as the the combustion calculations are concerned, the initial in-cylinder pressure and temperature at IVC have been evaluated accounting for the internal EGR following Senecal et al. [18] based on the target boost pressure, the external EGR rate and the air-to-fuel ratio. The in-cylinder pressure and the species densities have been assumed to be uniform at the beginning of the computation at the intake valve closure.

The hydraulic injection profile has been defined according to the target ECU controlling data (SOI, excitation time, injection pressure) using a modified version of the AMESim model developed by Bianchi et Al. [19].

4. Results and discussion

In this section the results of each step of the optimization process are presented and discussed together with the experimental-numerical 'a posteriori' validation of the work.

4.1. Assessment of the influence of the injector and bowl offset

The effect of both the piston bowl offset with respect to the cylinder axis (called A distance) and the injector offset with respect to the piston bowl axis (B distance) were analyzed in order to find out the best balance between mixing efficiency and the cost of the proposed solution. The simulated configuration are a combination of three values of the piston bowl offset and three values of the injector location offset as listed in Tab. 3. It must be underlined that the most external radial location of both the injector and bowl axis are those adopted in the baseline engine and that the most internal ones are the minimal that are feasible due to engine structural constraints which can not be avoided because of cost issues.

The simulations showed (Fig. 1) that varying the offset of bowl relative to the cylinder axis, an increase of the raw soot emissions of about 6% together with the decrease of NOx concentration of about 0.5% are obtained. Both small, compared to the effect of the variation of the offset of the injector location relative to the piston bowl axis that show

Table 3. Bowl-injector relative position

Bowl offset (A) [mm]	2	3.25	4.5
Injector offset (B) [mm]	1.5	2.5	3.5

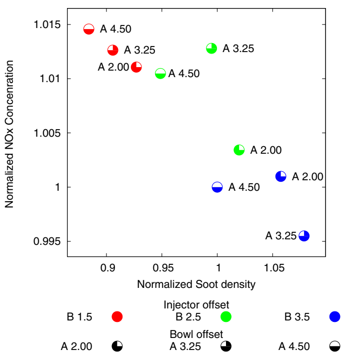


Fig. 1. Influence of piston bowl offset and injector offset combination on raw nox and soot emission

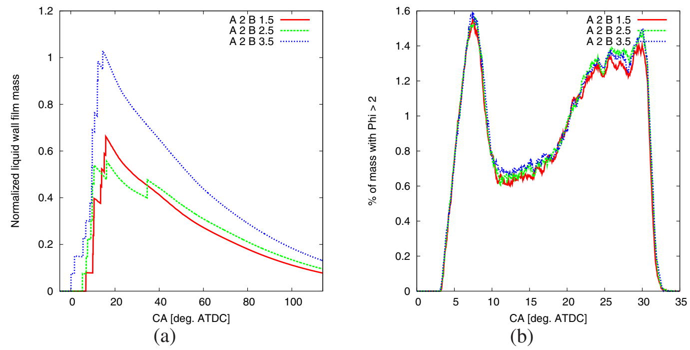


Fig. 2. Influence of the injector offset (B) for a fixed bowl offset (A=2.0 mm) on the in-cylinder value of: (a) normalized liquid wall film mass; (b) percentage of the mass having an equivalence ratio larger than 2.0

a decrease of the raw soot emissions of about 12.0% together with the increase of NOx concentration of about 1.3%. Both variation give a negligible effect on the IMEP.

So the offset of the piston bowl axis with respect to the cylinder axis (A) exhibits a lower influence on engine raw emissions and performance than the offset of the injector location (B), this effect can be explained by the fact that the piston bowl offset will results in different squish velocities, as a result of different squish areas, but their interaction with the swirl flow provide a weaker non axis-symmetric flow conditions. On the contrary, the injector offset result in different free paths available to each fuel spray and to each corresponding fuel vapor clouds before they impact against the piston walls, thus, the swirl flow will have different convective effect on each spray.

Centering the injector position with reference to the piston bowl axis, all the spray free paths becomes more uniform in length. The distance increase for the sprays that are near the bowl wall reduce the mass impacting the wall (Fig. 2(a)), reducing the late rich mixture spot formation (Fig. 2(b)). Also with longer and more uniform spray free path, the given swirl can play its convective effect more uniformly: this results in a lower in-cylinder soot concentration peak and a better circumferential oxygen exploitation that promote the soot oxidation during the expansion stroke. All this occurrence concur to reduce the final raw soot emission (Fig. 3(a)).

As far as the raw NOx emissions are concerned, though the increased combustion efficiency results in a slightly higher mean chamber temperature (Fig. 3(b)), the more uniform development of the combustion due to the more uniform air-fuel mixing and the more uniform spray path length leads to a decrease of the mixture "hot spots" (Fig. 3(c)) in the domain having a temperature higher than 2200 K. As a result, the raw NOx emission concentration records only a slight increase.

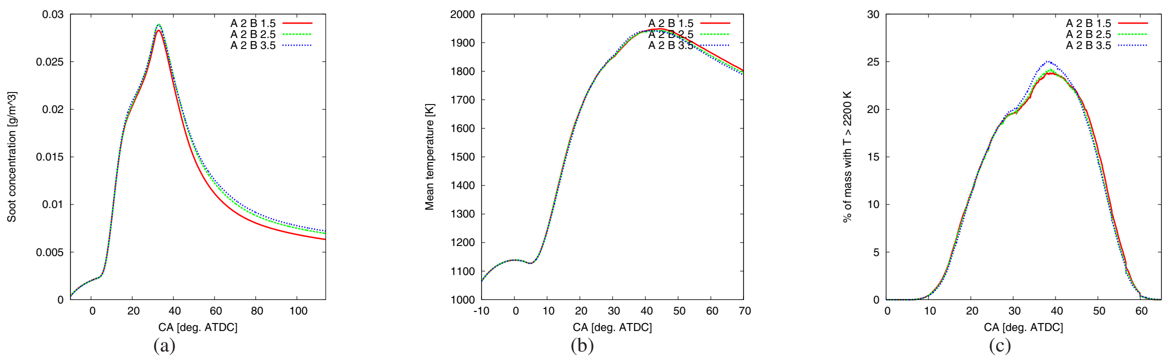


Fig. 3. Influence of the injector offset (B) for a fixed bowl offset (A=2.0 mm) on the in-cylinder value of: (a) soot concentration; (b) mean temperature; (c) percentage of in-cylinder mass having a temperature above 2200 K

Table 4. Swirl ratio at IVC

SR	0.9	1.1	1.3	1.5	1.8

The results achieved were very useful in addressing the design of the engine head with respect to the baseline layout. In particular, the injector location is fundamental to promote a more uniform mixing of both fuel and soot with air and usage of the oxygen in the bowl. Moreover, as mentioned, a more axis-symmetric flow and a more uniform spray path length are fundamental to the commitment of improving mixing and to the design solutions conceived performing at their best efficiency in improving engine performance and lowering raw emissions. Thus, this first part resulted in a revision of the engine head, founding a proper layout able to properly accommodate the exhaust and the intake ports, cooling ducts and the injector in a more central position.

4.2. *Influence of the swirl ratio*

Once the best centering of the piston bowl and the injector relative to it was found, the second goal was to identify the best matching between in-cylinder flow, injection and combustion phasing to enhance fuel and soot mixing with oxygen.

The idea that comes up was primarily based on the consideration that the retarded injection timing required by the absence of the De-NOx after treatment device changes the perspective: the reverse squish, which takes place during the expansion stroke, becomes a reference flow the swirl must match with. In fact due the occurrence of the combustion process after TDC the mixing between fuel and oxygen or soot occurs partly inside the bowl and partly in the periphery of the cylinder where centrifugal effects and the reverse squish shifts the oxygen and fuel vapor clouds. Thus, author thought that the first and most important commitment was to assess the influence of the swirl flow angular momentum on the emission and performance.

The effect of the swirl ratio at IVC (in the following indicated as SR) has been investigated reducing its value from the original engine value, as shown in Tab. 4. This part of the project is quite demanding since the design of the intake port cannot be further changed once defined. On the contrary, the piston bowl shape, the spray targeting and the injection specifications and calibration can be more easily revised during the steps of the engine design process or development.

Figure 4 show the influence of the swirl ratio at IVC and the piston bowl offset combination on raw NOx and soot engine out emissions. As it could be seen, varying SR from 1.8 to 1.3, a decrease of the raw soot concentration of about 32% together with an increase of NOx concentration of about 44% are obtained, the IMEP follow the NOx behavior and is of an order of magnitude lower than the increase of NOx (about 5%).

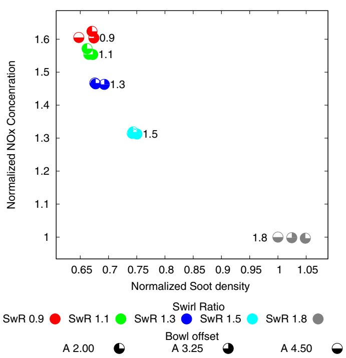


Fig. 4. Influence of the swirl ratio at IVC and the piston bowl offset combination on raw NOx and soot emission

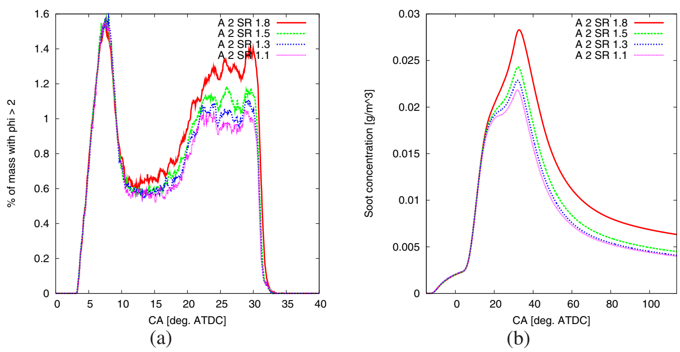


Fig. 5. Influence of the swirl ratio at IVC, for a fixed piston bowl offset (A=2.0 mm), on the in-cylinder: (a) percentage of the in-cylinder mass having an equivalence ratio larger than 2.0; (b) soot concentration

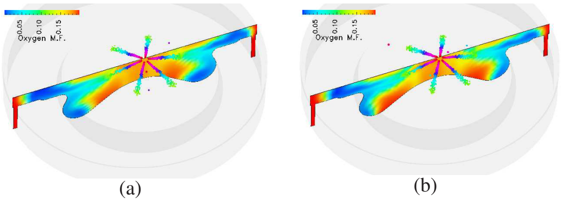


Fig. 6. Oxygen distribution on a plane passing through the cylinder axis at 20° CA ATDC for two different swirl ratios at IVC (SR): (a) SR 1.3; (b) SR 1.8

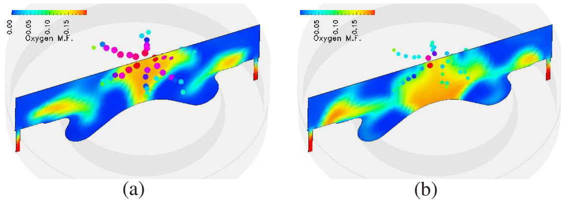


Fig. 7. Oxygen distribution on a plane passing through the cylinder axis at 40° CA ATDC for two different swirl ratios at IVC (SR): (a) SR 1.3; (b) SR 1.8

The high soot production of the high (original) swirl ratio is due to increase of fuel rich zones (Fig. 5(a)) during the tail of the combustion (from 20° ATDC to 35° ATDC), this result in a significant increase of in-cylinder soot concentration peak that cannot be reduced enough by the following oxidation (5(b)).

As it could be seen comparing the Fig. 6 and Fig. 7, the tangential (circumferential) convection operated by the swirl flow at SR of 1.8 (Figure 6(b)) limits the penetration of the fuel vapor in the bowl and prevent the central oxygen from flowing toward the bowl or cylinder periphery (Figure 7(b)), reducing the oxidation capabilities. The SR of 1.3 allows a better penetration of the fuel inside the bowl (Fig. 6(a)) and the flow of the central oxygen, following the reverse squish, toward the fuel rich zones (Fig. 7(a)), this exploitation of the bowl oxygen both for combustion and for soot oxidation causes a reduction of the soot and increase of the NOx.

A further reduction of the swirl ratio below 1.3 provides only a limited decrease of the raw soot emission coupled with a significant increase of the NOx concentration. Since the particulate matter levels are considered as a major concern for this engine in moving to Tier 4i fulfillment, the target swirl ratio at IVC of 1.3 is thus chosen as deign intake port target because it provides the best trade-off between NOx and soot raw emissions.

For the target swirl ratio at IVC chosen (i.e., SR =1.3), the piston bowl offset variations results in negligible differences of the engine raw emissions and engine performances. Thus, the piston bowl offset A=2 mm is chosen, since it offers a good compromise between the general engine performances, the design issues linked to the accommodation of the intake and exhaust ducts for a given injector location and the cost linked to a revised design of the piston. It must be considered that more relevant piston modifications to accommodate the bowl in the target location would require a costly revised design of the piston in order to accomplish with its thermo-mechanical structural resistance limits.

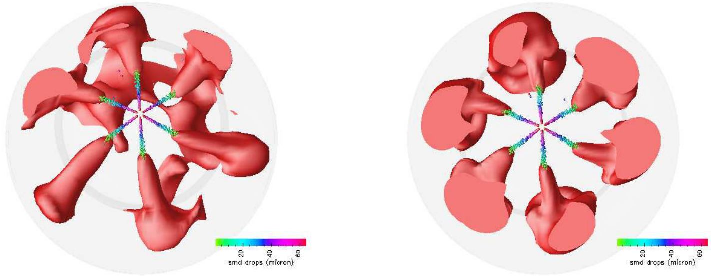
To summarize, under retarded combustion process, the weaker circumferential convective effect played by lower swirl ratios provides multiple benefits:

1. It promotes the air-fuel mixing inside the piston bowl where combustion could propagate backwards towards the piston bowl center due to the convection action played by the liquid and vapor fuel momentum. This allows an efficient use of the oxygen in the central part of the piston bowl and cylinder.
2. It provides a better matching between the tangential air-fuel mixing, which is a commitment of the swirl motion, and the radial outward convection of the fuel and the soot inside the cylinder periphery, which is a commitment of reverse squish because of the retarded fuel injection time chosen to accomplish with NOx emission limits.

This achievement is quite important since it demonstrates that low cost combustion concept, and therefore low-cost engines complying with upcoming regulations, might be found from a detailed investigation of the thermo-fluids process.

4.3. Fluid dynamics comparison between the final design and the baseline design: the experiments and the simulations

The development of the engine has been finalized through a spray targeting optimization by the use of complete CFD calculations which were based on a intake stroke performed by using FIRE code, that provide the fluid-dynamics



(a) (b)

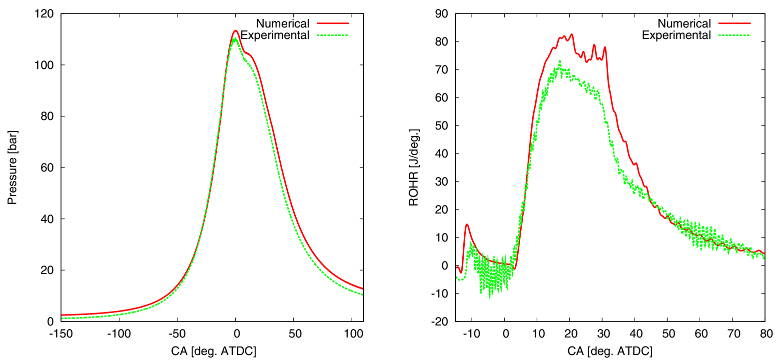
Fig. 8. Isosurface of equivalence ratio $\phi = 1$ at MFB 50% for: (a) the baseline engine; (b) the final engine

solution at IVC, solution that is used as initial condition for the simulation of combustion process performed by the KIVA3D code, as previously pointed out.

Using the complete CFD simulation where intake and combustion phases were simulated, we can have a brief and effective look on the in-cylinder mixing conditions comparing the iso-surface of stoichiometric equivalence ratio (i.e., $\phi = 1$)for the baseline engine (Fig. 8(a)) and the final engine (Fig. 8(a)), plotted at the crank angle corresponding to the same mass of fuel burnt fraction of 50%. As it could be seen, the original configurations with an higher injector relative offset coupled with a higher SR of 1.8, provides a non uniform behavior between different the fuel jets. The three fuel jets oriented towards the longer side of the piston bowl results in a longer penetration of fuel, due to the spray momentum, but they were lowly diffusive in circumferential direction though the swirl ratio was high. The three fuel jets oriented towards the shorter side of the piston bowl suffered of overlapping. It is thus clear that the designed swirl ratio is too weak for half the spray jets and too strong for the remaining. On the contrary, as a result of the optimization process, the final design presents a uniform development of combustion which is similar to those of a four valve engine.

Due to a decreased injector relative offset and a more uniform flow field, it has been possible to target each spray towards piston bowl lip where the interactions between spray, swirl motion, reverse squish motion and chamber contribute to enhance the fuel diffusion inside the cylinder, inside the piston bowl and back to cylinder center. As a result, the soot raw emission is drastically dropped as requested by Tier 4i emission target fulfillment.

Once that CFD calculation have been completed providing the confirmation that the new emission, power and BSFC target of the engine was achieved under the final and feasible for manufacturing design, as the final stage of the work, a real engine has been built and successfully tested at the bench since its first run.



(a) (b)

Fig. 9. Numerical and experimental comparison: (a) in-cylinder pressure; (b) in-cylinder derived rate of heat release

Table 5. Experimental comparison

Engine	NOx [g/kWh]	soot [g/kWh]	Power [kW]
Baseline	1.95	0.134	70
TIER 4i	2.14	0.069	77
Var	+9.7%	-48.5%	+7.1%

After the real engine run, a comparison between the numerical and experimental in-cylinder pressure and derived rate of heat release rate traces are has been made. As it could be seen, though the rate of heat release is slightly overestimated (Fig. 9(b)), resulting in a slight overestimation of chamber pressure (Fig. 9(a)), a good agreement between experimental and numerical results is achieved having the evidence that the final engine running at the test bench was accurately simulated in advance by the CFD approach.

In Tab. 5 the experimental comparison between the original engine and the Tier 4i engine is shown. Particularly, the new engine recorded an increase of NOx emission of +9.7% and a decrease of soot emission of -48.5% together with an increase of the maximum rated power of +7.1%. The final engine is easily complying with both Tier 4i emission limits and the required power target, achieving a reduction of BFSC too.

5. Conclusion

In this paper the numerical methodology adopted in order to aid the development of a two-valve engine into a Tier 4i emission complying engine has been described.

By means of three dimensional CFD simulations, the influences of many parameters, such as the swirl ratio at IVC, the injector location and the piston bowl offset were jointly investigated leading to the development of a new combustion concept. The numerical simulations have been shown the need of decreasing both injector relative offset and piston bowl offset in order to promote a more uniform flow field, a more uniform development of the combustion and to achieve an axis-symmetric flow condition as similar as possible to a four-valve engine.

Moreover, it was demonstrated that engines working with retarded fuel injection strategy should operate with lower swirl ratio at IVC. The latter, together with a proper use of the spray momentum by promoting longer spray free paths, avoid the overlapping of the jets providing a significant decrease of the raw soot emissions due to the better exploitation of the oxygen both for combustion, which leads to a decrease of fuel rich zones, and for soot oxidation.

The baseline two-valve engine we started from, presented a piston bowl offset of 4.5 mm and injector relative offset is 3.5 mm, with a swirl ratio at IVC of the original engine of 1.8. The final two-valve engine lay-out, thus, presents a piston bowl offset of 2.0 mm and an injector relative offset of 1.5 mm, with a swirl ratio at IVC of the original engine of 1.3.

Furthermore, the final real engine has been tested and resulted perfectly able to fulfill the Tier 4i emission standard at a higher maximum power providing a BSFC reduction. Finally, the good agreement between numerical and experimental indicating data of the final engine proved the accuracy and the reliability of the developed methodology of design based on CAE/CFD approaches.

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