The Effect of Swirl on the Combustion Process on a D.I. Diesel Engine

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ABSTRACT

In this work, a study about the influence of the air flow motion upon the combustion process of a one-liter per cylinder direct-injection Diesel engine is presented. In order to achieve different swirl intensities in the cylinder, three design parameters have been changed: the cylinder head, characterized by its swirl number, the piston, characterized by the piston chamber diameter, and the inlet duct, characterized by the frequency parameter. In this way, a wide range of swirl speeds at the end of the compression stroke could be tested by keeping all the other influencing parameters constant. The characterization of the air motion produced by the cylinder head has been contrasted using the paddle-wheel and the laser-Doppler anemometry techniques, and also the influence of turbulence has been examined through this technique. Finally, the results are discussed and the swirl influence on the engine performance is analysed.

1. INTRODUCTION

It is well known that the in-cylinder air motion strongly affects the combustion process of fast direct injection Diesel engines, and consequently their performances. In fact, the swirl intensity increases the velocity of the air entraining into the fuel jet, which means an acceleration of the mixing process. Nevertheless, the higher swirl intensities do not undefinitely provide the better performances, and usually, for a given injection system, an optimal swirl intensity can be found. Although Henein (1) justifies this fact because excessive swirl might cause overlap of the sprays and an increase in unburned hydrocarbon emission, it seems difficult to establish theoretical reasons for that penalty.

The objective of this work is to evaluate the influence of swirl intensity on the quality of the combustion process, and on the engine performances, namely, specific fuel consumption, smoke emission and Nitric Oxide emission. The study has been focused on the engine behaviour at high speed and full load conditions.

The results discussion must take into account that other parameters which have their own influence on the combustion process, do vary when swirl intensity does, that is, the turbulent intensity, and consequently the mixing efficiency, the intake air mass flow rate, and consequently the axial velocity distribution, the squish velocities, etc.

2. METHODOLOGY

This study has started from the results of a parametric experimental work on a single-cylinder D.I. Diesel engine. In order to obtain different swirl intensities, three design parameters were changed in the engine: the cylinder head, the piston, and the inlet duct (see table 1). Intake conditions, fuel, oil and water temperatures, and air and fuel mass flow rates were kept constant during the tests, so that accurate comparisons were possible.

Cylinder head

During the intake process the cylinder head throuh its inlet duct geometry imposes the in-cylinder tangential velocity distribution, which can be characterized by the angular momentum flux or the swirl intensity, the axial velocity distribution and the turbulent intensity.

A wave action model (2) is used to calculate the angular momentum that is generated in each case. This calculation is performed by integrating the following expression all along the intake process:

$$\frac{d\,M_{ang}}{dt} = \frac{\pi}{8} \cdot \frac{D^2}{V_D} \cdot SN \cdot \dot{G}^2$$

where SN represents the swirl number for a given cylinder head and for a given valve lift. The swirl numbers, as well as the discharge coefficients, are measured at the steady flow rig, as described in the next paragraph. G is the instantaneous mass flow rate, which is evaluated in relation to the discharge coefficient and the manifold characteristics. Also the engine running conditions are inputs of the model.

Inlet manifold

The inlet manifolds are characterized by their frequency parameter, which expresses the relation between the manifold natural frequency and the engine exciting frequency, that is:

$$Q = \frac{a/2L}{n/60} = \frac{30}{n}\,\frac{a}{L}$$

The manifolds can also modify the total intake air mass, m (their geometry is an input to the mentioned wave-action model), its distribution, the angular momentum flux and the turbulent intensity with respect to the cylinder head without manifolds.

Piston

The piston geometry, which is characterized by the combustion chamber diameter (D_b), modifies the velocity field along the compression and expansion strokes. Not only the swirl velocities, but also the squish, the spray inpingement, the air entrainment, and other phenomena change as consequence.

If a forzed vortex is supposed to develop in the combustion chamber, the air velocity at the bowl wall at T.D.C. can be considered as a characteristic parameter to evaluate the swirl intensity during the combustion process. It can be obtained from:

$$u = \frac{M_{ang}(TDC)}{1/4\ D_b\ K\ m}$$

where a friction coefficient depending on the bowl shape and on its offset (3) has reduced the angular momentum along the compression stroke from its initial value at the intake valve closure.

Table 1. Design parameters

cylinder head			piston		inlet duct		
	CDM	SNM		Db(mm)		Q	L(mm)
C1	0.407	2.941	P1	50	D1	13	150
C2	0.385	3.290	P2	55	D2	6	477
C3	0.439	2.705			D3	3	1084
C4	0.388	3.443					

In this way, a wide range of swirl intensities at the end of the compression stroke could be tested by keeping all the other control parameters constant.

A phenomenological model (4), which also uses the mentioned inlet flow data and running conditions as input data, obtains the parameters that characterize the air motion during the combustion process, as well as the combustion process itself. Those are the following:

- "Relative Spray Penetration Correlation" (RSPC). It was defined by Ball (5) as the relation between the spray penetration at the ignition and the spray center line lentgth until the wall inpingement. It would be equal to one if the combustion started at the same time as the spray inpinges on the wall. The same author estimates an optimum for his parameter between 0.9 and 1.1. Lower values indicate not to take advantage of the entrained air, and higher values anounce the presence of unburnt fuel in contact with the wall, what may delay the combustion process and increase hydrocarbon emissions.

- "Air-to-Fuel Momentum Ratio Correlation" (AFMRC). Proposed by Bassoli (6), it relates the in-cylinder angular momentum flux at T.D.C. with the injected fuel momentum:

$$AFMRC = \frac{K \cdot m \cdot \omega \cdot D_b/2}{m_f/N_h \cdot U_0}$$

The optimum estimated value for I.D. Diesel engines turns around 7 and 10, or a bit higher in the case of re-entrant chamber.

- "Croswind Velocity Correlation" (CWVC). Timoney (7) defined it as the difference between the air velocity of the supposed forzed vortex at the wall and the tangential component of the mean spray velocity at the inpingement:

$$CWVC = \omega D_b/2 - u_{it}$$

Although this parameter is not non-dimensional, some studies performed on differently sized engines have settled on about 8 m/s the optimum value for small Diesel engines.

3. STEADY FLOW RIG TESTS

To measure the flow characteristics produced by the inlet port geometry of the cylinder heads, tests in a steady flow test rig have been carried out. This experiments have been performed by means of the paddle wheele technique, as proposed by the A.V.L. method (8), and with Laser-Doppler Anemometry (LDA).

To understand the effect of the air motion in the combustion process, the study has been focused on cylinder heads C3 and C4, because of their significative behaviour with respect to the combustion process, as will be exposed later.

The measurements were performed first at a section of the duct in the test rig, placed at a distance of 1.75 times the bore of the engine (see fig. 1). In this case the A.V.L. swirl number was obtained, and the axial and tangential velocity fields and the turbulence intensities were measured for eight valve lifts with the LDA technique.

To study the effect on the combustion process of the turbulence produced in the inlet flow, which some authors consider important (9), the turbulence production by the inlet flow has been measured with LDA.

This second set of measurements were carried out at the discharge section of the valves. In this tests, both the tangential and radial components of the velocity and their turbulence intensities were measured at eight equispaced zones around the valve, the total number of measurement points being dependent on the valve lift. To reduce this data the geometric mean value of the axial and tangential turbulence intensity was calculated at each measurement point. A plot of this results for both cylinder heads is shown later in figure 11.

The LDA results of the first set of measurements with the two cylinder heads showed that at lower valve lifts, a complex pattern for the tangential and axial velocities is established. Figure 2 shows the measured flow pattern for some significative valve lifts of cylinder head C4. Zones of reverse flow (shadowed areas), and two counterrotating vortexes are present at the lower valve lifts. At higher lifts, a fairly uniform swirl is generated, but the axial flow field is still non-uniform, the higher axial velocities appearing near the walls, where the tangential ones are higher too.

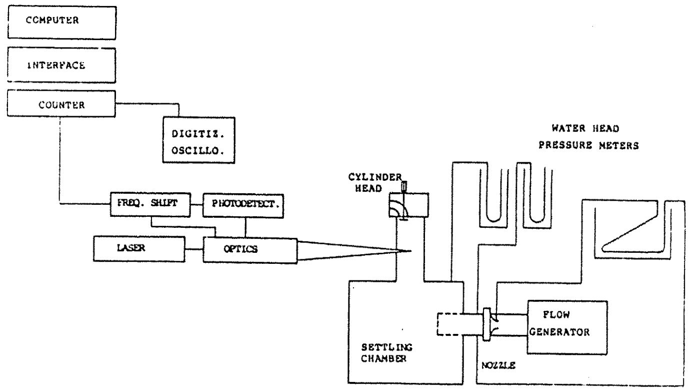

Figure 1. Flow test rig and LDA system.

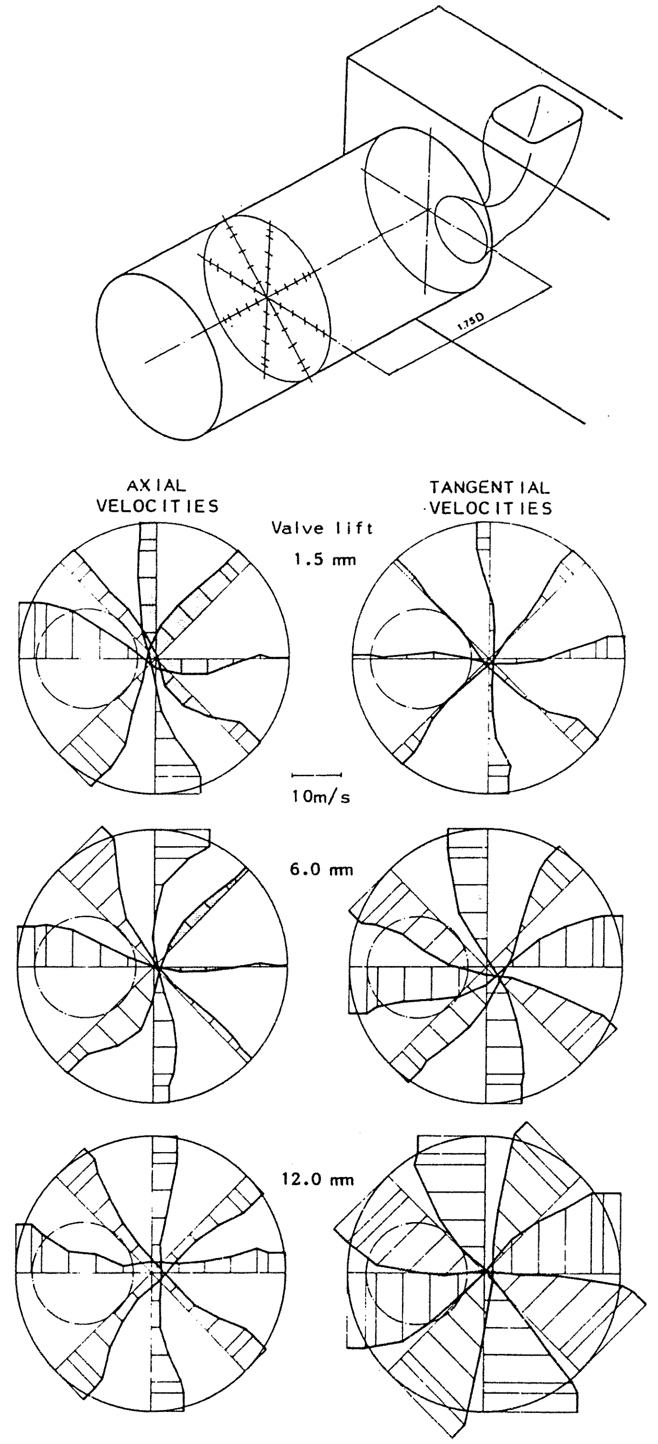

Figure 2. Axial and tangential velocity fields for cylinder head C4.

With this experimental results the equivalent AVL swirl number of the two cylinder heads were calculated. The results of this calculation yielded always higher values of the LDA swirl number, and an almost linear relation between those and the paddle-wheel values was observed. Therefore, the AVL swirl numbers were multiplied by the proper coefficient and the corrected values were used in the following calculations.

4. ENGINE TESTS

All the experimental tests have been carried out on a one-liter single-cylinder D.I. Diesel engine. The main characteristics of the experimental set-up (figure 3) are described next.

The following parameters are kept under very narrow margins around the indicated values during the tests in order to guarantee valuable results and adequate comparisons:

- Engine speed:	2800 rpm
- Water temperature:	90 ºC
- Oil temperature:	100 ºC
- Fuel temperature:	35 ºC
- Intake temperature:	30 ºC
- Exhaust differential pressure:	1020 mm H_2O
- Intake pressure:	970 mbar
- Relative fuel/air ratio:	0.725

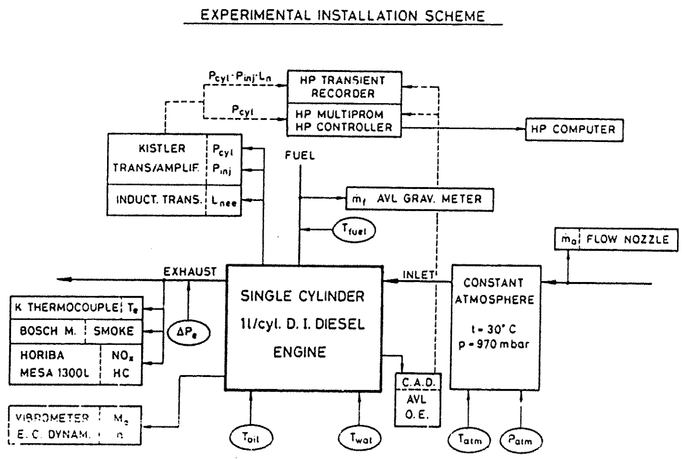

Figure 3. Experimental set up, including the engine test bench and the instrumentation.

Also the injection parameters (injection timing, injection pressure) have been maintained, and a 4 holes injector with 0.3 mm diameter has been used.

This experimental procedure finds an exception in the tests with manifolds (Q=6 and Q=3), in which a total intake air mass flow rate similar to that provided in the tests without any manifold (Q=13) is fixed, instead of controlling the intake pressure.

5. DISCUSSION OF RESULTS

In the following diagrams the swirl intensity at T.D.C. can always be found in relation to u_0, which means the swirl intensity parameter for the reference test (C1, P1, D1).

In terms of specific fuel consumption and smoke emissions, figures 4 and 5 show that cylinder head C1 behaves better than C4 with pistons P1 and P2 and without any manifold. This result already proves the existence of a limitation in the swirl optimization.

It is also remarkable that piston P2 provides higher consumptions and much higher smoke emissions than P1 (about 2 Bosch units for the same swirl intensity). This result suggests that the modification of the combustion chamber diameter has stronger influence upon the combustion process by itself, than by means of the associated variation of swirl intensity. It may be due to the variation of the squish intensity and of the wall impingement.

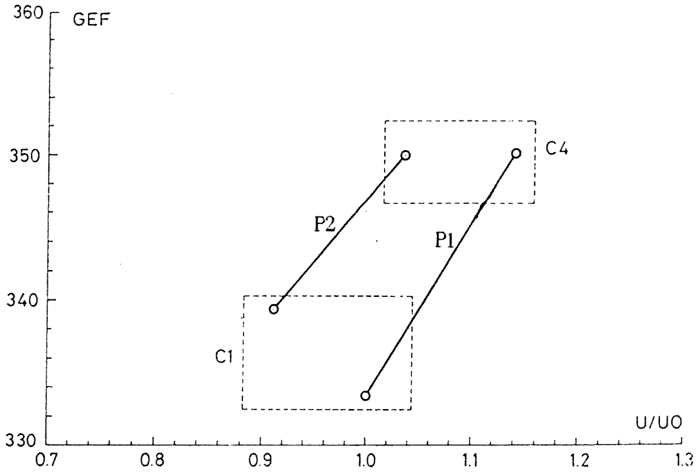

Figure 4. Specific fuel consumption vs. swirl intensity, when no manifold was equiped.

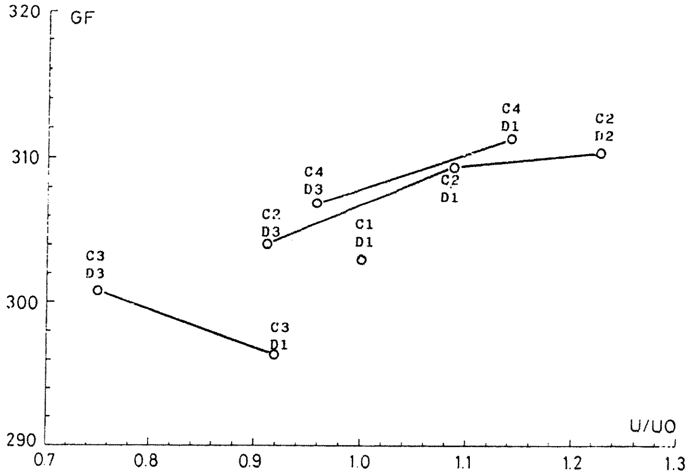

Figure 6. Specific fuel consumption vs swirl intensity, when piston P1 was equiped.

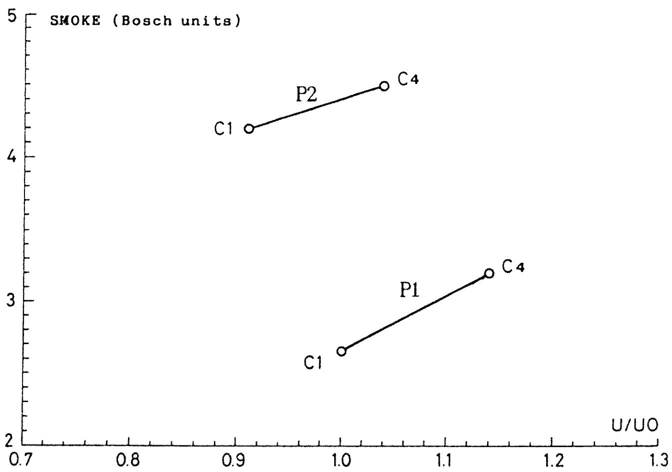

Figure 5. Smoke emission vs swirl intensity, when no manifold was equiped.

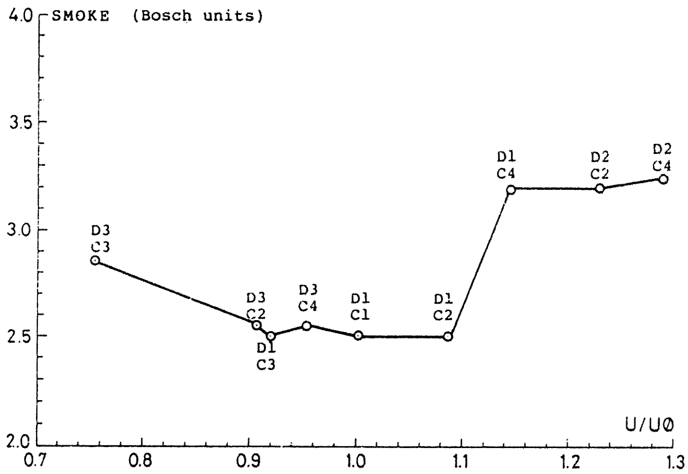

Figure 7. Smoke emission vs swirl intensity, when piston P1 was equiped.

Figures 6 and 7 show all the results of specific fuel consumption (including pump losses) and smoke emissions for piston P1. A minimum value of about 2.5 B.U. can be observed for every cylinder head in the case of smoke emissions, which is located between 0.9 and 1.1 of u/u_0. However, no common minimum can be found for fuel consumption, although each cylinder head has its own minumum around a u/u_0 value of 0.9. The lowest of these minima correspond to cylinder head C3, and C2 and C4 have the higher ones, with similar values.

Justly cylinder heads C1 and C3, which present the lowest fuel consumptions, have also similar SN curves, as shown in figure 8: They have lower mean SN than cylinder heads C2 and C4, but higher SN values for low valve lifts.

Turning to the engine performance, it seems interesting to relate the specific fuel consumption to the mentioned parameters characterizing the combusting flow field, as AFMRC (figure 9) and CWVC (figure 10). It has only been done for the tests without manifolds because the combustion model does not record their influence. Both graphics show that although cylinder heads C1 and C3 maintain low consumptions in a wide range of these parameters, they reach the minimal values for AFMRC=7.5 and CWVC=7.7, which quite agrees with literature (5) (6) (7). A different behaviour

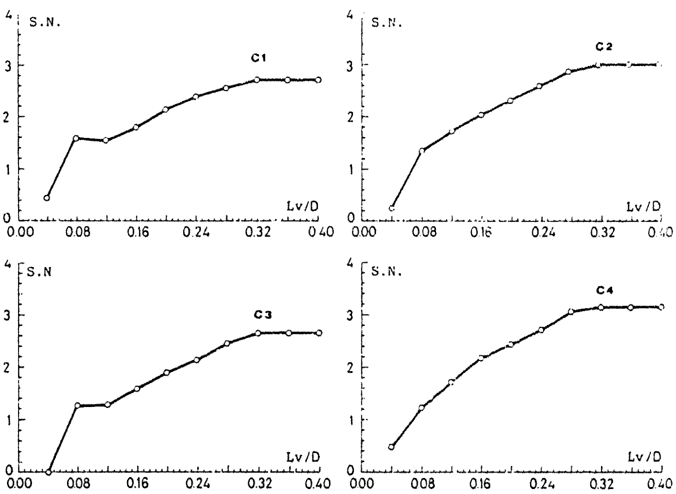

Figure 8. Swirl numbers vs valve lift for each cylinder head.

can be noticed for C2 and C4, whose high fuel consumption values seem to be insensitive to the variations of these parameters, even when they get next to the supposed optimal values. Finally, no correlation was observed with parameter RSPC.

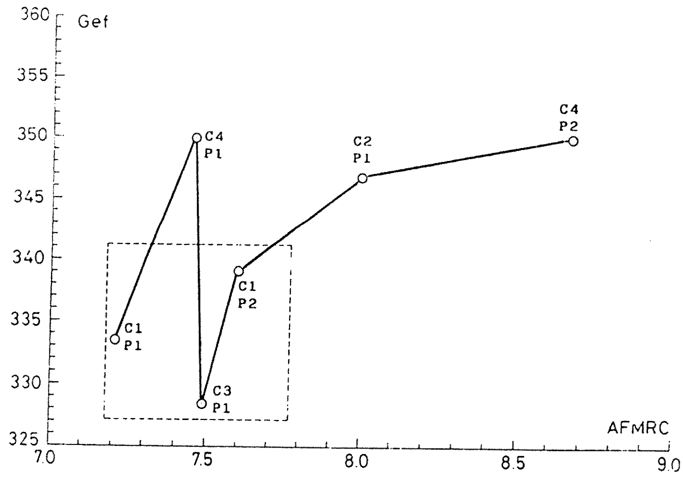

Figure 9. Specific fuel consumption vs AFMRC.

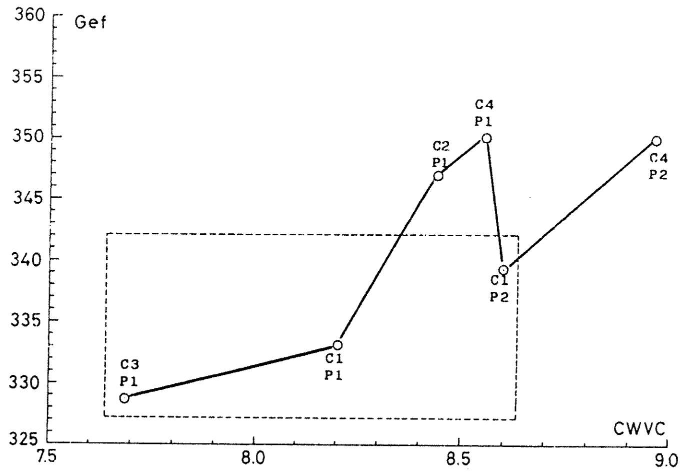

Figure 10. Specific fuel consumption vs CWVC.

These conclusions ratify the cuantitative differences which the Gef vs u/u_0 curves showed between the couples C1-C3 and C2-C4, and prove that the optimization of swirl intensity is not enough to reach an efficient combustion process. In fact, it seems that the characteristics of the flow during the injection and combustion processes are greatly affected by their history, that is to say, by the characteristics of the intake flow.

In order to justify the differences between these two couples of cylinder heads, figure 11 shows the quadratic mean value of the turbulent intensities for cylinder heads C3 (as representative of the couple C1-C3) and C4 (as representative of C2-C4). It can be noticed that C3 provide higher turbulence intensities than C4. If these measured data are used as boundary conditions of a k-ϵ turbulence model, the evolution of the averaged turbulent intensity proves, as figure 12 shows, that the differences are maintained all along the compression stroke. Smaller turbulence levels provide in this case lower smoke formation and better efficiency for cylinder heads C1 and C3.

Finally, figure 13 shows that in the case of the tests with piston P1 the NO emissions decrease with swirl intensity. In the figure, the test with high intake pressure (970 mbar) have been plotted separately from those with lower intake pressure. The mentioned tendency, which is the same in both cases, is opposite to that of smoke emissions since its minimum. The tests with piston P2 provide lower NO emissions, which is also coherent with the higher smoke levels that they also

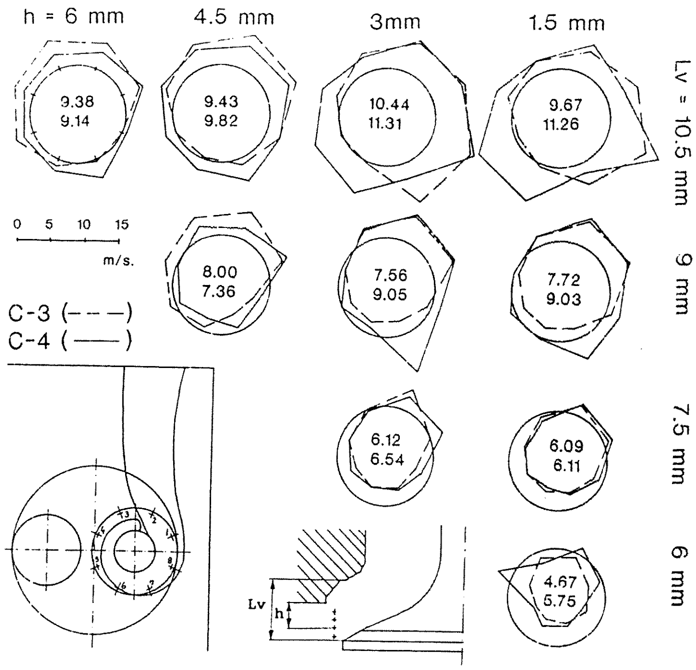

Figure 11. Turbulent intensities at the valve discharge section, for C3 and C4. Numbers inside the circles indicate the averaged turbulent intensities for the eight points (C3 above, C4 below).

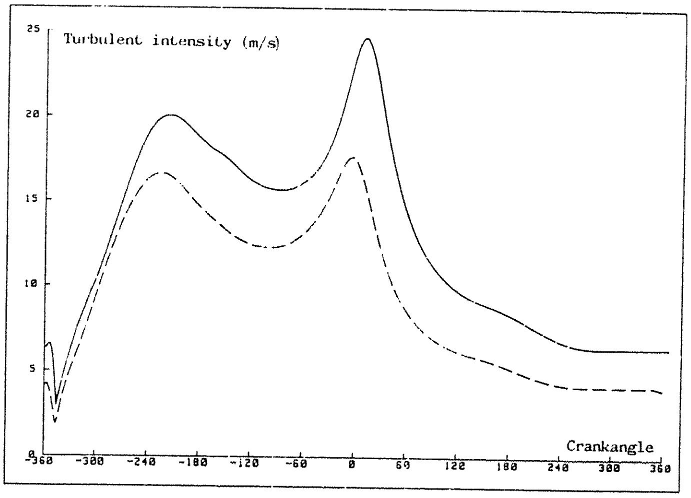

Figure 12. Evolution of averaged turbulent intensities with the engine crankangle.

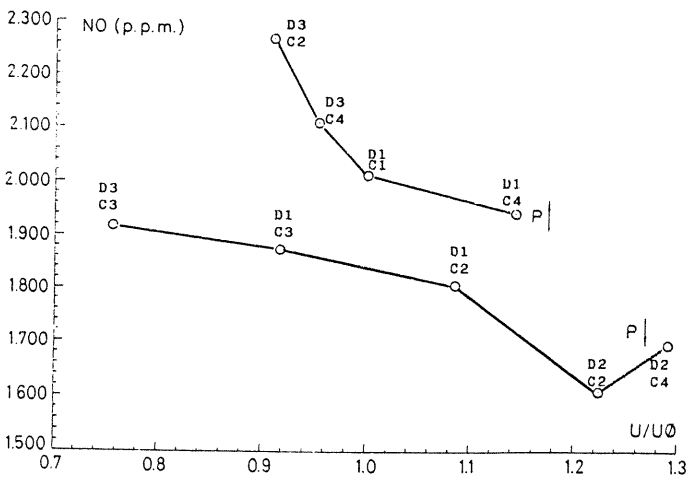

Figure 13. NO emission vs. swirl intensity, when piston P1 was equiped.

provide, as observed in figure 5. Summarizing, NO emissions trend to decrease with swirl but they are affected by two other variables, namely the chamber diameter and the intake pressure.

6. CONCLUSIONS

- The AVL method and the LDA technique have been applied to characterize the flow produced by the inlet duct. An almost linear relation between swirl numbers calculated by both techniques was found, and these values were used as input data for in-cylinder flow models.

- Engine tests with different cylinder heads, inlet manifolds and pistons have been carried out, and fuel consumption, smoke and NO emissions have been measured as main performances.

- A phenomenological combustion model has been applied to calculate several parameters that characterize the evolution of the combustion process.

From the study, following conclusions can be drawn out:

- Increasing the combustion chamber diameter provides higher smoke levels for a given u/u_0. It proves that changes on swirl intensity have other effects which affect greatly the combustion process.

- Smoke vs u/u_0 curves for a given piston find an overall minimum, while several minima are found in fuel consumption curves, depending on the cylinder head. Nevertheless, these minima are all settled around 0.9-1 of u/u_0, and belong to the ranges that literature considers optimal for parameters AFMRC and CWVC.

- Cylinder heads C1 and C3 present lower specific fuel consumptions than the others, even when piston or manifolds are changed. This fact coincides with a higher turbulence production during the intake process. These differences on turbulence level do not disappear during the compression stroke and may affect the combustion process.

- Nitric Oxide emissions present a light decrease when u/u_0 increases, and depend on the intake pressure in the sense of increasing when intake pressure does. Tests with 55 mm chamber diameter provide much lower NO emissions.

7. NOMENCLATURE

D = cylinder bore
Db = bowl diameter
CDM = mean discharge coefficient
Gef = specific fuel consumption
Gf = idem, including pump losses
K = bowl volume/total volume at TDC
L = duct length
m = total mass in the cylinder
m_f = total fuel mass/cycle
N_h = number of holes of the injector
Q = frequency parameter of the inlet duct
SNM = mean swirl number
u/u_0 = tangential velocity at the bowl wall, related to reference test (P1-C1-D1)
u_{it} = tangential component of the jet inpingement velocity
U_0 = injection velocity
V_D = swept volume
ω = swirl velocity

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