SPRAY EVOLUTION IN SMALL BORE D.I. DIESEL ENGINES WITH DIFFERENT INJECTION STRATEGIES

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ABSTRACT

Engine downsizing is one of the most promising alternatives to improve fuel economy, while maintaining good emissions and NVH behavior. In this paper the spray evolution in a small combustion chamber, adopting different injection strategies (pilot-main and pre-main), are analysed by means of experiments and numerical simulations. To this aim an optical research engine with the combustion chamber size of a small (0.3L) DI diesel engine was employed. Multidimensional simulations were performed with KIVA3V code. Once assessed the code predictivity in the case of different injection strategies, numerical results are validated comparing spray evolution and wall impact with optical engine visualizations.

INTRODUCTION

Extensive research and development over the last ten years, especially in the injection technology, have demonstrated that diesel engines are the most indicated to equip small passenger cars with low emission and high efficiency [1]. In fact, amongst the measures for reducing exhaust emissions which are already under development (high-pressure injection and accurate fuel metering), the engine downsizing (reduction of displacement) provides an additional advantage, namely, the reduction of frictional losses and the engine operation at higher loads. Consequently, the outcome will be a reduction in fuel consumption and CO_2 emissions.

At the moment, European car manufacturers are present on the market with four cylinder diesel engines of about 1200÷1400 cc. of total displacement [2][3][4]. This is due to the fact that in this range significant fuel consumption reductions were obtained assuring good performances in B class vehicles. To realize efficient small diesel engines able to meet 2005 and 2008 regulations further efforts are required. Up to now the nozzle injection rate, spray distribution and combustion characteristics are developed and optimised for piston displacements corresponding to about 0.5 L. With a mere scale reduction of all components the optimisation problem cannot be solved. This is due to a very complex interaction between injection, ignition and combustion and the boundary of a small size combustion chamber. Therefore, the design of the whole combustion system and the management of the injection rate is of great importance in order to improve both emissions and performance.

The fuel mixture preparation process is one of the aspects that must be deepened to achieve the objective of reducing raw emissions of a small size engine, while maintaining advantages in terms of fuel economy. Once fixed a sufficient swirl level, the spatial fuel distribution optimization can be achieved via a suitable design of the injector nozzle as well as through an improved fuel distribution with multiple closely injections.

The last generation of CR systems becomes very useful in small size engine applications. In fact the further development of the solenoid injector and of the Electronic Control Unit of

Common Rail injection systems (C.R.) has permitted the use of multiple sequential injections. Thanks to this feature, such systems are capable to perform up to five consecutive injections in one engine cycle (with a minimum dwell time of 150μs) thus improving the control of the fuel injection rate in the combustion chamber [5].

In this paper the spray evolution in a small combustion chamber, adopting a ‘Microsac’ 6 holes injector and different injection strategies (standard pilot main injection and close coupled pre injection), are analysed by means of experiments and numerical simulations. To this aim an optical research engine was employed whose some combustion system characteristics were designed similar to a small (0.3L) DI diesel engine.

Multidimensional simulations with KIVA3V code were carried out with small displacement engine (0.3L) to evaluate how the injection strategy may be usefully adapted to a reduced bowl volume, considering the different effect of the spray/wall interaction. Computational results were compared with spray visualisation obtained from the optical engine using an high speed CCD camera. The analysis indicates that, also with new injection CR systems, an appropriate injector nozzle design and reduced combustion chamber bowl, the global spray behaviour does not show evident liquid fuel impact on the walls. Numerical results show that reduced bowl size increases the tangential air velocity improving the fuel transport in the combustion chamber. This effect coupled with a high hole to hole spray symmetry and an enhanced atomisation level, derived from an adequate nozzle design (microsac nozzle with rounded conical hole shape), justified the previous analysis.

THE EXPERIMENTAL APPARATUS

The experimental and numerical measurements of combustion and emissions were performed on a 4 cylinder turbocharged engine equipped with a second generation Common Rail system and with total displacement equal to 1.25L. The engine characteristics are reported in Table 1.
The visualization of spray evolution was realized in a single-cylinder optically accessible prototype engine with the same main characteristics of the real engine (inlet and exhaust valves, cam profile, etc.). Figure 1 shows the engine transversal section with a classical elongated piston and the optical path.

The engine design allows the use of different optical diagnostics: in these tests a high-speed CCD camera, with a maximum resolution of 480x420 pixels and maximum speed of 8000f/s, and a copper vapour laser for spray illumination were used. The optical engine has a modular structure in order to reproduce different engines characteristics. In particular this structure permits to vary the combustion chamber geometry and also the compression ratio.

Concerning the combustion system design, the adoption of a simplified combustion chamber is obviously necessary in the transparent engine. Figure 2 shows the real engine combustion chamber shape (a) as simulated for numerical computations, and the chamber chosen in the transparent engine version (b) for the 0.3L engine configuration. Difficulties due to the adoption of the quartz window lead to this last chamber profile in the optical engine as the best choice to reproduce as close as possible the real in-bowl air motion behavior.

THE NUMERICAL CODE

The diesel engine combustion simulations were performed by an improved version of the Kiva3V code [6] with the main models described in the following. The atomisation process was computed using the hybrid model [7]. The droplet evaporation model was improved describing the drop internal heating [8]. The ignition model is based on the Hiroyasu method [9] using the Handerberg and Hase correlation [10].

Further improvements in the description of initial conditions for spray computations were added in the present set of applications. In particular an in-house hydraulic code was set-up by Beatrice et al. [11], in order to calculate the fuel injection rate starting from experimental measurements of the needle lift. Accurate measurements of pressure in the injection line demonstrated a good predictive capability of this hydraulic code. Due to high-pressure injection, cavitating flows may occur within the nozzle holes. Therefore, a nozzle flow model as reported by Kong et al. [12] was used allowing the calculation of the KIVA initial spray conditions (droplet diameter and velocity and discharge coefficient) starting from the calculated rate of injection. The fuel injection routine was also modified to obtain an on-line control of the mass conservation on the fuel mass splitted between the various sequential injections each characterized by a different injection duration and velocity profile. Finally an off-line pre-processing allows an easy definition of the initial conditions in presence of EGR and turbo-charging.

1.25L 4 cyl. Engine	
Bore x stroke [mm]	69.6 x 82.0
Displacement [cm^3]	1250
Compression ratio	18.0:1
Swirl ratio	3.8
Injection System	Multijet Common Rail
Nozzle	Microsac 6 holes ∅ 0.125mm cone angle 149°
Intercooler	Air-Water
E.G.R.	Cooled

Table 1. The four cylinder 1.25L engine

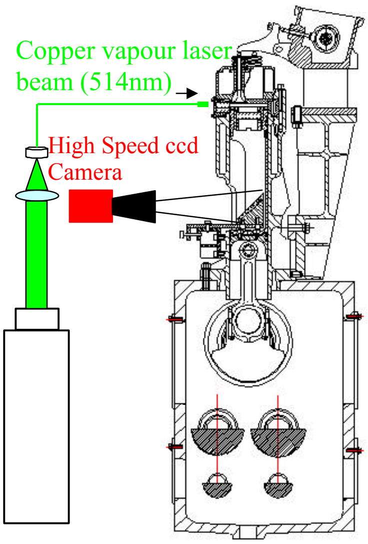
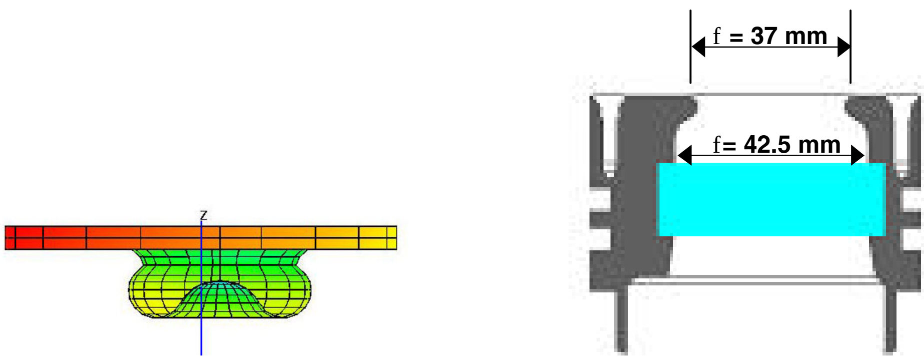


Fig. 1: The transparent engine optical path



a) Real engine 1.25L b) Transparent engine 0.3L

Fig.2: The Combustion System Design for the real engine (left) and the transparent engine (right)

As regard to the combustion model, it is coupled with the soot formation and oxidation mechanism trough a reduced six steps mechanism in the form developed by Belardini et al.[13], following Leung [14] and Fairweather [15]. In this model the acetylene is assumed as crucial pyrolytic specie for the nucleation and surface growth processes. The high temperature combustion mechanism is applied not only to the injected fuel but also to the formed acetylene. The global formulation of the model is a right compromise between a comprehensive description of the in-cylinder soot loading process and the computational cost of full 3D diesel combustion calculations.

RESULTS AND DISCUSSION

The 1.25L engine was settled at 1500 rpm, 5 bar of mean effective pressure, corresponding to 14.5 mm^3/stroke of injected fuel mass. Tests were performed with rail pressure equal to 600bar and without EGR. The injector used was a Microsac, 6 holes, ϕ=0.125mm. The same setting was used for the four cylinder engine as well as for the transparent engine.

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Experimental activity [5] on the real engine has outlined that, in the class of small engines and with the new generation of Common Rail systems, with respect to more complex injection strategies (multiple injection as pre-main-after) a simpler one (one injection before the main one) may be sufficient to obtain good results both in terms of fuel economy and emissions levels. Therefore the attention was focused on using a pre injection very close to the main one that is more convenient in terms of fuel economy with respect to using the standard pilot-main strategy. On the contrary some problems deriving from the fuel wall interaction must be considered adopting the pre injection strategy due to the injection timing very close to the TDC.

So the present study is limited to testing the pilot-main and the pre-main injection cases both experimentally and numerically as reported in Table 2.

Numerical computations were initially performed and used to address the experimental work. As first the code was tuned on the experimental results of the real engine, as shown in fig.3, where, experimental and numerical values of combustion pressure are reported for a pre-main injection strategy test. In Figure 4 the different behaviors of pilot and pre injection strategies obtained from numerical applications are shown in terms of soot and NOx emissions confirming the convenience of the pre-main injection with respect to the pilot-main one. In order to understand these results relative to the real engine, a deepen analysis of the fuel spray behavior in the pre-main injection test case is necessary and can be accomplished on the optical engine.

Test cases	Start First E.C. [c.a.]	First injected mass	Start Main E.C. [c.a.]	Main injected mass	First-Main Dwell Time [μs]
Pilot-Main Injection	18 BTDC	10%	5 BTDC	90%	1150
Pre-Main Injection	9 BTDC	10%	5 BTDC	90%	150

Table 2: Selected injection strategies (E.C.= Energizing Current of the Electro-injector)

Therefore the transparent engine was used to analyze the cold phase and the interaction of the liquid fuel jets with the combustion chamber walls, that is the main topic to be investigated in the class of small size engines [14].

In particular, aim of this study is investigating on the different effect of the pre injection in a small combustion chamber when compared to the pilot injection one. Fig.5-a shows that the pilot injection produces higher penetration and lower vaporization; while the pre injection, is characterized by lower penetration and higher vaporization. As expected this different behavior of the pre and pilot injections is a consequence of the different in-bowl pressure and temperature conditions at the moment of their injection timing. No wall impingement is evidenced in the pre-main test, while the fuel impact shown in the pilot-main test is probably ascribed to some droplets deriving from pilot fuel that remain unburnt on chamber walls. In Fig. 5-b it is evidenced that the pre injection case is more favorable to promote the fuel vaporization because of its combustion closer to the incoming main fuel jets.

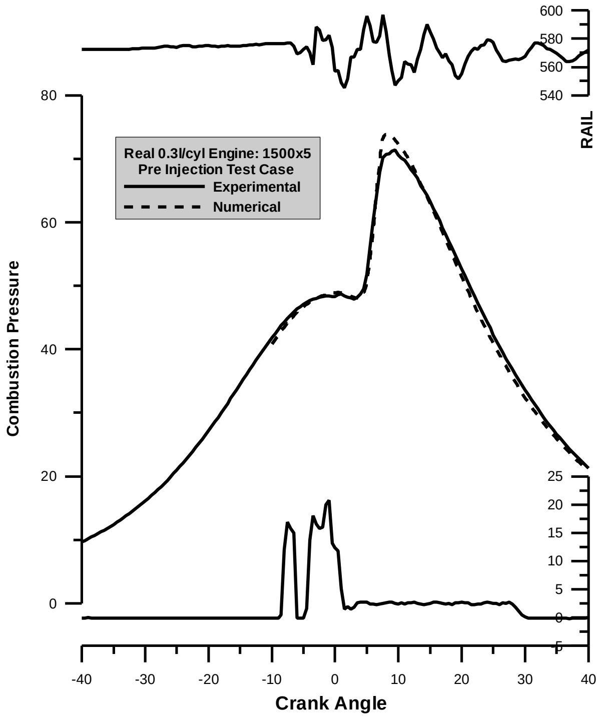


Fig.3: Comparison between Experimental and Numerical Combustion Patterns for real 1.25L Engine.

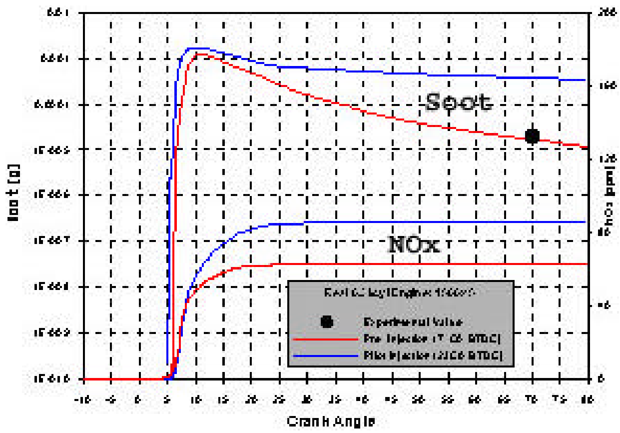
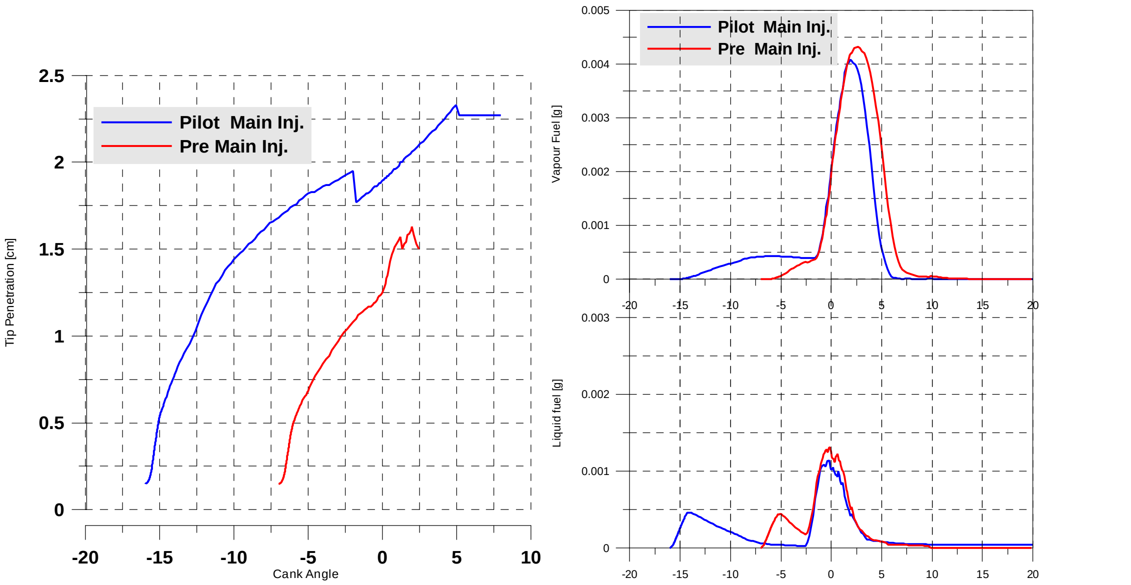


Fig.4: Comparison between Pilot-Main and Pre-Main injection Strategies.

The different interaction between the small size combustion chamber and the first injection is also outlined in Fig. 6, where the computed fuel vapour distribution (in a plane orthogonal to the Z axis) in the bowl is shown for both injection strategies, at the same time step after the first injection starting. It is possible to notice that, when using the pilot injection the higher penetration and the possible spray/wall impact justifie results of fig. 6-a where higher fuel vapour quantity remains unburnt near the chamber wall. On the contrary, in the case of pre injection (fig. 6-b) there is a better distribution of the fuel vapour and a consequent better air utilization in the chamber bowl. In this way also the evolutions of liquid fuel and vapour for both test cases shown in fig. 5-b are confirmed. Therefore the numerical computations confirms that for this class of small engines, the simple strategy of Pre-Main injection may be of promise.

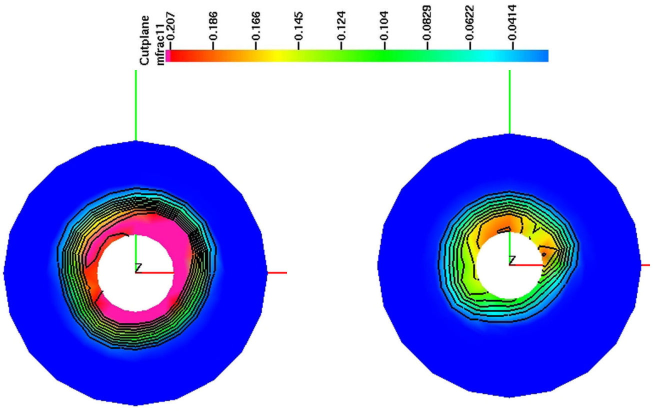
To validate numerical results, experimental visualizations were performed on the optical engine in both injection strategies. In Fig 7 visualizations of the whole chamber of a Pre-Main injection sequence are displayed: pictures have been taken at 8000 frames/s with a temporal resolution of 1.125 crank angle. The nozzle injector of new technology (hole diameter and hole type) is very suitable for small displacement engines, producing a good atomisation level and high jet momentum near the nozzle exit also at a moderate rail pressure. It can be noted that, as regard to the pre injection, there is no kind of wall impingement and a full evaporation of liquid jets.

Also concerning the main fuel jets, the liquid fuel doesn't splash on the wall before the start of combustion. Different repetitions of the same test case confirmed this behaviour. Fig. 8 shows the same evolution for the pilot-main strategy, confirming also in this case the numerical results. In fact, the penetration of pilot injection is quite higher with respect to the pre injection test. This could probably lead to the impact of some fuel droplets on the chamber wall as carried out from computations. Unfortunately the adopted optical technique doesn't permit to investigate on this aspect. Taking always in mind that the temporal uncertainty is of 1.125 c.a., until 2.25 c.a. BTDC the main injection presents almost the same evolution. Subsequently, the main injection penetrates more before the start of combustion that is not so rapid than in the previous case and some initial spots of combustion are evident at the fuel jets tip downstream the swirl direction. This different evolution of the ignition process can be justified by the more favorable fuel vaporization (fig. 5) and the better fuel vapour distribution (fig. 6) of the pre-main test case with respect to the pilot-main one. In this way previous numerical results are again completely confirmed.



5-a) Liquid Tip Penetration 5-b) In Cylinder Liquid and Vapour Fuel Mass

Fig.5: The 0.3L Transparent Engine Comparison of Pilot-Main and Pre-Main Injection Strategies



a) Pilot Injection b) Pre Injection

Fig. 6: Fuel Vapour distribution in the chamber bowl at the same timing from the first injection start

Due to high temporal resolution of previous visualizations and because of the fouling window phenomena during combustion the spatial resolution and image quality are limited to investigate on a detailed analysis of the fuel spray impact. In fact, in the pilot-main test, even if no macroscopic impact of the main injection is visible it could be interesting increasing the spatial image resolution for a detailed analysis at the spray tip. Therefore some visualizations were performed at the same CCD speed but zooming the image on a single jet of the main injection. Fig. 9 evidences a very slight liquid contact on the bottom part of the bowl lips at 2.25 c.a. BTDC with the jet tip shape weakly deflected by the swirl.

CONCLUSION

In the present paper, experiments and numerical analysis have been performed in the class of small size engines. A real 1.25L 4 cylinder engine and a single cylinder 0.3L transparent engine were adopted. In the class of small size engine a simpler strategy with the injected fuel mass splitted over only two close-coupled injections, that is a pre-main strategy, produces good results in terms of emissions. For small combustion chamber a proper nozzle design assures good levels of mixing: also at moderate rail pressure, fuel jets have a good level of atomization, probably due to the high values of the jet momentum near the nozzle exit. In any case the fuel wall interaction is moderate or avoided.

Concluding: With new injection CR systems and with an appropriate design of injector nozzle one of the main problems in the class of small DI diesel engines is exceeded, increasing their potentiality in equipping the future low emission and low fuel consumption passenger cars.

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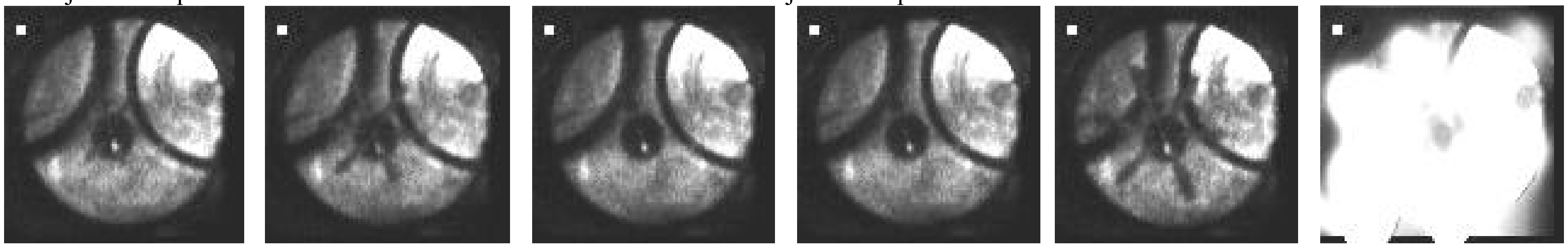
Pre Injection Sequence Main Injection Sequence



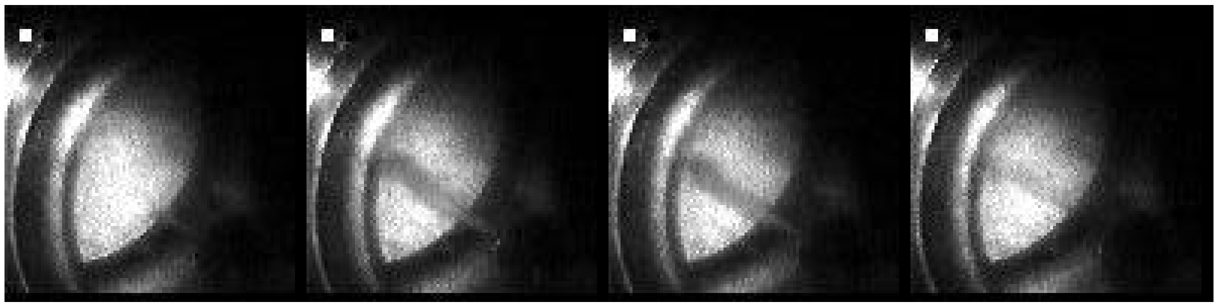
6.75° BTDC ±1.125° 5.625° BTDC 4.5° BTDC 3.375° BTDC 2.25° BTDC 1.125° BTDC

Fig. 7: Pre-Main Test Case at 8000f/s Visualizations

Pilot Injection Sequence Main Injection Sequence

15.75° BTDC ±1.125° 14.625° BTDC 13.5° BTDC 3.375° BTDC 2.25° BTDC 1.125° BTDC TDC

Fig. 8: Pilot-Main Test Case at 8000f/s Visualizations



3,375° BTDC 2,25° BTDC 1,125° BTDC TDC

Fig.9: Visualization of the main injection in the Pilot-Main Test Case at 8000f/s.