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Résumé — Aide au développement d'un moteur à injection directe d'essence par modélisation 3D — Depuis quelques années, l'injection directe d'essence apparaît comme l'une des voies d'investigation les plus performantes pour améliorer le rendement des moteurs à allumage commandé. Cependant, cette technologie est, actuellement, plutôt réservée aux moteurs de cylindrée assez importante, et les travaux de R&D présentés dans la littérature font surtout état de développement pour des moteurs d'alésage supérieur à 80 mm. En ce qui concerne les petits moteurs (alésage inférieur à 75 mm), implanter un injecteur dans une culasse déjà largement encombrée, avec un fort potentiel de combustion en mélange très pauvre stratifié, tout en maintenant de bonnes performances moteur et un circuit de refroidissement bien dimensionné, constitue un véritable défi. Pour un tel moteur, l'*Institut français du pétrole (IFP)* développe une solution à 3 soupapes par cylindre (NSDI-3), basée sur un concept *narrow-spacing*. Une chambre a donc été dessinée avec une culasse comprenant une bougie et un injecteur proches l'un de l'autre, et un piston à bol devant permettre de conserver le carburant au voisinage de la bougie au moment du déclenchement de la combustion. Cet article décrit la contribution de la modélisation 3D au développement de ce moteur, depuis les travaux effectués tout au début de l'étude pendant la phase de conception au Bureau d'études, jusqu'à ceux réalisés pendant la phase de mise au point du moteur prototype au banc d'essais moteur, afin d'aider à la compréhension de son fonctionnement et proposer des améliorations de forme de chambre.

Mots-clés : CFD, modélisation, développement moteur, injection directe, essence.

Abstract* — *CFD Aided Development of a SI-DI Engine — *Gasoline Direct Injection (GDI) appears to be the most relevant way to improve fuel efficiency of SI engines. But today, GDI is essentially regarded as a suitable technology for relatively high displacement engines, and the literature shows that the R&D effort on GDI engines is generally made for bores larger than 80 mm. For small bore engines (bore below 75 mm), locating an injector in already congested cylinder heads, with ultra lean stratified combustion capability while maintaining high engine specific power and proper cylinder head cooling is a real challenge. For such an engine, IFP is developing a 3-valve per cylinder engine (NSDI-3), based on a "narrow spacing" concept, with a spark-plug-close-to-the-injector design and a suitable piston to confine the gasoline spray within the vicinity of the ignition location. This paper describes the contribution of 3D modeling to the development of this engine, from the initial work during the design of the prototype combustion chamber, to the development and tuning of the prototype engine on the test bench.*

Keywords: CFD, modeling, engine development, direct injection, gasoline.

INTRODUCTION

The development of performing GDI engines is undoubtedly a great challenge for the automotive industry. Recent work on GDI engines has shown their great potential in terms of fuel consumption economy and performance in comparison to port fuel injection engines. Direct injection enables realization of stratified charges and seems the most promising way [1-3] of improving fuel economy of spark-ignition (SI) engines in comparison to other solutions like homogeneous lean-burn conditions, variable valve timing, supercharging with displacement reduction or with a "start-stop device". Extensive research made by manufacturers showed numerous advantages of GDI in comparison to Port Fuel Injected (PFI) engines [4-8]. Most are linked to the direct injection itself and the lean-burn combustion. However several technical difficulties still remain. The most challenging aspect of GDI is probably the specific stratified charge mixture operation at part-load which means that an adequate combustion chamber design and a suitable injection strategy are required to avoid excessively rich or lean areas around the spark-plug. In order to meet these targets and also to satisfy production constraints, various concepts are now proposed.

Side injection from an injector located between two intake valves with a curved-top piston to control the shape of the air-fuel mixture defines the "wide spacing" concept and has been adopted by several engine manufacturers. This approach is strongly influenced by a large number of parameters and the role of the in-cylinder charge motion (swirl or tumble) as well as the injection strategy were found to be essential in the mixing process. However, the suitable mixture strategy over a wide range of operating conditions requires a careful optimization of the piston shape and the in-cylinder flow. Both aspects create additional design costs and optimizing the fluid motion can reduce the GDI potential for increased volumetric efficiency. This is especially true in the case of swirl which may also require an auxiliary throttle leading to a more costly and more complex engine.

Since the objective is to obtain a rich enough mixture around the spark-plug at spark timing, the "narrow spacing" concept, characterized by an injector close to the spark-plug position while the charge motion is used to constrain the mixture around this position, can be another way to stratify the charge.

Though it represents an interesting solution, the "narrow spacing" suffers from heavy requirements on injector spray (fine atomisation and perfect axial-symmetry of the plume). Because of possible liquid fuel deposits on the spark plug, the risks of spark plug fouling, which may lead to misfiring, are also high. Nevertheless, *IFP* experience showed that the ultra lean-burn potential of this concept is high, with already proven stratified operation with lambda value greater than 8. While for "wide spacing", the removal of the spark plug from the injector often causes fuel dispersion, which does not allow such lean burn operating conditions.

Undoubtedly, GDI will involve the whole range of engine sizes, although today most of the effort of automotive manufacturers focuses on relatively large scale engines with bores greater than 80 mm. The *IFP* proposal for a small bore engine applies to a GDI combustion chamber with a bore below 75 mm. It is an in-line four cylinder engine and a "narrow spacing" concept was chosen to reach ultra lean-burn capabilities. The basic concept, the combustion chamber layout and 3D modeling optimization stages are reported here. More details about the whole engine development, such as 1D calculations and 3D water cooling simulations, can be found in [9].

1 ENGINE SPECIFICATIONS

A three valves per cylinder arrangement is selected to deal with the small bore. No further auxiliary valve to control swirl, no variable valve timing nor variable length of the induction system are needed. To take full advantage of the direct injection fuel economy potential, the concept engine designated by the name NSDI-3 has to be able to run in stratified charge mode.

Intake and exhaust valves are sized to reach a 50 kW/l power target. Thus, two intake valves are located at a nearly central position on the cylinder head in order to allow for a large valve diameter. Intake valve stems are vertical and as a result valve seats are located in a plane perpendicular to the cylinder axis. The single exhaust valve, located in a pentroof of the combustion chamber, should give quicker catalyst light-off than a two exhaust valve layout.

To reduce heat transfer from the flame to the wall and to improve knock resistance the spark plug is conventionally centrally placed but because of the small bore an eccentric location is less constraining. As a result, and to deal with a congested cylinder head, the "narrow spacing" is realized by placing the spark plug on the intake side and the injector underneath, both far away from high temperature zones and easily accessible in a vehicle with the exhaust on the back side.

For ultra lean burn operation, the confinement of the fuel spray around the spark plug is ensured by a lateral piston bowl facing the injector (Fig. 1).

The bowl has an inclined axis to restrict the fuel-air mixture around the ignition position, and not to guide it to the spark-plug.

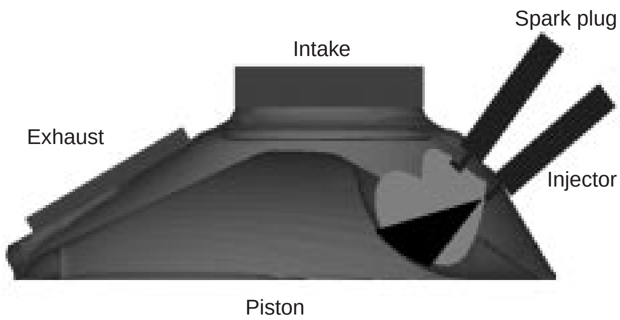

Figure 1

Basic concept.

2 PRE-SCREENING WITH THREE-DIMENSIONAL MODELING

The development of favorable internal fluid dynamics and the formation of a suitably stratified mixture before ignition are essential stages of the NSDI-3 combustion chamber design. During the compression phase, the flow characteristics in the chamber are essentially governed by the interaction between the previous bottom dead centre (BDC) flow pattern and the piston bowl shape. This interaction determines the air-fuel distribution around the spark plug near TDC.

Basically three aspects must be taken into account:

– the in-cylinder air flow and the piston bowl profile characteristics should contribute to the mixture confinement;
– the fuel vaporization process during injection should be adequately fast to avoid presence of liquid on the chamber's wall near TDC and finally;
– the stratification obtained at the spark-plug location should not include as much as possible excessively rich or lean areas.

Taking these aspects into account, three-dimensional calculations were performed in a step-by-step iteration cycle with the *Design Office* to accurately define the adequate combustion chamber design for ultra lean burn operation. Special emphasis was given to the piston-bowl shape and the in-cylinder flow. The CFD code used is KMB, a multi-block version of Kiva-II [9, 10]. It solves the full 3D averaged compressible Navier-Stokes equations coupled with spray and combustion equations in a finite volume formalism. The multi-block approach developed at *IFP* allows computation of complex meshes with a low number of inactive cells. The Wave-Fipa breakup model [11, 12] is used to describe the injection process, while the Extended Coherent Flame Model (ECFM) [13] allows stratified combustion computation.

In order to reduce meshing and calculation times, only the compression phase is computed in this validation stage. Simulations started at BDC and ended at TDC. The in-cylinder flow at BDC is initialized by imposing a centered uniform rotating flow in the chamber which is described by a mesh with 110 000 vertices. Swirl, conventional and reverse (clock-size motion) tumble with various intensities (weak or strong) were imposed at BDC in order to characterize the future design of suitable intake pipes. Based on experimental visualizations with an existing injector, injection is simulated at the injector nozzle by a 70° hollow cone and the Sauter mean radius of the injected droplets is set to 10 mm. Injection started at about 60 cad BTDC. Engine operating conditions used in this stage are summarized in Table 1.

Several combustion chambers were designed, and for each in-cylinder air flow imposed at BDC, injection timing and injector inclination were adjusted to achieve the most suitable distribution of fuel in the vicinity of the spark plug.

TABLE 1

Speed (tr/min)	2000	Chamber mean turbulence at BDC	30 m^2/s^2
Volumetric efficiency	0.90	Chamber pressure at BDC	0.9 bar
Chamber temperature at BDC	350 K	Fuel/air equivalence ratio	0.34
Injection duration	12 cad	Fuel mass rate (kg/h)	1.75

The latest iteration of the combustion chamber design provides a good air-fuel equivalence ratio at the spark-plug as shown in Figure 2 and is obtained with an initial reverse tumble at BDC. Indeed, during the compression phase, this reverse tumble motion moves toward the piston bowl side. Near TDC, the vortex remains over the bowl. This structure contributes to confining the rich air-fuel mixture around the spark-plug (Fig. 3). In contrast, a conventional tumble vortex at BDC moves toward the exhaust side during compression. As a result, air-fuel stratification around the spark plug near TDC seems not to be so adequate for a stable ignition.

According to these conclusions, upright intake ports, well suited to develop a reverse tumble, were designed. The resulting combustion chamber is shown in Figure 4.

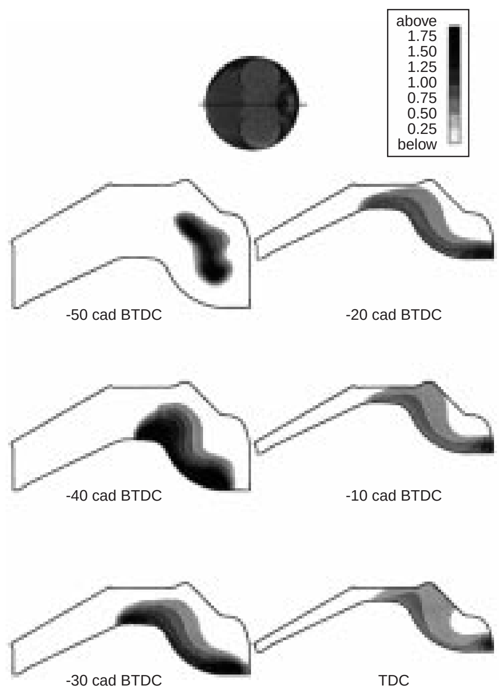

Figure 2

NSDI-3 prototype chamber design: fuel/air equivalence ratio fields after injection (60 cad BTDC).

3 3D MODELING OF INTAKE, INJECTION AND COMBUSTION

In order to reach the fuel economy goal, the engine has to be able to work at a very low Fuel-Air equivalence ratio. Thus, we have to verify as a preliminary phase to the engine prototype building that the complete geometry studied allows stratified operating conditions by taking into account the whole engine cycle.

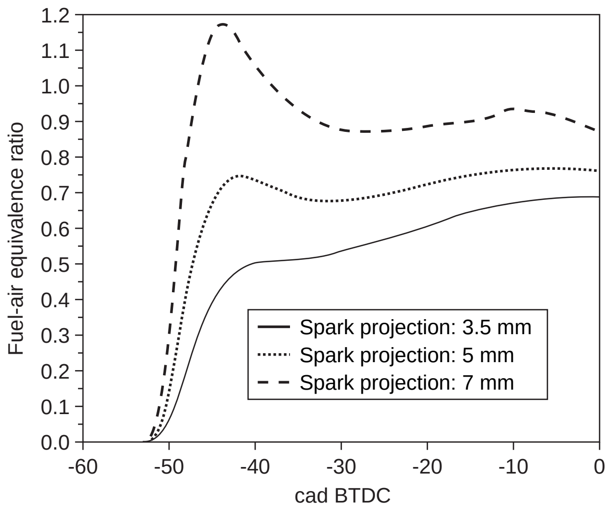

Figure 3

Fuel/air equivalence ratio at the spark plug location for various spark projections.

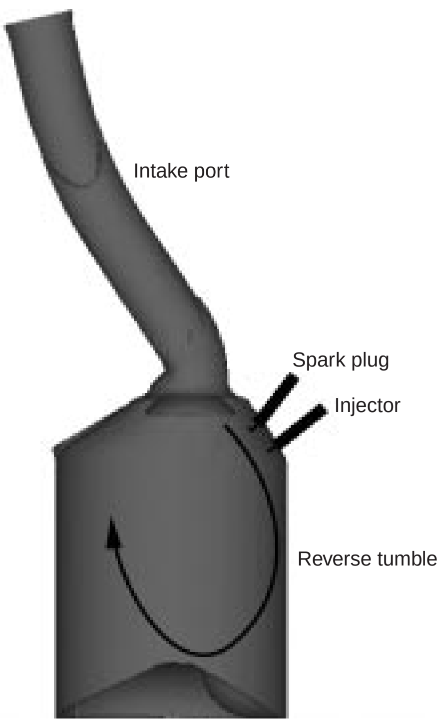

Figure 4

Chamber side view.

Three-dimensional computation of injection and combustion is an effective way to validate the choices made as it enables to understand how the combustion occurs in the combustion chamber.

Because of the symmetry of the retained solution, only one half of the chamber is now considered. The mesh is composed of two blocks. About 120 000 hexahedric cells are used to describe the half combustion chamber while 25 000 cells describe the inlet pipe.

3.1 Spray Modeling

The gasoline stratification in the cylinder is, to a large extent, a function of the characteristics of the spray. Since the combustion process widely depends on the shape of the fuel cloud, the modeling of fuel injection has to be done carefully.

The spray model parameters were fitted on experimental results: visualizations of the spray in a pressurised vessel were performed for various injection pressures, various injection duration and various vessel pressures. Then, the key parameters of the model were adjusted to reproduce the spray measurements.

The new simulated injector is a *Siemens* Deka DI with an injection pressure of 10 MPa. The resulting injection speed is around 120 m/s. For the operating conditions, in correspondence with spray visualization, we consider that the injected mean Sauter diameter is about 10 mm and the initial cone angle is 90 degrees. We used a Rosin Rammler distribution for droplet size [13].

Since the piston is rather close to the injectior nozzle, droplets impinge upon it. We describe this impingement according to Naber and Reitz [15] with heat flux between wall and droplets modeled as in Eckhause and Reitz [16].

3.2 Combustion and Pollutant Modeling

As underlined previously [13], the ECFM model does not require any specific tuning for this engine. It has just been used with the same parameters as in [13]. The ECFM model includes a conditioned description of burned/unburned gases and a pollutant model.

In this phase, NO_x is the only kinetically computed pollutant. CO is calculated through an equilibrium with CO_2 (the cut off temperature is 1200 K). Unburned hydrocarbons originate only from a reduction in the rate of combustion caused by excess dilution or temperature decrease during the expansion stroke. There is no activated crevice model. Anyway, fuel trapping by crevices is less likely to happen under stratified operating conditions, and so the computed HC should not be very far from what is measured.

3.3 Operating Conditions

The stratified lean burn operating conditions retained for this study, corresponding to the test rig engine parameters, are summarized in Table 2.

TABLE 2

Speed (tr/min)	2000
Volumetric efficiency	~0.9
Fuel/air equivalent ratio	0.35
Injection timing (cad BTDC)	~45
Spark timing (cad BTDC)	~15

In comparison to the *Mitsubishi* GDI “wide spacing” engine, the NSDI-3 engine requires later injection timing with leaner operating conditions.

The complete process (intake, compression, injection, ignition, combustion) is computed for two inlet pipe shapes. The first tests of the engine showed that improvements of the cylinder head were required, especially for homogeneous operating conditions. Thus, both computed and experimental results led to propose some modifications in the inlet pipe design. Two cases were computed. The first one, corresponding to the actual engine, is compared to the engine test rig results. The second one is compared to the first one, before the realization of a new prototype.

3.4 Results and Discussion

In case one, during the intake phase, the average flow motion is rather weak and the resulting flow is composed of two vortices: the first one over the bowl and the second one on the exhaust side (Fig. 5).

This configuration has been optimized for volumetric efficiency and the natural drawback is the lack of intensity in the mean flow motion. Consequently, turbulence intensity is not very high during the whole process (Fig. 5).

Near combustion TDC, a small reverse tumble motion appears over the bowl (Fig. 6), but this vortex is mainly due to the shape of the piston bowl and to the effect of air entrainment by the spray.

Experimentally, it has been observed in this case, that the engine worked better at high injection pressures. This might be directly linked to the flow motion and the turbulence generated by the spray. Indeed, the higher injection pressures should generate stronger flow motion, and thus, faster combustion. For load stratification, as expected, because of the late injection timing, the fuel is rather concentrated in the piston bowl. The reverse tumble motion also helps to confine the fuel. Despite the important initial cone angle of the spray, there is no fuel that directly impacts the bottom of the piston or the cylinder head. Indeed, since the density in the cylinder is rather high when injection starts, the spray tends to narrow.

The fuel distribution is presented in Figure 7. The maximum of fuel-air equivalence ratio is located at the top of the bowl. Even if the stratification observed is satisfying, one would wish for a smaller peak equivalence ratio, with a wider area in the bowl occupied by the fuel, and a smaller amount of fuel located in the immediate vicinity of the piston.

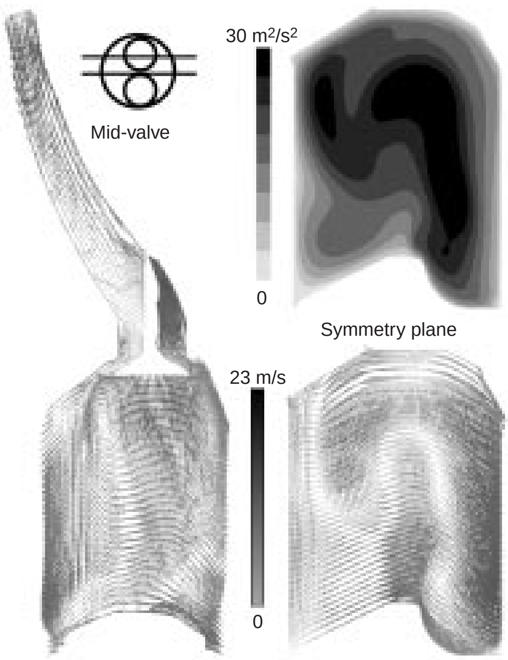

Figure 5

Velocities and turbulent kinetic energy at BDC for case 1.

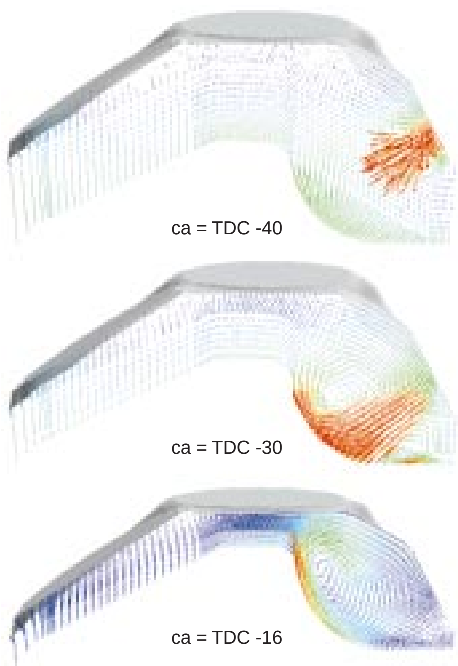

Figure 6

Velocity field near TDC (case 1).

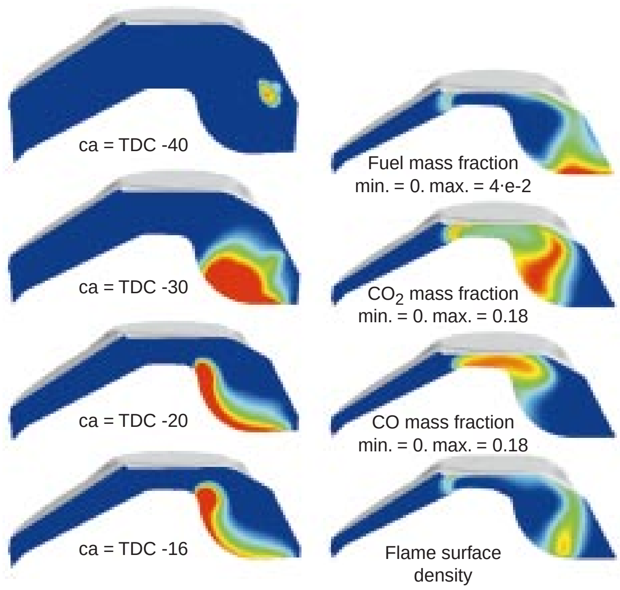

Figure 7

Fuel mass fraction during the compression phase in case 1 (min. = 0 and max. = 0.2) and combustion characteristics 5 cad ATDC (case 1).

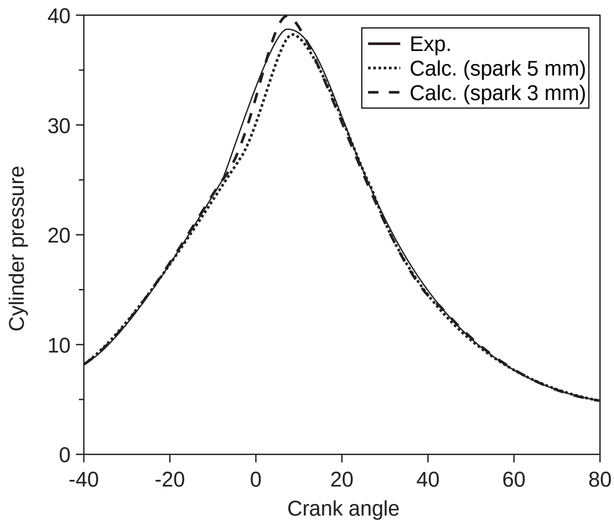

Figure 8

Cylinder pressure evolution (case 1).

Compared with preliminary computations, the fuel is more confined. This is partly due to the later injection timing needed by the engine to work well on the test bench. This timing probably prevents the fuel to flow out of the bowl.

Thus, limiting fuel dispersion also limits fuel dilution and consequently, HC emissions. The increase of the confinement might also be caused by the narrowing of the spray. The preliminary computations (Fig. 2) did not exhibit the strong narrowing of the spray observed here: the injection speed and the density of the medium at the moment of injection were smaller.

The ignition delay is small owing to the location of the spark plug that is in a rather rich area and to the high cylinder pressure. As the flame grows (Fig. 7), it pushes the gases, including fuel, out of the bowl and a great amount of CO is created just at the top of the piston. The spot of CO is the direct image of the spot of high fuel concentration that existed at the top of the piston bowl before ignition.

For the combustion phase, as measurements of cylinder pressure were already available, combustion and pollutant computations are used here to understand what happens in the cylinder when the flame appears and propagates. The cylinder pressure evolution for case one (Fig. 8) shows some differences between computation and measurements, but the IMEP are rather close (less than 5%). For that case, we have also plotted a computation result for a different spark plug location (2 mm closer of the head) and as a result the quicker increase of the cylinder pressure is due to a better ignition.

As we said before, the behavior of the engine under homogeneous operating conditions (with injection during intake process) needed to be improved. This was supposed to be mainly due to the weak fluid dynamics motion and the lack in turbulence. Moreover, we have seen in Figure 5, that the fluid motion was composed of two counter-rotating vortices, when we wanted to generate a main reverse tumble motion in the chamber.

Thus, in order to enforce the only desired vortex, new inlet pipes were designed, and a new calculation was performed with the same operating stratified conditions. Although, the volumetric efficiency is close to 0.9 for this stratified case, the observation of fluid dynamics gives a good idea about what happens for weaker flow rates of homogeneous configurations. The resulting flow and turbulence fields at BDC in this new engine (case 2) are presented in Figure 9.

Compared to Figure 5, the flow is much stronger in this case, with a turbulence level doubled, and a main reverse tumble that is well established in the chamber. The second counter-rotating vortex at the exhaust side of the cylinder has almost disappeared. This new design, which is more favorable for homogeneous operating conditions, has also been investigated for stratified conditions.

Figure 10 presents the cylinder pressure evolution in case 2, compared to case 1, for the deeper spark projection. We have seen before that the spray generates strong flow motion and turbulence during injection. Nevertheless, we can see for case 2 that stronger fluid motion and higher turbulence intensities generated during the intake process are favorable to increased combustion rates. Ignition delay is shorter, and the peak of pressure higher, but the computed IMEP is identical.

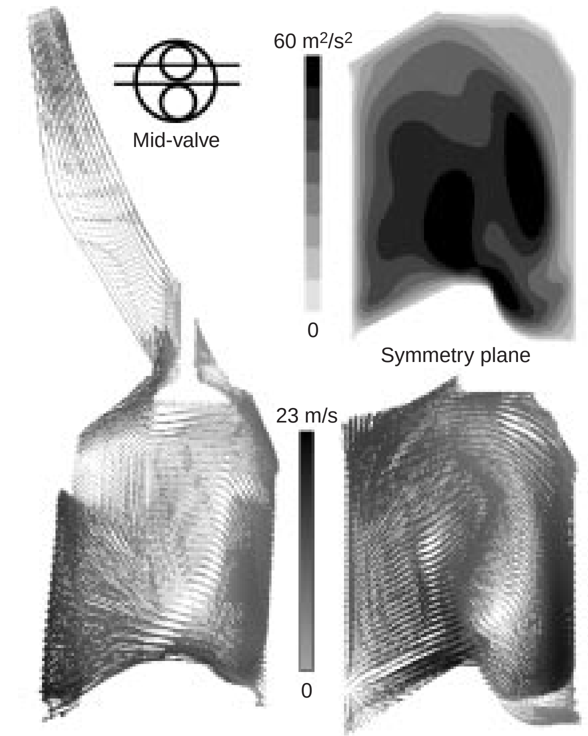

Figure 9

Velocities and turbulent kinetic energy at BDC for case 2.

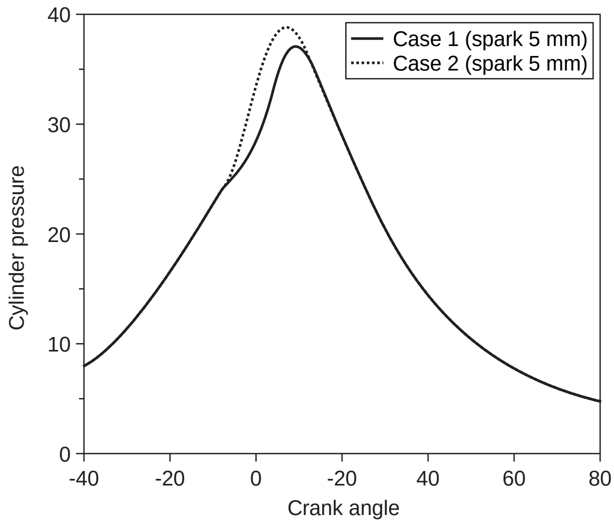

Figure 10

Cylinder pressure evolution.

That means that this new configuration seems to be comparable to the previous one for stratified combustion, while it should improve homogeneous behavior of the engine. Future engine results, as well as homogeneous calculations, should confirm that point.

Table 3 summarizes pollutant emissions computed and measured for case 1. For the case studied, the computation under-predicts NO_x and HC levels. NO_x under-prediction might be due to the stratification which might be not very accurately described in the computation (as NO_x formation is extremely sensitive to the local fuel/air ratio). HC levels might be underestimated by the computations because of the current chemical model and by the fact that crevices are not taken into account. However, computed and experimental results are of the same order.

TABLE 3

	NO_x (ppm)	HC (ppmc)
Computed-1 (electrode projection 3 mm)	310	2950
Computed-2 (electrode projection 5 mm)	340	2750
Experiments	480	3500

CONCLUSION

Three dimensional modeling tools have been used to help to define the global characteristics of a small bore GDI engine with a well adapted three valve per cylinder design. The engine was designed to reach 50 kW per displacement liter with relatively high BMEP over the entire engine speed range. The IFP proposal is a combination of “narrow-spacing” layout and piston bowl confinement. The “spark-plug close to the injector” arrangement seems to be the most promising way of achieving ultra lean-burn conditions.

A step-by-step process demonstrated the adequation of computational tools to the design of the combustion chamber in order to obtain a good stratification capability and potentially high performances. The computational tools can be very helpful during the first stage of the design of the engine, to validate a new concept, as well as during the tuning of the engine on the test bench, to understand its behavior.

Computations of the complete engine cycle showed interesting results operating reproducing the measured trends. It helped the engine designers in making appropriate decisions for the evolution of their design. For example, in this study, new intake pipes have been designed after 3D calculation iterations.

A future small car, fitted with an improved engine is now under active development at *IFP*. Its objectives are to meet emission standards with only a conventional three-way catalyst for year 2000. For 2005 regulations, a DeNox catalyst with 70% efficiency would probably be required.

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