IN-CYLINDER SWIRL ANALYSIS OF DIFFERENT STRATEGIES ON OVER-EXPANDED CYCLES

Jorge Martins, jmartins@dem.uminho.pt
Bernardo Ribeiro, bernardor@dem.uminho.pt
Senhorinha Teixeira, st@dps.uminho.pt
Universidade do Minho, Escola de Engenharia, Guimarães, Portugal

Abstract. *Innovative series hybrid vehicles have a small Internal Combustion Engine used as a "range extender". This engine should work at steady conditions whenever is running. Therefore its development is quite different from engines intended to be driven (unsteady working conditions, such as accelerations). For this application the engine development seeks optimisation on a specified speed/load condition.*
Previous work proved the concept of over-expansion, as a mean to enhance thermal efficiency of spark ignition engines, even beyond the efficiency of common Diesel engines. There are two ways of performing over-expansion through variation of inlet valve timing: early inlet valve closure (EIVC) and late inlet valve closure (LIVC).
To further enhance efficiency, these engines should work with lean or extra lean mixtures. which are very sensitive to in-cylinder charge motion. In order to improve fast burning rates at lean conditions one should optimise turbulence at the end of the compression stroke.
A small engine geometry and mesh was created in GAMBIT and later FLUENT was used to perform calculations of the in-cylinder fluid dynamics during intake and compression strokes.
These calculations were performed with both described over-expanded valve strategies. The results are compared with engine performance data using the same strategies.

Keywords*: internal combustion engines; over-expanded cycle; fluid dynamics; FLUENT; engine optimization*

1. INTRODUCTION

Emissions regulations and recent crisis in oil prices created the conditions for the increased interest of companies and general public for electric vehicles. However, the sate of development of these cars, specially referring to the batteries capacity and price, made it clear that a full replacement of conventional ICE car can only be made by series hybrid cars. These cars work only on electricity on the short travels, but have an IC engine to extend its range of operation. This type of car will have much less batteries than a conventional full electric car, as the range is no longer an issue. Therefore one of the main obstacles of the introduction of electric cars (price of batteries) can be minimized, without the sacrifice of range, but on the contrary, significantly extending it.

Several examples exist of technical solutions for hybrid cars (Hodkinson and Fenton, 2001). Internal combustion engines used on these applications are set to work at few or just one working point of load and speed. The design of engines for these applications is made considering just that working conditions allowing a better use of the energy potential of the fuel used. Among the technical solutions available and used there are over-expansion (Yamaguchi and Jost, 2004) and homogenous charge compression ignition (Sun et al, 2004). The latter is still under development, while the first one is already on the market.

Over-expansion in a reciprocating internal combustion engine is achieved when the length of the compression stroke is shorter than that of the expansion stroke. Several design concepts were created to make variations on the stroke length (Ribeiro 2006). The most easier and most conventional is using variable valve timing. By changing the intake valve closure timing it is possible to change the length of the compression stroke, making it shorter, keeping the expansion stroke as in a conventional engine. Two strategies may be adopted: early intake valve closure (EIVC) or late intake valve closure (LIVC). On the first the intake valve closes during the intake stroke well before the piston reaches the bottom dead center, causing the pressure within the cylinder to reduce to a value lower than the intake manifold pressure. On LIVC the intake valve closes during the compression stroke, making the compression to start just after its closure. During this operation the air/fuel mixture is sent back to the intake manifold during the initial upward movement of the piston.

Lean combustion has also been presented as a way to improve engine thermal efficiency, and also as a way of significantly reduce emissions from IC engines. Using an air/fuel mixture much leaner than the conventional stoichiometric reduces significantly NOx production since the formation of these oxides is very dependent on temperature during the combustion process. Using lean mixtures the flame temperature is lower once the excess of air works as a heat sink. On the other hand lean and extra-lean combustion is less stable due to fuel dispersion on the combustion chamber volume.

Considering also the speed of combustion on a spark ignition engine it is critical that the cylinder charge has enough and correct motion to assure a complete and stable combustion. To promote rapid combustion, sufficient large-scale turbulence (kinetic energy) is needed at the end of the compression stroke because it will result in a better mixing process of air and fuel and it will also enhance flame development. However, too much turbulence leads to excessive heat transfer from the gases to the cylinder walls, and may create problems of flame propagation (Miller and Lieberherr, 1957), (Wang *et al*, 2007). The key to efficient combustion is to have enough turbulence in the combustion chamber prior to ignition. This turbulence can be created by the design of the intake port.

In cylinder motion is commonly divided in two types: swirl and tumble. Swirl is the rotational movement of the charge around the cylinder axis, while tumble is the movement of the cylinder charge from the upside to the down side of the cylinder regions and back again, around an axis perpendicular to the cylinder axis. Due to the leanness of the mixture there is a narrow range of motion values that allow stable combustion. If motion is reduced the combustion may not be complete due to flame extinguish due to lack of flame propagation speed and on the other hand if motion is too high, combustion may not be complete due to turbulent and unstable flame propagation.

On this study a computer analysis is performed on the pattern and intensity of cylinder charge motion on the two over-expansion strategies described above. The results of this analysis are used to compare with engine test results using the same two strategies.

2. FLOW MODELING

2.1. Swirl number

The swirl number is used to estimate the swirl level. The swirl number is a dimensionless value and can be defined in different ways (Stone and Ladommatos, 1992). A paddle wheel is usually used to measure the level of swirl on an engine (with open end and steady flow of air), using the swirl parameter (Thien, 1965):

$$C_{sp} = \frac{B^2 S_t \omega_s}{4Q} \tag{1}$$

where:
C_{sp}: swirl parameter, dimensionless
B: diameter of the bore [m]
S_t: length of the stroke [m]
ω_s : angular velocity of the swirl [rad.s^{-1}]
Q: flow rate [m³/s]

2.2. Mesh generation

After several attempts, an optimized cylinder head design was developed so its geometry (Fig. 1-a) would generate the intended swirl. The principle of this geometry is that the air flow would be deflected to the right when it passes the bend (Fig. 1-b). Then, the air flows around the valve, creating an angular momentum just prior of entering the cylinder, producing a circular motion within the cylinder.

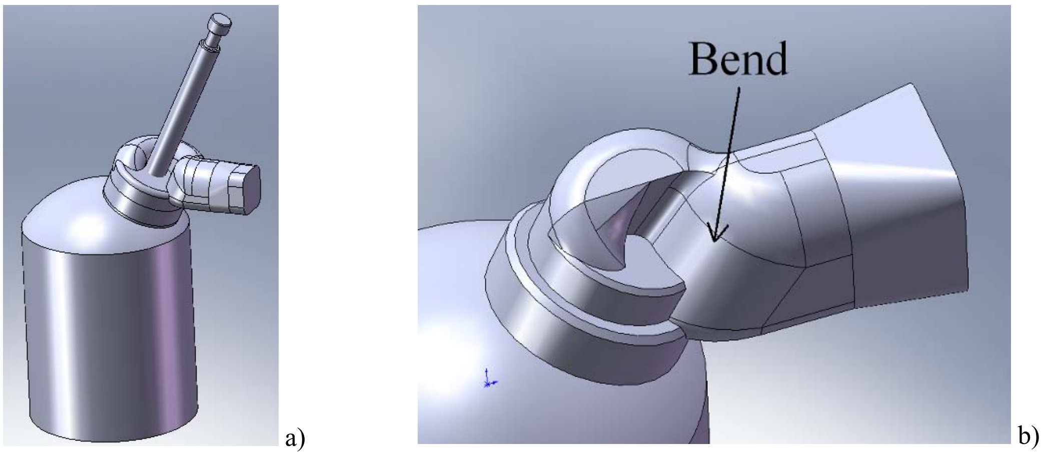

Figure 1. a)Design of intake port; b) Bend on the intake port.

Using this geometry a mesh was generated using GAMBIT. On the simulation of a working internal combustion engine, one of the boundaries, the piston, moves and so the mesh has to be modified as the boundary moves. On the case of the internal combustion engine the layering option was used and layers of cells were added or removed as the piston moved downward or upward, respectively. The cells are split or merged (Fig. 2) using the *constant height* or *constant ratio* option. When using the *constant height* option the cells are split to create two new layers, one with a height of H_{ideal} and one with a height of H - H_{ideal}. H is the height in cell layer j (Fig. 2) and H_{ideal} is a constant value and can be set separately for each region. When using the *constant ratio* option the cells are split so that the ratio of the cell height is α_S.

The layers are split according to Eq. 2:

$$H_{min} > (1+ \alpha_S)H_{ideal} \tag{2}$$

With H_{min} the minimum cell height and α_S is the split factor.

A layer is collapsed according to Eq. 3:

$$H_{min} < \alpha_c H_{ideal} \tag{3}$$

α_c is the collapse factor.

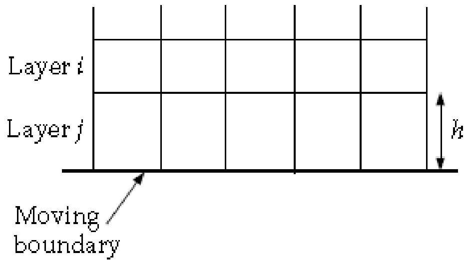

Figure 2. Layering.

2.3. Boundaries

The boundary types that have to be defined are given in Tab. 1. The faces that are not defined are automatically considered as walls.

Table 1: Boundary conditions piston movement

Boundary name	Boundary type	Faces
Piston	Wall	Bottom of the cylinder, this is only one face
Inlet	Pressure inlet	The inlet of the model

The mesh for the movement is shown at different crank angles (Fig. 3). The mesh created in the cylinder as the piston moves downwards is created according to the layering method (as described above), so this has to be a structured mesh. In this case a hex/wedge cell type was used to be able to connect the piston zone to the unstructured mesh in the combustion zone.

The mesh is build up out of 36 429 elements at 0° crank angle, and 91 116 elements at 180° crank angle.

3. FLOW MODEL RESULTS

Simulations of engine fluid flow were preformed at 2000 rpm at defined cross sections of the cylinder as represented on Fig. 4 (Vanhaelst, 2008). Three engine cycles were simulated, firstly the Otto cycle and then the Miller cycle EIVC and LIVC. The simulation on each cycle was limited to the intake and compression strokes, which correspond to the downward movement of the piston and the reverse movement until the combustion takes place.

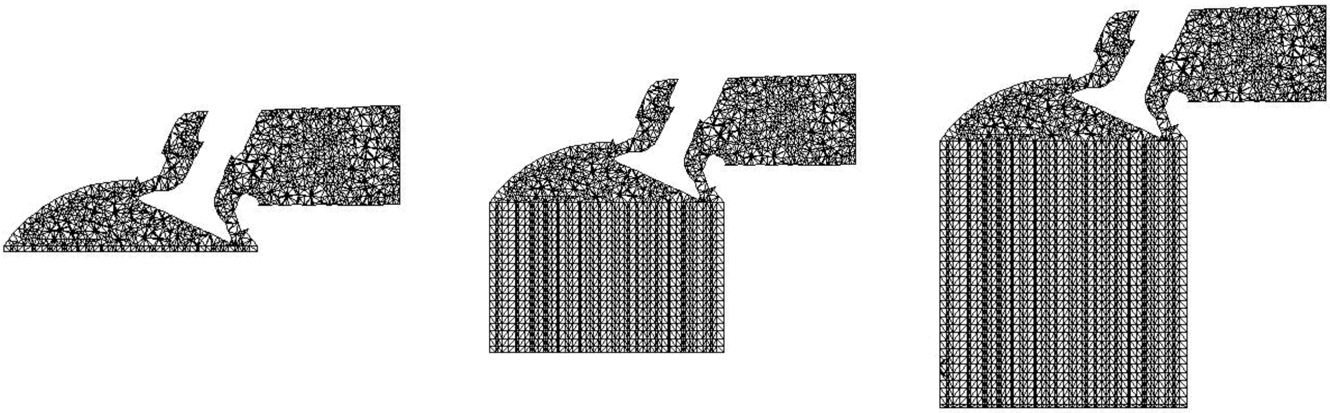

Figure 3. Mesh motion at 0°, 90°, 180°.

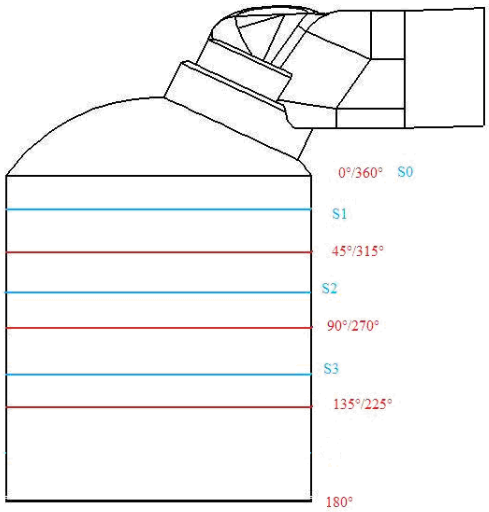

Figure 4. Cross sections for swirl measurement.

3.1. Otto cycle

On the simulation of the Otto cycle, it is assumed that the intake valve opens at 0° crank angle, at Top Dead Centre (TDC), which marks the beginning of the inlet stroke. The valve closes at 180° crank angle, at Bottom Dead Centre (BDC), when the compression stroke begins.

During intake and at 45° crank angle in section S0 there are two vortices visible (Fig. 5). This is because of the air that is sucked in through the valve. But the vortex in the upper left corner disappears in section S1. At 90° crank angle the vortices at section S0 are not so visible, but in the sections S1 and S2 the vortices are clearer, with one big vortex that creates a swirl motion in the cylinder. When the crank angle is 135° the swirl motion is visible in the entire cylinder. At 135° the vortex is approximately in the centre of the cylinder. At 180° is visible that the flow speed increases at the bottom of the cylinder.

At 180° crank angle the velocity of the swirl was around 10m/s and the swirl motion was still visible (Fig. 5). After BDC the velocity of the swirl is lower and decreases as the piston reaches TDC. However, the swirl motion is still clearly visible at the end of the compression stroke in section S0 (Fig. 6).

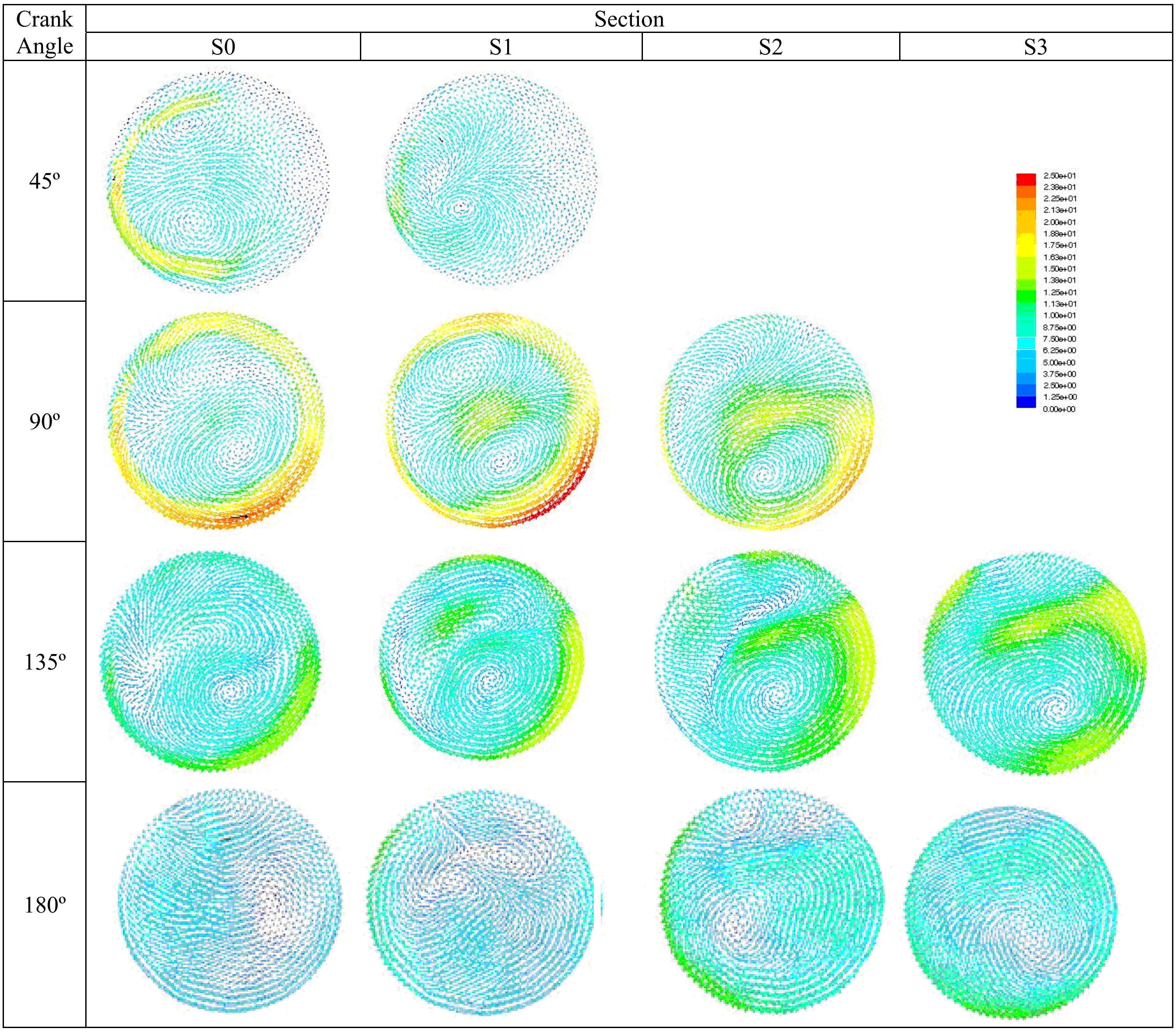

Figure 5. Swirl for the Otto cycle during intake stroke.

3.2. Miller EIVC

On the Miller EIVC cycle the inlet valve closes at 90º after TDC, and the influx of air finishes at that location, therefore reducing the available swirl. The result for 180º can be seen in Fig. 7. The velocity at 180° crank angle was already lower in this cycle compared to the velocity at the same point in the Otto cycle (Fig. 5). At the beginning of the compression stroke (Fig. 8) the velocity is comparable to the velocity at 180° crank angle. It decreases as the piston nears the TDC. At TDC the swirl motion is low, when compared with the Otto cycle.

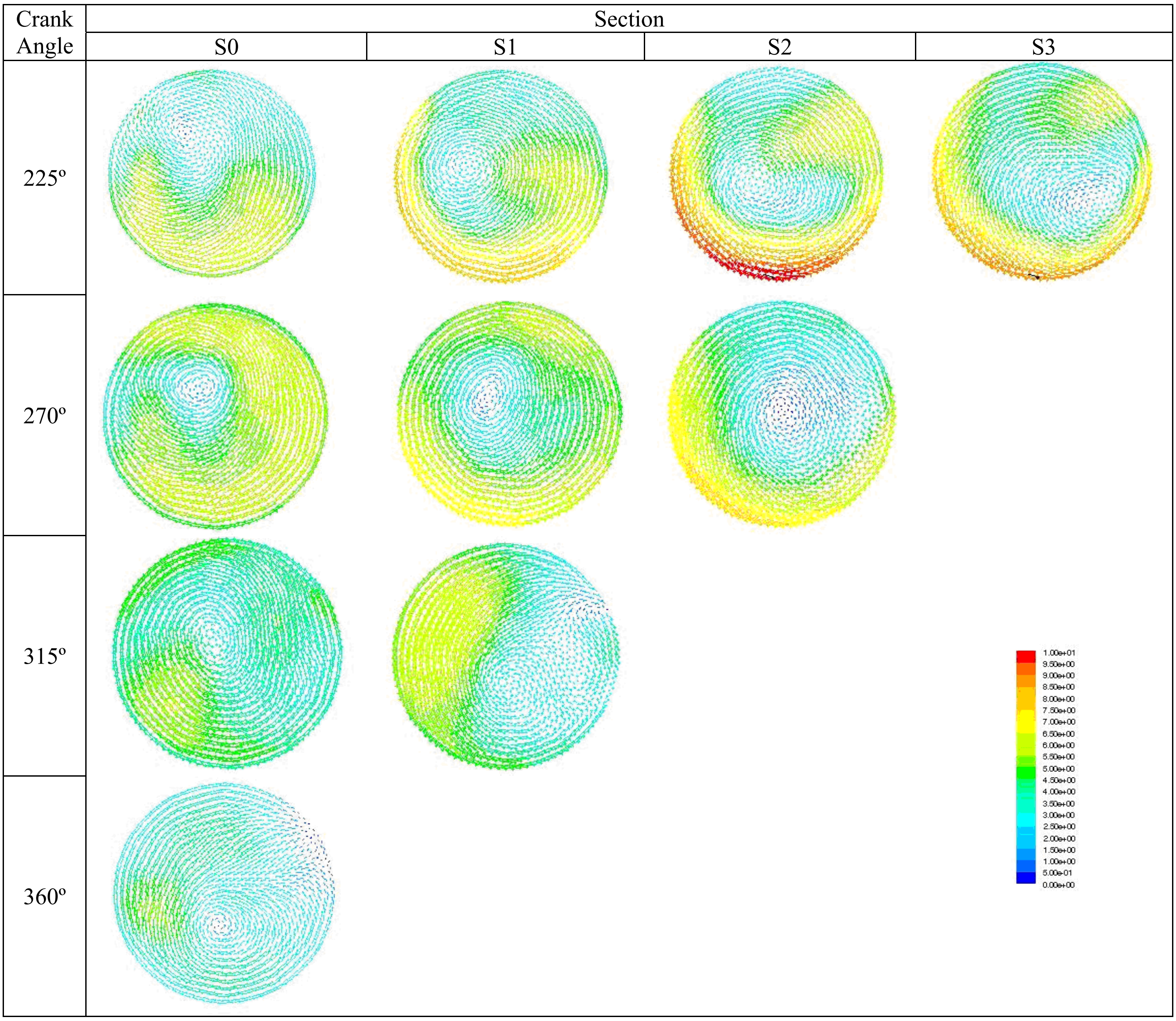

Figure 6. Swirl for the Otto cycle during compression stroke.

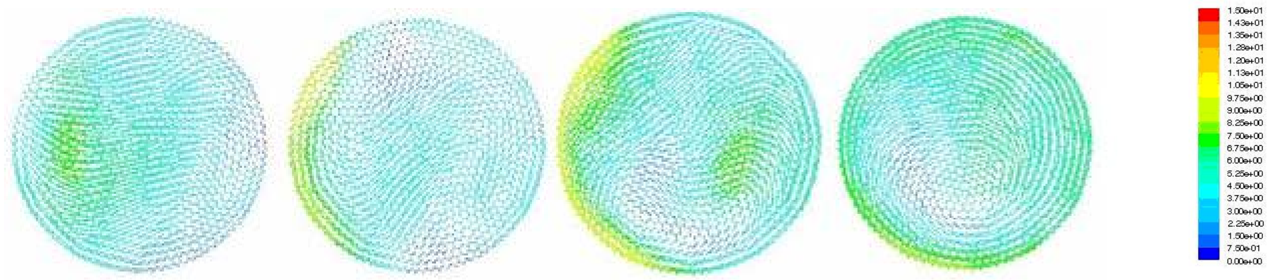

Figure 7. Swirl for Miller EIVC cycle, during intake stroke at 180º

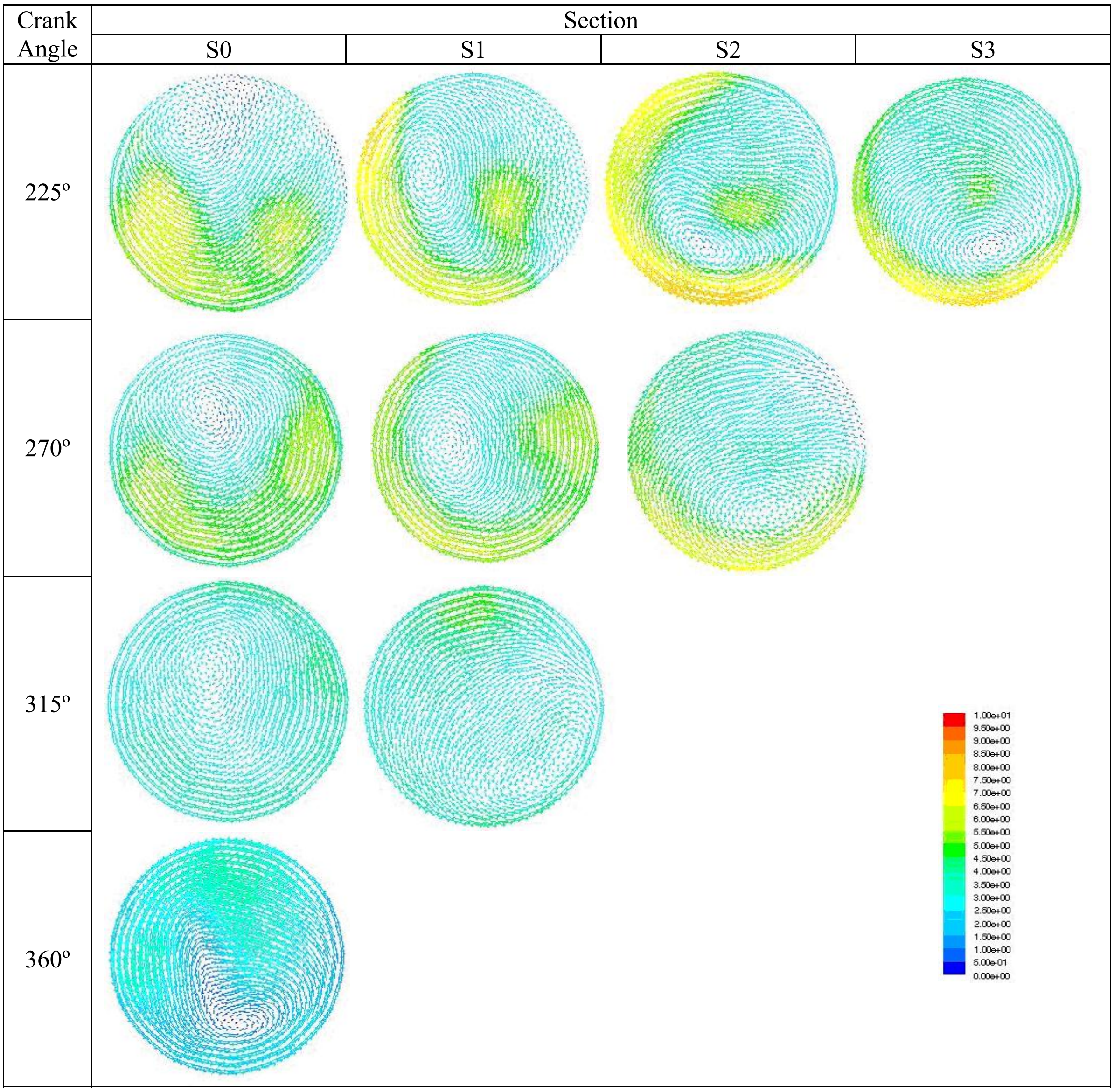

Figure 8. Swirl for the Miller EIVC cycle during compression stroke.

3.3. Miller LIVC

In the case of the Miller LIVC cycle, the intake stroke is similar to the intake stroke of the Otto cycle, once the valve opening event is equal in both cycles, until piston reaches BDC. The upward movement of the piston on the Miller LIVC cycle does not represent an effective compression because the intake valve is still open and so there is a back flow ınto the intake manifold. The valve closes just at 280º, and only at that instant the effective compression starts.

At 225° crank angle the velocity (Fig. 9) is higher compared to the velocity at 180° crank angle (Fig. 5). This is the result of the back flow into the inlet manifold. At 280° crank angle the compression stroke starts and the velocities significantly reduce when compared to the velocity level before 280° crank angle. Locally the velocity increases as the piston nears the TDC. The swirl motion becomes more visible as the piston nears the TDC (360° on Fig. 9).

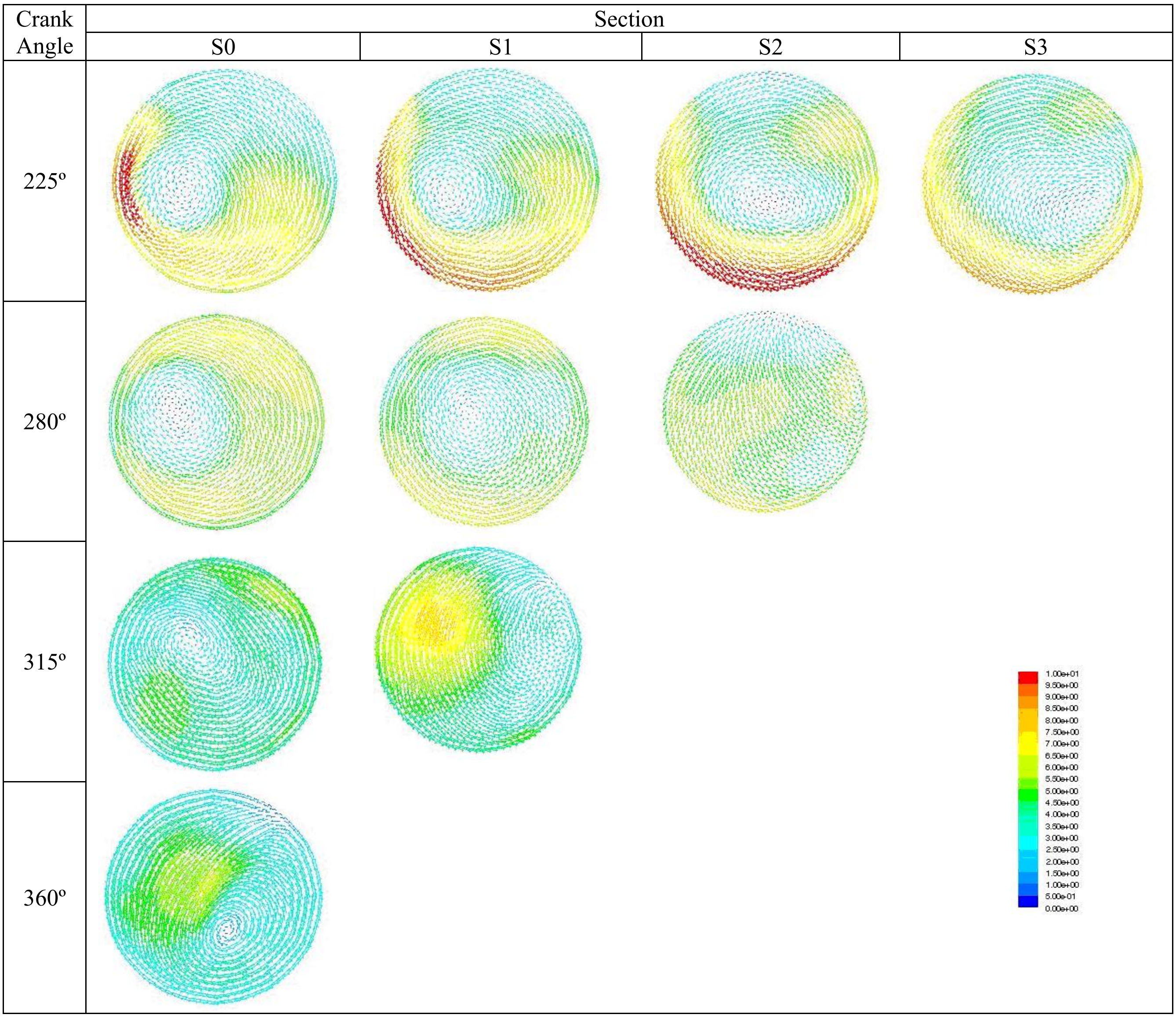

Figure 9. Swirl for the Miller LIVC cycle during compression stroke.

3.4. Comparison between cycles

As the inlet valve is opened for both cycles, the swirl in the cylinder is identical for the Otto cycle and the Miller LIVC cycle during the initial 180° of the piston movement. The Miller EIVC cycle has a different swirl after 90° crank angle, as the inlet valve closes at that position. At 180° crank angle there is almost no swirl motion visible in the Miller EIVC cycle (Fig.7). The swirl after 180° crank angle has different motions and velocities for the three cycles. The Otto cycle has the stronger swirl motion for crank angles until 315° crank angle. However the motion is comparable at 315° crank angle in section S0 between the Otto cycle and the Miller LIVC cycle, with the velocity in the Otto cycle higher than in the Miller LIVC cycle. But at 360° crank angle the swirl motion and velocity are higher in the Miller LIVC cycle (Fig. 9). At the end of the compression for the Miller EIVC cycle there is a swirl motion with very low speeds compared to the other two cycles. The Miller LIVC cycle gives the highest results for the swirl at the end of the compression stroke.

3.5. Steady state simulations

The simulations presented above allow knowing the more favorable strategy for the Miller cycle to obtain the better combustion conditions. To confirm that the intake duct configuration is the optimal, swirl intensity under steady state conditions had to be assessed in a running engine. These assessments were firstly made with computer models (using FLUENT) and later on the real engine head that had been optimized during running tests (Coene, 2008). Under these conditions the engine is tested without the piston and a constant air flow is forced through the intake valve.

Fig. 10 shows the evolution in swirl (calculated with Eq. (1)) at different locations (from TDC) for a valve lift of 2,5 mm for a speed of 4000 rpm.

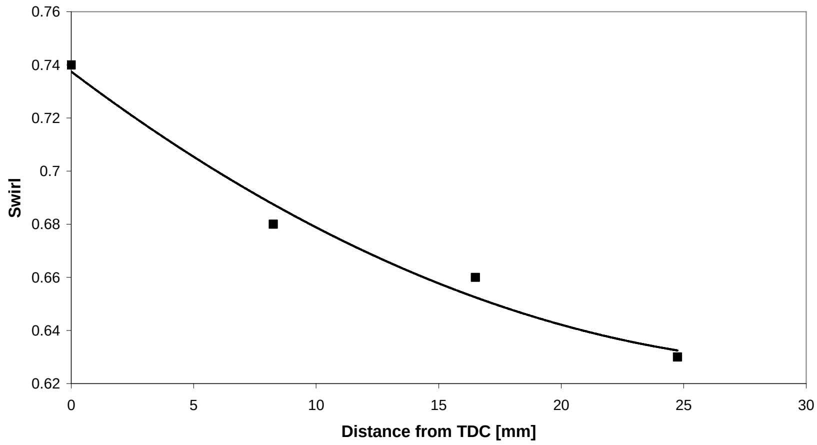

Figure 10. Evolution in swirl (2.5mm valve lift).

4. ENGINE TESTS

Tests in a running engine were performed on a single cylinder engine. This engine was a commercial Diesel engine Yanmar L48 AE, a 211 cc direct injection air cooled Diesel engine. The use of an original Diesel engine allowed for a direct comparison between the Diesel cycle and the Otto cycle with and without over-expansion presented in an earlier study (Ribeiro and Martins, 2007). Several modifications were introduced in order for the engine to run under S.İ. cycle, such as engine head modification to have a spark plug, introduction of a new intake manifold with throttle valve and fuel injector. The original compression ratio of approximately 20:1 also had to be changed. Different pistons with different combustion chamber sizes were used. The combustion chamber on the piston was of cylindrical shape with constant height and different diameters, allowing compression ratio variation from 11.5:1 up to 17.5:1.

4.1. Over-expansion variation

On the laboratory tests, over-expansion of the engine was achieved through the use of different camshafts with different timings for the intake valve events, corresponding to different levels of over-expansion. As a baseline the original intake cam profile was used and considered as the Otto cycle with no over-expansion. After this, the intake valve closure timing was delayed for several crank angle degrees, performing LIVC. More camshafts were used but changing the intake valve closure event to earlier crank angles, performing the EIVC. All the cam profiles tested are represented on Fig. 11.

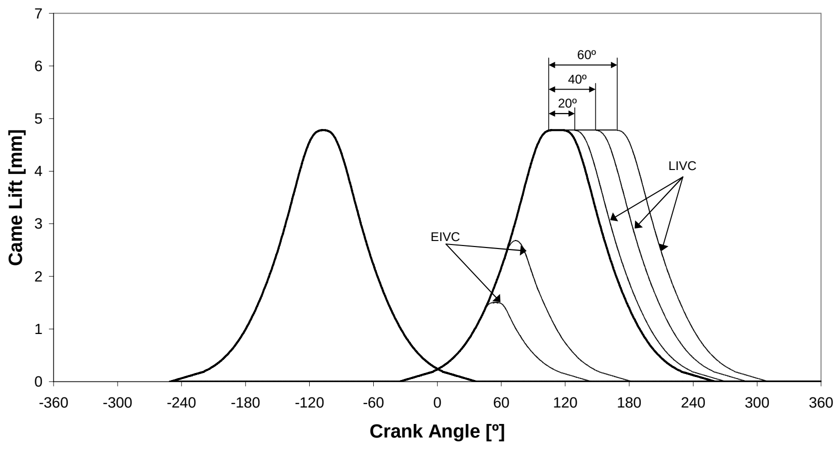

Figure 11. Different cam profiles tested.

4.2. Swirl measurement

Once the original engine was Diesel cycled the intake port was of helical shape to produce intense swirl to the cylinder charge, in order to improve fuel mixture on the air during fuel injection. This was the cause for poor performance of the engine during the first spark ignition tests. The intake port had to be redesigned to reduce the swirl and allow for better burning conditions. Swirl levels are presented on Fig. 12 for different air flows into the cylinder and for different distances from TDC. Due to the poor combustion conditions the engine head was modified in order to reduce swirl. After modification, the swirl reduction goes from 60% up to 80%, depending on the distance at which it is measured and the air flow. For the same speed the swirl measured on the modified engine head under steady state conditions and using a paddle wheel was around 0.3 for an engine speed of 2000 rpm (the test rig could not go beyond this). Using the similarity theory (Martins, 2006) to compare the two engines, the relation between the swirl values should be around 50%. The rig tests results compare well with the data from the computer model presented on Fig. 10. As referred by Heywood (1988), a difference of up to 30% from calculations to real tests may exist.

As the results from the steady state modelation are in agreement with the “moving piston” modelation, and are also in agreement with the measured conditions for optimized experimental combustion, we can conclude that the intake design shown in Fig.1 should create an optimized swirl on the new engine.

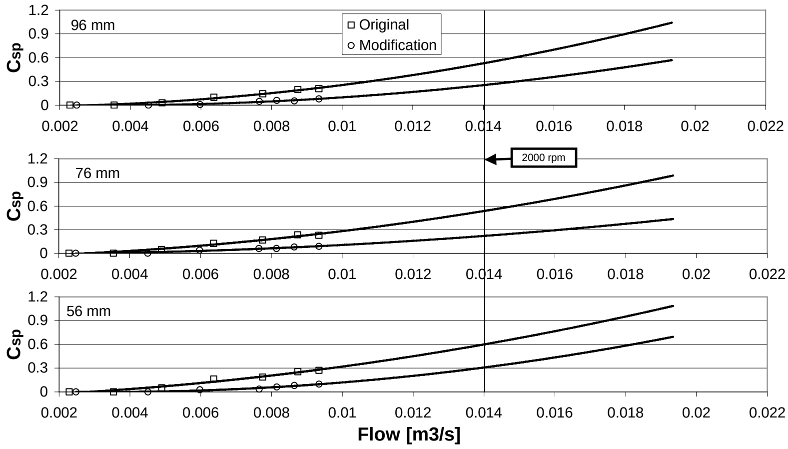

Figure 12. Swirl results after and before the intake duct modification.

4.3. Tests results

The running engine was tested on the Otto configuration at several positions of the throttle valve. On the Miller configuration the runs were done with the different camshafts referred above. With each camshaft the piston was changed so that an optimal compression ratio was used. Fig. 13 presents the results of these tests, showing the specific fuel consumption as a function of load or break mean effective pressure.

As it can be seen, there is a significant improvement of the engine thermal efficiency as load is controlled via the intake valve closure event instead of using a throttle valve, as it occurs on the Otto cycle. Comparing just the two versions of the Miller cycle, the one with late intake valve closure (LIVC) gets a better improvement than the one with the early intake valve closure (EIVC). The reasons for this difference to exist are the increase of pumping work and fluid flow losses due to the total valve lift reduction (and consequently the valve opening section) and also the better control of swirl induced by this strategy.

5. CONCLUSIONS

A computer model created in GAMBIT was used for FLUENT simulations. Simulations were made to measure the swirl on a spark ignition internal combustion engine working under the Otto cycle and the Miller cycle, with early and late intake valve closure. From these simulations the Miller cycle with late intake valve closure was the one with better swirl conditions.

Steady state swirl was also measured firstly on a computer model using FLUENT and later on a swirl measuring rig. The results were equivalent. The engine tested on the swirl measuring rig was used for engine tests, operating again on the Otto and the two referred Miller cycles. Under real tests, the Miller cycle with late intake valve closure had the best performance in terms of fuel consumption. The reasons for these results can be justified due to less fluid flow losses, but also due to better burning conditions provided by the increased swirl at the combustion period.

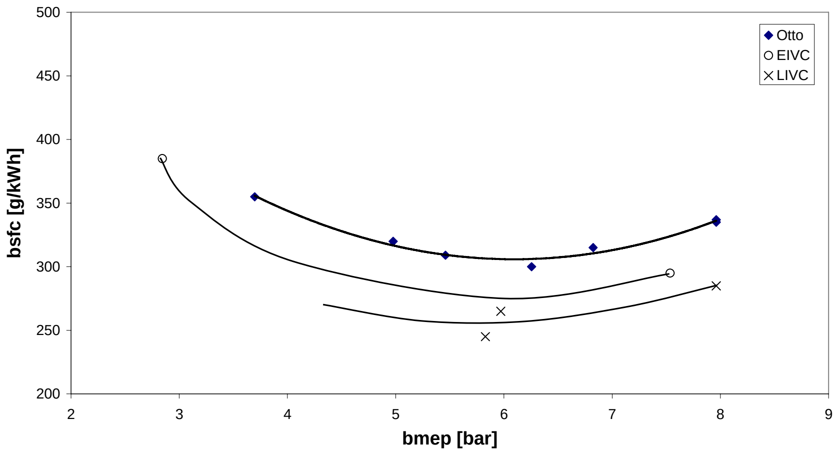

Figure 13. Test results at 2000 rpm.

6. ACKNOWLEDGEMENTS

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